SECTION
A

PUMPS
AND
PUMPING
MACHINERY

CHAPTER 1

Water

As an introduction to the study of hydraulics and pumps, the reader should first consider water and its characteristics.

Water is a most remarkable substance.

Its behavior under different conditions is very extraordinary. That is to say, under the influence of temperature water may be converted into a gas and employed as a medium for developing power, as steam applied to a steam engine.

Again when subjected to low temperatures, it is converted into a solid (ice) which because of its peculiar characteristic of expanding during its change of state, causes pipes to burst and does other damage.

By definition, water is *a compound of hydrogen and oxygen in the proportion of* 2 *parts by weight of hydrogen to* 16 *parts by weight of oxygen.*

Ques. What is the most remarkable characteristic of water?

4°C

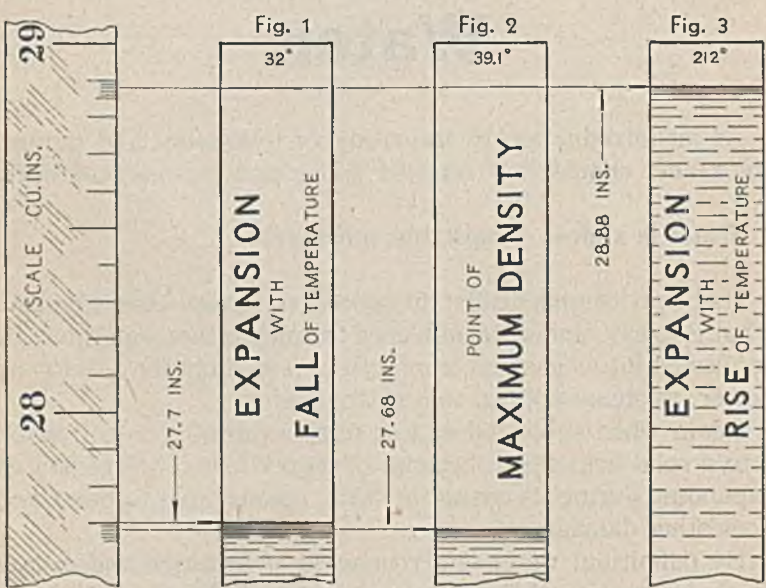
Ans. At maximum density (39.1° Fahr.) water will expand as heat is added and it will expand slightly as the temperature falls from this point, as shown in figs. 1 to 3.

Ques. What is the freezing and boiling points of water at atmospheric pressure at the sea level?

Ans. It will freeze at 32° Fahr. and boil at 212° Fahr. when the barometer reads 29.921 ins. (1 atm)

Ques. What is the reading 29.921 ins.?

Ans. The standard atmosphere.



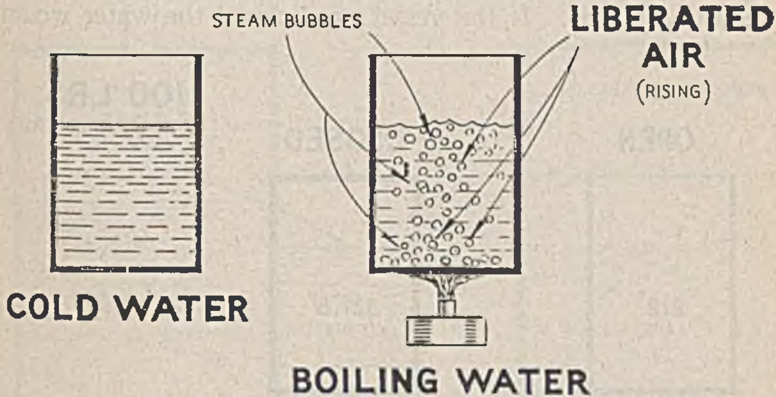
Figs. 1 to 3.—The most remarkable characteristic of water: **expansion below and above its temperature or "point of maximum density"** 39.1° **Fahr.** Imagine one pound of water at 39.1° Fahr. placed in a cylinder having a cross sectional area of 1 sq. in. as in fig. 2. The water having a volume of 27.68 cu. ins., will fill the cylinder to a height of 27.68 ins. If the liquid be cooled it will expand, and at say the **freezing point** 32° Fahr., will rise in the tube to a height of 27.7 ins., as in fig. 1, before freezing. Again, if the liquid in fig. 2 be heated, it will also expand and rise in the tube, and at say the **boiling point** (for atmospheric pressure 212° Fahr.), will occupy the tube to a height of 28.88 cu. ins. as in fig. 3.

Ques. What is its equivalent in pounds per square in.?

Ans. 14.696 lbs. per sq. in. $1\ kg/cm^2$

Ques. What else besides hydrogen and oxygen is contained in water?

Ans. Water contains mechanically mixed with it about 5 per cent of air by volume.



Figs. 4 and 5.—Cold and boiling water illustrating the liberation of air **"mechanically mixed"** in the cold water.

For this reason condensing steam engines must have air pumps attached to the condenser otherwise the necessary *vacuum* could not be maintained.

Ques. Give a striking example of the air contained in water.

Ans. The operation of steam heating plants.

Since this air is liberated when the water boils it passes into the radiators with the steam and accordingly automatic air valves must be provided to rid the system of this air otherwise the radiators would become air bound and rendered ineffective.

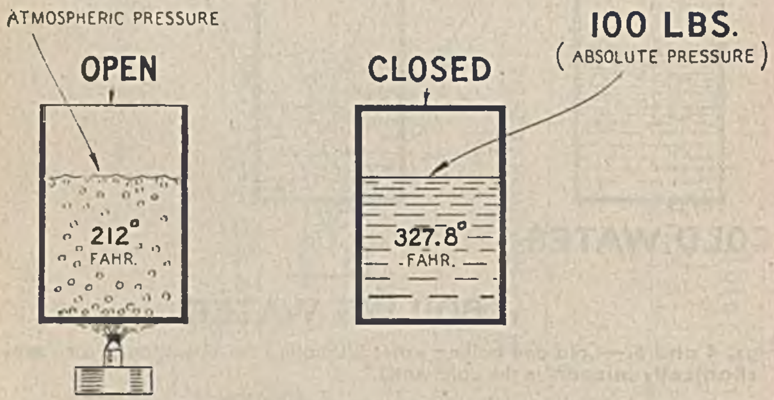
Ques. How does pressure affect the boiling point of water?

Ans. The boiling point rises with the pressure.

Thus, it is 212° Fahr. at sea level atmospheric pressure, and at say 100 lbs. (absolute) pressure it is 327.8° Fahr.

Ques. Why is the water "boiling" in fig. 6 and not boiling in fig. 7?

Ans. Fig. 6 shows water boiling (as in a tea kettle) by the addition of heat. If the vessel were closed the water would



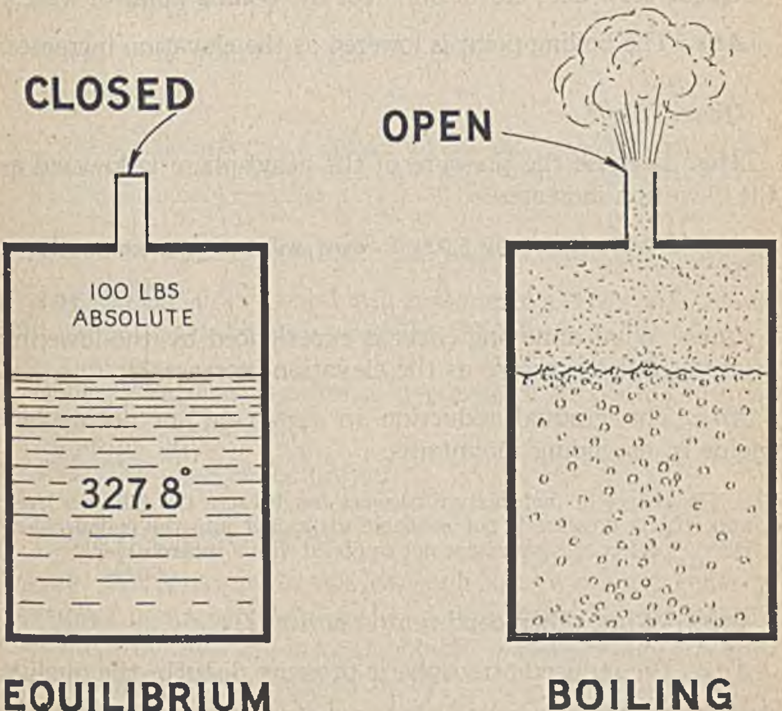
Figs. 6 and 7.—Water boiling at atmospheric pressure and at 100 lbs. pressure absolute. **Note temperatures.**

continue to boil which would cause the pressure to rise. Now in fig. 7, if no more heat be added when the pressure rose to say 100 lbs. the water would cease boiling and the pressure would remain constant if no heat were lost.

The temperature of the water, steam and pressure are said to be in a state of equilibrium. The least variation of temperature (either up or down) would destroy the "state of equilibrium" and cause a change.

Ques. How could the water (fig. 8) be made to boil again?

Ans. By letting some of the confined steam escape, as in fig. 9.



Figs. 8 and 9.—Equilibrium between temperature and pressure (fig. 8) and equilibrium upset by reduction of pressure (fig. 9) resulting in **"boiling."**

Ques. What happens?

Ans. On the escape of steam a reduction of pressure takes place and equilibrium of the system is disturbed. The water (containing excess heat) immediately begins to boil and tends to keep the pressure constant.

If this process be continued a gradual reduction in temperature and pressure will result until all the heat originally put into the system is used up.

Ques. How does elevation affect the boiling point of water?

Ans. The boiling point is lowered as the elevation increases.

Ques. Why?

Ans. Because the pressure of the atmosphere is lowered as the elevation increases.

At an elevation of say 5,000 ft. water will boil at a temperature of 202° Fahr.

Ques. What annoying effect is experienced by the lowering of atmospheric pressure as the elevation increases?

Ans. The gradual reduction in power of an automobile engine in ascending mountains.

The charge of fuel mixture becomes *less* because the full (sea level) atmospheric pressure is not available to push it into the cylinders—an inherent defect of a gas engine not equipped with a supercharger.

Ques. What other condition is noticed?

Ans. The reduced atmospheric pressure disturbs the quality of the mixture.

That is, the proportion of air entering the carburetor is reduced. The author in an ascent of Smoky Mountain had to adjust the secondary air valve four times before reaching the top.

Ques. What domestic operation is impossible at high altitudes?

Ans. Eggs cannot be boiled.

Ques. What cooking utensil depends for its operation on the effect of pressure on the boiling point?

Ans. The pressure cooker.

Ques. Must water be "hot" to boil?

Ans. No.

The popular idea that a liquid must be "hot" to boil is a wrong conception. For instance water under a 28 in. vacuum will boil at 100° Fahr.; if the vacuum be increased to 29.74 it will boil at 32° Fahr.

Ques. At what temperature can water both boil and freeze?

Ans. At 32° Fahr. water will boil under a 29.74 in. vacuum and freeze at atmospheric pressure (14.7 lbs.).

Ques. What cooking utensil depends for its operation upon a variation of pressure?

Ans. The glass coffee brewer.

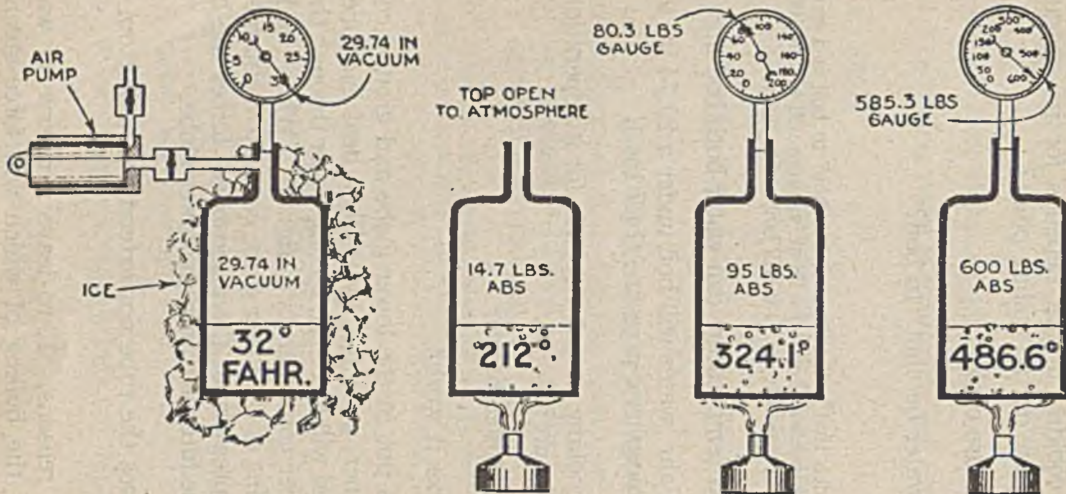
Ques. How does it work?

Ans. Water is put in the lower globe and ground coffee in the upper container, as in fig. 14. When heat is applied, the pressure generated will force the boiling water through the filter and into the upper container as in fig. 15. When the heat is shut off, the cooling of the lower globe causes a vacuum to form therein and the pressure of the atmosphere forces the brewed coffee through the filter and into the lower globe.

Ques. Why does the tube not extend to the bottom of the lower globe?

Ans. For two reasons. 1, to leave a reserve of water in the lower globe, the boiling of which will force steam into

Effect of Pressure on Boiling Point



Figs. 10 to 13.—Variation of the boiling point with change of pressure. Taking water to illustrate, when the pressure is—

Vacuum	Absolute	Gauge	Gauge
29.74 ins.	14.7 lbs.	80.3 lbs.	585.3 lbs.

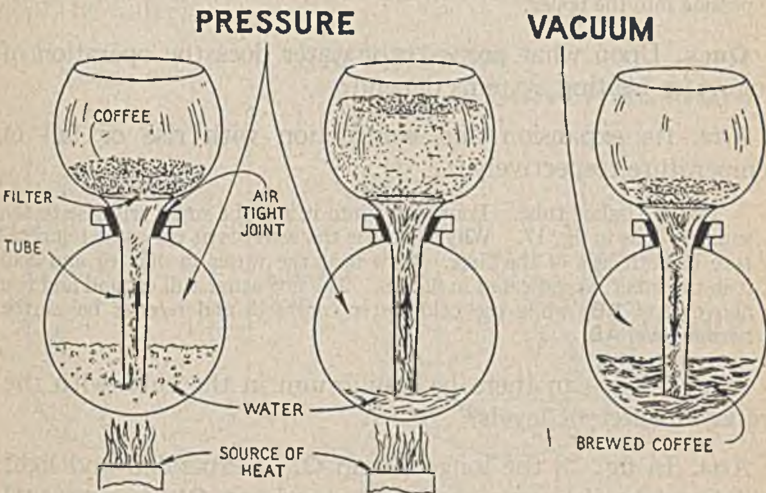
The water will boil at a temperature (Fahr.) of—

32°	212°	324.1°	486.6°

Highly different values are obtained for other liquids.

the container and cook the coffee; 2, to prevent the lower globe becoming dry at once with probable breakage due to temporary inattention.

Ques. Why is it some waters give considerable trouble in boiler operation?



Figs. 14 to 16.—The familiar glass coffee brewer illustrating the variation of pressure upon which its operation depends. Fig. 14, heat applied, pressure generated and the water being forced into the upper container; fig. 15, completion of the upflow part of the cycle; fig. 16, cooling period producing vacuum which causes excess pressure of atmosphere to force the liquid down into the lower globe.

Ans. All waters are not pure, and in most cases contain ingredients that form scale which is precipitated on heating and adheres to the heating surfaces of the boiler.

Scale in boilers may be of hard rock-like nature, or of soft greasy, or powdery nature, according to its chemical and mechanical composition or formation.

Ques. What is the effect of scale in boilers?

Ans. It is a very poor conductor of heat, which results in waste of coal and overheating of the metal of the heating surface.

For these reasons boilers have to be cleaned frequently and in some cases special chemical treatment must be given to the feed water before passing into the boiler.

Ques. Upon what property of water does the operation of hot water heating systems depend?

Ans. Its expansion and contraction with rise or fall of temperature respectively.

Take a U-glass tube. Pour water into it and the water will rise to the same level as in fig. 17. Why? Because the water is at the same temperature in both legs of the tube. Now heat the water in one leg and cool it in the other, as indicated in fig. 18. The hot water will expand and rise above level **AB**, while the cold water contracts and recedes below the normal level **AB**.

Ques. How can there be equilibrium in the tube with the water at different levels?

Ans. In fig. 18 the long column **C**, of expanded and light water weighs the same as the short column **C′** of contracted and heavy water.

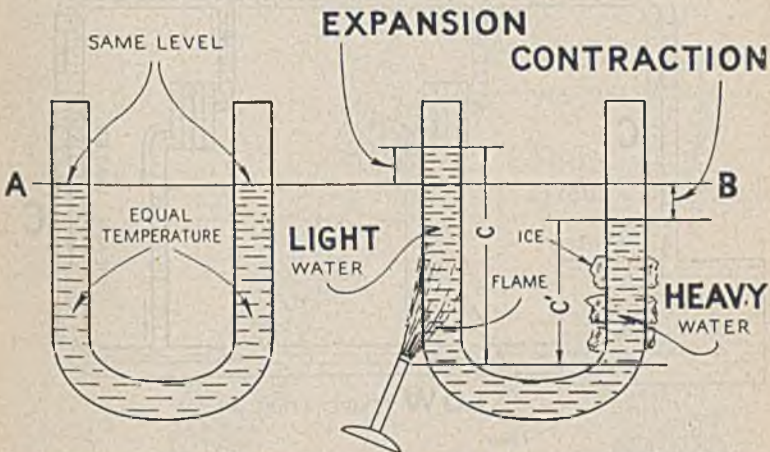
Ques. Explain why the expansion and contraction of water is the operating principle of hot water heating systems.

Ans. Fig. 19 shows an elementary hot water heating system. The weight of the hot and expanded water in the up flow column **C**, being less than that of the cold and contracted water in the down flow column **C′** upsets the equilibrium of the system and results in a continuous circulation of water as indicated by the arrows.

Ques. What is the kind of circulation in a hot water heating system called?

Ans. Thermo circulation, which is just another way of saying circulation due to heat.

Ques. What would happen to a boiler were there no circulation of the water?



Figs. 17 and 18.—Glass U-tube partially filled with water illustrating expansion and contraction of water with variation in temperature and resulting change in weight per unit volume.

Ans. There would be practically no generation of steam except for a film of steam separating the water from the heating surface. The latter would become red hot and the boiler probably destroyed.

Ques. What is the state just described called?

Ans. The spheroidal state.

Thus in fig. 21 pour a small quantity of water on a red hot plate. The water will separate into drops and ride all around the plate being supported by a thin film of steam. Thus the water (after steam has formed) not being in contact with the plate has practically no cooling effect on the plate.

Weight of Water.—The property of water of varying in weight (lbs. per unit volume) due to changes in temperature,

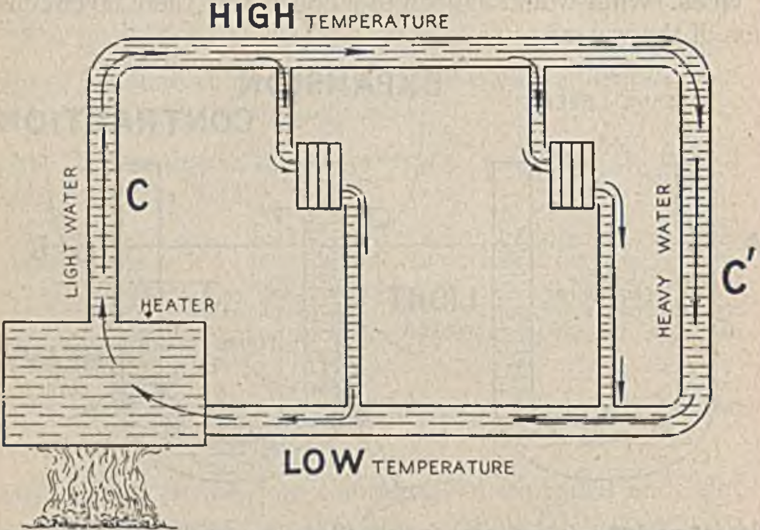


Fig. 19.—Elementary hot water heating system illustrating **thermo-circulation.**

giving rise to circulation in boilers and heating systems as has just been explained, is evidently a very important property.

Ques. Is the statement that a U. S. gallon of water (231 cu. ins.) weighs 8.33111 lbs. (ordinarily expressed as 8⅓ lbs.) accurate?

Ans. No.

Ques. Why?

Ans. The U. S. gallon of water weighs 8.33111 lbs. only at the standard temperature of 62° Fahr.; at any other temperature the weight is different.

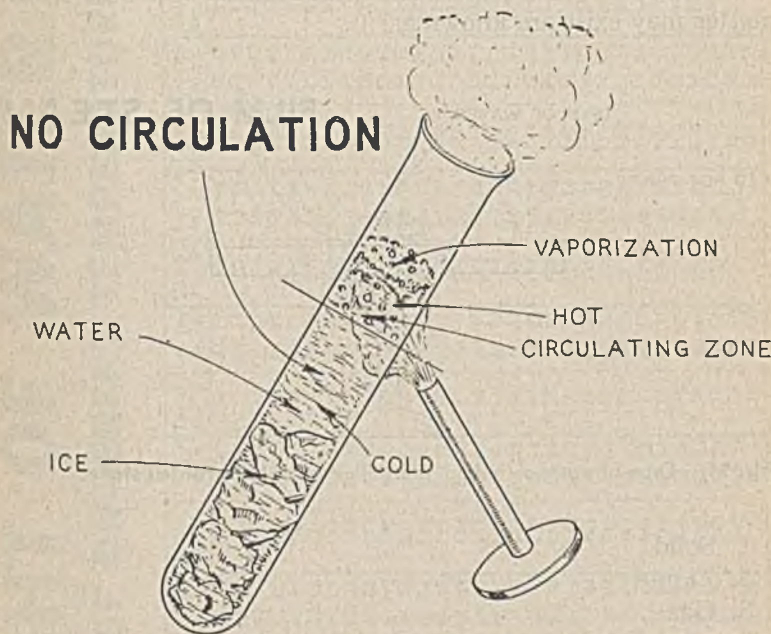


Fig. 20.—Experiment illustrating effect of no circulation. Anchor some ice at the bottom of a test tube and fill to near the top with water. Apply heat near the surface of the water. This will cause the water at that point to boil and no heat will reach the ice. Why? The cold water around the ice is heavier than the hot water at the top which condition prevents thermo circulation. If the heat be applied at the bottom of the tube circulation will begin, the ice will melt, and all the water will be vaporized if the heat be applied long enough.

For instance a gallon of water at 34° Fahr. weighs — lbs. and at 200° Fahr. only — lbs.

For ordinary calculations at ordinary temperatures, the weight is taken at 8⅓ lbs. which is close enough in most cases. However, it should be understood this is only approximate. Evidently for precision, the weight at the given temperature should be taken.

The Three States of Matter.—The three states in which matter may exist are known as:

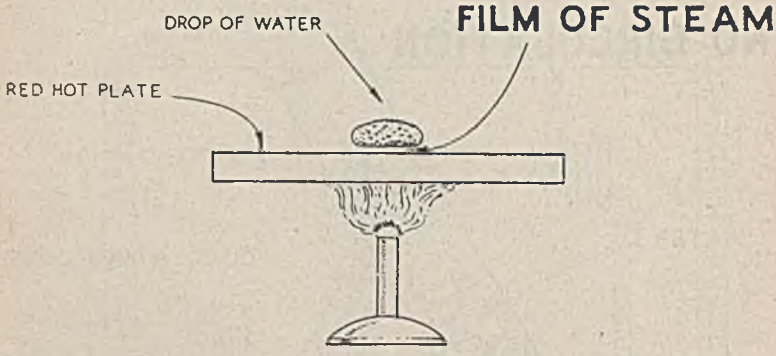


Fig. 21.—Drop of water on red hot plate illustrating **spheroidal state.**

1. Solid
2. Liquid
3. Gas

A familiar example of one substance existing in the three states is

1. Ice
2. Water
3. Steam

as shown in figs. 22 to 24.

Change of State.—*By sufficiently increasing the temperature,* **solids** *are converted into* **liquids** *and liquids into* **gases.**

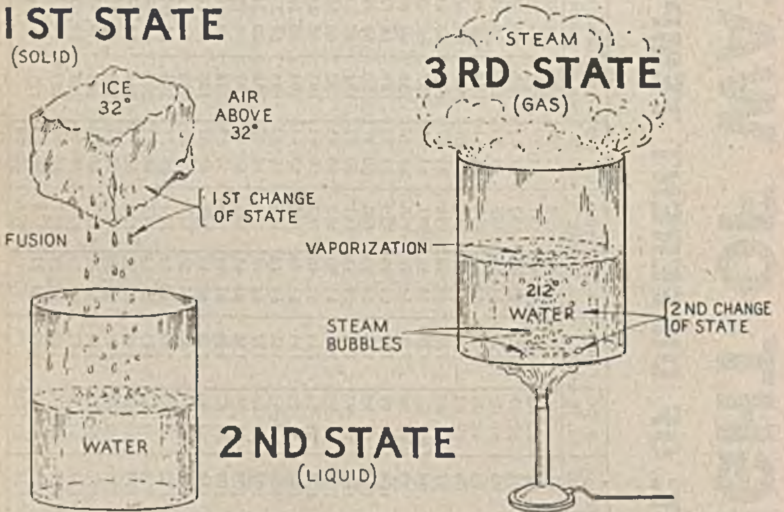
WEIGHT OF WATER

PER CU. FT. OF DIFFERENT TEMPERATURES

Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.	Temp., deg. F.	Lb. per cu. ft.
32	62.41	62	62.35	91	62.10	121	61.69	150	61.19	180	60.57	208	59.92	440	51.87
33	62.41	63	62.34	92	62.08	122	61.68	151	61.17	181	60.55	209	59.90	450	51.28
34	62.42	64	62.34	93	62.07	123	61.66	152	61.15	182	60.53	210	59.87	460	51.02
35	62.42	65	62.33	94	62.06	124	61.64	153	61.13	183	60.51	211	59.85	470	50.51
36	62.42	66	62.32	95	62.05	125	61.63	154	61.11	184	60.49	212	59.82	480	50.00
37	62.42	67	62.32	96	62.04	126	61.61	155	61.09	185	60.46	214	59.81	490	49.50
38	62.42	68	62.31	97	62.02	127	61.60	156	61.07	186	60.44	216	59.77	500	48.78
39	62.42	69	62.30	98	62.01	128	61.58	157	61.05	187	60.42	218	59.70	510	48.31
40	62.42	70	62.30	99	62.00	129	61.56	158	61.03	188	60.40	220	59.67	520	47.62
41	62.42	71	62.29	100	61.99	130	61.55	159	61.01	189	60.37	230	59.42	530	46.95
42	62.42	72	62.28	101	61.98	131	61.53	160	60.99	190	60.35	240	59.17	540	46.30
43	62.42	73	62.27	102	61.96	132	61.51	161	60.97	191	60.33	250	58.89	550	45.66
44	62.42	74	62.26	103	61.95	133	61.50	162	60.95	192	60.30	260	58.62	560	44.84
45	62.42	75	62.25	104	61.94	134	61.48	163	60.93	193	60.28	270	58.34	570	44.05
46	62.41	76	62.25	105	61.93	135	61.46	164	60.91	194	60.26	280	58.04	580	43.29
47	62.41	77	62.24	106	61.91	136	61.44	165	60.89	195	60.23	290	57.74	590	42.37
48	62.41	78	62.23	107	61.90	137	61.43	166	60.87	196	60.21	300	57.41	600	41.49
49	62.41	79	62.22	108	61.89	138	61.41	167	60.85	197	60.19	310	57.08	610	40.49
50	62.40	80	62.21	109	61.87	139	61.39	168	60.83	198	60.16	320	56.75	620	39.37
51	62.40	81	62.20	110	61.86	140	61.37	169	60.81	199	60.14	330	56.40	630	38.31
52	62.40	82	62.19	111	61.84	141	61.36	170	60.79	200	60.11	340	56.02	640	37.17
53	62.39	83	62.18	112	61.83	142	61.34	171	60.77	201	60.09	350	55.65	650	35.97
54	62.39	84	62.17	113	61.81	143	61.32	172	60.75	202	60.07	360	55.25	660	34.48
55	62.38	85	62.16	114	61.80	144	61.30	173	60.73	203	60.04	370	54.85	670	32.89
56	62.38	86	62.15	115	61.78	145	61.28	174	60.71	204	60.02	380	54.47	680	31.06
57	62.38	87	62.14	116	61.77	146	61.26	175	60.68	205	59.99	390	54.05	690	28.82
58	62.37	88	61.13	117	61.75	147	61.25	176	60.66	206	59.97	400	53.62	700	25.38
59	62.37	89	62.12	118	61.74	148	61.23	177	60.64	207	59.95	410	53.19	706.1	19.16
60	62.36	90	62.11	119	61.72	149	61.21	178	60.62	.....	......	420	52.74	.....	......
61	62.35			120	61.71			179	60.60			430	52.33		

Thus in fig. 22, if the temperature of the air surrounding the cake of ice rise above 32° Fahr. the ice will melt, that is, **fusion** takes place resulting in a change of state from solid to liquid, as in fig. 23.

Again if heat be added to the water as in fig. 24 raising its temperature to 212° Fahr. a second change of state will take place at atmospheric pressure, converting the water into steam, that is, **vaporization** takes place.



Figs. 22 to 24.—The three states of matter: solid, liquid and gas, as explained in the text. Note the first change of state from solid to liquid is called **fusion** and the second change of state from liquid to gas **vaporization.**

In this case 212° Fahr. is the boiling point but as before pointed out, it should be remembered that the boiling point depends upon the surrounding pressure. In this connection water can boil and freeze at the same time. See fig. 25.

Ques. Give a familiar example of cooling by *re-evaporation.*

Ans. In steam engine operation, as shown in figs. 26 to 28.

Ques. What is the mistaken idea about *re-evaporation* even among some engineers.

Ans. They regard it as a loss which in fact is not the case.

Re-evaporation, since it increases the area of the indicator diagram from the point L, fig. 26, up to the point of *pre-release* represents a *gain*.

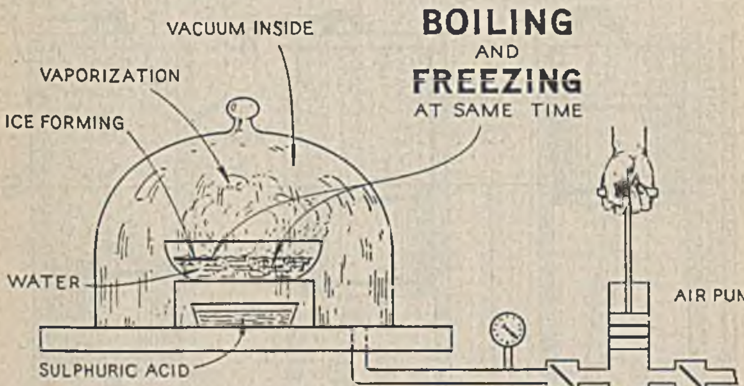


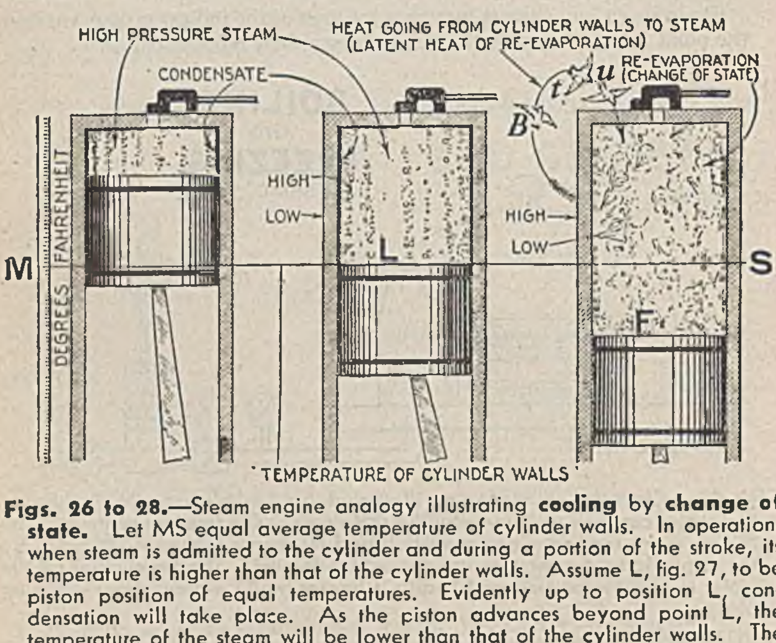
Fig. 25.—Change of state, as illustrated by Leslie's experiment showing that water can boil and freeze at the same time. On removal of the air with the air pump the water begins to boil and the vapor formed is absorbed by the sulphuric acid almost as rapidly as it is formed. The temperature of the water is quickly lowered and finally it freezes while vaporization is taking place.

Ques. How do they get the mistaken idea?

Ans. It is the price or cost of re-evaporation which is the loss and not re-evaporation considered alone.

That is, re-evaporation robs the cylinder walls of an amount of heat corresponding to the *latent heat of re-evaporation*. This extra cooling of the cylinder walls increases the amount of condensation during the first part of the stroke, and this is the loss. Since this loss is greater than the gain due to re-evaporation they ignorantly call re-evaporation a loss. A rather slipshod way of talking.

Expansion of Water.—The table on page 20 gives the relative volume of water at different temperatures compared with its volume at 4° C. according to Kopp as corrected by Porter.



Figs. 26 to 28.—Steam engine analogy illustrating **cooling** by **change of state.** Let MS equal average temperature of cylinder walls. In operation, when steam is admitted to the cylinder and during a portion of the stroke, its temperature is higher than that of the cylinder walls. Assume L, fig. 27, to be piston position of equal temperatures. Evidently up to position L, condensation will take place. As the piston advances beyond point L, the temperature of the steam will be lower than that of the cylinder walls. The excess heat of the cylinder walls will cause the condensate to boil, that is re-evaporation will take place, which will rob the cylinder walls of some of its heat.

Nature of Water with Respect to Pump Design.—Those who have had experience in the design of pumps, soon found out that water is practically an unyielding substance when confined in pipes and pump passages, thus necessitating very substantial

Pounds Per Square Inch to Feet (Head) of Water

Based on water at its greatest density

Pressure Pounds Per Square Inch	Feet Head	Pressure Pounds Per Square Inch	Feet Head	Pressure Pounds Per Square Inch	Feet Head	Pressure Pounds Per Square Inch	Feet Head	Pressure Pounds Per Square Inch	Feet Head	Pressure Pounds Per Square Inch	Feet Head	Pressure Pounds Per Square Inch	Feet Head
1	2.31	53	122.43	105	242.55	157	362.67	209	482.79	261	602.91	365	843.15
2	4.62	54	124.74	106	244.86	158	364.98	210	485.10	262	605.22	370	854.70
3	6.93	55	127.05	107	247.17	159	367.29	211	487.41	263	607.53	375	866.25
4	9.23	56	129.36	108	249.48	160	369.60	212	489.72	264	609.84	380	877.80
5	11.55	57	131.67	109	251.79	161	371.91	213	492.03	265	612.15	385	899.35
6	13.86	58	133.98	110	254.10	162	374.22	214	494.34	266	614.46	390	900.90
7	16.17	59	136.29	111	256.41	163	375.53	215	496.65	267	616.77	395	912.45
8	18.48	60	138.60	112	258.72	164	378.84	216	498.96	268	619.08	400	924.00
9	20.79	61	140.91	113	261.03	165	381.15	217	501.27	269	621.39	405	931.55
10	23.10	62	143.22	114	263.34	166	383.46	218	503.58	270	623.70	410	947.10
11	25.41	63	145.53	115	265.65	167	385.77	219	505.89	271	626.01	415	958.65
12	27.72	64	147.84	116	267.96	168	388.08	220	508.20	272	628.32	420	970.20
13	30.03	65	150.15	117	270.27	169	390.39	221	510.51	273	630.63	425	981.75
14	32.34	66	152.46	118	272.58	170	392.70	222	512.82	274	632.94	430	993.30
15	34.65	67	154.77	119	274.89	171	395.01	223	515.13	275	635.25	435	1004.85
16	36.96	68	157.08	120	277.20	172	397.32	224	517.44	276	637.56	440	1016.40
17	39.27	69	159.39	121	279.51	173	399.63	225	519.75	277	639.87	445	1027.95
18	41.58	70	161.70	122	281.82	174	401.94	226	522.06	278	642.18	450	1039.50
19	43.89	71	164.01	123	284.13	175	404.25	227	524.37	279	644.49	455	1051.05
20	46.20	72	166.32	124	286.44	176	406.56	228	526.68	280	646.80	460	1062.60
21	48.51	73	168.63	125	288.75	177	408.87	229	528.99	281	649.11	465	1074.15
22	50.82	74	170.94	126	291.06	178	411.18	230	531.30	282	651.42	470	1085.70
23	53.13	75	173.25	127	293.37	179	413.49	231	533.61	283	653.73	475	1097.25
24	55.44	76	175.56	128	295.68	180	415.80	232	535.92	284	656.04	480	1108.80
25	57.75	77	177.87	129	297.99	181	418.11	233	538.23	285	658.35	485	1120.35
26	60.06	78	180.18	130	300.30	182	420.42	234	540.54	286	660.66	490	1131.90
27	62.37	79	182.49	131	302.61	183	422.73	235	542.85	287	662.97	495	1143.45
28	64.68	80	184.80	132	304.92	184	425.04	236	545.16	288	665.28	500	1155.00
29	66.99	81	187.11	133	307.23	185	427.35	237	547.47	289	667.59	525	1212.75
30	69.30	82	189.42	134	309.54	186	429.66	238	549.78	290	669.90	550	1270.50
31	71.61	83	191.73	135	311.85	187	431.97	239	552.09	291	672.21	575	1328.25
32	73.92	84	194.04	136	314.16	188	434.28	240	554.40	292	674.52	600	1386.00
33	76.23	85	196.35	137	316.47	189	436.59	241	556.71	293	676.83	625	1443.75
34	78.54	86	198.66	138	318.78	190	438.90	242	559.02	294	679.14	650	1501 50
35	80.85	87	200.97	139	321.09	191	441.21	243	561.33	295	681.45	675	1559.25
36	83.16	88	203.28	140	323.40	192	443.52	244	563.64	296	683.76	700	1617.00
37	85.47	89	205.59	141	325.71	193	445.83	245	565.95	297	686.07	725	1674.75
38	87.78	90	207.90	142	328.02	194	448.14	246	568.26	298	688.38	750	1732.50
39	90.09	91	210.21	143	330.33	195	450.45	247	570.57	299	690.69	775	1790.25
40	92.40	92	212.52	144	332.64	196	452.76	248	572.88	300	693.00	800	1848.00
41	94.71	93	214.83	145	334.95	197	455.07	249	575.19	305	704.55	825	1905.75
42	97.02	94	217.14	146	337.26	198	457.38	250	577.50	310	716.10	850	1963.50
43	99.33	95	219.45	147	339.57	199	459.69	251	579.81	315	727.65	875	2021.25
44	101.64	96	221.76	148	341.88	200	462.00	252	582.12	320	739.20	900	2079.00
45	103.95	97	224.07	149	344.19	201	464.31	253	584.43	325	750.75	925	2136.75
46	106.26	98	226.38	150	346.50	202	466.62	254	586.74	330	762.30	950	2194.50
47	108.57	99	228.69	151	348.81	203	468.93	255	589.05	335	773.85	975	2252.25
48	110.88	100	231.00	152	351.12	204	471.24	256	591.36	340	785.40	1000	2310.0.
49	113.19	101	233.31	153	353.43	205	473.55	257	593.67	345	796.95	1500	3465.
50	115.50	102	235.62	154	355.74	206	475.86	258	595.98	350	808.50	2000	4620.
51	117.81	103	237.93	155	358.05	207	478.17	259	598.29	355	820.05	3000	6930.
52	120.12	104	240.24	156	360.36	208	480.48	260	600.60	360	831.60		

construction to withstand the pressure and especially the periodic shocks or water hammer. Accordingly in pump design a liberal factor of safety should be used.

Pressure of Water at Different Depths.—The pressure of water varies with the *head* and is equal to .43302 lbs. per sq. in. for every foot of static head. Thus a head of 2.31 ft. gives a pressure of $2.31 \times .43302 = 1$ lb. as per accompanying table.

Expansion of Water

Cent.	Fahr.	Volume.	Cent.	Fahr.	Volume.	Cent.	Fahr.	Volume.
4°	39 1°	1.00000	35°	95°	1.00586	70°	158°	1.02241
5	41	1.00001	40	104	1.00767	75	167	1.02548
10	50	1.00025	45	113	1.00967	80	176	1.02872
15	59	1.00083	50	122	1.01186	85	185	1.03213
20	68	1.00171	55	131	1.01423	90	194	1.03570
25	77	1.00286	60	140	1.01678	95	203	1.03943
30	86	1.00425	65	149	1.01951	100	212	1.04332

Compressibility of Water.—Water is very slightly compressible. According to Kent its compressibility is from .00004 to .000051 for one atmosphere, decreasing with increase of temperature. For each foot of pressure, distilled water will be diminished in volume .0000015 to .0000013. Water is so incompressible that even at a depth of a mile a cubic foot of water will weigh only about ½ lb. more than at the surface.

CHAPTER 2

Physics

By definition, physics is *that branch of science which treats of the laws and properties of matter and the forces acting upon it.* Only such items will be here considered as are necessary for a proper understanding of the chapters which follow. This may be regarded as an introduction to Physics, additional basic principles being given in various parts of the book.

Pressure.—By definition, pressure is *a force of the nature of a thrust, distributed over a surface.* In other words, it is *the kind of force which when applied to a body tends to compress it*

For instance, fig. 1 shows a spring in its natural state. If a force be applied to it in the direction of its axis, it will be compressed, as in fig. 2. The resistance offered by the spring constitutes an opposing force, equal and opposite in direction to the force applied.

Ques. When a certain force (say 2 lbs.) is applied to the spring, how much will the spring be compressed?

Ans. Since the resistance of the spring increases with the degree of compression, it will be compressed to a point such that its resistance equals the pressure applied.

Ques. What is the condition of the system shown in fig. 2?

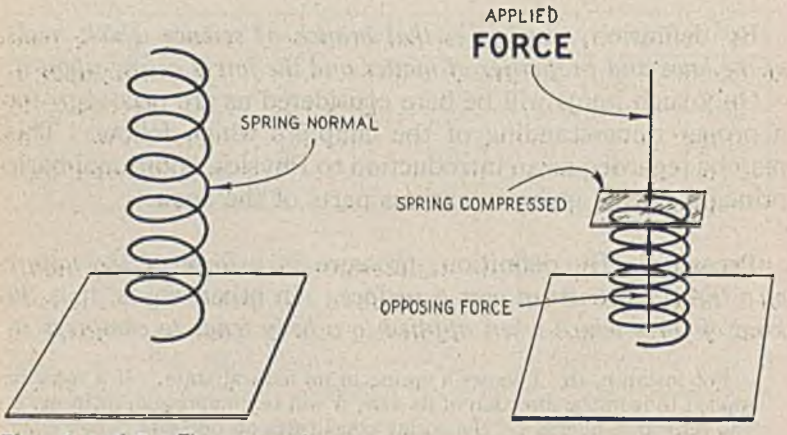
Ans. It is said to be in a state of equilibrium.

Ques. What should be carefully noted about pressure?

Ans. It is considered as being distributed over a surface.

Ques. How is the pressure distributed over a surface usually stated?

Ans. It is stated in terms of the pressure distributed over a



Figs. 1 and 2.—The nature of pressure.

unit area *of the surface*, most usually as pounds per square inch (abbreviated as lbs. per sq. in.).

Example.—A pump plunger has an area of 10 sq. in. What is the total pressure acting on the plunger when pumping against 125 lbs. per sq. in. as in fig. 3?

$$10 \text{ sq. ins.} \times 125 \text{ lbs. per sq. in.} = 1250 \text{ lbs.}$$

that is, 125 lbs. pressure acts on each square inch of the plunger and since its acting face has an area of 10 sq. ins. the total pressure is 1250 lbs.

Ques. Why is the ball peen of a machinist's hammer made spherical?

Ans. To reduce the area of surface upon which the blow acts so as to apply a terrific amount of force to a very small area of surface. This is what the ball peen hammer does.

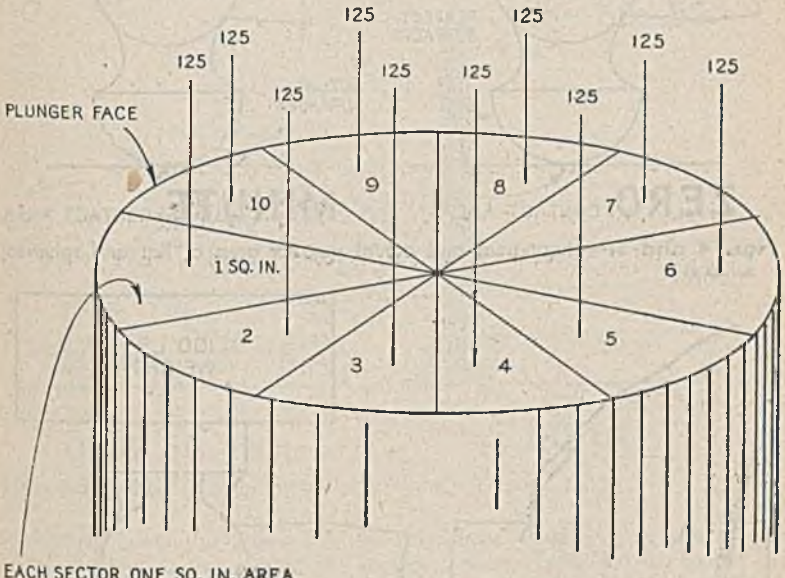
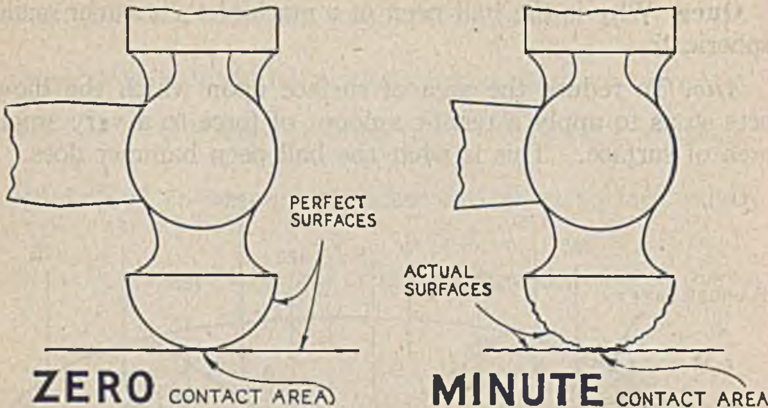


Fig. 3.—Pressure per square inch (sq. in.).

The ball peen hammer is generally used for peening* or riveting operations.

Figs. 4 and 5 show theoretical and actual conditions. In fig. 4 if the flat and spherical surfaces were ideal or perfect, when brought into contactt the contact area would be zero. However, there are no actual perfec, surfaces. The most polished surfaces (as seen under the microscope) look

*NOTE.—***By definition,** peening is the operation of hammering metal so as to indent or compress it in order to expand or stretch that portion of the metal adjacent to the indentation.



Figs. 4 and 5.—Theoretical and actual contact area of flat and spherical surfaces.

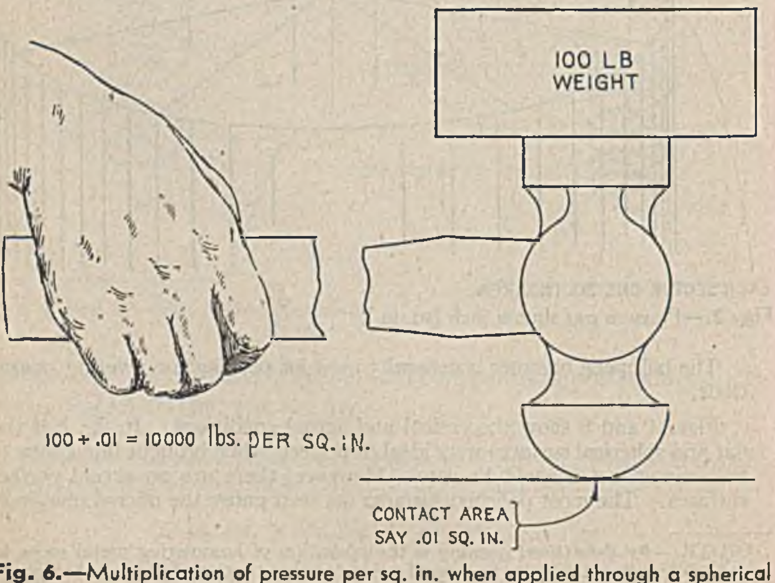
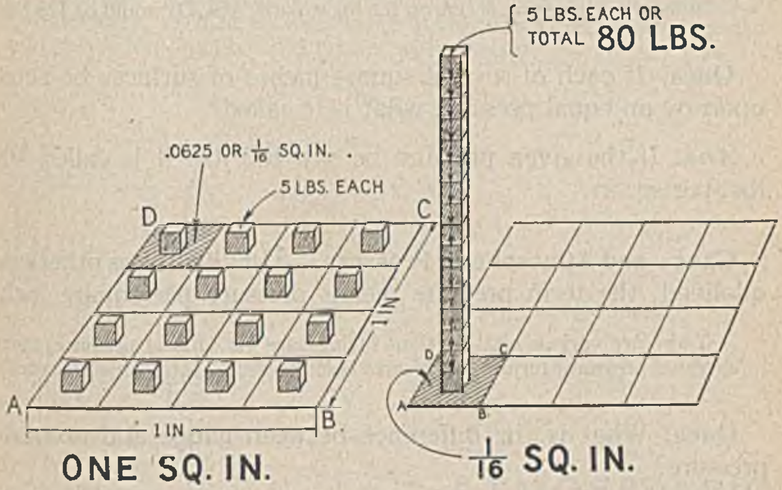


Fig. 6.—Multiplication of pressure per sq. in. when applied through a spherical contact surface.

something like emery paper. Accordingly, there would be a minute contact area as in fig. 5.

Example.—If the ball peen of a machinist's hammer be placed in contact with a flat surface (as in fig. 6) and 100 lbs. weight be placed on the hammer (neglecting weight of hammer) what pressure will be exerted at the point of contact if the contact area be say .01 sq. in.?



Figs. 7 and 8.—Diagram illustrating pressure per sq. in.

If the contact area were 1 sq. in. then there would be a pressure of 100 lbs. for the sq. in. on the flat surface. Now, if the 100 lb. weight or pressure act on only .01 sq. in. as in the figure, there would be a pressure of

$$100 \div \frac{1}{100} = 100 \times \frac{100}{1} = 10{,}000 \text{ lbs. per sq. in.}$$

Another example will perhaps make the matter plainer:

In fig. 7 suppose a surface ABCD measure one sq. in. Divide this into 16 little squares and put a 5 lb. weight on each little square. Now the area of each little square is 1/16 or .0625 sq. in. Pile up all the 5 lb. weights on one little square as in fig. 8. Then the weight or pressure acting on this little square is

$$5 \times 16 = 80 \text{ lbs.}$$

In fig. 7 the 16 weights are distributed over one sq. in. corresponding to 80 lbs. per sq. in. In fig. 8, the 16 weights or 80 lbs. are distributed over only $\frac{1}{16}$ sq. in. corresponding to

$$80 \times 16 = 1{,}280 \text{ lbs. per sq. in.}$$

That is, if in fig. 8 a pile of 16 weights be placed on each little square the total weight or pressure acting on the big square (ABCD) would be 1280 lbs.

Ques. If each of several square inches of surfaces be acted upon by an equal pressure what is it called?

Ans. If the given pressure be say 100 lbs. it is called 100 lbs. per sq. in.

Gauge and Atmospheric Pressure.—Usually unless otherwise qualified, the term pressure means pressure per square inch.

There are various qualifications of pressure viz.: initial pressure; mean effective pressure; terminal pressure; back pressure, total pressure, etc.

Ques. What is the difference between gauge and absolute pressure?

Ans. Gauge pressure is *pressure* **above** *that of atmospheric pressure*, and absolute pressure is *pressure measured* **above** a perfect vacuum.

Ques. What is a perfect vacuum?

Ans. A space devoid of matter and one in which the pressure is zero.

Ques. What is the pressure of the atmosphere?

Ans. The "standard atmosphere" is 14.696 lbs. per sq. in. at sea level ordinarily taken as 14.7 lbs. per sq. in.

Thus, in fig. 9, if an air-tight and frictionless piston having an area of 1 sq. in. be connected to a weight by a string running over a pulley, as shown, it will require a weight weighing 14.696 lbs. to draw the weight upward from the bottom of the cylinder against the pressure of the atmosphere, which is 14.696 lbs. distributed over the top face of the piston whose area is one sq. in. Strictly speaking, the system is in a "*state of equilibrium*," the weight exactly balancing the resistance or weight of the atmosphere. A little excess pressure would be required for any movement of the piston, but it would remain where placed.

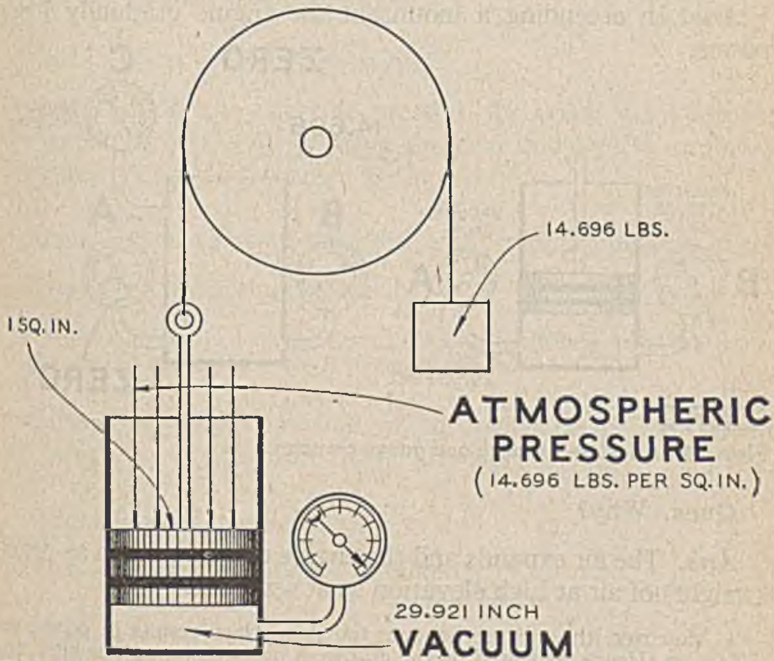


Fig. 9.—Diagram illustrating atmospheric pressure.

Ques. Why do we not feel the pressure of the atmosphere?

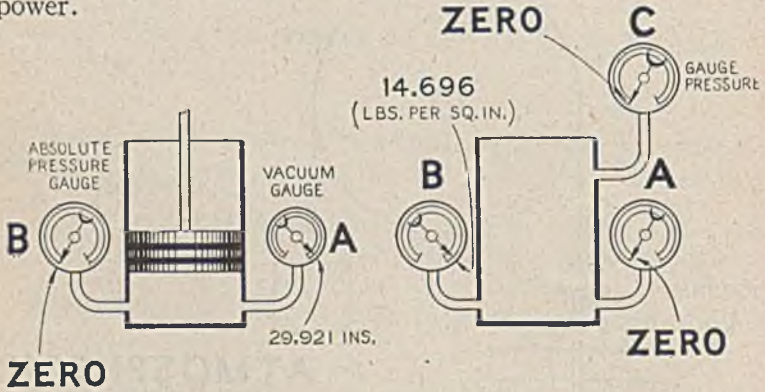
Ans. Because air presses the body both externally and internally so that the pressures in different directions balance.

Ques. How does atmospheric pressure vary with the elevation?

Ans. It decreases approximately ½ lb. for every 1,000 ft. of ascent.

Ques. What effect has this decrease in atmospheric pressure on gas engine operation?

Ans. In ascending a mountain the engine gradually loses power.



Figs. 10 and 11.—Absolute and gauge pressures.

Ques. Why?

Ans. The air expands and the engine cannot take in as much (weight) of air at high elevation as at sea level.

Moreover, the mixture becomes too rich, which results in poor combustion. Hence, as stated by the author in his book on automobiles, the need of super-chargers on all automobiles, especially for mountain climbing, and moreover on account of the inherent defect of the gas engine in not being able to take in a full charge.

Ques. Does the pressure of the atmosphere remain constant in any one place?

Ans. No. It continually varies depending upon the conditions of the weather.

Fig. 10 shows a cylinder and piston with a perfect vacuum below the piston registered by the gauge **A**, as 29.921 ins. of mercury (later explained). The equivalent of this in absolute pressure is zero lbs. per sq. in. as registered by gauge **B**. Now remove the piston as in fig. 11, and air will rush into the cylinder. That is, the vacuum will be replaced by air atmospheric pressure. Under this condition the vacuum gauge **A**, will drop to zero and the absolute pressure gauge will register 14.696 lbs.

Ques. What is "gauge pressure?"

Ans. This is the effective pressure for doing work against the pressure of the atmosphere as measured by the ordinary pressure gauge, as by gauge **C**, fig. 11.

Ques. How is absolute pressure expressed as gauge pressure?

Ans. By subtracting 14.696 lbs.

In figs. 10 and 11 comparing gauges **B** and **C**, it will be noted that gauge **C** gives the reading of gauge **B** minus 14.696 lbs.

Barometer.—By definition, a barometer is an instrument for measuring the pressure of the atmosphere.

Ques. How is it measured?

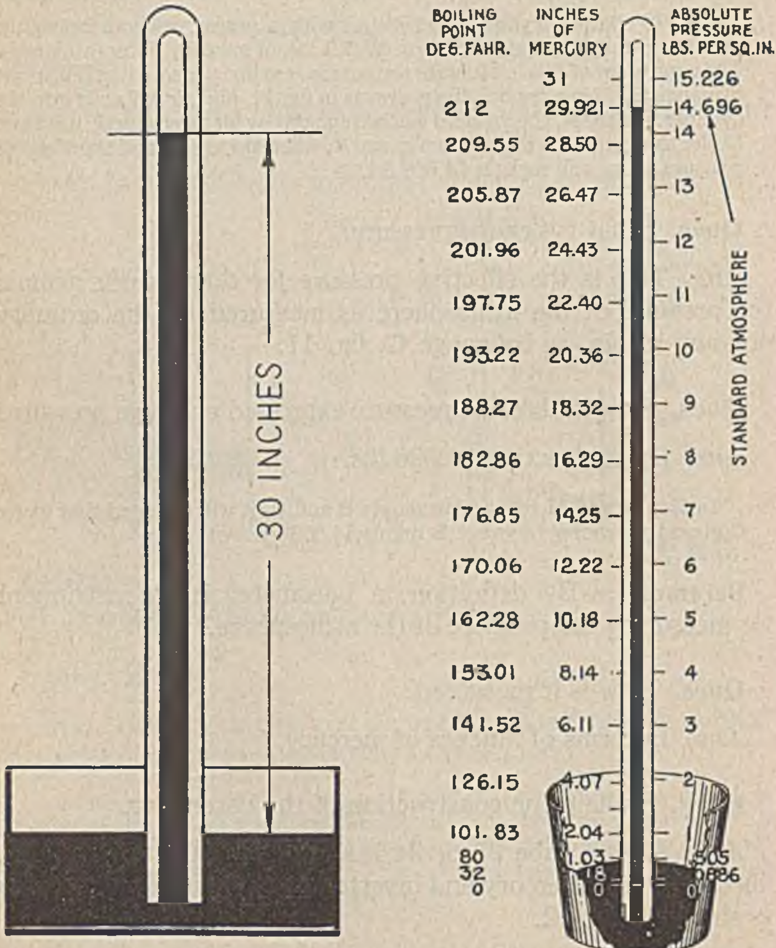
Ans. In terms of "inches of mercury."

Ques. Explain the construction of the barometer.

Ans. A glass tube 33 or 34 ins. long is sealed at one end, filled with pure mercury and inverted in an open cup of mercury as shown in fig. 12.

Ques. How does it work?

Ans. The mercury will fall in the tube until its height above the mercury in the cup is about 30 ins. as in fig. 12.



Figs. 12 and 13.—Principle of the barometer and barometer with scales of boiling points, inches of mercury and absolute pressure per sq. in.

Ques. Why does it remain suspended at this height?

Ans. Because the weight of a column of mercury 30 ins. high is the same as the weight of a like column of air about 50 miles high.

Ques. Does the pressure of the atmosphere remain constant in any one place?

Ans. No. It continually varies, depending upon the conditions of the weather.

Ques. Is the pressure of the atmosphere the same in different places?

Ans. It varies with the elevation as previously explained.

Ques. How are pressures below that of the atmosphere usually expressed?

Ans. As pounds per square inch absolute in making calculations, or the equivalent in inches of mercury in practice.

Thus in the engine room, the expression "24 in. vacuum" would signify an absolute pressure in the condenser of .946 lbs. per sq. in. absolute, that is the mercury in a mercury column connected to a condenser having a 28 in. vacuum would rise to a height of 28 ins. representing the difference between the pressure of the atmosphere and the pressure in the condenser, or

$$14.75 - .946 = 13.804 \text{ lbs.}$$

Ques. How is the pressure in pounds per square inch obtained from the barometer reading?

Ans. Multiply the barometer reading by .49116. .49116 is the pressure per square inch corresponding to a column of mercury 1 in. high having a cross section of 1 sq. in.

Example.—What is the absolute pressure corresponding to a 20 in barometer reading?

$$.49116 \times 20 = 9.82 \text{ lbs.}$$

The following table gives the pressure of the atmosphere in pounds per square inch for various readings of the barometer.

Pressure of the atmosphere per square inch for various readings of the barometer:

Rule.—*Barometer in inches of mercury* × .49116 = *lbs. per sq. in.*

Barometer (ins. of mercury)	Pressure per sq. ins., lbs.	Barometer (ins. of mercury)	Pressure per sq. ins., lbs.
28.00	13.75	**29.921**	**14.696**
28.25	13.88	30.00	14.74
28.50	14.00	30.25	14.86
28.75	14.12	30.50	14.98
29.00	14.24	30.75	15.10
29.25	14.37	31.00	15.23
29.50	14.49		
29.75	14.61		

The above table is based on the standard atmosphere, which by definition = **29.921** ins. of mercury = **14.696** lbs. per sq. in., that is 1 in. of mercury = 14.696 ÷ 29.921 = **.49116** lbs. per sq. in.

Matter.—By definition matter is any substance or material that can be weighed or measured.

Ques. In what three forms does matter exist?

Ans. As a solid, liquid or gas.

Ques. What is the difference between matter and a body?

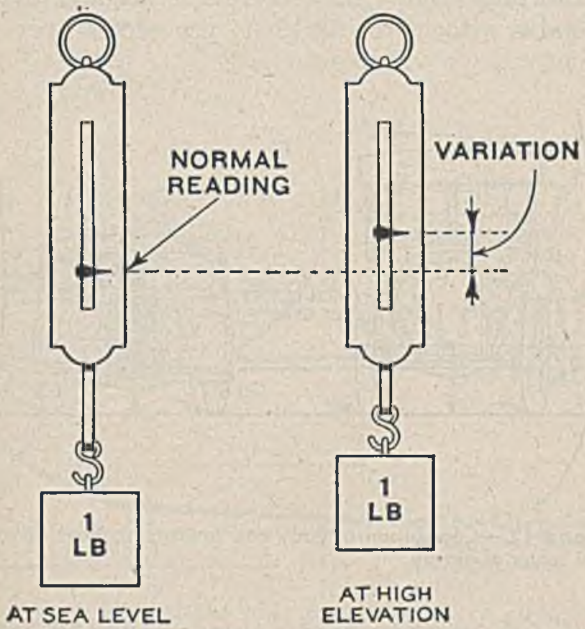
Ans. A body is a definite quantity of matter.

Ques. How is the quantity of matter in a body determined?

Ans. By weighing on a lever or platform scale or on a spring scale.

Ques. Which scale gives a true reading in all locations?

Ans. The lever or platform scale.



Figs. 14 and 15.—Variations in reading of a spring scale for latitude and elevation. Since the value of **g** (attraction due to gravity increases with latitude and decreases with the elevation) evidently there will be more or less variation of a spring scale reading for different localities.

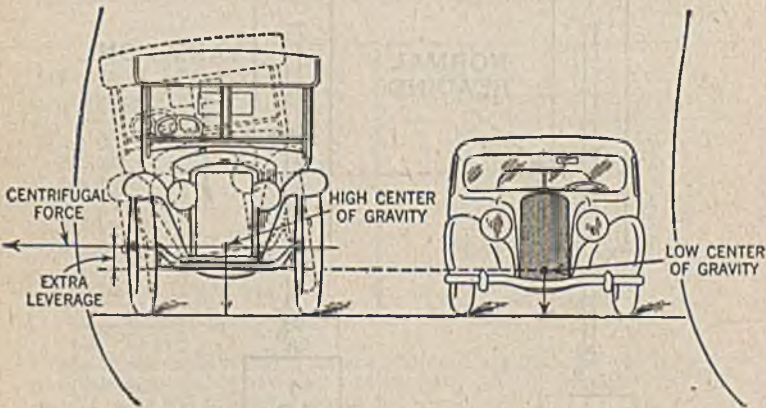
Ques. Why does the reading of a spring scale vary?

Ans. Since *weight* depends upon *gravity* and since gravity varies with latitude and elevation the reading of a spring scale will vary as in figs. 14 and 15.

Gravity.—By definition, gravity is *a force that attracts bodies, at or near the surface of the earth toward the center of the earth.*

Ques. What effect has gravity upon a body free to fall?

Ans. Starting from a state of rest it acquires during the first second a velocity of 32.16 ft. per second per second;



Figs. 16 and 17.—Comparison of early and late automobiles with respect to height of center of gravity.

at the end of the second second a velocity of $32.16 + 32.16 = 64.32$ ft. per second and so on.

Ques. What is the symbol for 32.16?

Ans. *g*.

Center of Gravity.—By definition the center of gravity is *that point of a body about which all its parts are balanced, or* *which being supported, the whole will remain at rest, though acted upon by gravity.*

High and low center of gravity is very forcibly illustrated in early and late construction of automobiles, figs. 16 and 17.

Centrifugal Force.—By definition, centrifugal force is, *that force which acts upon a body moving in a circular path tending to force it farther from the axis or center of the circle described by its path.*

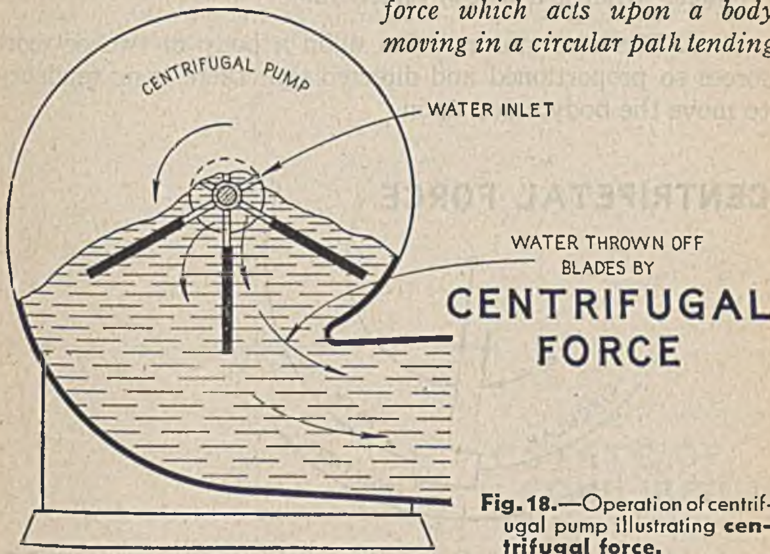


Fig. 18.—Operation of centrifugal pump illustrating **centrifugal force.**

The most familiar example of centrifugal force is the centrifugal pump as shown in fig. 18.

If the centrifugal force be just sufficient to balance the attraction of the mass around which it revolves, the moving body will continue in a uniform path.

Centripetal Force.—By definition, *that force which draws or impels a body toward some point as a center.*

The centripetal force is *that force which resists the centrifugal force and when these opposing forces are equal, the moving body will revolve in a circular path as shown in fig.* 19, that is the system is in a state of equilibrium.

Ques. What is a *state of equilibrium?*

Ans. The resultant reaction upon a body of two or more forces so proportioned and directed that there is no tendency to move the body acted upon.

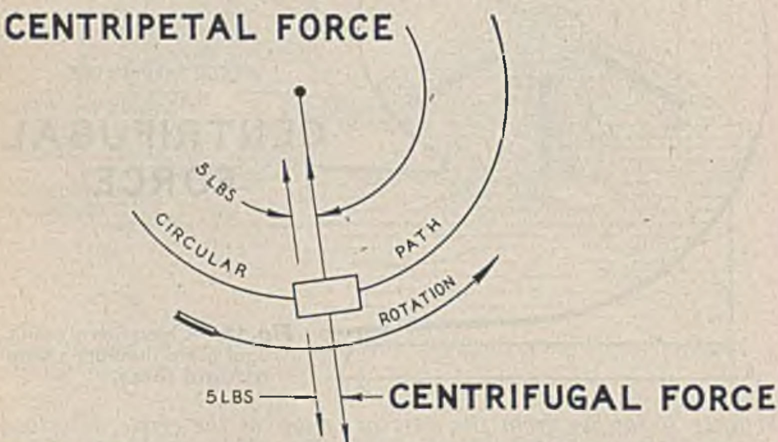


Fig. 19.—State of equilibrium between centrifugal and centripetal forces.

Thus if a body **O**, as in fig. 20, be acted upon by two equal opposed forces **OA** and **OC**, and two equal opposed forces **OB** and **OD**, these various forces will balance each other and the resultant reaction upon the body **O**, is zero, that is, the body will remain in a state of rest.

Forces, Their Composition and Resolution.—A force is completely defined when its direction, magnitude and point of application are defined. It is possible to represent all these requirements by a line so that its direction length and location corresponds to any given conditions.

Thus, in fig. 21, to represent a force of 4,000 lbs. select any convenient scale, as 1 in. = 1,000 lbs. Draw the line **AB** 4 ins. long, in the direction and at the point where the force acts. Then **AB** will completely represent the force and its application. Note the arrow head is placed at the point where the force acts.

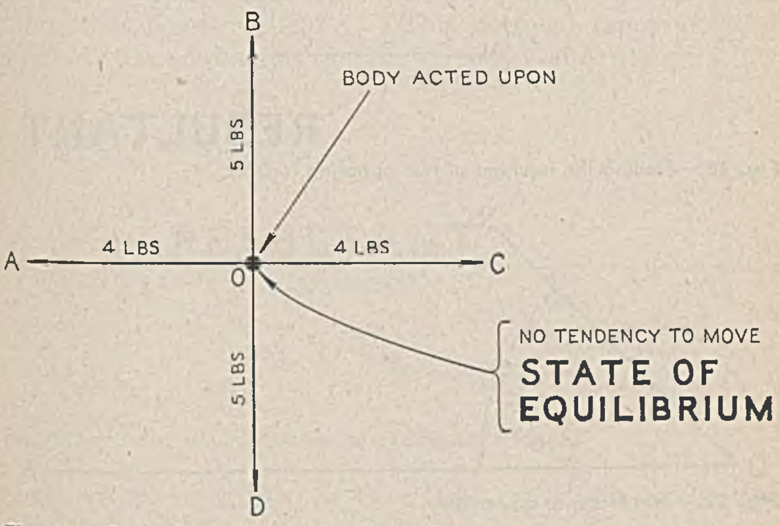


Fig. 20.—State of equilibrium of opposing forces.

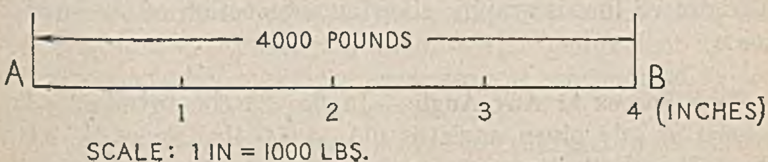


Fig. 21.—Representation of a force, its intervals and direction by a line. The arrow head denotes the point of application of the force and direction.

Two Opposed Forces.—In fig. 22 let OA and OB be the two forces acting in opposed directions on the point O. Suppose force OA = 4,000 lbs. and OB 2,000 lbs. then these two opposed forces could be replaced by a single force equal to

$$4{,}000 - 2{,}000 = 2{,}000 \text{ lbs.}$$

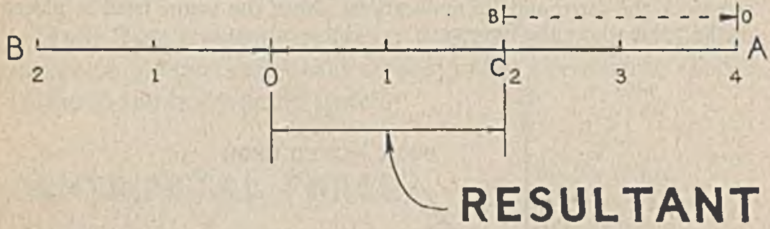


Fig. 22.—Finding the resultant of two opposing forces.

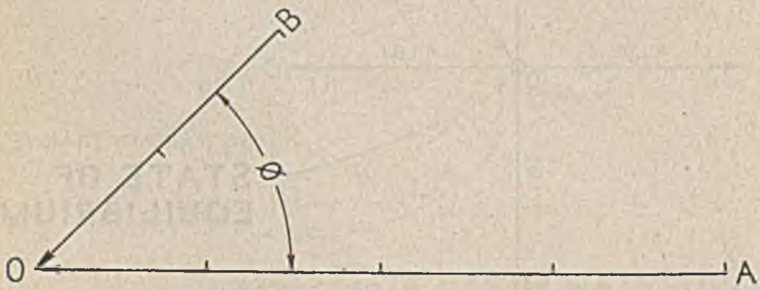


Fig. 23.—Two forces at any angle.

represented by OC. This is the *resultant* of forces OA and OB. The dotted line is graphic showing subtraction of the smaller force.

Two Forces at Any Angle.—In fig. 23 the two forces are shown at any given angle as OA, 4,000 lbs. and OB, 2,000 lbs. Here both forces act at the common point O at an angle to each other equal to the given angle Θ.

Ques. What must be done to find the resultant of these forces?

Ans. Construct a parallelogram with the two given forces as adjacent sides.

Parallelogram of Force.—Let the two forces OA and OB in fig. 24 be the same as in fig. 23. Now draw the dotted line BC, parallel with OA and AC with OB. Draw the diagonal connecting points O and C. This *diagonal* represents the *resultant* force which acting in direction and intensity (as

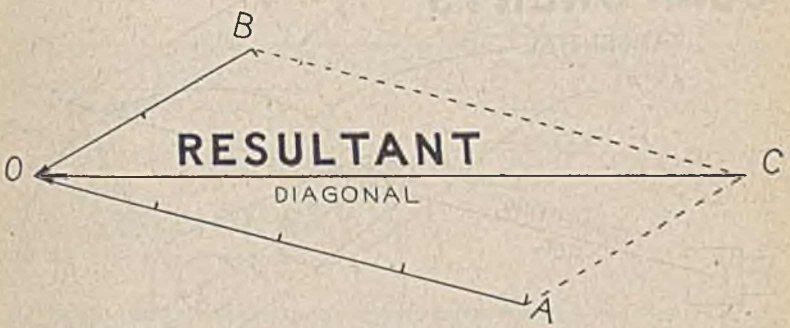


Fig. 24.—Finding resultant by the parallelogram of forces.

measured by its length) at O_1 will be equivalent to the two forces OA and OB.

Components of a Force.—This is the reverse of finding the resultant of two forces. By definition, a component is *one of the forces out of which the whole may be compounded by the principle of the parallelogram of force.*

As a familiar example, consider the reaction due to the thrust of a connecting rod on the crank pin as in fig. 25. This thrust is split up into two component forces.

One force acts in the direction of a tangent to the circle described by the crank pin which causes the crank to turn, and the other acts in the direction of the axis of the crank arm, which causes the shaft to press against its bearing.

In construction, from **O** project to **C** in length equal to the thrust of the connecting rod. Complete the parallelogram of forces, thus obtaining the points **B** and **A,** their lengths **OB** and **OA** representing the components in direction and intensity.

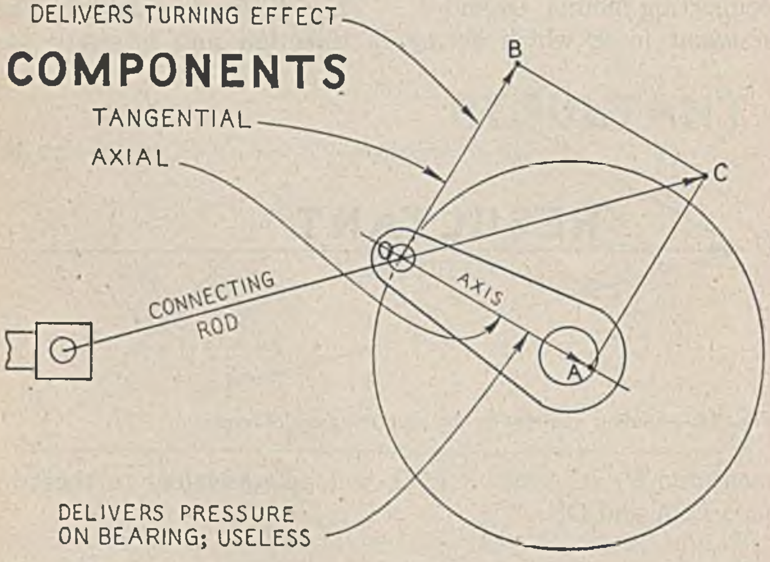


Fig. 25.—Method of finding the components of a force.

Motion.—The author has explained so many times "what is motion" that it seems unnecessary to go into this again, However: One definition of motion is *a change of position in relation to some assumed fixed point.* However, the old definition best describes it viz.:

Motion is purely a relative matter.

Ques. Why is motion a relative matter?

Ans. Because something must be regarded as being stationary in order that there be motion.

Thus in fig. 26 a man is rowing a bateau at a speed of 4 miles per hour against a current flowing 2 miles per hour in the opposite direction. Unless

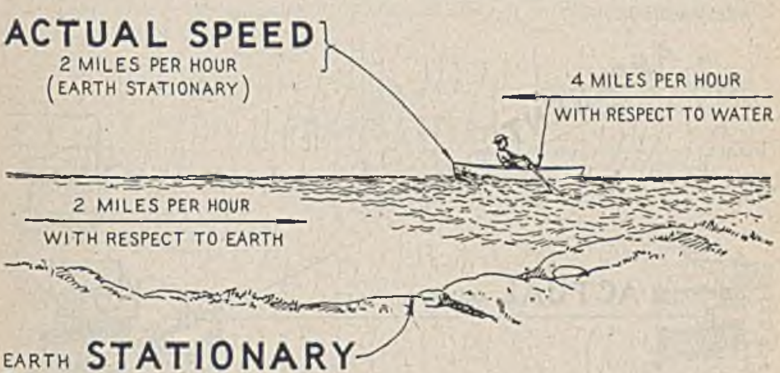


Fig. 26.—Graphic of motion.

something be regarded as stationary, these statements would mean nothing. Thus in fig. 26

Stationary	*Actual speed of boat*
Earth	2 miles per hour
Water	4 " " "

Again in fig. 27 shows the familiar example of a ferryboat crossing a river and pointed up steam to counteract the motion of the water. **OA** is the apparent motion of boat (both distance and direction) but regarding the earth as stationary **OB**, gives the actual direction in which the boat is moving.

If the water be regarded as stationary the boat would be actually moving in the direction **OA.**

Ques. Name two kinds of velocity.

Ans. Linear and rotary.

Thus a train is running at a rate of 30 miles per hour (abbreviated *m.p.h.*) and a line shaft is rotating at a rate of 125 revolutions per minute (abbreviation *r.p.m.*).

To illustrate further, a speed boat is shown in fig. 28 making 30 *m.p.h.* or ½ mile per miunte. This is *linear motion.* To drive the boat at this rate of speed the propeller must turn at a rate of 1,000 *r.p.m.* This is *rotary motion.*

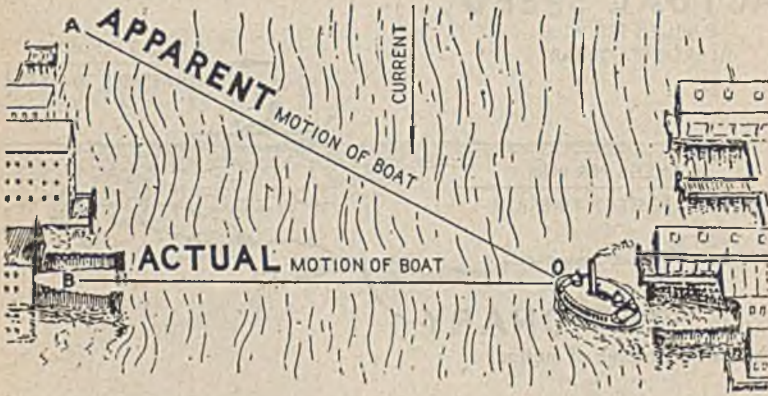


Fig. 27.—Apparent and actual motion.

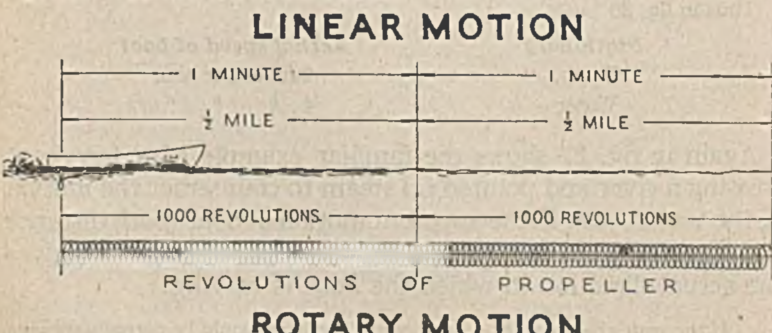


Fig. 28.—Linear and rotary motion.

Ques. What is tangential motion?

Ans. The equivalent of rotary motion regarded as moving in a straight or tangential direction.

Tangential motion or speed is used especially in belting calculation. Thus in fig. 29, a pulley one foot in diameter has a circumference of 3.1416 ft. Accordingly for each revolution of the pulley the belt will travel a distance **AB**, or 3.1416 ft.

Example.—A 4-ft. pulley is rotating at 100 *r.p.m.* What is the tangential speed of the belt?

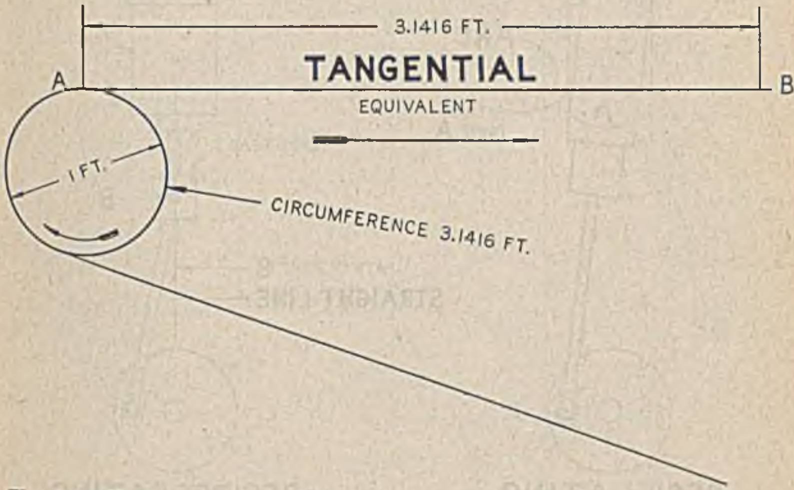


Fig. 29.—The tangential equivalent of rotary motion.

$$\text{Tangential speed} = \text{circumference} \times r.p.m.$$
$$= 4 \times 3.1416. \times 100$$
$$= 12.566 \times 100 = 1{,}256.6 \text{ ft. per minute.}$$

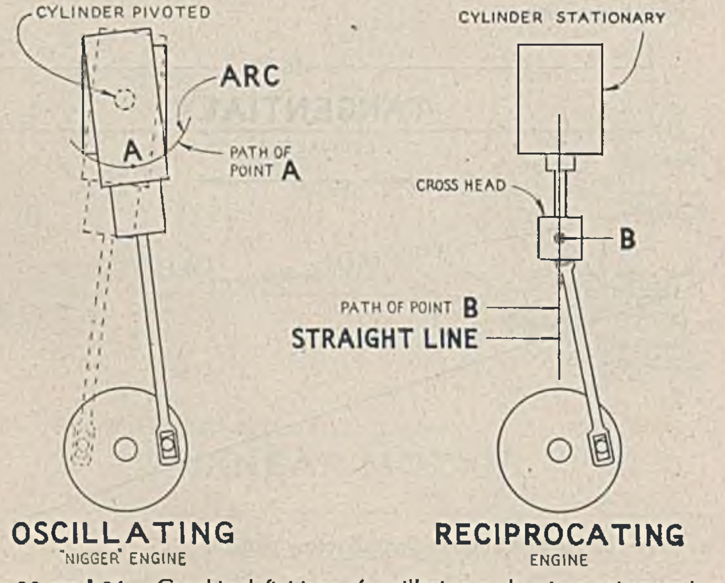
Ques. What is the difference between oscillating and reciprocating motion?

Ans. By definition oscillating motion is vibrating motion in the path of an arc of a circle. Reciprocating motion is vibrating motion in the path of a straight line.

The motion of a clock pendulum is a familiar example of oscillating motion. The movement of an engine cross head illustrates reciprocating motion.

Further illustrated by fig. 30 showing an oscillating or "nigger" engine employed sometimes on shipboard as a capstan engine.

Note the to and fro path of any point as **A**, lies in the arc of a circle whose center is the center of the pivot about which the cylinder oscillates.



Figs. 30 and 31.—Graphic definitions of oscillating and reciprocating motions.

Note in fig. 31, the cylinder is stationary and the movement of a point as **B**, on the cross head moves up and down or reciprocates in a straight line path.

Ques. What is the difference between *constant* and *variable* motion?

Ans. Constant motion is the movement of point through equal *spaces in equal intervals of time.* Varying motion is the movement of a point through **unequal** spaces in equal intervals of time.

A familiar illustration of this is seen in the movements of the crank pin and piston of an engine as shown by the diagram fig. 32. If **A,** be the position of the crank pin at the beginning of the stroke, its rate of motion

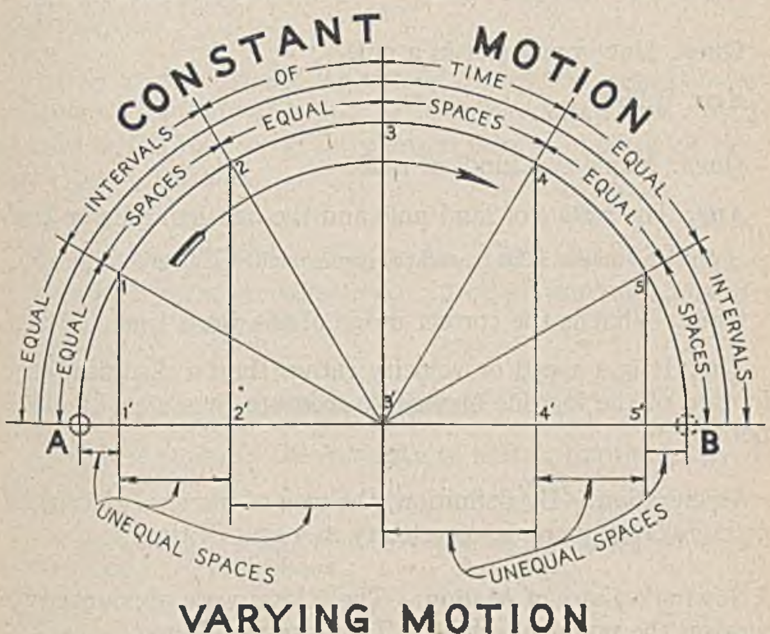


Fig. 32.—Constant and varying motions.

is constant as it rotates to **B.** That is, it passes through equal arcs in equal intervals of time. Perpendiculars let fall at the end of each arc (as 1, 2, 3, etc.) give points 1′ 2′, 3′, etc., which represent the corresponding positions of the piston. Evidently the spaces traversed (**A**1′, 1′, 2′, etc.) are unequal and the motion is *varying.**

*NOTE.—This diagram represents the true relation where there is no distortion as with a Scotch yoke connection, but in the case of a connecting rod there is distortion due to the "angularity of the connecting rod." For explanation of this see Audels Engineers and Mechanics Guide Volume 1, pages 154 to 156.

Velocity.—By definition, velocity is *the speed at which a body moves, that is the rate of motion of a body at any instant.*

Ques. How is velocity measured?

Ans. Usually as feet per minute or miles per hour.

Ques. How many feet in a mile?

Ans. 5,280.

Ques. Name two kinds of mile.

Ans. The *statute* or land mile and the nautical mile or *knot.*

The land mile is 5,280 ft. and the nautical mile 6,080.28 ft.

Ques. What is the correct usage of the word *knot.*

Ans. It is a speed or velocity rather than a distance, each division of the log line serving to measure the rate of a ship's motion.*

Acceleration.—By definition, the rate of increase of velocity or the average increase of velocity in a unit of time.

Newton's Laws of Motion.—These laws were announced by Newton the noted physicist. They are as follows:

1st Law.—

If a body be at rest, it will remain at rest, or if in motion it will move uniformly in a straight line until acted upon by some force.

---

*NOTE.—Each knot on the line bears the same proportion to the mile that 30 seconds do to an hour. The number of knots which run off from the reel in half a minute therefore shows the number of nautical miles the vessel sails in an hour.

2nd Law.—

If a body be acted on by several forces, it will obey each as though the others did not exist, and this whether the body be at rest or in motion.

3rd Law.—

If a force act to change the state of a body with respect to rest or motion, the body will offer a resistance equal and directly opposed to the force; or to every action there is opposed an equal and opposite reaction.

Momentum.—By definition, *the power of a body of overcoming resistance by virtue of its motion.* In other words *the* **quantity** *of motion in a moving body.*

Ques. How is momentum measured?

Ans. It is equal to the quantity of matter multiplied by its velocity.

Numerically it is equal to *the force in pounds steadily applied that will stop a moving body in one second.*

Hence momentum is equal to the mass multiplied by the velocity in feet per second or

$$\text{Momentum} = \frac{\text{weight}}{32.16} \times \text{velocity in feet per second}$$

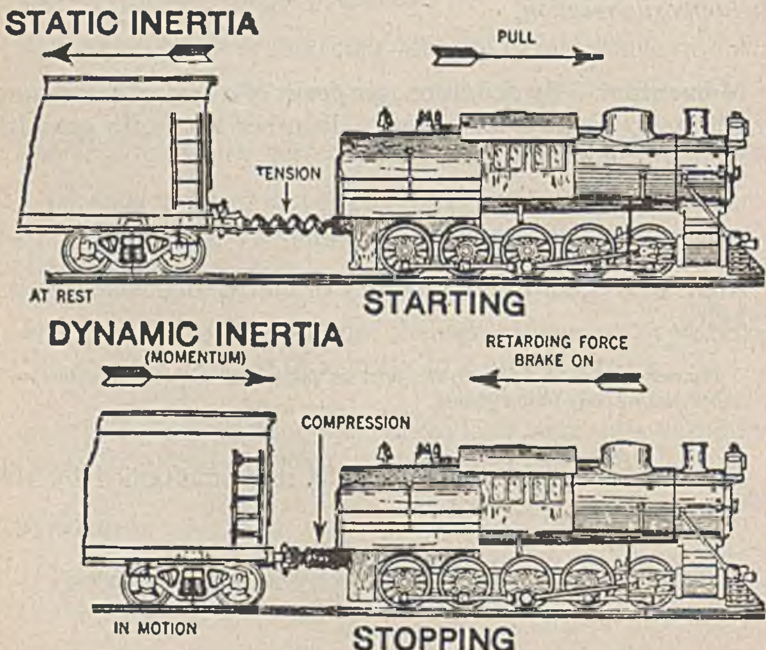
$$= \frac{WV}{g}$$

in which

W = weight in lbs.
V = velocity in feet per second
g = attraction due to gravity.

Inertia.—By definition, *that property of a body which causes it to remain in its state of rest or of uniform motion unless it be acted upon by some force compelling it to change that state.* This gives rise to two states of inertia:

1. Static inertia.
2. Dynamic inertia



Figs. 33 and 34.—Starting and stopping of train, illustrating inertia. This property may be called static inertia with respect to a body at rest (fig. 33) and dynamic inertia with respect to a body in motion (fig. 34).

Heat.—By definition heat is *a form of energy known by its effects.*

There are two kinds of heat—

1. Sensible
2. Latent $\begin{cases}\text{internal}\\\text{external}\end{cases}$

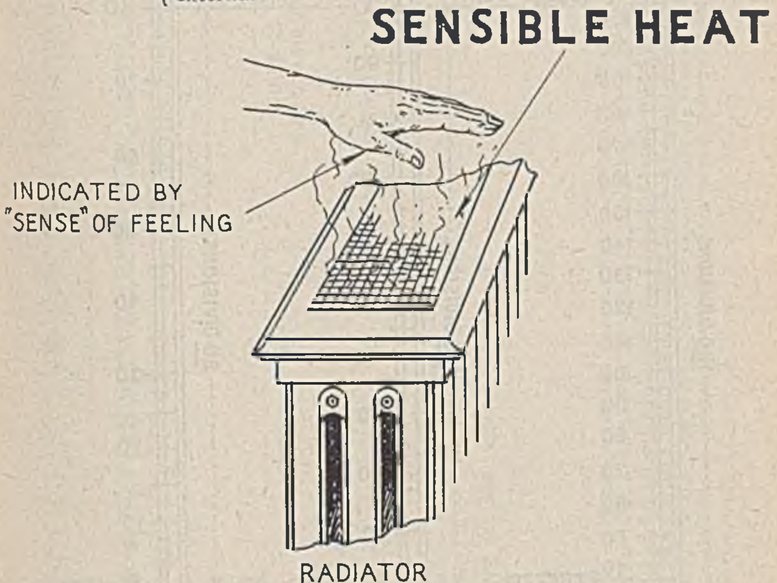


Fig. 35.—Familiar radiator example of sensible heat.

Sensible Heat.—By definition *that form of heat whose effect is indicated by the sense of feeling as for instance in fig.* 35.

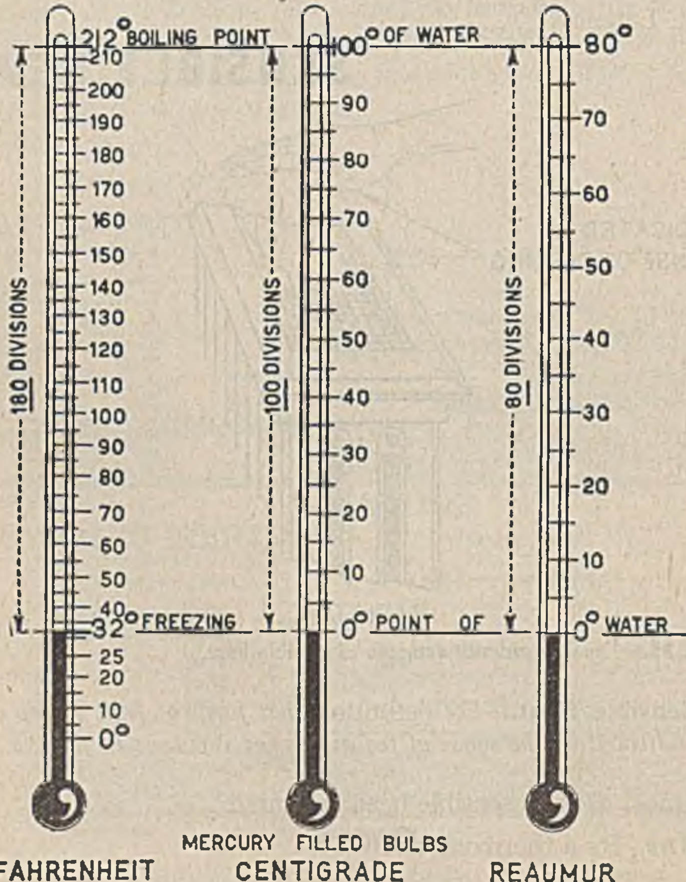
Ques. How is sensible heat measured?

Ans. By a thermometer.

Ques. What is a thermometer?

Ans. An instrument consisting of a glass tube terminating in a bulb, which is charged with a liquid, usually mercury or colored alcohol.

THERMOMETER SCALES



Figs. 36 to 38.—Fahrenheit, Centrigrade and Reaumur thermometer scales.

THERMOCOUPLE PYROMETER PRINCIPLE

HEAT AT JUNCTION OF TWO DISSIMILAR METALS PRODUCES ELECTRIC CURRENT

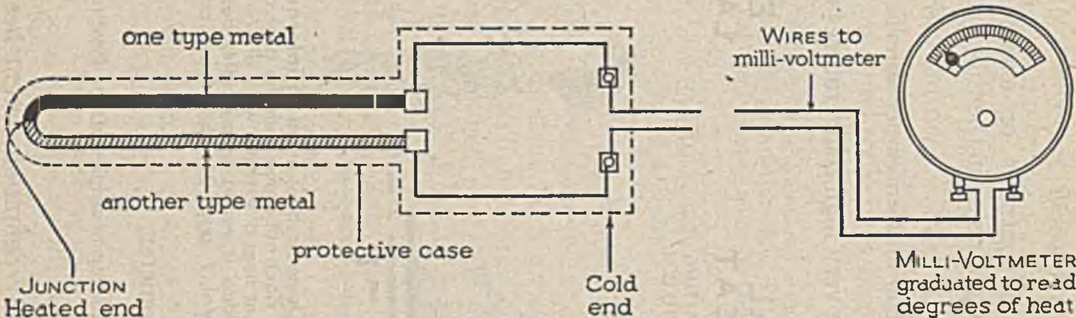


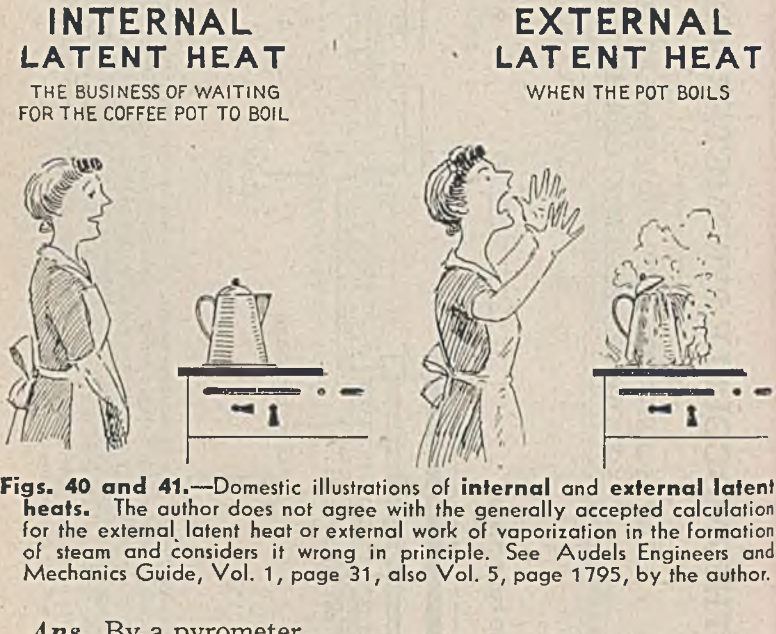
FIG. 39.—Elementary thermocouple thermometer used for measuring high temperatures. ***In principle,*** when heat is applied to the junction of two dissimilar metals, a current of electricity begins to flow in proportion to the amount of heat applied; this current can be brought to a meter and translated into terms of heat. The thermocouple for Diesel engine use is inserted in an opening in the cylinder exhaust and, by its temperature reading, denotes the burning condition of the fuel; by it, also, it is possible to compare the actions in the various cylinders. Where thermocouples are permanently installed in cylinder exhausts, the individual thermocouples may be connected to one indicating instrument by switches which connect one or another at will, for comparative readings.

Ques. How does it work?

Ans. The liquid contracts or expands with changes of temperature, falling or rising in the tube against which is placed a graduated scale.

Figs. 36 to 38 show the thermometer scales used in different countries.

Ques. How are very high temperatures measured?



Figs. 40 and 41.—Domestic illustrations of **internal** and **external latent heats.** The author does not agree with the generally accepted calculation for the external latent heat or external work of vaporization in the formation of steam and considers it wrong in principle. See Audels Engineers and Mechanics Guide, Vol. 1, page 31, also Vol. 5, page 1795, by the author.

Ans. By a pyrometer.

A diagram illustrating principles of one type of pyrometer is shown in fig. 39.

Latent Heat.—By definition, latent heat is *that quantity of* *heat which becomes concealed or disappears in a body while producing some change in it other than a rise of temperature.*

When water at atmospheric pressure has been heated to 212° Fahr. no further rise in temperature takes place, although the supply of heat be continued. Instead vaporization takes place and considerable heat must be added to the liquid to transform it into steam, this total heat being made up of the **internal** and **external** latent heats*.

Thus in the case of water at 212° Fahr. and atmospheric pressures considerable heat must be added to start the water boiling (this is the *internal* latent heat) and additional heat must be added to boil it, the latter being the *external* latent heat.

A familiar but not very dignified example of these heats is shown in figs. 40 and 41 without accompanying remarks.

The Unit of Heat.—This is based upon the amount of heat necessary to raise the temperature of one pound of water one degree. Various units have been given in the past, but the present generally accepted heat unit, called the British thermal unit (*B.t.u.*) is defined as 1/180 *of the heat required to raise the temperature of water from* 32° *Fahr. to* 212° *Fahr.*§

The old definition of the heat-unit (*Rankine*), viz., the quantity of heat required to raise the temperature of 1 lb. of water 1° Fahr., at or near its temperature of maximum density (39.1° Fahr.) was the standard till 1909.

By Peabody's definition, the heat required to raise 1 lb. of water from 32° Fahr. to 212° Fahr. is 180.3 instead of 180 units, and the latent heat at 212° Fahr. is 969.7 instead of 970.4.

Example.—How many heat units (*B.t.u.*) are required to raise the temperature of 25 lbs. of water from 60° Fahr. to 212° Fahr.

$$\text{temperature rise } 212 - 60 = 152° \text{ Fahr.}$$

$$152° \text{ Fahr.} \times 25 \text{ lbs.} = 3{,}800\ B.t.u.$$

*NOTE.—For a full explanation of the phenomena involved see Audels Engineers and Mechanics Guide No. 5. Chapter 55 "From Ice to Steam" by the author.

§NOTE.—It should be noted that this is the definition adopted in this work and other books by the author corresponding to the unit used in the Marks and Davis Steam Tables, which is now the recognized standard.

Specific Heat.—By definition specific heat is *the capacity of any substance for receiving heat as compared with another which is taken as a standard.*

Ques. What is the standard?

Ans. Water usually from 62° to 63° Fahr.

Example.—The same quantity of heat that will raise 1 lb. of water 1° Fahr. will raise about 8.4 lbs. of cast iron 1° Fahr. Accordingly the specific heat of water being taken as 1, that of cast iron would be only

$$1 \div 8.4 = .1189$$

That is, it is the ratio between the two heats. The specific heat of a few substances are as follows:

Specific Heat of Various Substances

Solids

Copper	.0951
Wrought iron	.1138
Glass	.1937
Cast iron	.1298
Lead	.0314
Tin	.0562
Steel { Sott	.1165
Steel { Hard	1175
Brass	.0939
Ice	.504

Liquids

Water	1.
Sulphuric Acid	.335
Mercury	.0333
Alcohol (nnn)	.7
Benzine	.95
Ether	.5034

Gases

	Constant pressure	Constant volume
Air	.23751	.16847
Oxygen	.21751	.15507
Hydrogen	3.409	2.41226
Nitrogen	.2438	.17273
Ammonia	.508	.299
Alcohol	.4534	.399

Transfer of Heat.—There are three ways in which heat may be transferred from one body to another at lower temperature, as by:

1. Radiation
2. Conduction
3. Convection.

Radiation, Conduction and Convection.—When heat is transmitted by radiation the hot body, as burning fuel for instance, sets up waves in the ether. In a boiler furnace, heat is given off by *radiation* in which *rays* radiate in straight lines in all directions being transferred to the crown sheet and sides of the furnace by radiation.

Ques. What is conduction?

Ans. The transference of heat from the hotter to the colder parts of a body.

Ques. Upon what does conduction depend?

*NOTE.—*Specific heat of gases.* Experiments by Mallard and Le Chatelier indicate a continuous increase in the specific heat at constant volume of steam, carbon dioxide, and even the perfect gases, with rise of temperature. The variation is inappreciable at 212° F., but increases rapidly at the high temperatures of the gas engine cylinder.

Ans. Upon the fact of inequality in temperature in the several portions of a body.

The transference of heat through solids, as through boiler plates, is due to conduction contrary to popular opinion. The temperature of a boiler furnace plate is only a few degrees hotter than the water in contact with the plate. This is due to the extremely rapid conductivity of the plate.

Ques. How is heat transferred by convection?

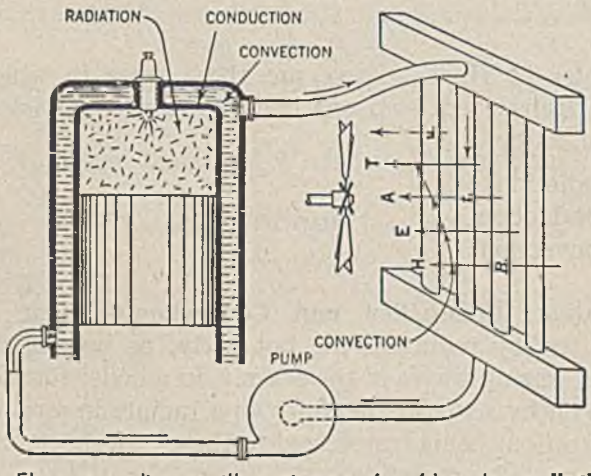


Fig. 42.—Elementary diagram illustrating transfer of heat by **radiation, conduction** and **convection.** It should be noted that air is the cooling agent and not the water as the water is only the medium for transferring the heat to the point where it is extracted and dissipated by the air. Accordingly, the term water cooled engine is a misnomer, but nothing can be done about it.

Ans. By the motion of the heated matter itself.

Ques. In what classes of substances can heat be transmitted by convection?

Ans. In liquids and gases.

High Temperature Judged by Color.—The temperature of a body can be approximately judged by the experienced eye unaided. M. Poillet constructed a table which has been generally accepted giving colors and corresponding temperatures as follows:

	Deg. C	Deg. F
Incipient red heat..	525	977
Dull red heat......	700	1,292
Incipient cherry red heat.............	800	1,472
Cherry red heat....	900	1,652
Clear cherry red heat	1,000	1,832

	Deg. C	Deg. F
Deep orange heat..	1,100	2,021
Clear orange heat..	1,200	2,192
White heat.......	1,300	2,372
Bright white heat..	1,400	2,552
Dazzling white heat	1,500 to 1,600	2,732 to 2,912

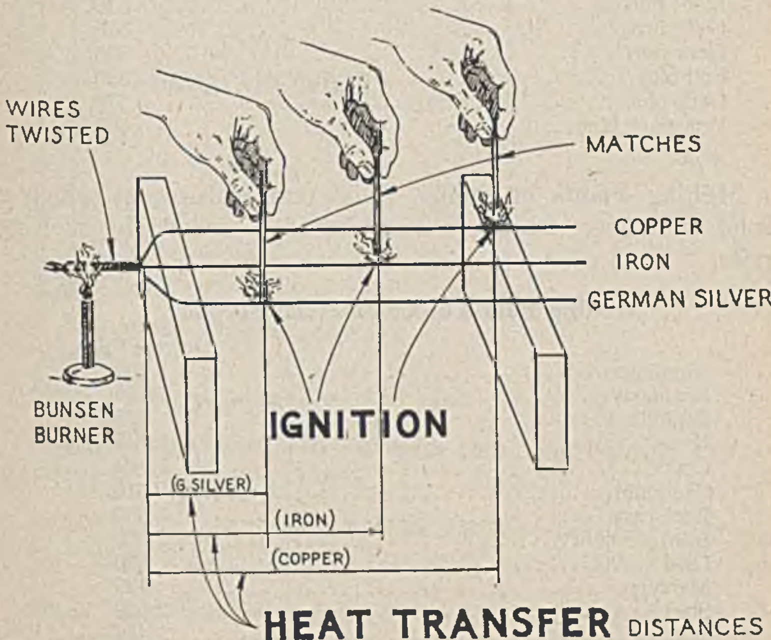


Fig. 43.—Experiment illustrating heat conductivity of various metals.

The following table gives temper colors of steels and heats corresponding according to the Halcolm Steel Co.

Colors	Temperatures Fahrenheit
Very pale yellow	430
Light yellow	440
Pale straw-yellow	450
Straw-yellow	460
Deep straw-yellow	470
Dark yellow	480
Yellow-brown	490
Brown-yellow	500
Spotted red-brown	510
Brown-purple	520
Light purple	530
Full purple	540
Dark purple	550
Full blue	560
Dark blue	570
Very dark blue	600

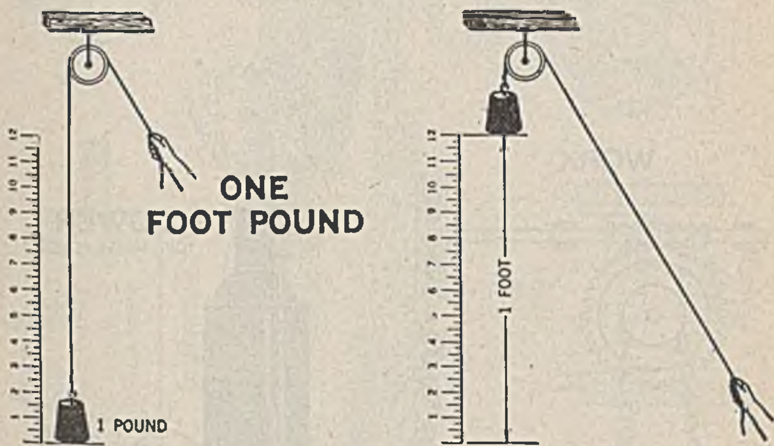
Melting Points of Solids.—The temperatures at which a solid substance changes into a liquid is called the melting point.

Melting Points of Commercial Metals

	Degrees Fahr.
Aluminum	1,200
Antimony	1,150
Bismuth	500
Brass	1,700–1,850
Copper	1,940
Cadmium	610
Iron, cast	2,300
Iron, wrought	2,900
Lead	620
Mercury	139
Steel	2,500
Tin	446
Zinc, cast	785

Work and Power.—By definition, *work is the overcoming of resistance through a certain distance by the expenditure of energy.*

Most people excluding engineers, don't know the difference between the terms *work* and *power*. Without knowing the exact meaning of these two words, it would be ridiculous for such people to talk about *horse power*.



Figs. 44 and 45.—One foot pound.

Ques. What is power?

Ans. The **RATE** *at which work is done, that is, work divided by the* **time** *in which it is done.*

To fix in mind the difference between *work* and *power* study figs. 42 and 43 carefully, then study them again.

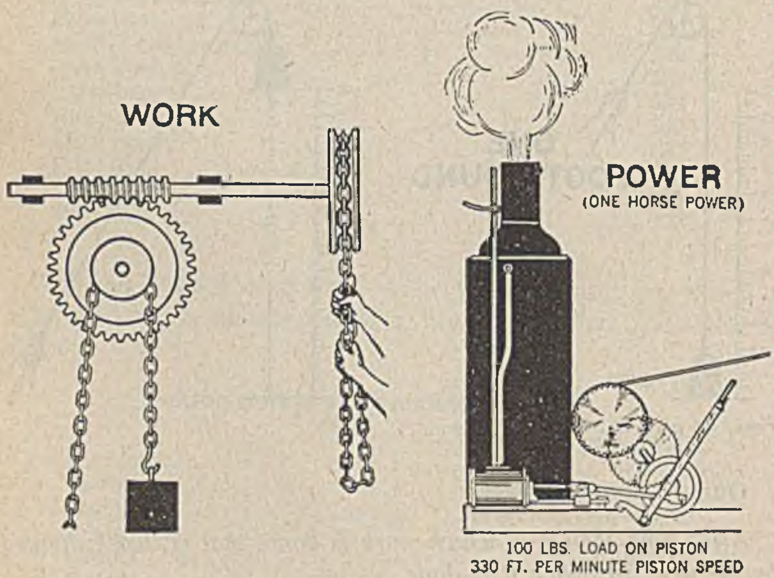
Ques. How is work measured?

Ans. By a standard unit called a *foot pound.*

Ques. What is a foot pound?

Ans. The amount of work done in raising one pound one foot, or in overcoming a pressure of one pound through a distance of one foot as shown in figs. 44 and 45.

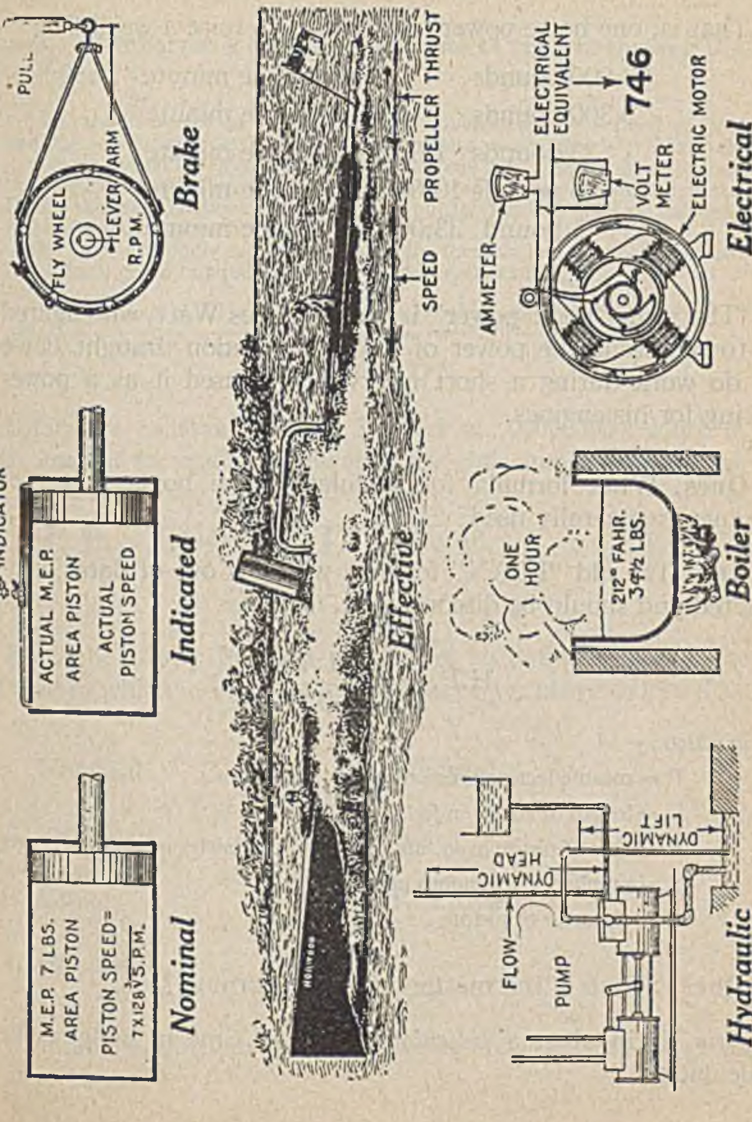
$$\text{power} = \frac{\text{work}}{\text{time}}$$



Figs. 46 and 47.—The difference between **work** and **power.**

Horse Power.—The unit of power is one horse power, which is defined as

33,000 foot pounds per minute.



Figs. 48 to 54.—Principal factors involved in the calculation of various kinds of horse power.

That is, one horse power* is required to raise a weight of

33,000 pounds	1 foot in one minute
3,300 pounds	10 feet in one minute
33 pounds	1,000 feet in one minute
3.3 pounds	10,000 feet in one minute
1 pound	33,000 feet in one minute
	etc.

*The term "horse power" is due to James Watt, who figured it to represent the power of a strong London draught horse to do work during a short interval, and used it as a power rating for his engines.

Ques. What formula for calculating the horse power of engines is generally used?

Ans. The old "PLAN" formula which is out of date, antiquated and should be discontinued. It is

$$\text{H.P.} = \frac{2\ \text{PLAN}}{33{,}000}$$

in which

P = mean effective pressure in lbs. per sq. ins.;

L = length of stroke in *feet;*

A = area of piston in sq. ins. = .7854 × diameter of piston squared;

N = number of *revolutions* per minute;

D = diameter of piston.

Ques. What is the matter with this formula?

Ans. It involves a *ridiculous* waste of time in making the calculation.

Ans. It involves a considerable waste of time in making the calculation.

Since the stroke of an engine is usually given in inches instead of feet, and the revolutions per minute instead of the piston speed, the formula just given evidently involves extra calculations for these items as well as the extra multiplication and division introduced because of the constants. Its use therefore is about as laborious as multiplying and dividing fractions without reducing them to their lowest terms.

The author strongly recommends that the formula just given be not used in the form given but reduced to its lowest terms as follows:

$$\text{H. P.} = \frac{2\,P\,L\,A\,N}{33{,}000} = \frac{2 \times P \times \frac{L}{12} \times .7854 \times D^2 \times N}{33{,}000} = \frac{.1309 \times PLD^2N}{33{,}000}$$

$$= .00000396\,7\,P\,L\,D^2N$$

Using the constant .000004 instead of .000003966 which is near enough for ordinary calculations, and changing the order of the factors, the formula becomes

$$\text{H. P.} = .000004\,D^2\,L\,N\,P \quad . \quad . \quad . \quad (2)$$

Kinds of Horse Power.—According to definitions and the manner in which it is determined, horse power may be classed as:

1. Nominal
2. Indicated
3. Brake
4. Effective
5. Hydraulic
6. Boiler
7. Electrical

Graphical definitions of these various horse powers are given in figs. 48 to 54.

The Mechanical Powers.—This term for a long time ha[d] been in popular use for certain basic mechanical contrivance[s] that enter into the composition or formation of all machines. A better term is *fundamental machines* because these mechanica[l] contrivances are regarded more in a static sense than dynamic[;] that is, the consideration of opposing forces in equilibrium

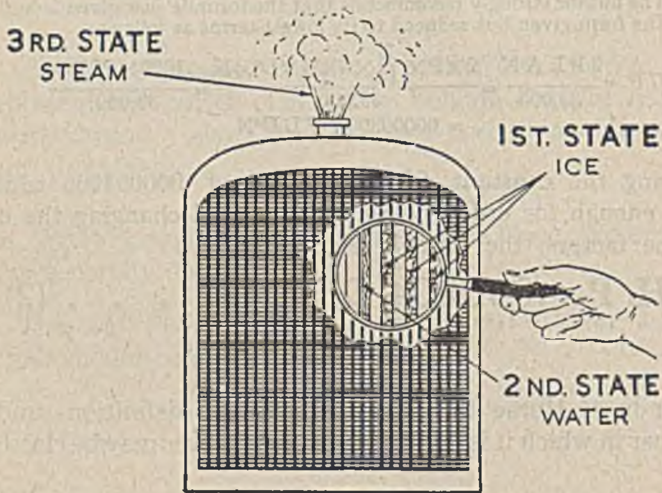


FIG. 55.—Frozen radiator operation illustrating the *three* states. Water freezes at 32° Fah[r]. As the ice forms in the radiator a *change of state* takes place; the ice stops circulation an[d] thus cuts out part of the cooling surface, hence when the car is started there is not enoug[h] cooling surface to carry off the heat and the radiator begins to steam. This results in [a] change of state—water to steam. Moreover due to expansion of the water as it freezes some of the tubes burst, resulting in leaks.

rather than tending to produce motion. As in the above title the word *force* is popularly spoken of as *power*.

Strictly speaking, the term power is a dynamic term relatin[g] to the *time rate of doing work*, but when the elements of a ma[-]chine are in equilibrium no work is done, in which case *powe[r]* is incorrectly used.

It should be understood that the so-called mechanical powers all depend for their action upon what is known as the *principle of work*, that is: *The applied force, multiplied by the distance through which it moves, equals the resistance overcome, multiplied by the distance through which it is overcome.*

The mechanical powers or basic machines are:

1. The lever
2. The wheel and axle
3. The pulley
4. The inclined plane
5. The screw
6. The wedge

These can in turn be reduced to three classes:

1. A solid body turning on an axis
2. A flexible cord
3. A hard and smooth inclined surface

The lever.—By definition, *a bar of metal, wood or other substance, used to exert a pressure or sustain a weight, at one point of its length by receiving a force at a second point, and free to turn at a third, or fixed point called the* **fulcrum.** Its application is based on the principle of moments.

Ques. What is a moment?

Ans. A measure of the turning effect of a force which tends to produce rotation around an axis, as around the fulcrum of a lever.

Principle of Moments.—*When two or more forces act upon a rigid body and tend to turn it about an axis, then equilibrium will exist if the sum of the moments of the forces which tend to turn the body in one direction equals the sum of the moments of those which tend to turn it in the opposite direction about the same axis.*

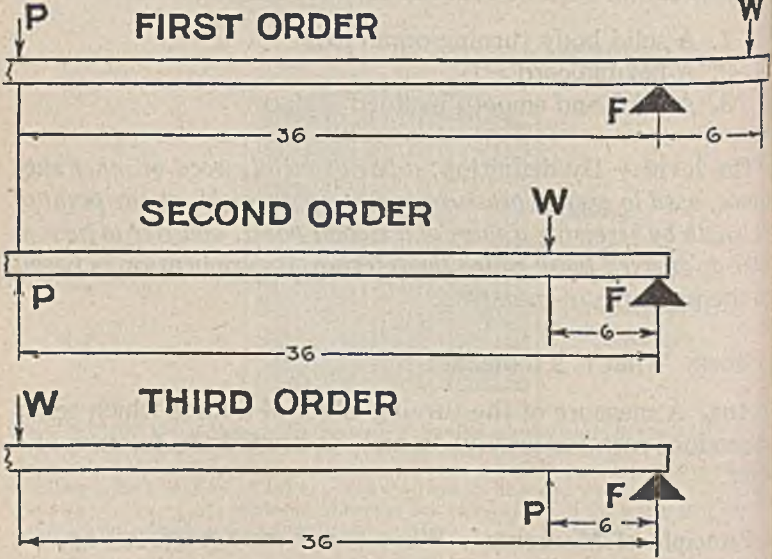
The lever safety valve when at the point of blowing off is a good illustration of the above principle.

The Lever.—The following general rule holds for all classes of lever:

Rule. *The force P, multiplied by its distance from the fulcrum, is equal to the load W; multiplied by its distance from the fulcrum.* That is:

$$\textit{Force} \times \textit{distance} = \textit{load} \times \textit{distance} \ldots\ldots\ldots (1)$$

Example.—What force applied at 3 ft. from the fulcrum will balance



Figs. 56 to 58.—Diagrams of the *three orders of lever* illustrating the accompanying examples.

a weight of 112 lbs. applied at 6 ins. from the fulcrum? Here the distances or "leverages" are 3 feet and 6 inches.

The distance must be of the same denomination; hence reducing ft. to ins., $3 \times 12 = 36$ ins.

Applying the rule

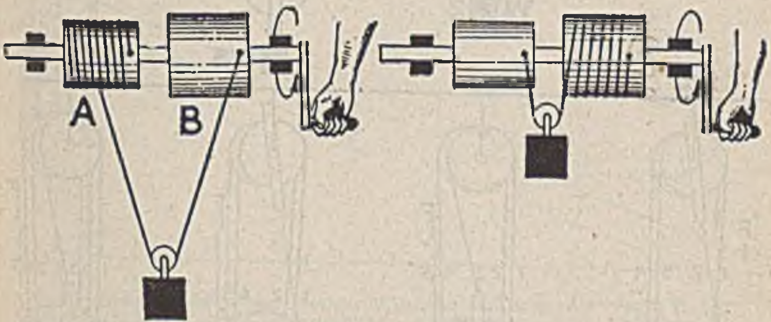
$$\text{Force} \times 36 = 112 \times 6$$

Solving

$$\text{Force} = \frac{112 \times 6}{36} = 18.67 \text{ or } 18\tfrac{2}{3} \text{ lbs.}$$

This solution holds for all levers as illustrated in FIGS. 56 to 58.

Wheel and Axle.—Comparison of the wheel and axle with a 1st order lever shows that in principle they are the same thing. The general equation (1) on page 66 applies to the wheel and axle.



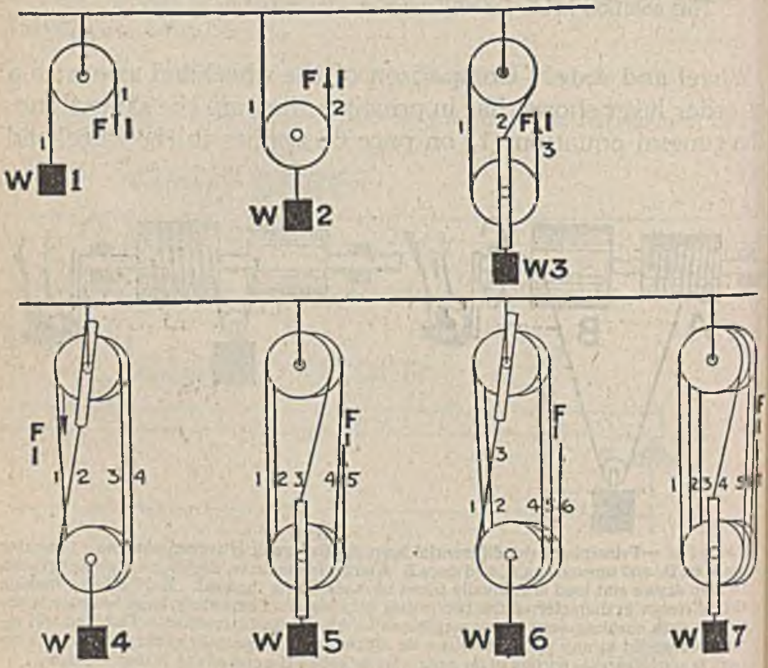
FIGS. 59 and 60 —Principle of the differential hoist. As the crank is turned clockwise the cable winds on B, and unwinds on A, and since B is larger in diameter, the length of cable between the two drums and load is gradually taken up, thus lifting the load. Evidently by making the difference in diameter of the two drums very small an extremely large leverage is obtained, thus enabling very heavy weights to be lifted with little effort. The load will remain suspended at any point, *because the difference in the diameter of the two drums is too small to overbalance the friction of the parts.* FIG. 60 shows the end of the lifting operation.

Chinese Wheel and Axle.—This is a modification of the wheel and axle and is used for obtaining extreme degree of leverage. Its principle and construction are shown in figs. 59 and 60.

The Pulley.—Pulleys are classed as *fixed* or *movable.*

In the fixed pulley no mechanical advantage is gained, but its use is of the greatest importance in accomplishing the work appropriate to the pulley, such as raising water from a well.

The *movable* pulley, by distributing the weights into separate parts, is attended by mechanical advantages proportional to the number of points of support.



FIGS 61 to 67.—Elementary pulley combinations - illustrating accompanying rule for relation between force applied and load lifted and showing how the load may be increased from 1 to 7 times per unit of force applied. Of course a greater range may be secured by additional pulleys, but there is a limit in practice to which it is mechanically expedient.

The following rule expresses the relation between the force and load.

Rule.—*The load capable of being lifted by combination of pulleys is equal to the force×the number of ropes supporting the lower or movable block.*

The Inclined Plane.—By such substitution of a sloping path for a direct upward line of ascent, a given weight can be raised by another weight weighing less than the weight to be raised.

The inclined plane becomes a *mechanical power* in consequence of its supporting part of the weight, and of course leaving only a part to be supported by the power.

Rule.—*As the applied force* P, *is to the load* W, *so is the height,* H, *to the length of the plane* W.

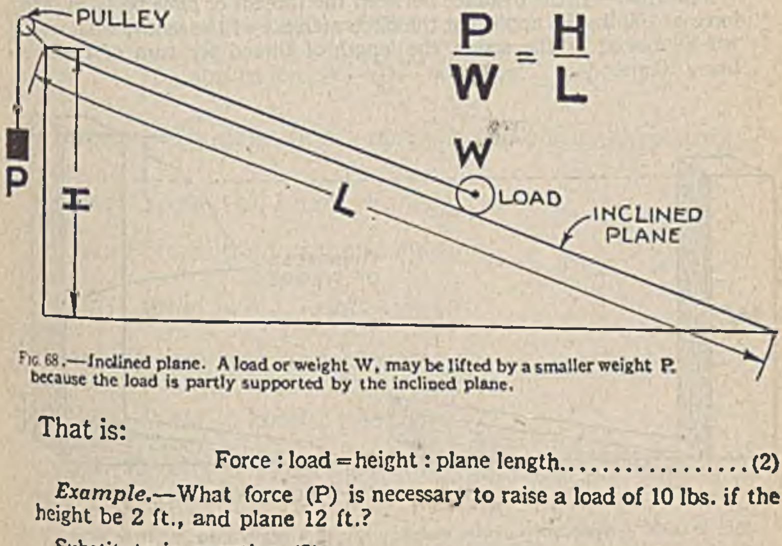


FIG. 68.—Inclined plane. A load or weight W, may be lifted by a smaller weight P, because the load is partly supported by the inclined plane.

That is:

Force : load = height : plane length. . . . .(2)

Example.—What force (P) is necessary to raise a load of 10 lbs. if the height be 2 ft., and plane 12 ft.?

Substitute in equation (2)

$$P : 10 = 2 : 12$$

$$P \times 12 = 2 \times 10$$

$$P = \frac{10 \times 2}{12} = \frac{20}{12} = 1\tfrac{2}{3} \text{ lbs.}$$

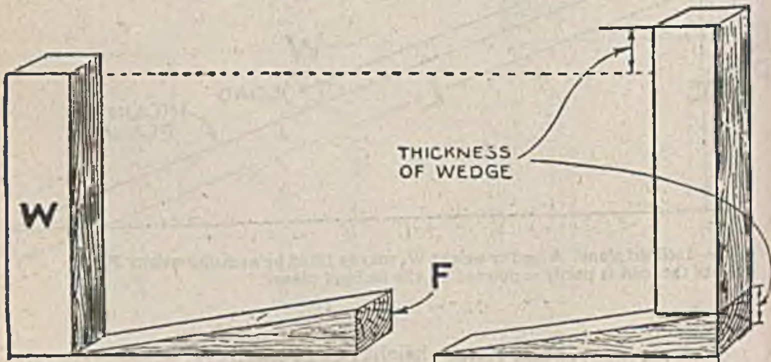
The Screw.—This is simply *an inclined plane wrapped around a cylinder.*

The screw is generally employed when severe pressure is to be exerted through small spaces; being subject to great loss from friction it usually exerts but a small power of itself, but derives its principal efficacy from the lever or wheel work with which it is very easily combined.

Rule.—*As the applied force is to the load so is the pitch to the length of thread per turn*, that is:

Applied force : load = pitch : length of thread per turn....... (3)

Example:—If the distance between the threads or *pitch* be ¼ in. and a force of 100 lbs. be applied at the circumference of the screw, what weight will be moved by the screw, the length of thread per turn of the screw being 10 ins.



Figs. 69 and 70.—Application of the wedge in raising a heavy load.

Substituting in equation (3)

$$100 : \text{load} = \tfrac{1}{4} : 10$$
$$\text{load} \times \tfrac{1}{4} = 10 \times 100$$
$$\text{load} = \frac{10 \times 100}{\tfrac{1}{4}} = 4{,}000 \text{ lbs.}$$

The Wedge.—This is *virtually a pair of inclined planes in contact along their bases or back to back.*

Rule.—*As the applied force is to the load so is the thickness of the wedge to its length;* that is:

Applied force : load = thickness : length of wedge....................(4)

Example.—What force is necessary to apply to a wedge 20 ins. long and 4 ins. thick to raise a load of 2,000 lbs.?

Substituting in equation (4)

$$\text{Applied force} : 2{,}000 = 4 : 20$$

$$\text{applied force} \times 20 = 4 \times 2000$$

$$\text{applied force} = \frac{4 \times 2000}{20} = 400 \text{ lbs.}$$

Energy.—By definition, energy is the *ability to do work.*

Ques. Name two kinds of energy.

Ans. Potential and kinetic energy.

Ques. What is potential energy?

Ans. Energy due to *position*.

Ques. What is kinetic energy?

Ans. Energy possessed by a moving body due to its momentum.

Fig. 71 illustrates potential and kinetic energy.

Ques. Give another definition for potential energy.

Ans. Stored capacity for performing work possessed by a body at rest due to its elevation.

Water, as in fig. 71, stored in an elevated reservoir represents potential energy, as its liberation to a lower level may be utilized to do work.

Ques. Give another definition of kinetic energy.

Ans. The *dynamic inertia* possessed by a moving body.

Conservation of Energy.—The doctrine of physics that energy can be transmitted from one body to another or transformed in its manifestations, but *may neither be created nor destroyed.*

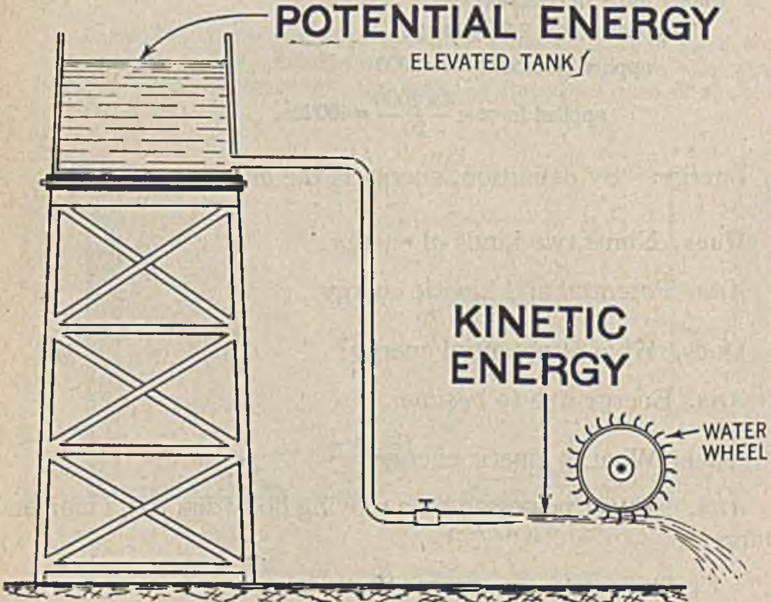


Fig. 71.—Potential and kinetic energy.

Energy may be dissipated, that is, converted into a form from which it cannot be recovered, as is the case with the great percentage of heat escaping with the exhaust of a locomotive, or the condensing water of a steamship, but *the total amount of energy in the universe*, it is argued, *remains constant and invariable.*

Ques. Give a classic illustration of the conservation of energy.

Ans. Joule's experiment.

Joule's Experiment.—In 1843 Dr. Joule of Manchester, England, performed his classic experiment which revealed to the world the *mechanical equivalent of heat*.

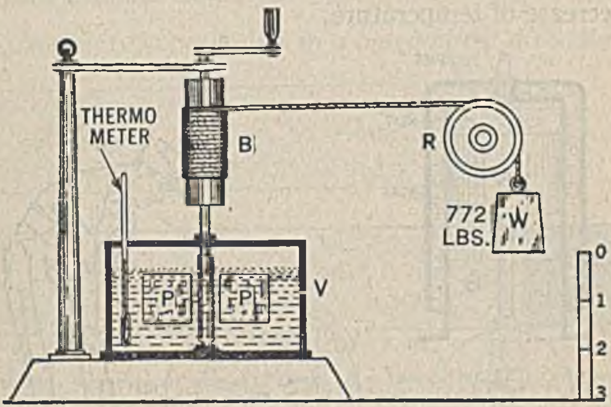


Fig. 72.—Joule's experiment revealing the mechanical equivalent of heat.

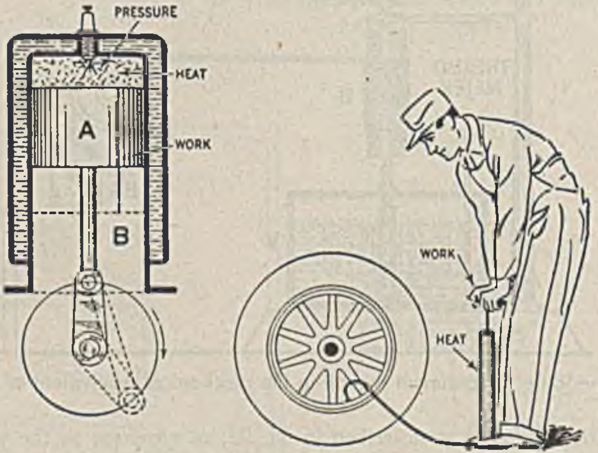
With apparatus as described in fig. 72, *in operation* as the weight W, falls, the paddles rotate in the water, the water itself being kept from rotating by fixed pieces not shown. It was discovered that the work done by the weight in descending, was not lost but appeared as heat in the water, the agitation of the paddles having increased the temperature of the water by an amount which can be measured by a thermometer.

According to Joule, when 772 foot pounds of *work* energy had been expended on the pound of water, the temperature of the latter had risen 1° Fahr. This is known as *Joule's equivalent*, that is 1 unit of heat equals 772 units of work.

More recent experiments by Prof. Rowland (1880) and others give higher figures; 778 is generally accepted, but 777.5 is probably more nearly correct, the value 777.52 being used by Marks and Davis in their steam tables.

The value 778 is sufficiently accurate for ordinary cacula-tions.

Expansion and Contraction.—Practically all substances expand with increase in temperature and decrease or shrink with decrease of temperature.



Figs. 73 and 74.—Heat converted into work (fig. 73) and work converted into heat (fig. 74).

Ques. What substance does not obey this law for all changes in temperature?

Ans. Water.

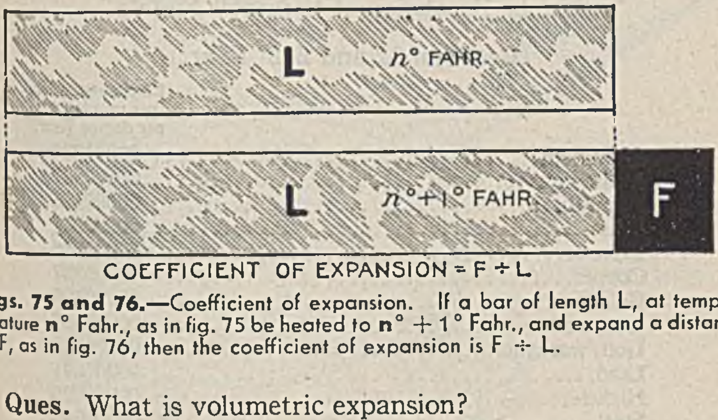
The most remarkable characteristic of water, as previously pointed out, is that at its point of maximum density (39.1° Fahr.) water will expand as heat is added and it will also slightly expand as the temperature falls from this point.

Ques. Why does heat cause expansion?

Ans. An increase of heat is due to an increase in the velocity of motion of the molecules. Accordingly the molecules by their more frequent violent collisions become separated a little farther from one another, and as a result the body expands, as shown in the experiments illustrated in the accompanying cuts.

Ques. What is linear expansion?

Ans. Expansion of solid bodies *in a longitudinal direction.*



Figs. 75 and 76.—Coefficient of expansion. If a bar of length L, at temperature **n**° Fahr., as in fig. 75 be heated to **n**° + 1° Fahr., and expand a distance F, as in fig. 76, then the coefficient of expansion is F ÷ L.

Ques. What is volumetric expansion?

Ans. Expansion in volume.

Ques. Define the coefficient of linear expansion.

Ans. It is the *ratio of the increase in length produced by a rise of temperature of* 1° *Fahr. to the original length*, as illustrated in fig. 76.

Ques. State some advantages and disadvantages of expansion and contraction due to heat.

Ans. Boiler plates are fastened with red hot rivets. When the rivets cool they contract and bind the plates together with great force.

Iron tires are first heated and then put onto the wheel. When the iron cools, the tire contracts and binds the wheel. A short space must be left between the rails of a railroad to permit expansion and contraction without injury.

For linear expansion the table here given is used for calculation.

Linear Expansion of Common Metals

(Between 32 and 212° Fahr.)

	Linear expansion per unit length per degree Fahr.
Aluminum	00001234
Antimony	.00000627
Bismuth	00000975
Brass	.00000957
Bronze	.00000986
Copper	.00000887
Gold	.00000786
Iron, cast	.00000556
Iron, wrought	.00000648
Lead	.00001571
Nickel	.00000695
Steel	.00000636
Tin	.00001163
Zinc, cast / Zinc, rolled	.00001407

Volumetric expansion = 3 × linear expansion.

Friction.—By definition, friction is *that force which acts between two bodies at their surface so as to resist their sliding on each other.* In other words, *the resistance existing between two bodies in contact which tends to prevent their motion on each other.*

Ques. What are the causes of friction?

Ans. It is partly due to the natural adhesion of one body to another but chiefly to the roughness of the surfaces in contact.

Ques. What does a surface polished as fine as possible look like when viewed under a powerful microscope?

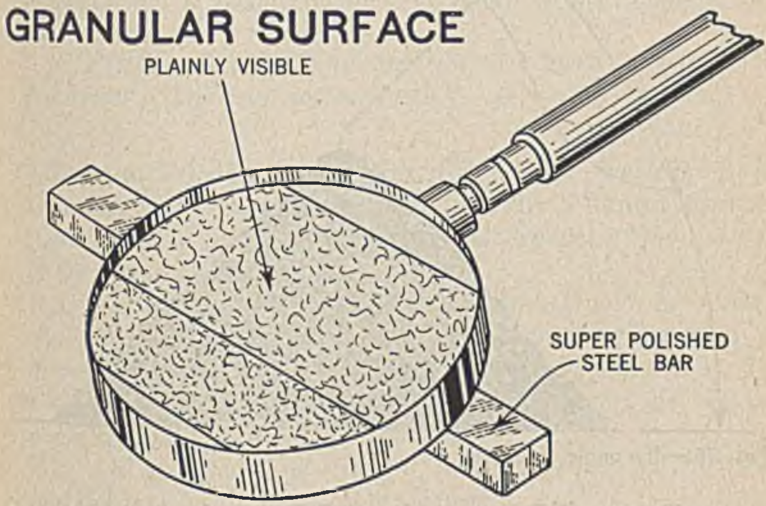


Fig. 77.—The nature of a polished surface.

Ans. Like a piece of coarse emery paper as in fig. 77.

Here a magnifying glass is shown instead of a microscope simply for simple illustration—imagine the magnifying glass to have the power of a microscope.

Ques. What are the characteristics of friction?

Ans. It may be either harmful or useful.

In most cases it is harmful in that there is a waste of power to overcom friction in any mechanism. An outstanding example of friction use brakes.

Ques. What is the most harmful effect of friction.

Ans. The wear of bearings.

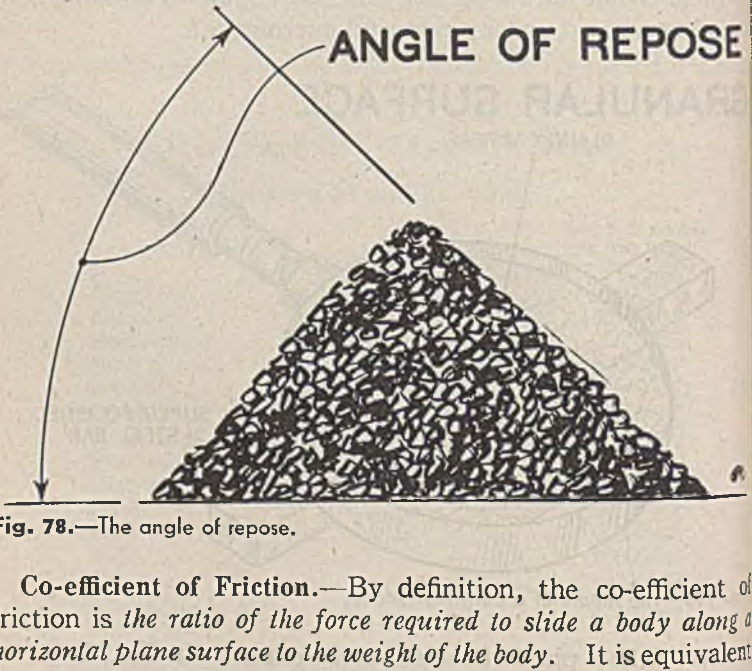


Fig. 78.—The angle of repose.

Co-efficient of Friction.—By definition, the co-efficient of friction is *the ratio of the force required to slide a body along a horizontal plane surface to the weight of the body.* It is equivalent to *the tangent of the angle of repose.*

Ques. What is the angle of repose?

Ans. The greatest angle with the horizontal at which a mass material as in an embankment or coal pile will lie without sliding. The angle varies for different materials.

Laws of Friction.—The first laws of friction were given by Morin about 1830 but have since been modified by the results of later experiments. As summarized by Kent, the laws are:

1. *Friction varies approximately as the normal pressure with which the rubbing surfaces are pressed together.*
2. *Friction is approximately independent of the area of the surfaces, but is slightly greater for small surfaces than for large surfaces.*
3. *Friction decreases with increase of velocity, except at very low velocity, and with soft surfaces.*

Laws of Friction for Lubricated Surfaces.—Perfect lubrication, i.e. surfaces completely separated by a film of lubricant:

4. *The co-efficient of friction is independent of the materials of the surfaces.*
5. *The co-efficient of friction varies directly with the viscosity of the lubricant, which varies inversely with temperature of the lubricant.*
6. *The co-efficient of friction varies inversely as the unit pressure, and directly as the velocity.*
7. *The co-efficient of friction varies inversely as the mean film thickness of the lubricating medium.*
8. *Mean film thickness varies directly with velocity and inversely as the temperature and unit pressure.*

Imperfect Lubrication.—Surfaces partially separated by a film of lubricant may range from almost complete separation of the surfaces to almost complete contact.

9. *The co-efficient of friction increases with increase of pressure between surfaces.*

10. *The co-efficient of friction decreases with increase of relative velocity between the surfaces.*

Lubrication.—On account of minute irregularities of a smooth metal surface, it is impossible to run machinery without lubrication of some kind notwithstanding the alleged "anti-friction" metals.

Ques. What should be noted by the term "anti-friction metals"?

Ans. It is misleading and ridiculous—in common language, a lie.

Ques. What is the duty of a lubricant?

Ans. Its office is to keep the rubbing parts separated by a thin film of oil, thus preventing as far as possible actual contact.

Ques. What is the peculiarity of graphite?

Ans. It does not lubricate but fills up the minute pores in the bearing surface.

CHAPTER 3

Hydraulics

Leading up to the study of pumps, it is helpful to have a knowledge of *hydraulics*, hence this chapter.

Although the generally accepted definition of hydraulics considers it *as treating of liquids, usually water, in motion*, this is altogether too narrow a definition, and it should include *the consideration of liquids under all conditions, whether in motion or at rest.*

That is to say, hydraulics should be defined as *the science which treats of liquids especially water and the forces acting on the liquid whether it be in a state of rest or motion.*

Accordingly, broadly speaking, there are two general divisions of the subject:

1. Hydrostatics
2. Hydrodynamics.

1. Hydrostatics

Ques. What is hydrostatics?

Ans. That division of hydraulics which treats of liquids especially water at rest.

Ques. What governs the pressure of a liquid on a surface?

Ans. It is proportional to the area of the surface, as shown in fig. 1.

Ques. Is much water required to balance a heavy weight?

Ans. Any quantity of water however small may be made to balance any weight however heavy as shown in fig. 2.

Ques. Upon what does the pressure of water at any point below the surface depend?

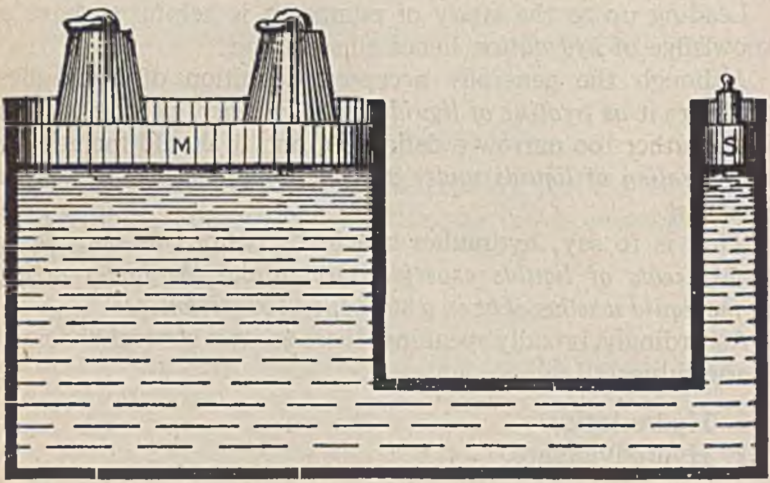


Fig. 1.—Hydraulic principles 1.—The pressure exerted by a liquid on a surface is proportional to the area of the surface. Two cylinders of different diameter are joined by a tube and filled with water. On the surface are the two pistons M and S, which hermetically close the cylinders, but move without friction. Let the area of the large piston M be. say thirty times that of the smaller one S, and let a weight, say of 2 lbs., be placed upon the small piston. The pressure will be transmitted to the water and to the large piston, and as this pressure amounts to 2 lbs. in each portion of its surface equal to that of the small piston, the large piston must be exposed to an upward pressure thirty times as much, or 60 lbs. If now a 60 lb. weight be placed upon the large piston, both pistons will remain in equilibrium, but if the weight be greater or less, the equilibrium will be destroyed.

Ans. It is proportional to the depth of the point below the surface.

In fig. 4 is shown a number of 1 lb. weights piled on top of each other. Evidently if the pile be placed on a scale the total weight or pressure would be 11 lbs. and if resting on 1 sq. in. of surface it would be 11 lbs. per sq. in.

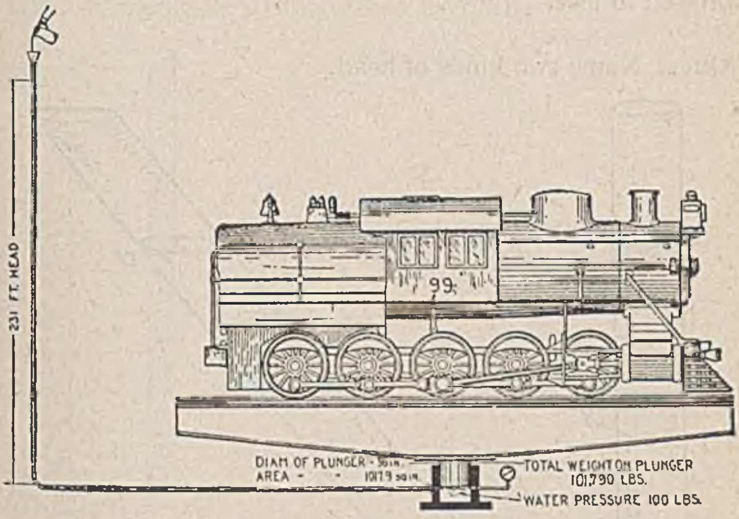


Fig. 2.—Hydraulic principles 2.—Any quantity of water however small may be made to balance any weight however great. This illustration shows a locomotive on a hydraulic elevator. Assuming no leakage or friction at the joint, and that the vertical pipe leading to the plunger cylinder is very small, it is evident that it could be filled to the elevation shown with a very small quantity of water—say one quart. If the weight of the locomotive be 101,790 lbs. and area of plunger such that it requires 100 lbs. per sq. in. pressure on plunger to balance locomotive, then the load will be balanced when the pipe is filled with water to a height $100 \times 2.31 = 231$ ft.

Now in fig. 5 a similar condition exists. That is, a column of water (1 sq. in. in cross section) and 2.31 ft. high weighs 1 lb. Accordingly if a gauge be placed at the bottom of the column which in the figure is $2.31 \times 2 = 4.62$ ft. in height it would exert a pressure of 2 lbs. per sq. in. Hence the pressure of water at any depth equals

$$\text{depth in feet} \div 2.31$$

Head and Pressure.—These are two primary considerations in hydraulics. By definition head is *the depth of water in a vessel pipe or conduit which is the measure of the pressure upon any given point below the surface.* That is, the word head signifies *the difference in level of water between two points*, and is usually expressed in feet.

Ques. Name two kinds of head.

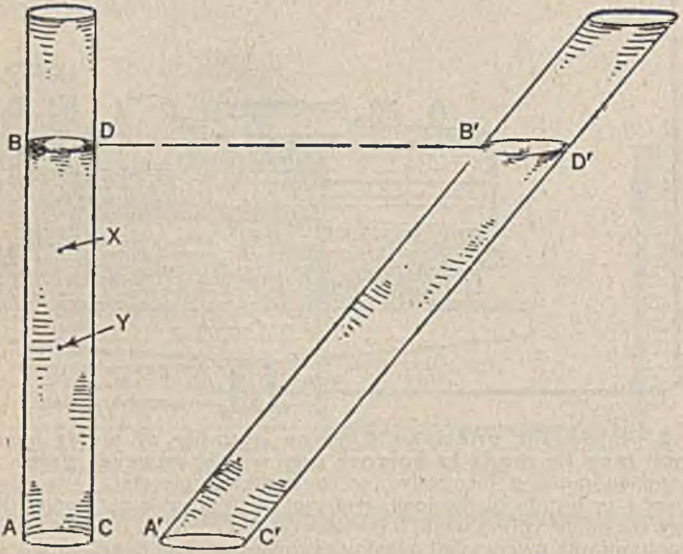


Fig. 3.—Hydraulic principles 3.—The pressure upon any particle of a fluid of uniform density is proportional to its depth below the surface.

Ans. Static and dynamic head.

Dynamic head is treated in Section 2 on Hydrodynamics.

Ques. What is static head?

Ans. The height from a given point of a column or body of water at rest, considered as causing or measuring pressure.

Ques. With respect to pumps, what should be noted about head?

Ans. It should be distinguished from lift.

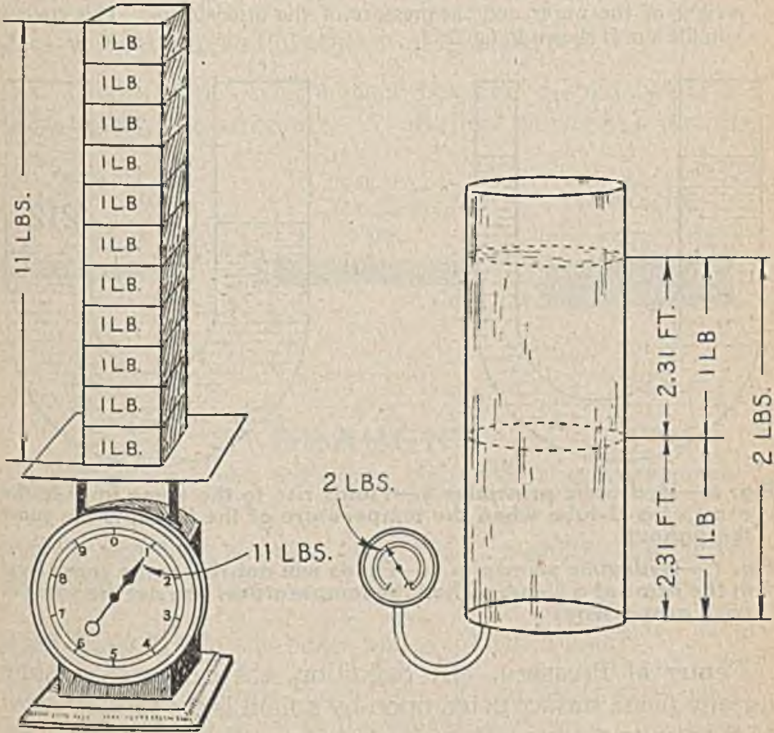
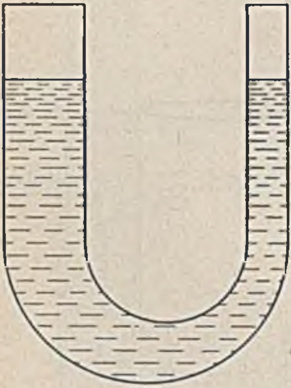


Fig. 4.—Assembly of 1 lb. weights on scale to show that **the pressure of water at any point below the surface is proportional to the depth of the point below the surface.**

Fig. 5.—Relation between water pressure at any point below the surface and the depth in feet.

Static Lift.—When the barometer reads 30 ins. at sea level, the pressure of the atmosphere at that elevation is 14.74 lbs. per sq. in., that is, this pressure will maintain or balance a column of water 34.042 ft. high, when the column is completely exhausted of air, and the water is at a temperature of 62° Fahr.

In other words the pressure of the atmosphere then "*lifts*" (hence the name) the water to such height as will establish equilibrium between the weight of the water and the pressure of the atmosphere. This state of equilibrium is shown in fig. 26.



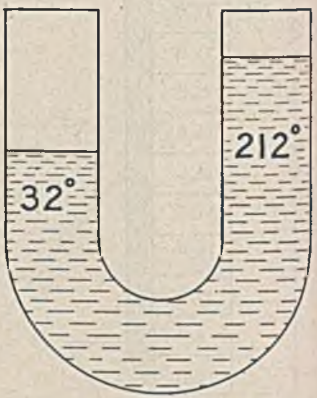


Fig. 6.—Hydraulic principles 4.—Fluids rise to the same level in the arms of a U-tube when the temperature of the liquid is the same throughout.

Fig. 7.—Hydraulic principles 5.—Fluids will not rise to the same level in the arms of a U-tube when the temperatures are not the same in each arm. Why?

Center of Pressure.—By definition, the center of pressure for any plane surface acted upon by a fluid is *the point of action of the resultant pressure acting upon the surface.*

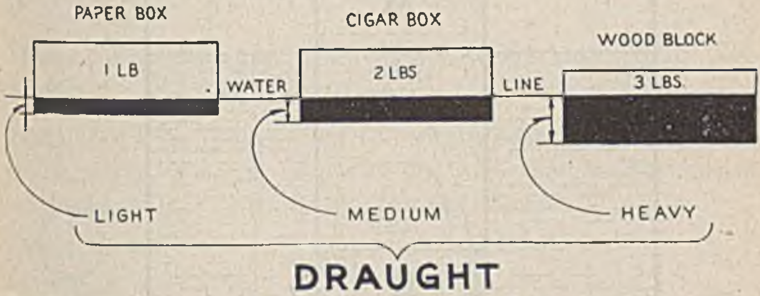
Archimede's Principle.—The resultant pressure of a fluid on a body immersed in it *acts vertically upward through the center* *of gravity of the displaced fluid and is equal to the weight of the fluid displaced.*

Displacement.—By definition, displacement is *the weight of water pushed aside (displaced) by the flotation of a vessel.*

This and a few paragraphs following is a marine digression, but it presents very forceably some principles of hydrostatics.

Ques. Give a simple illustration of displacement.

Ans. Take a paper box, a cigar box and a solid block of wood all exactly the same size. If all three be put in a pan of



Figs. 8 to 10.—Various bores and wooden blocks illustrating **displacement** and **draught.**

water they will float, as in figs. 8 to 10, each sinking in degree proportional to its weight.

Ques. How far do the boxes and solid block sink?

Ans. Each sinks until the weight of the water that they occupy or displace is equal to the weight of each.

Thus in figs. 8 to 10, suppose the paper box weigh 1 lb. the cigar box 2 lbs. and the block 3 lbs. Then the cigar box will sink twice as deep as the paper box and the wooden block three times as deep as plainly shown in the illustrations.

Ques. What is draught?

Ans. A marine term meaning the depth of water to which a vessel will sink, at which depth the weight of water displaced is equal to the weight of the vessel.

Ques. Why is the draught of a boat less in salt water than in fresh water?

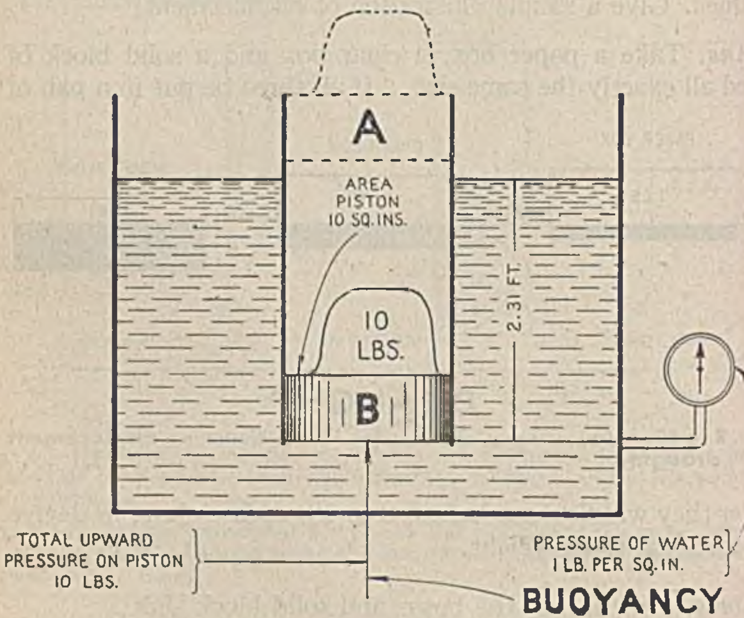


Fig. 11.—Cylinder submerged in water and containing a frictionless piston illustrating **buoyancy.**

Ans. Because salt water weighs more than fresh water.

Ques. What is buoyancy?

Ans. The power or tendency of a liquid to keep a vessel afloat. The upward pressure exerted upon a floating body by a fluid.

Thus in fig. 11, consider a cylinder open at both ends and submerged in water to a depth of 2.31 ft. Now if an air-tight and frictionless piston be inserted in the cylinder at **A**, as shown by dotted lines and released, it would sink to position **B** and remain suspended at that point.

Ques. In fig. 11 what happens during the descent of the piston?

Ans. The pressure of the water acting upward on the lower face of the piston gradually increases.

Ques. Why does the piston not descend lower than position **B**?

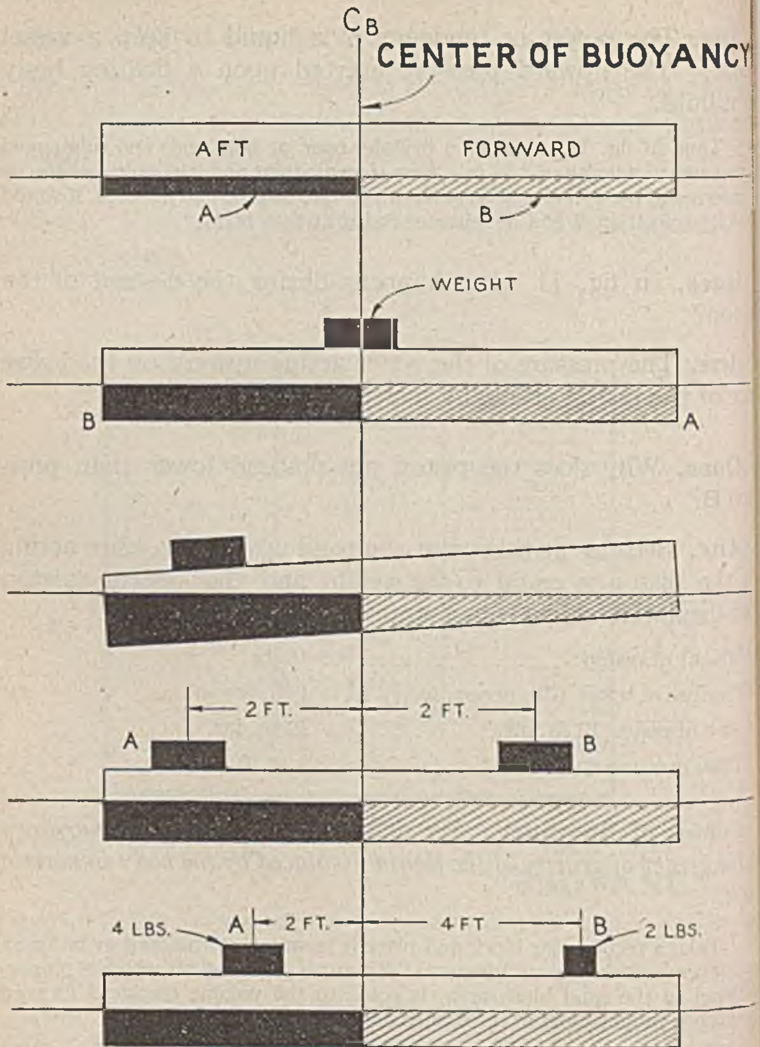
Ans. Because at this point the total upward pressure acting on the piston is equal to its weight and the system (piston and displaced water) is in a state of equilibrium.

Weight of piston	10 lbs.
Pressure of water 1 lb. per sq. in..........	1 lb. per sq. in.
Area of piston 10 sq. ins..................	10 sq. ins.
Total pressure on piston..................	$1 \times 10 = 10$ lbs.

Center of Buoyancy.—By definition, *the center of buoyancy is the center of gravity of the liquid displaced by the body immersed in it.*

Take a rectangular block and place it in water, it will float even (or on an even keel as they say) because the volume displaced aft which is proportional to the solid block area, is equal to the volume displaced forward which is proportional to the shaded area **B**, fig. 12.

The center of buoyancy is clearly shown at the middle point of the rectangular block and is indicated in fig. 12 by the axis C_B.



Figs. 12 to 16.—Diagrams illustrating **center of buoyancy** and effect of dispositions of weight.

In fig. 13 if a weight be placed at the center of buoyancy the block would sink as much forward as aft. That is, points **A** and **B** would be immersed to the same depth.

Consider now placing the weight aft as in fig. 14; this will cause the block to be immersed more aft than forward. Under these conditions the center of buoyancy no longer remains in the middle of the length of the block, but will shift to a point which *depends upon the position of the weight and its size.*

Now in fig. 15 if equal weights as **A** and **B** were placed at equal distances the center of buoyancy would be at the middle point and the block would remain level.

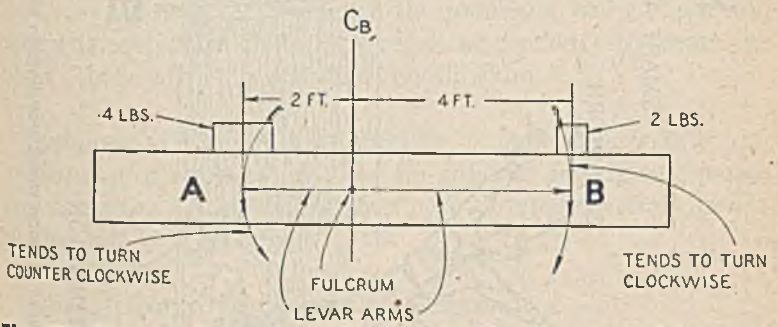


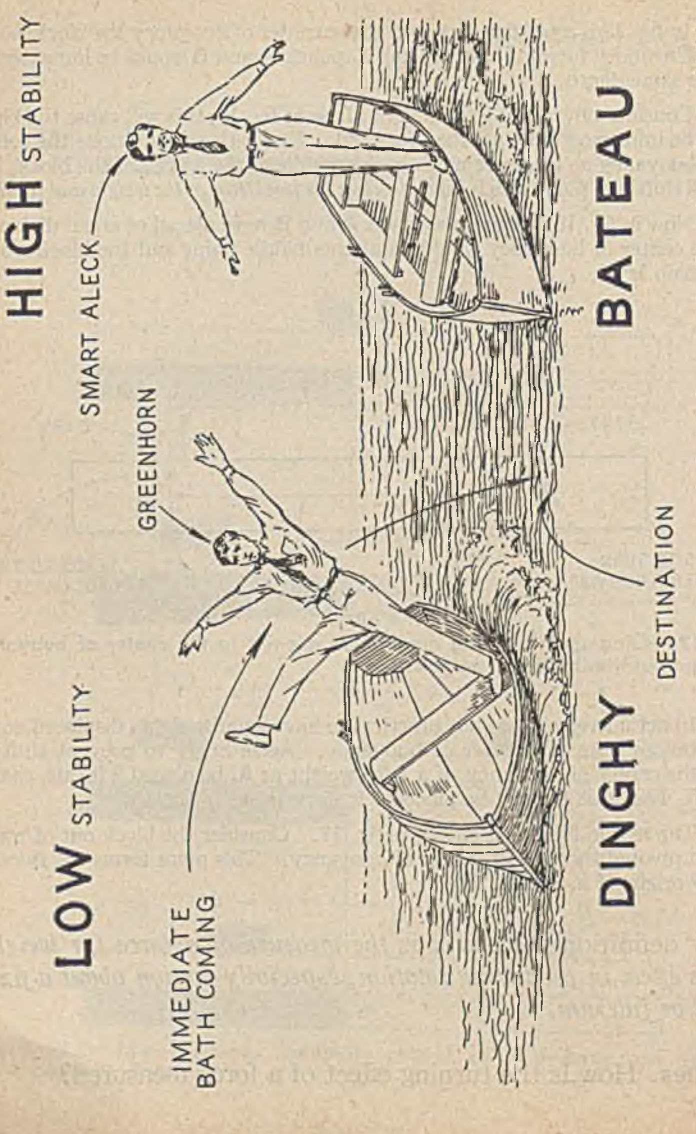
Fig. 17.—Diagram illustrating equilibrium referred to the center of buoyancy by proper distribution of weights.

In actual vessels it is not practical to have equal weights displaced equal distances from the center of buoyancy. Accordingly to prevent shifting of the center of buoyancy, if a 4 lb. weight as **A**, be placed 2 ft. aft, then a 2 lb. weight as **B** must be placed 4 ft. forward as in fig. 16.

The reason for this is shown in fig. 17. Consider the block out of water and pivoted through its center of buoyancy. This pivot forms the *fulcrum* or "origin of moments."

By definition a *moment* is *the measure of a force* (*or weight*) *by its effect in producing rotation, especially motion about a fixed point or fulcrum.*

Ques. How is the turning effect of a force measured?



Figs. 18 and 19.—Results obtained by greenhorn or smart aleck standing on side of two boats having highly different characteristics.

Ans. In foot pounds preferably called pound feet, when the force is measured in pounds and the distance in feet.

Ques. Describe the conditions in fig. 17.

Ans. The 4 lb. weight with a 2 ft. lever arm tends to turn the block counter-clockwise in amount equals $4 \times 2 = 8$ pound feet. Opposed to this the 2 lb. weight with a 4 ft. lever arm tends to turn the block clockwise in amount equals $2 \times 4 = 8$ pound feet. Hence the moments being equal and opposing each other there is no resulting tendency to rotate the block, that is, it is in a state of equilibrium.

Stability.—By definition, stability is *that characteristic (due to shape) of a vessel which gives her capacity for righting herself and assuming her normal upright position after a roll or oscillation caused by a heavy sea.*

Ques. Upon what does stability depend?

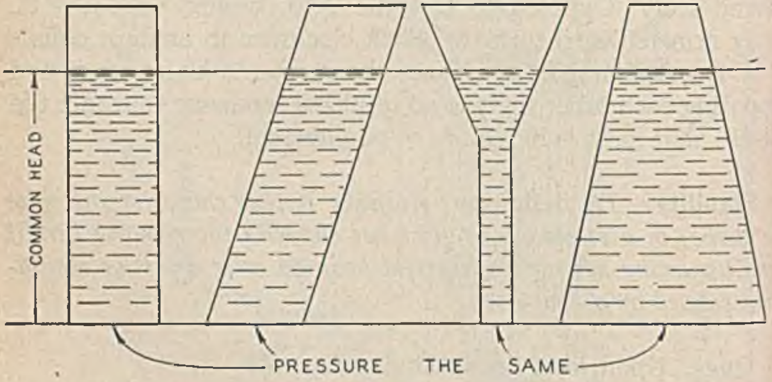
Ans. The body of cross sectional shape of the immersed surface.

For instance the stability of a round bottom yacht's tender or dinghy is very low compared to that of a bateau, that is a flat bottom row boat. The results of a greenhorn and smart aleck standing on the side of these two types are shown in figs. 18 and 19.

Center of Buoyancy.—By definition, *the center of gravity of the displaced fluid and is the point of application of the resultant of all the upward forces acting on the body.*

Metacenter.—In a floating body at rest on the water, *the line joining the center of gravity of the body and center of buoyancy is always vertical and is known as the axis of equilibrium.*

If an external force cause this axis of equilibrium to occupy an inclined position, then if a vertical line be drawn upward from the new center of buoyancy to this axis, the point where it intersects the axis is called the *metacenter*. That is, the metacenter of a vessel's hull is determined by *location of center of gravity or buoyancy of immersed bottom of hull*, for it is that point of transverse section of hull where a vertical line raised from its center of gravity or buoyancy intersects a line passing through the center of gravity of hull.



Figs. 20 to 23.—Hydraulic principles 6.—Any quantity of fluid however small may balance any weight however great—hydrostatic paradox.

Ques. What should be noted about the position of the metacenter?

Ans. If the metacenter is above the center of gravity, equilibrium is stable; if it coincide with it, equilibrium is indifferent and if it be below it, equilibrium is unstable.

Hydrostatic Paradox.—This is the principle that *any quantity of fluid however small may balance any weight however great.*

When water is contained in containers of various shapes as in figs. 20 to 23, *the intensity of pressure in lbs. per sq. in. is the same at the bottoms of the variously shaped containers, but the* *total liquid pressures against the various bottoms are proportional to the areas of the bottoms.*

The amount of liquid or its total weight makes no difference in either the intensity of pressure or the total pressure, so long as the *head* remains the same.

The fact that *the total liquid pressure against the bottom may be many times greater or less than the total weight* of the liquid is termed the ***hydrostatic paradox.***

Archimedes' Principle; Hydrostatic Balance.—A principle stated by Archimedes relating to hydraulics states that: *A body immersed in a fluid loses an amount of weight equal to that of the fluid it displaces.*

When a body is immersed in a liquid it is acted upon by two forces:

1. Gravity

   Which tends to lower it.

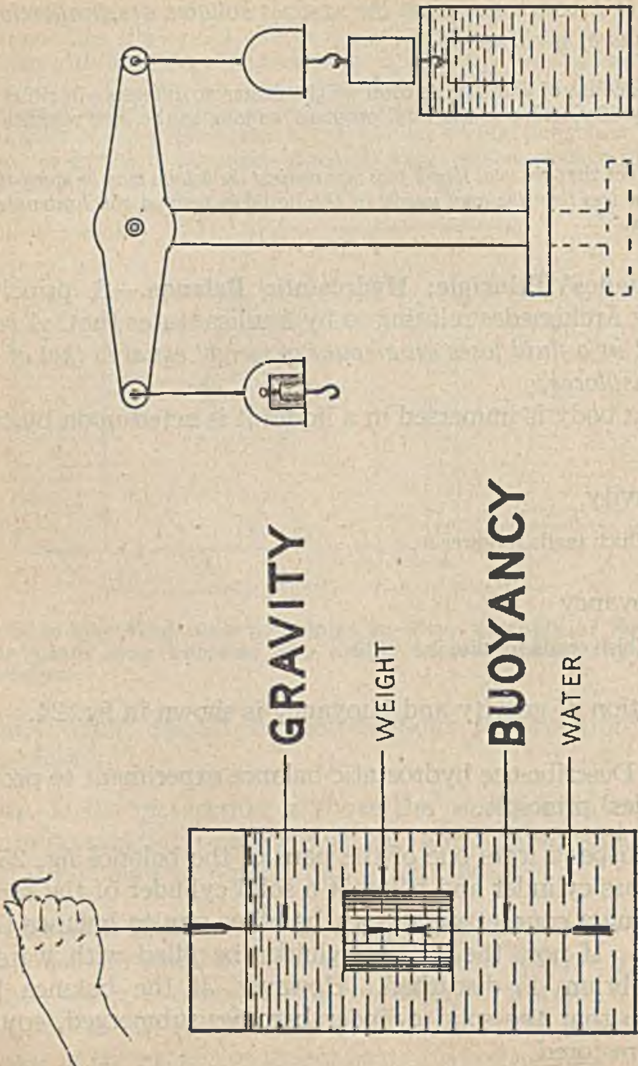
2. Buoyancy

   Which tends to raise it.

The action of gravity and buoyancy is shown in fig. 24.

Ques. Describe the hydrostatic balance experiment to prove Archimedes' principle.

Ans. Suspend from one of the pans of the balance fig. 25 a hollow brass cylinder and below it a solid cylinder of the same size, placing a counter weight on the other pan to balance the assembly. If now the hollow cylinder be filled with water, the equilibrium is disturbed. However, if the balance be lowered so that the solid cylinder becomes submerged, equilibrium is restored.



Figs 24 and 25.—Hydraulic principles 7.—A body immersed in a fluid loses an amount of weight equal to that of the fluid it displaces.—Archimede's principle.

Ques. What happens when the solid cylinder is submerged?

Ans. It loses a portion of its weight equal to that of the water in the hollow cylinder.

Since the value of the hollow cylinder is the same as the solid cylinder the experiment proves Archimedes' principle.

2. Hydrodynamics.

By definition hydrodynamics is *that branch of hydraulics which treats of liquids, especially water, and the forces acting upon it, causing it to be in a state of motion.*

Dynamic Lift.—By definition dynamic lift is *an equivalent or virtual lift of water in motion which represents the resultant pressure necessary to lift the water from a given point to a given height and to overcome all frictional resistance.*

Ques. What is the practical actual lift in pump operation?

Ans. From 20 to 25 feet.

Ques. What kind of piping requires shorter lifts?

Ans. Long inlet lines, multiplicity of inlet elbows and pipes too small.

Ques. Name another condition that reduces the practical limit of lift.

Ans. High altitudes.

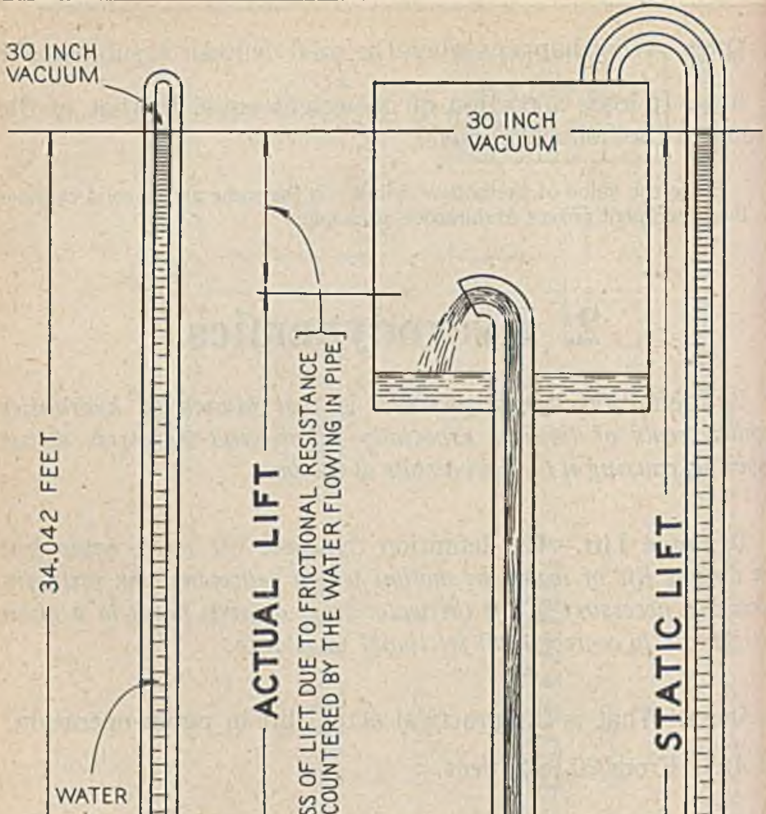


Fig. 26.—Theoretical lift for a pump. This corresponds to static lift for a given barometer reading, but not obtained in practice.

Fig. 27.—Actual lift and the corresponding dynamic lift representing the actual lift plus all frictional resistance.

Effect of Temperature on Dynamic Lift.—Pumps handling water at elevated temperatures must work on reduced actual lift because the boiling point corresponds to the pressure. Thus at 212° Fahr. a pump would not lift water at all because on the admission strokes the cylinder would fill with steam.

Theoretically a perfect pump will draw or "lift" water to a height of 34.042 ft. when the barometer reads 30 ins. but since a perfect vacuum cannot be obtained on account of valve leakage, air contained in the water and the vapor of the water itself, the actual height is generally less than 30 ft. and for warm or hot water, considerably less.

The following table shows the theoretical maximum lift for different temperatures, leakage not considered.

Theoretical Lift for Various Temperatures

Temp. Fahr.	Absolute pressure of vapor lbs. per sq. ins.	Vacuum in inches of mercury	Lift in feet	Temp. Fahr.	Absolute pressure of vapor lbs. per sq. ins	Vacuum in inches of mercury	Lift in feet
102.1	1	27.88	31.6	182.9	8	13.63	15.4
126.3	2	25.85	29.3	188.3	9	11.6	13.1
141.6	3	23.83	27	193.2	10	9.56	10.8
153.1	4	21.78	24.7	197.8	11	7.52	8.5
162.3	5	19.74	22.3	202	12	5.49	6.2
170.1	6	17.70	20	205.9	13	3.45	3.9
176.9	7	15.67	17.7	209.6	14	1.41	1.6

When the water is warm, the height to which it can be lifted decreases, on account of the increased pressure of the vapor. That is to say, for illustration, a boiler feed pump taking water at say 153° Fahr., could not produce a vacuum greater than 21.78 ins., because at that point the water would begin to boil and fill the pump chamber with steam. Accordingly, the theoretical lift corresponding would be

$$34 \times \frac{21.78}{30} = 24.68 \text{ ft. approximately}$$

The result is approximate because no correction has been made for the 34 which represents a 34 foot column of water at 62° Fahr.; of course, at 153° Fahr. the length of such column would be slightly increased.

It should be noted that the figure 24.67 ft. is the *approximate* theoretical lift for water at 153° Fahr.; the *practical* lift would be considerably less.

Dynamic Head.—By definition, dynamic head is *an equivalent or virtual head of water in motion which represents the resultant pressure necessary to force* the water from a given point to a given height and to overcome all frictional resistance.

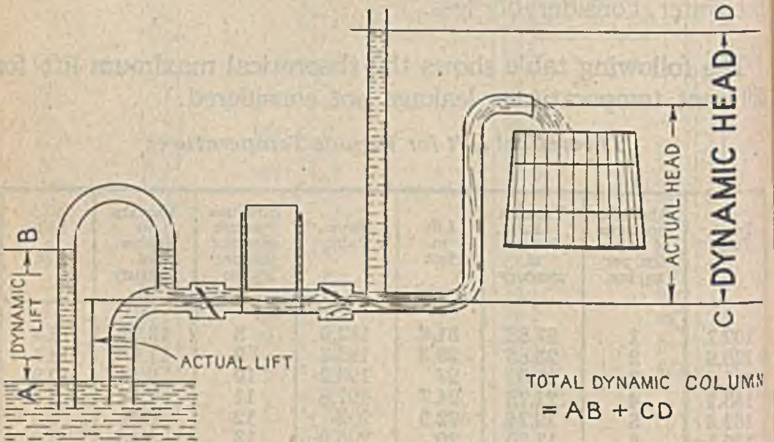


Fig. 28.—Pump diagram illustrating actual lift, actual head and corresponding dynamic lift and dynamic head.

The dynamic or virtual head operating to cause flow is divided into three parts:

1. Velocity head.

This is the height through which a body must fall in a vacuum to acquire the velocity with which the water flows into the pipe equals $v^2 \div 2g$ in which v equals velocity in ft. per sec. and $2g = 64.32$.

THEORETICAL HORSE POWER

REQUIRED TO

RAISE WATER TO DIFFERENT HEIGHTS

Feet Elevation	5	10	15	20	25	30	35	40	45	50	60
Gallons Per Min.											
5	.006	.012	.019	.025	.031	.037	.044	.05	.06	.06	.07
10	.012	.025	.037	.050	.062	.075	.087	.10	.11	.12	.15
15	.019	.037	.056	.075	.094	.112	.131	.15	.17	.19	.22
20	.025	.050	.075	.100	.125	.150	.175	.20	.22	.25	.30
25	.031	.062	.093	.125	.156	.187	.219	.25	.28	.31	.37
30	.037	.075	.112	.150	.187	.225	.262	.30	.34	.37	.45
35	.043	.087	.131	.175	.219	.262	.306	.35	.39	.44	.52
40	.050	.100	.150	.200	.250	.300	.350	.40	.45	.50	.60
45	.056	.112	.168	.225	.281	.337	.394	.45	.51	.56	.67
50	.062	.125	.187	.250	.312	.375	.437	.50	.56	.62	.75
60	.075	.150	.225	.300	.375	.450	.525	.60	.67	.75	.90
75	.093	.187	.281	.375	.469	.562	.656	.75	.84	.94	1.12
90	.112	.225	.337	.450	.562	.675	.787	.90	1.01	1.12	1.35
100	.125	.250	.375	.500	.625	.750	.875	1.00	1.12	1.25	1.50
125	.156	.312	.469	.625	.781	.937	1.094	1.25	1.41	1.56	1.87
150	.187	.375	.562	.750	.937	1.125	1.312	1.50	1.69	1.87	2.25
175	.219	.437	.656	.875	1.093	1.312	1.531	1.75	1.97	2.19	2.62
200	.250	.500	.750	1.000	1.250	1.500	1.750	2.00	2.25	2.50	3.00
250	.312	.625	.937	1.250	1.502	1.875	2.187	2.50	2.81	3.12	3.75
300	.375	.750	1.125	1.500	1.875	2.250	2.625	3.00	3.37	3.75	4.50
350	.437	.875	1.312	1.750	2.187	2.625	3.062	3.50	3.94	4.37	5.25
400	.500	1.000	1.500	2.000	2.500	3.000	3.500	4.00	4.50	5.00	6.00
500	.625	1.250	1.875	2.500	3.125	3.750	4.375	5.00	5.62	6.25	7.50

THEORETICAL HORSE POWER

REQUIRED TO

RAISE WATER TO DIFFERENT HEIGHTS

TABLE—Continued

Feet Elevation	75	90	100	125	150	175	200	250	300	350	400
Gallons Per Min.											
5	.09	.11	.12	.16	.19	.22	.25	.31	.37	.44	.50
10	.19	.22	.25	.31	.37	.44	.50	.62	.75	.87	1.00
15	.28	.34	.37	.47	.56	.66	.75	.94	1.12	1.31	1.50
20	.37	.45	.50	.62	.75	.87	1.00	1.25	1.50	1.75	2.00
25	.47	.56	.62	.78	.94	1.09	1.25	1.56	1.87	2.19	2.50
30	.56	.67	.75	.94	1.12	1.31	1.50	1.87	2.25	2.62	3.00
35	.66	.79	.87	1.08	1.31	1.53	1.75	2.19	2.62	3.06	3.50
40	.75	.90	1.00	1.25	1.50	1.75	2.00	2.50	3.00	3.50	4.00
45	.84	1.01	1.12	1.41	1.69	1.97	2.25	2.81	3.37	3.94	4.50
50	.94	1.12	1.25	1.56	1.87	2.19	2.50	3.12	3.75	4.37	5.00
60	1.12	1.35	1.50	1.87	2.25	2.62	3.00	3.75	4.50	5.25	6.00
75	1.40	1.69	1.87	2.34	2.81	3.28	3.75	4.69	5.62	6.56	7.50
90	1.68	2.02	2.25	2.81	3.37	3.94	4.50	5.62	6.75	7.87	9.00
100	1.87	2.25	2.50	3.12	3.75	4.37	5.00	6.25	7.50	8.75	10.00
125	2.34	2.81	3.12	3.91	4.69	5.47	6.25	7.81	9.37	10.94	12.50
150	2.81	3.37	3.75	4.69	5.62	6.56	7.50	9.37	11.25	13.12	15.00
175	3.28	3.94	4.37	5.47	6.56	7.66	8.75	10.94	13.12	15.31	17.50
200	3.75	4.50	5.00	6.25	7.50	8.75	10.00	12.50	15.00	17.50	20.00
250	4.69	5.62	6.25	7.81	9.37	10.94	12.50	15.72	18.75	21.87	25.00
300	5.62	6.75	7.50	9.37	11.25	13.12	15.00	18.75	22.50	26.25	30.00
350	6.56	7.87	8.75	10.94	13.12	15.31	17.50	21.87	26.25	30.62	35.00
400	7.50	9.00	10.00	12.50	15.00	17.50	20.00	25.00	30.00	35.00	40.00
500	9.37	11.25	12.50	15.62	18.75	21.87	25.00	31.25	37.50	43.75	50.00

2. Entry head.

Head required to overcome the frictional resistance to entrance to the pipe. With sharp edged entrance the entry head equals about one-half the velocity head; with smooth rounded entrance the entry head is negligible.

3. Friction head.

This is due to the frictional resistance to flow within the pipe. In ordinary cases of pipes of considerable length, the sum of the entry and velocity heads required scarcely exceeds 1 ft. In the case of long pipes with low heads the sum of the velocity and entry heads is generally so small that it may be neglected.

The tables on pages 195 and 196 give the loss of head due to friction of water in pipes and elbows of various sizes and for various rates of flow.

Capillary Attraction.—By definition, *a measure of surface tension observed in liquids which "wet" the surface.*

In fine tubes and bores the surface tension is sufficient to balance a small column of liquid maintaining it at a level above the outside.

In fig. 29 is shown a series of glass capillary tubes of varying diameter; when immersed in water for instance the water in the tubes will rise to a higher level than in the container.

Ques. How do other fluids act?

Ans. They are raised to unequal heights by the same tube.

Ques. What effect has the diameter on height to which the liquid will rise?

Ans. The height to which a liquid will rise is inversely proportional to the diameters of the bores of the tubes.

Capillary Depression.—By definition, *a manifestation of surface tension observed in liquids which* **do not** *"wet."*

When a liquid like mercury does not wet the tube, the behavior is just the reverse of liquids which wet the surface. That is, the level is depressed and maintained at a level lower than that of the outside as shown in fig. 30.

The extent of the depression is inversely proportional to the diameters of the bores of the tube.

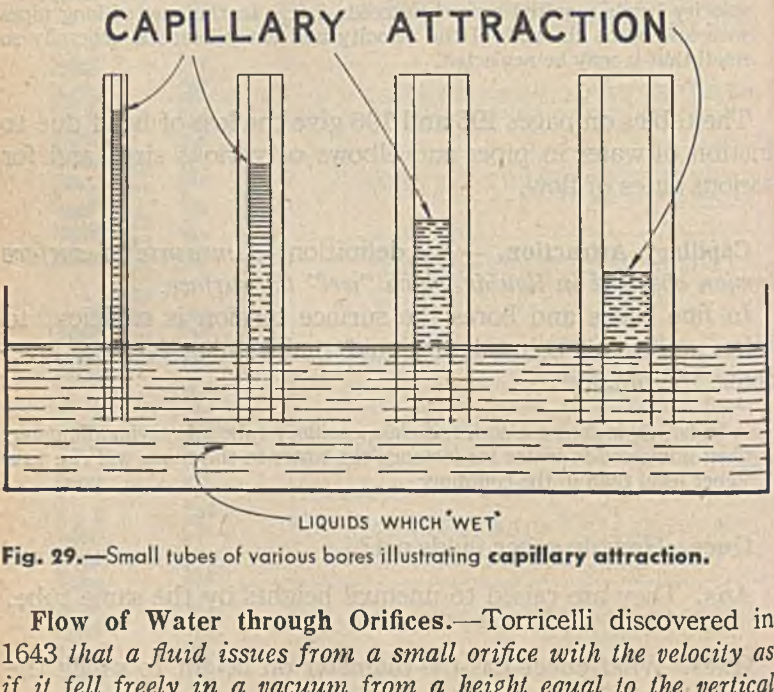


Fig. 29.—Small tubes of various bores illustrating **capillary attraction.**

Flow of Water through Orifices.—Torricelli discovered in 1643 *that a fluid issues from a small orifice with the velocity as if it fell freely in a vacuum from a height equal to the vertical distance from the surface to the center of the orifice.*

If a jet issuing from an orifice in a vertical direction have the same velocity as a body would have which fell from the surface of the liquid to that orifice, the jet ought to rise to the level of the liquid. It does not, however, reach this; for the particles which fall hinder it.

By inclining the jet at a small angle with the vertical it reaches about 9/10 of the theoretical height, the difference being due to friction and to the resistance of the air.

Ques. Are the sizes of orifices and quantities of water issuing from them proportional?

Ans. Very nearly so.

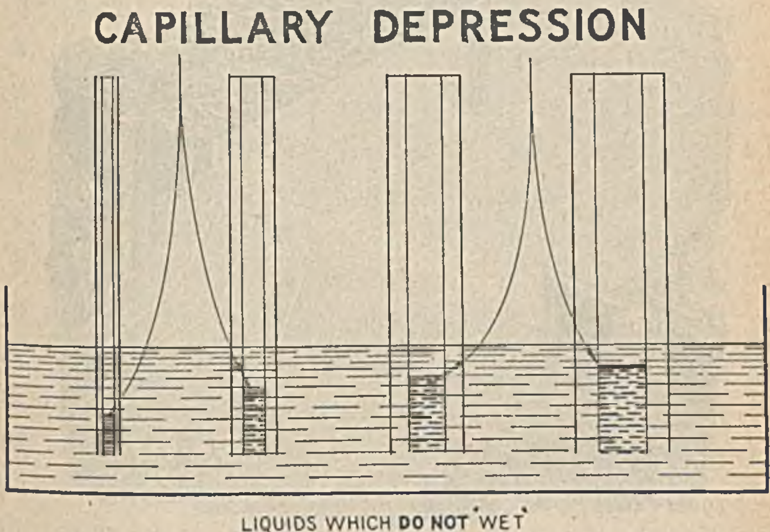


Fig. 30.—Small tubes of various bores illustrating **capillary depression.**

Water Flow Measurement.—A device commonly used for measuring water flow, especially that of small streams, is the weir—sometimes called tumbling bay.

The weir proper consists of a notched board as shown in fig. 31. To make a weir, place a board across the stream at some point which will allow a pond to form above. The board should have a notch cut in it with both side edges and the bottom sharply beveled, as shown in the cut.

The bottom of the notch, which is called the "crest" of the weir, should be perfectly level and the sides vertical.

In the pond back of the weir, at a distance not less than the length of the notch, drive a stake near the bank, with its top precisely level with the crest.

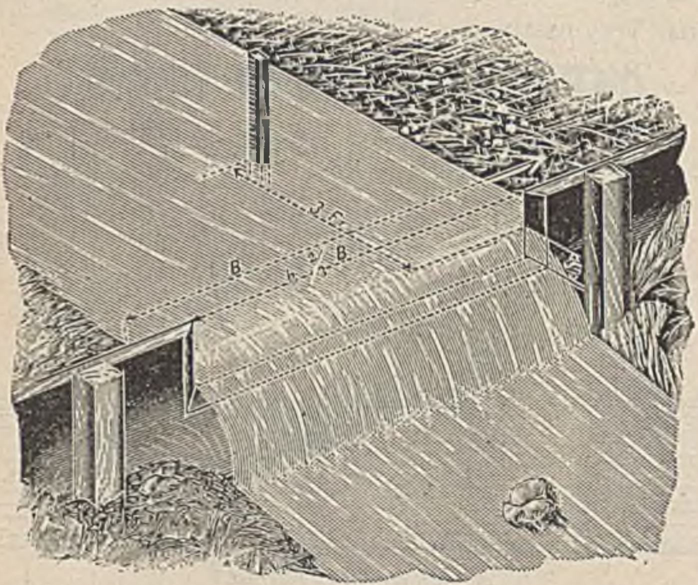


Fig. 31.—Example of a weir.

By means of a rule, or a graduated stake as shown, measure the depth of water over the top of stake, making allowance for capillary attraction of the water against the sides of the weir.

Ques. How is the depth measured for extreme accuracy?

Ans. By means of a hook gauge.

WEIR TABLE

INCHES		1/8	1/4	3/8	1/2	5/8	3/4	7/8
0	.00	.01	.05	.09	.14	.19	.26	.32
1	.40	.47	.55	.64	.73	.82	.92	1.02
2	1.13	1.23	1.35	1.46	1.58	1.70	1.82	1.95
3	2.07	2.21	2.34	2.48	2.61	2.76	2.90	3.05
4	3.20	3.35	3.50	3.66	3.81	3.97	4.14	4.30
5	4.47	4.64	4.81	4.98	5.15	5.33	5.51	5.69
6	5.87	6.06	6.25	6.44	6.62	6.82	7.01	7.21
7	7.40	7.60	7.80	8.01	8.21	8.42	8.63	8.83
8	9.05	9.26	9.47	9.69	9.91	10.13	10.35	10.57
9	10.80	11.02	11.25	11.48	11.71	11.94	12.17	12.41
10	12.64	12.88	13.12	13.36	13.60	13.85	14.09	14.34
11	14.59	14.84	15.09	15.34	15.59	15.85	16.11	16.36
12	16.62	16.88	17.15	17.41	17.67	17.94	18.21	18.47
13	18.74	19.01	19.29	19.56	19.84	20.11	20.39	20.67
14	20.95	21.23	21.51	21.80	22.08	22.37	22.65	22.94
15	23.23	23.52	23.82	24.11	24.40	24.70	25.00	25.30
16	25.60	25.90	26.20	26.50	26.80	27.11	27.42	27.72
17	28.03	28.34	28.65	28.97	29.28	29.59	29.91	30.22
18	30.54	30.86	31.18	31.50	31.82	32.15	32.47	32.80
19	33.12	33.45	33.78	34.11	34.44	34.77	35.10	35.44
20	35.77	36.11	36.45	36.78	37.12	37.46	37.80	38.15

Weir Table.—The weir table contains figures 1, 2, 3, etc., in the first vertical column which indicates the inches depth of water running over weir board notches. Frequently the depths measured represent also fractional inches, between 1 and 2, 2 and 3, etc.

The horizontal line of fraction at the top represents these fractional parts, and can be applied between any of the numbers of inches depth, from 1 to 21.

The body of the table shows the cubic feet, and the fractional parts of a cubic foot, which will pass each minute for each inch in depth, and for each fractional part of an inch by eighths for all depths from 1 to 21 ins. Each of these results is for only 1 in. width of weir.

To estimate for any width of weir the result obtained for 1 in. width must be multiplied by the number of inches constituting the whole horizontal length of water.

Ques. Having ascertained the depth of water over the stake, how is the amount of water flowing determined?

Ans. By referring to the accompanying table from which may be calculated the amount of water flowing over the weir.

Ques. In making a weir what proportions should be observed in the dimension of the notch?

Ans. Its length or width should be between four and eight times the depth of water flowing over the crest of the weir.

Ques. How about the pond back of the weir?

Ans. It should be at least 50 per cent wider than the notch and of sufficient width and depth that the velocity of flow or approach be not over one foot per second.

In order to obtain these results it is advisable to experiment to some extent.

CHAPTER 4

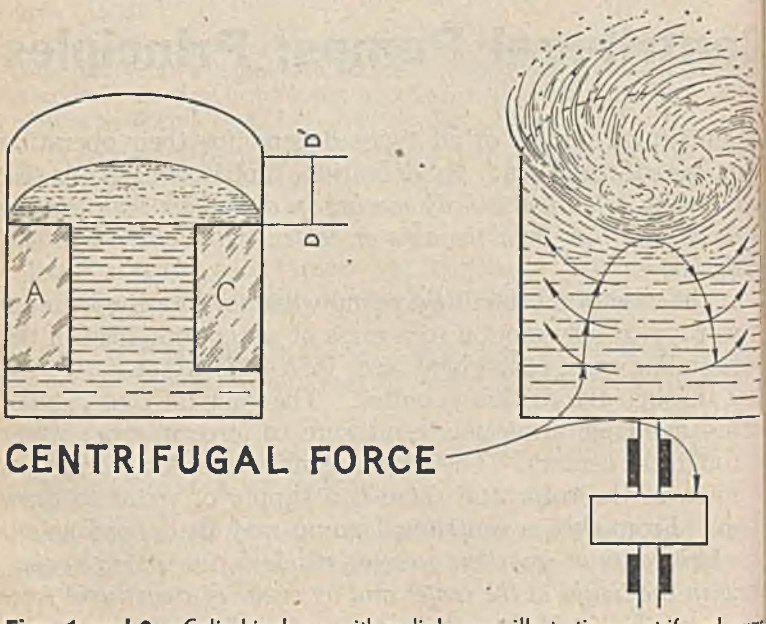
Centrifugal Pumps; Principles

Centrifugal pumps of all types depend for their operation upon centrifugal force. By definition, centrifugal force is *that force which acts upon a body moving in a circular path tending to force it farther from the axis or center of the circle described by the body.*

In the case of a centrifugal pump when its rotating member gives rapid rotary motion to a mass of water contained in the surrounding case, centrifugal force forces the water out of the case through the discharge outlet. The vacuum thus created makes available atmospheric pressure to force in more water (through the center). The process continues as long as motion is given to the rotor and there is a supply of water to draw upon. From this, a centrifugal pump may be defined as *one in which vanes or impellers rotating inside a close fitting casing, draw in the liquid at the center and by virtue of centrifugal force throw out the liquid through an opening at the periphery of the casing.*

How a Centrifugal Pump Works.—Take a cylindrical can as in fig. 1, having radial vanes A, C, to force the liquid in it to revolve when the can is rotated. In fig. 2, mount the can on a shaft and with pulley so as to rotate the can at great speed.

Centrifugal force acts upon the water (rotating at high speed to force the water out toward the walls of the can. This causes the water to press radially outward and since it cannot get past the walls, the pressure pushes the water upward and causes it to overflow along the circumference of the can at the same time the water near the center is drawn downward.



Figs. 1 and 2.—Cylindrical can with radial vanes illustrating centrifugal pump principles.

Ques. What causes the water near the center to be drawn downward?

Ans. The water moving outward creates a vacuum near the center and atmospheric pressure forces it downward.

Ques. What is the trouble with this primitive arrangement?

Ans. There is nothing to catch the water as it spills over the rim. No supply of additional water, and since the water which spills over the top has a high velocity equal to the rim speed, the kinetic energy thus generated is wasted.

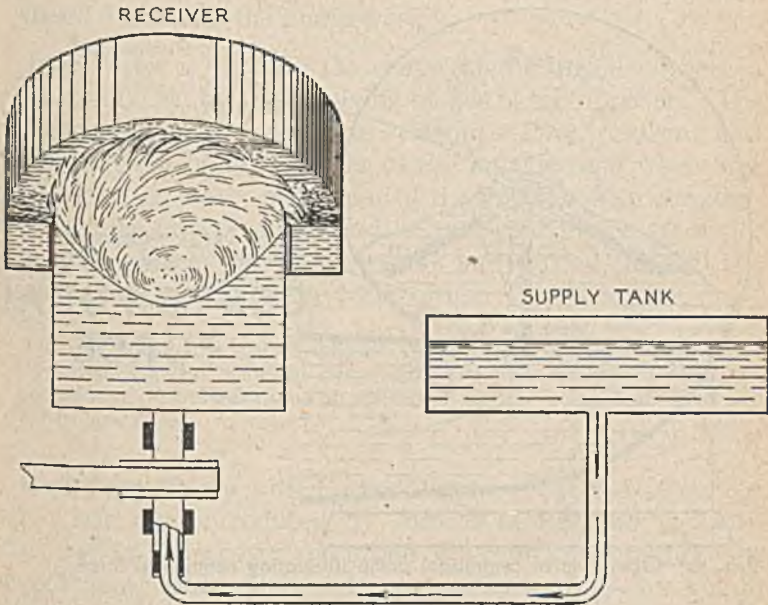


Fig. 3.—Elementary centrifugal pump consisting of cylindrical can with radial vanes, concentric receiver and supply tank.

Ques. How do figs. 1 and 2 show any pumping operation?

Ans. From fig. 2 it is evident that the water has been lifted a distance DD′ (fig. 1).

Ques. Describe the arrangement shown in fig. 3.

Ans. The illustration shows a receiver to catch the water when it spills over the top of the can and a supply tank connected with the hollow shaft to supply water to the can.

With these provisions, in operation, water will be pumped by this crude arrangement from the supply tank to the receiver. If, instead of rotating the whole can, the vanes only be rotated, the same result will obtain.

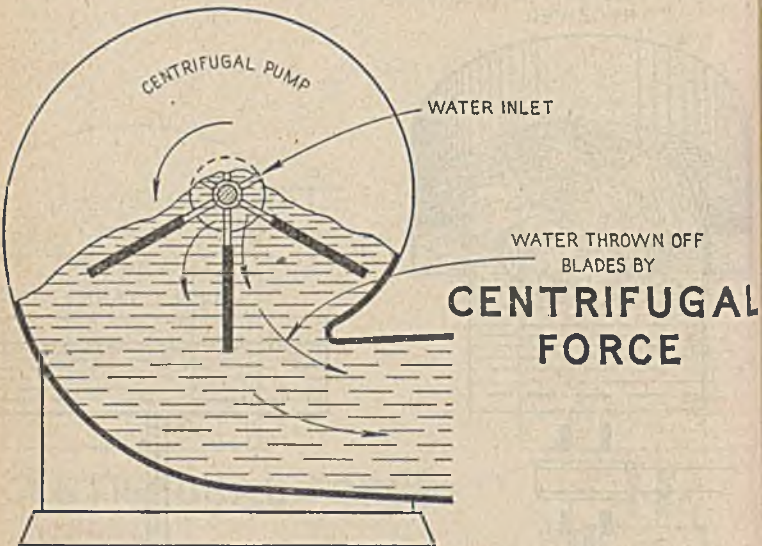


Fig. 4.—Operation of centrifugal pump illustrating centrifugal force.

Centrifugal Pump with Straight Vanes.—The first practical centrifugal pump was built with a rotor having straight (radial) vanes. The essential parts of a centrifugal pump are:

NOTE.—Many people are practically acquainted with the principle of the centrifugal pump; that is, the force by which a body revolving around a center tends to recede from the center and with a force proportional to its velocity. Thus mud is thrown from the rims of carriage wheels when they move rapidly over wet roads; a stone in a sling flies off the moment it is released. A bucket of water may be whirled like a stone in a sling and the contents retained even when the bucket is upside down.

1. Impeller, or rotating member.
2. Surrounding case.

Fig. 4 shows an early centrifugal pump with an impeller having straight vanes. Though a very inefficient type it will serve very plainly to show how centrifugal pumps work.

Ques. How does the pump work?

Ans. Water is led into the center of the impeller where it is set in rotation by the revolving blades of the impeller. The rotation of the water generates *centrifugal force*, resulting in a pressure at the outer diameter of the impeller and when flow takes place the water passes out of the impeller with considerable velocity and pressure, and is connected in the gradually expanding passageway of the casing and carried through the discharge connection to the point of use.

In the figure, the arrows indicate the direction of flow. Note water thrown off blades or vanes by *centrifugal force*. The analysis of what goes on inside a centrifugal pump in operation is quite complicated and will not be given here.

Centrifugal Pump with Curved Vanes.—The use of curved vanes was first introduced by Appold in England in 1849. Figs. 5 and 6 show cover and inside of a centrifugal pump with curved vanes and volute case, commonly known as a volute pump.

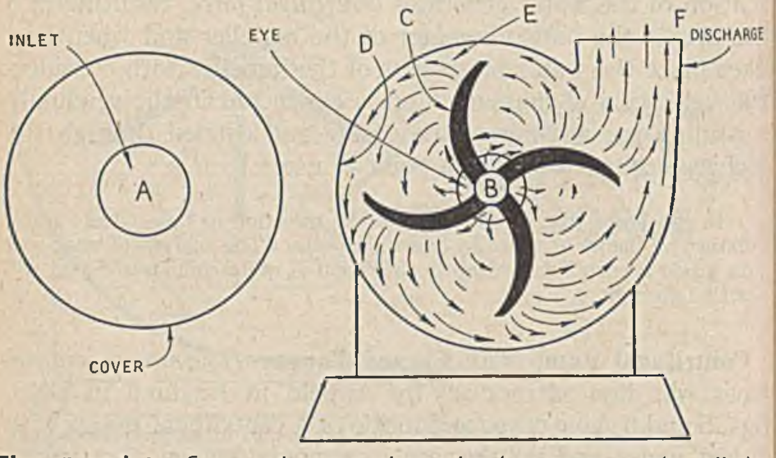
An inlet pipe connection **A**, to the cover (fig. 5) leads the water into the "eye" **B** of the rotating impeller.

The curved vanes **C**, of the impeller direct the water from the eye to discharge edge **D**, moving the water in a spiral path.

As the impeller revolves, the water moves toward the discharge edge **D**. The water then enters the volute shaped passageway **E**, where it is collected from all around the impeller which leads to the discharge connection **F**.

The Volute.—By definition, a volute is a *curve winding about and constantly receding from a center;* a spiral, which lies in a plane (as distinguished from a conical spiral). This is the shape given to the periphery of the case surrounding the impeller of a volute type centrifugal pump.

The casings made to this shape form *a* **progressively** *expanding passageway* into which the impeller discharges the water. In other words the volute passageway collects the water from the impeller and directs it to the discharge opening.



Figs. 5 and 6.—Cover and section of centrifugal pump commonly called a volute pump on account of the shape of the casing. The illustrations show the inlet, eye, discharge, etc.

Ques. How is the volute proportioned?

Ans. The volute shaped casing is so proportioned as to produce equal velocity flow all around the circumference and to reduce gradually the velocity of the liquid as it flows from the impeller to the discharge.

Ques. What is the object of this arrangement?

Ans. To change velocity head into pressure head.

Curvature of Impeller Vanes.—If the vanes were made to mathematically correct shape, there would be a different curve for every change of working conditions, but this is not

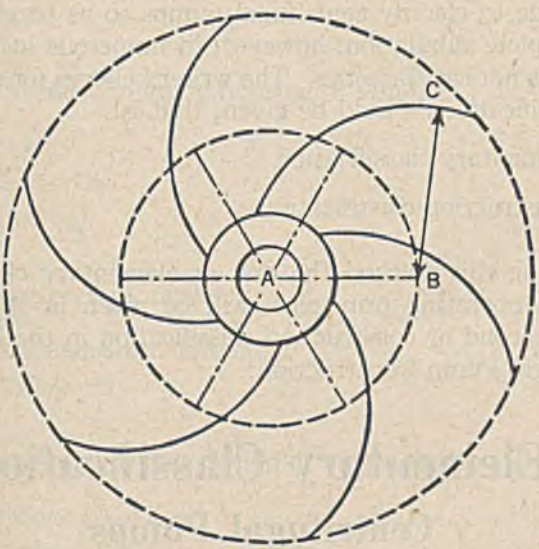


Fig. 7.—Method of laying out impeller vanes.

commercially feasible, because of the undue multiplicity of patterns to be carried in stock.

Fig. 6 shows a simple method of describing the curve of the vanes which works well for impellers up to large diameters and lifts of 60 ft. or over.

The method is as follows:

Divide the circle into say six number of arms. Bisect each radius. Then using this bisected point as a center and with a radius

$$BC = AB + \frac{1}{6} \text{ of } AB$$

describe the curves which represent the working faces of the vanes.

Classification of Centrifugal Pumps.—Many attempts have been made to classify centrifugal pumps so as to give a clear and complete tabulation; however, in numerous instances the result was not satisfactory. The writer believes for simplicity, two classifications should be given, that is:

1. Elementary classification
2. Construction classification

Adopting this method the first or elementary classification based on operating principles, will be given in this chapter and the second or construction classification in the chapter on Centrifugal Pump Construction.

Elementary Classification

Centrifugal Pumps

Considering the basic designs of centrifugal pumps corresponding to various principles of operations, centrifugal pumps may be classed:

1. With respect to the shape of the casting, as
   a. Conoidal
   b. Volute

2. With respect to the shape of the vanes, as

   a. Straight
   b. Curved

3. With respect to the number of vanes, as

   a. Single
   b. Multi

4. With respect to guide housing of vane, as

   a. Open
   b. Semi-open
   c. Enclosed

5. With respect to intake, as

   a. Single admission (suction *)
   b. Double " "

6. With respect to flow design of vanes, as

   a. Francis mixed flow
   b. Screw axial flow

7. With respect to secondary or diffusion vanes, as

   a. Turbine

*NOTE.—The author objects to the word "*suction*" under any circumstances, but virtually nothing can be done about it, otherwise the vernacular would be foreign to those "fellas" residing in districts remote from centers of learning.

8. With respect to balancing, as

   a. Single admission

   Balancing disc

   b. Double admission
   - Double admission impeller.
   - Back to back single admission impellers.
   - Uni-directional admission impellers with balancing drum.
   - Balancing chamber.

9. With respect to stage operation, as

   a. Single-stage

   b. Multi-stage

10. With respect to output, as

    a. Large volume—low head

    b. Medium volume—medium head

    c. Small volume—high head

Single Stage Pumps.—This type pump is adapted to installations for pumping against low or moderate heads. The single admission type pumps, as shown in fig. 8, are most commonly used for the small sizes. Single admission pumps are made in one or more stages. The double admission pump shown in fig. 9 may be either single or multi-stage.

Ques. What is the advantage of the double admission type?

Ans. The impeller is hydraulically balanced in an axial direction.

Ques. Why?

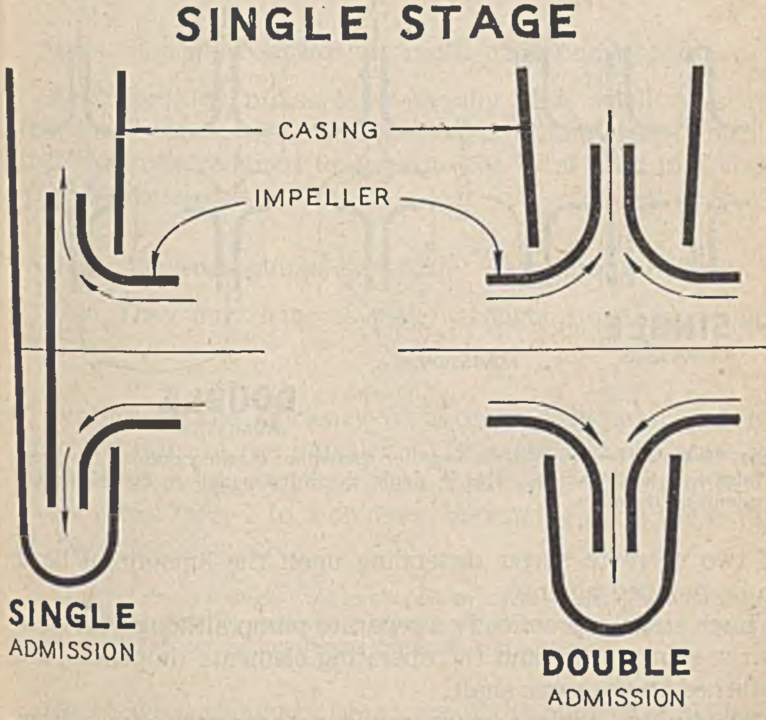
Ans. Because the thrust from one admission stream is counteracted by the thrust from the other.

Ques. Mention another feature of the double admission single stage pump.

Ans. It is adapted to elevating large quantities of water to moderate heights.

Ques. What is the disadvantage of the single stage pump?

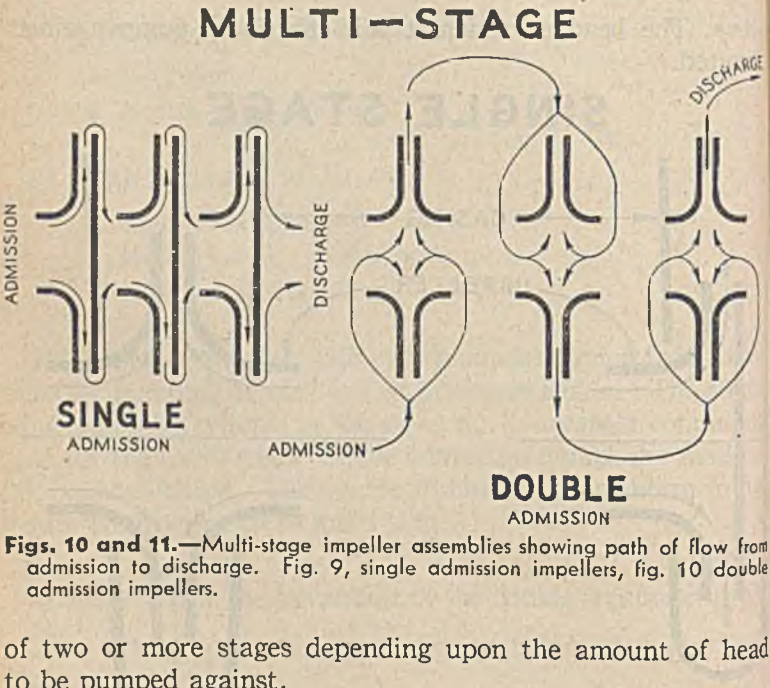
Ans. The head at which it will effectively pump against is limited.



Figs. 8 and 9.—Single and **double** admission impellers. Diagram showing direction of flow.

The head generated by a single impeller is a function of its tangenti[al] speed. It is possible and in some cases practicable to generate as muc[h] as 1,000 ft. head with a single impeller, but for heads exceeding 250 t[o] 300 ft. multi-stage pumps are generally employed.

Multi-Stage Pumps.—The multi-stage centrifugal pump is essentially a high head or high pressure pump and consists



Figs. 10 and 11.—Multi-stage impeller assemblies showing path of flow from admission to discharge. Fig. 9, single admission impellers, fig. 10 double admission impellers.

of two or more stages depending upon the amount of head to be pumped against.

Each stage is practically a separate pump although they are in the same casing and the operating elements (impellers) are attached to the same shaft.

The initial or first stage receives the water direct from the source through the admission pipe, and in operation builds up the pressure to a proper single stage amount, and passes it into the next succeeding stage.

Here the pressure is increased to the capacity of that stage, and is passed on to the next stage. This process continues throughout the entire succession of stages, the pressure being increased or built up in each stage until the water is discharged from the final stage, at such pressure and volume as the pump is intended to deliver.

The diagrams figs. 10 and 11 show water flow in single and double admission multi-stage.

Ques. For what service are multi-stage pumps adapted?

Ans. For high pressures, especially with small capacities or where, because of speed limitations, the diameter of impeller that would be required to generate the total head in a single stage is excessive.

Ques. For such pumps how many stages are provided?

Ans. They may have as many as eight stages in a single casing.

Impellers.—The efficiency of a centrifugal pump depends upon the form of the impeller. The vanes and all details are designed to meet given operating conditions. The number of vanes varies from 1 to 8 or more, depending upon the nature of the service, size, etc.

Fig. 12 shows a single vane semi-open impeller. Such design is adapted to special industrial pumping problems which require a rugged pump to handle liquids containing fibrous materials and some solids.

Ques. What should be noted about the number of blades in design?

Ans. The minimum number necessary for proper water guidance should be used.

Ques. Why?

Ans. Because a decrease in number of blades decreases the friction head due to vane shock.

Ques. For what service is the open type vane suitable?

Ans. For liquids clear of foreign matter which would be liable to clog between impeller and stationary side plates.

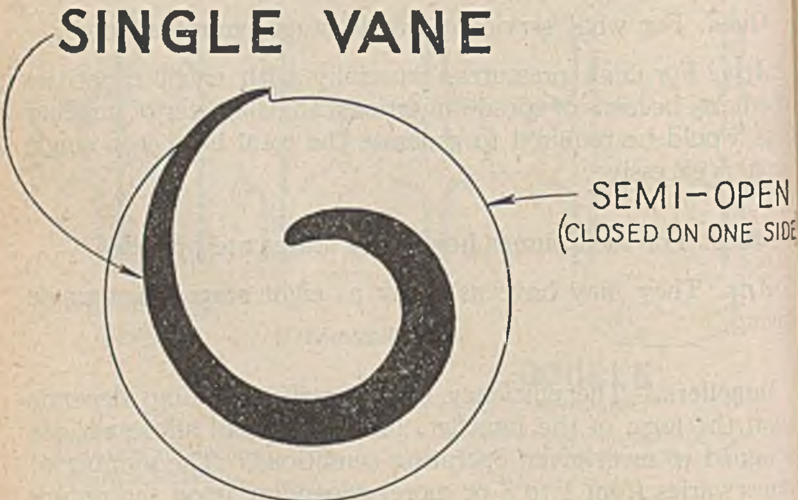
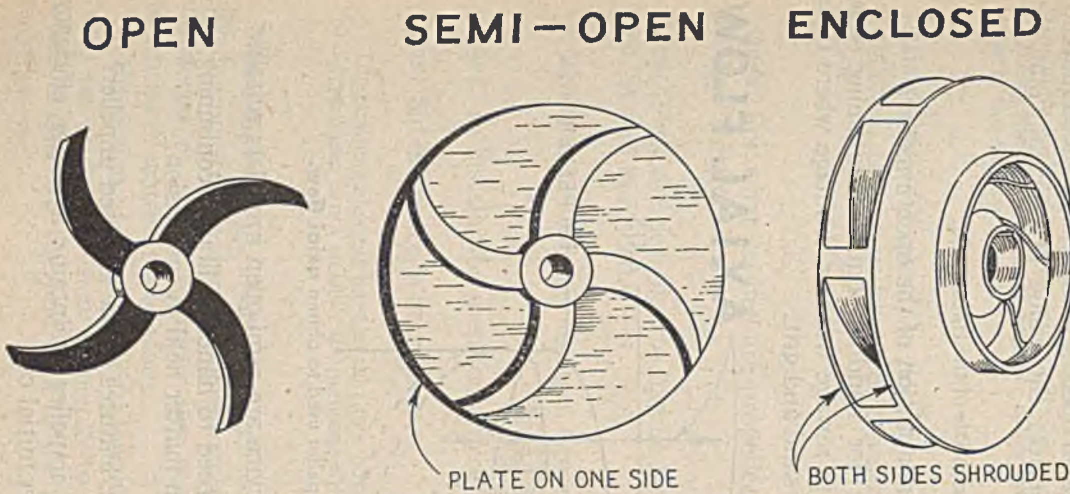


Fig. 12.—Single vane semi-open impeller.

Ques. For what are enclosed or "shrouded" impellers adapted?

Ans. They are designed for various applications. The shape and number of vanes being governed by the service conditions.



Figs. 13 to 15.—Impeller types. Fig. 12, open; fig. 13, semi-open; fig. 14, enclosed.

Shrouded impellers do not require wearing plates. Owing to the enclosed impeller construction, wear is reduced to a minimum, assuring full capacity operation with initial high efficiency for a prolonged period.

Enclosed impellers are not clogging in the sense that they do not depend upon close running clearances.

Ques. What governs the choice between the three types of impeller?

Ans. Whether to use an open, semi-open or enclosed impeller depends upon the service, the value of efficiency and the cost.

Figs. 13 to 15 show essentials of the three types.

Ques. What is the adaptation of the open impeller?

Ans. It is suitable for handling liquids containing some solids, such as found in sewage work or drainage where there is a limited amount of sand and grit.

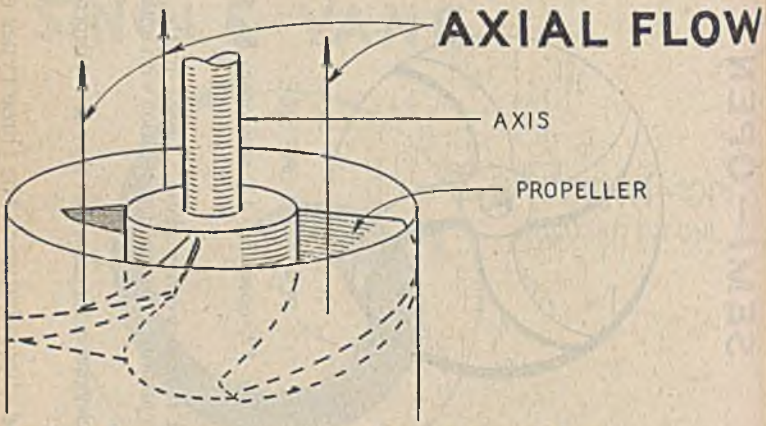


Fig. 16.—Propeller type impeller used to obtain **axial flow.**

Ques. For what service are semi-open impellers suitable?

Ans. They are adapted to handling liquids containing sediment and other foreign matter held in suspension.

Ques. What are the features of the enclosed impeller?

Ans. The enclosed impeller maintains a better efficiency but at a slightly greater initial cost.

In many cases where power costs are important items, the additional cost of the enclosed impeller pump will soon be paid out of savings, notwithstanding the life of the open impeller is likely to be somewhat longer than that of the enclosed type.

Axial Flow.—To obtain a flow in the direction of the axis of rotation, the propeller type impeller is used as shown in fig. 16.

Ques. For what service are propeller type impellers giving axial flow suited?

Ans. They are designed to handle large quantities of water at no lift and low heads.

Services such as drainage, irrigation, excavation, drainage, sewage, etc.

Ques. In installation mention a necessary requirement.

Ans. Submergence is necessary.

The pumping element must at all times be submerged. In other words this type pump is not suitable for pumping with lift.

Mixed Flow.—Fig. 17 shows the mixed flow impeller and the distribution of the water passing through same. This type was introduced as suited to meet the need for pumping large quantities of water at low heads.

In the design of large capacity low head pumps, they were based on the mixed flow principle to increase the rotative speeds, reduce the size and bulk of the pump, also to increase the efficiency.

Balancing Centrifugal Pumps.—The centrifugal pump is inherently an unbalanced machine. All centrifugal pumps are subject to end thrust and some means must be provided to counteract this load.

A single admission impeller experiences an unbalanced hydraulic thrust directed axially toward the admission side. This is due to the vacuum in the admission side causing atmospheric pressure to act on the impeller producing a thrust.

Various methods of balancing have been tried. Centrifugal pumps may be balanced:

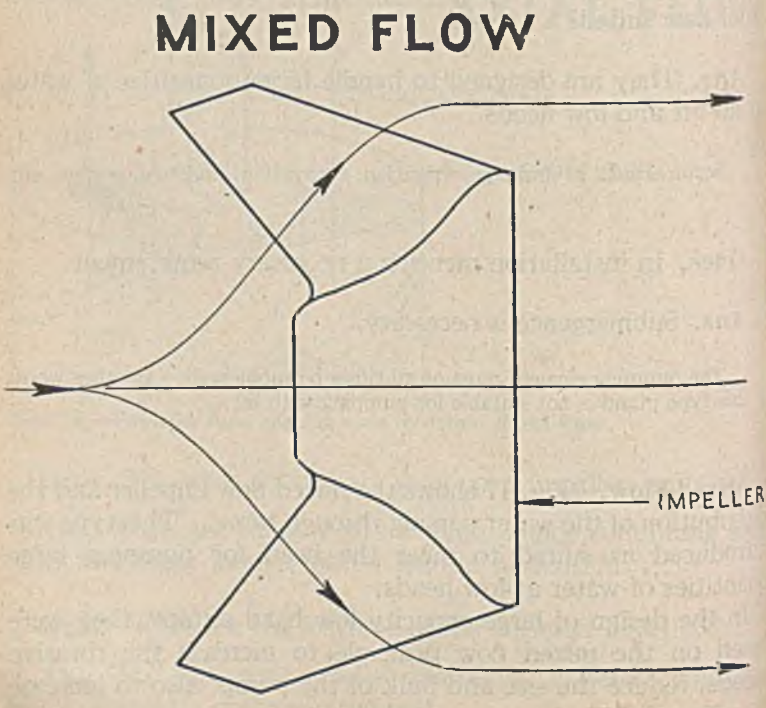


Fig. 17.—Mixed flow-type impeller.

1. Naturally

   As by opposing impellers.

2. Mechanically

   As by balancing disc, etc.

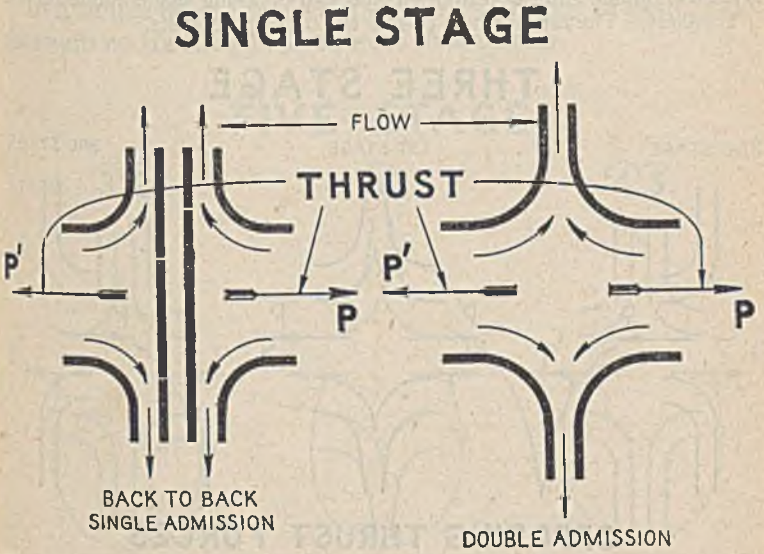


Fig. 18.—Natural balancing 1. Single stage with two single admission back to back impellers.

Fig. 19.—Natural balancing 2. Single stage with one double admission impeller.

Natural Balancing.—This method is generally employed in double admission, single stage pumps and on single or double admission multi-stage pumps.

Figs. 18 and 19 show back to back single admission impellers and a double admission impeller, being diagrams showing direction of the axial thrusts. In each combination there are two thrusts acting in an axial direction and opposing each other. Indicated in the figs. by **P** and **P′**.

The method of opposing impellers is applied also to multi-stage pumps by various impeller arrangements.

For instance, fig. 20 shows a three stage arrangement consisting of a central double admission impeller and two opposing single admission end impellers. The arrows show plainly the opposing thrusts.

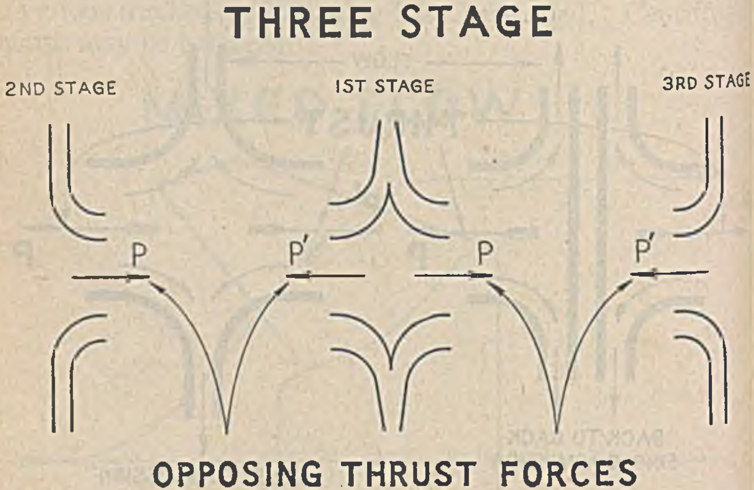


Fig. 20.—Natural balancing 3. Three stage with assembly of one double admission and two opposed single admission impellers.

A further development of the opposed impeller balancing is the five stage pump having a central double admission impeller for the first stage, two pairs of opposing back to back impellers for the second, third and the fourth, fifth stages.

This is shown in diagrammatic form in fig. 21. As seen there are three thrust forces **P,P,P,** counterbalanced by three equal opposing forces P′P′P′. This method extended to six stages is shown in fig. 22.

Mechanical Balancing.—End thrust being due to the dynamic force necessary because the liquid enters through the eye of the impeller in an axial direction and leaves in a radial direction, is also due to forces acting on the shrouds of the impeller. This is due to the fact that the liquid in the clearance spaces is under pressure.

In a simple open impeller these forces are not present, since there are no shrouds for the forces to act upon.

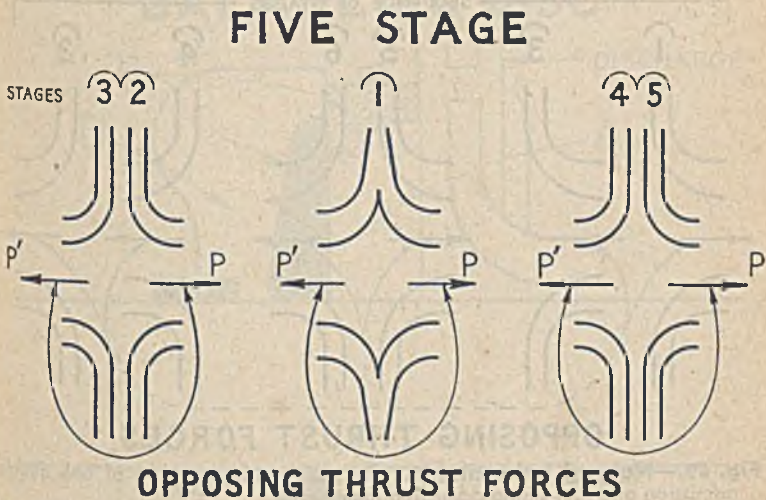


Fig. 21.—Natural balancing 4. Five stage with assembly of one double admission and two pairs of single admission opposing back to back impellers.

Fig. 23, will illustrate the conditions in a single admission, enclosed impeller pump.

Liquid from the case, being under pressure, leaks back through the clearance spaces A and D, past the sealing rings C and B, to the inlet. The impeller is usually cored in the rear shroud to permit the leakage accumulating, to pass on to the inlet without building up a pressure there,

There are, therefore, forces on the two shrouds equal to a pressure acting on the shroud areas.

Because of the fact that there is a difference in pressure intensity highest at the rim of the impeller, lowest at the sealing rings, this pressure is variable.

The pressure at the holes cored through the rear shroud is not quite the same as in the inlet chamber. Hence the forces on the two shrouds

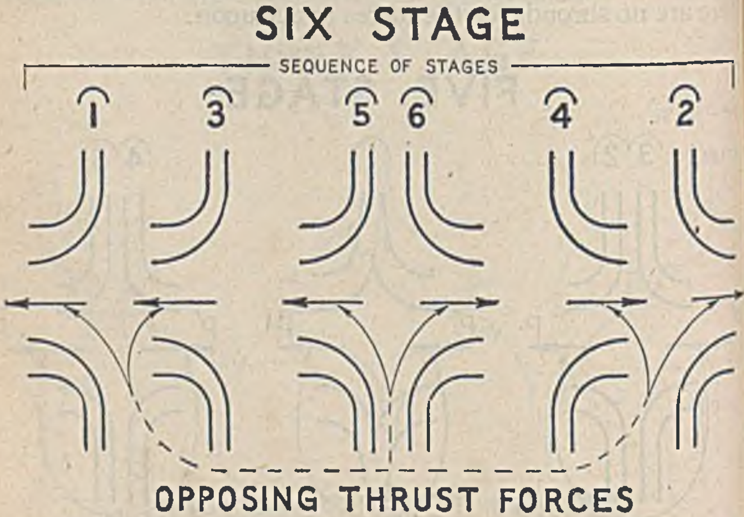


Fig. 22.—Natural balancing 5. Six stage with assembly of one double admission and four single admission impellers.

will be different. The resultant force is normally greater in clearance space D, than in A, so that the resultant thrust will be toward the inlet end of the pump.

As the clearances in the sealing rings increase due to normal wear in operation, these forces are changed and in different proportion, the end thrust increasing with wear. In order to balance this as nearly as possible in the design, engineers have changed the diameter at which the sealing ring B, is placed, increasing it to reduce the area upon which the high pressure liquid acts. Obviously, if this ring were moved out to the rim of the impeller, the force on the rear shroud could be so reduced as to change the resultant end thrust to the opposite direction, away from the admission.

In order to give the pump a minimum of end thrust over the length of its useful life, this ring is placed far enough out to reverse the thrust when the pump is new and has small clearances in the sealing rings. Then as these clearances increase with use, the thrust gradually goes down and finally reverses, toward the admission end. The proportions are so fixed that by the time this thrust becomes large enough to cause an undue load on the thrust bearing on the shaft, the leakage through the increased clearances is enough to seriously affect the pump efficiency. At this time,

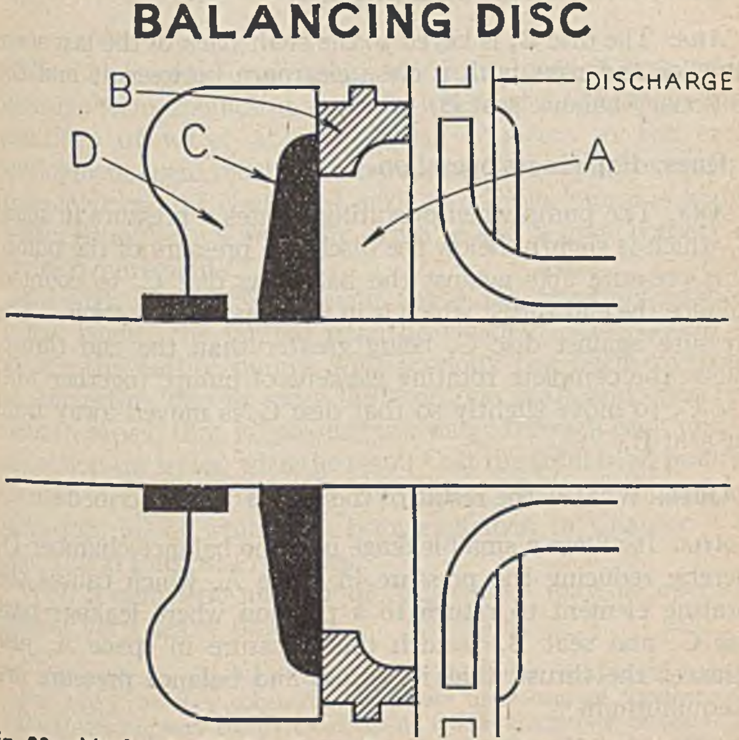


Fig. 23.—Mechanical balancing. Diagram of mechanical disc illustrating principles involved.

the clearance rings can be replaced at small cost, restoring the pump to nearly its new condition.

Balancing Disc Method.—As already explained, the thrust tendency is toward the admission side of the pump. The object of the balancing disc is to balance this thrust, by a counter pressure in the opposite direction, automatically maintained in proper proportions against the balancing disc.

Ques. Describe the balancing disc.

Ans. The disc **C**, is keyed to the shaft back of the last stage impeller and runs with a close clearance between it and the stationary balance seat B.

Ques. Explain its operation.

Ans. The pump when operating creates a pressure in space A, which is slightly below the discharge pressure of the pump. This pressure acts against the balancing disc C, to counterbalance the end thrust which is in the opposite direction. The pressure against disc C, being greater than the end thrust, causes the complete rotating element of pump, together with disc C, to move slightly so that disc C, is moved away from the seat B.

Ques. What is the result of the action just described?

Ans. It allows a small leakage into the balance chamber D, thereby reducing the pressure in space A, which causes the rotating element to return to a position where leakage past disc C, and seat B, is such that pressure in space A, just balances the thrust; that is, thrust and balance pressure are in equilibrium.

The leakage between disc C, and seat B, is very slight and not of sufficient amount to affect the efficiency or capacity of pump for a long time.

CHAPTER 5

Centrifugal Pumps; Construction

When the centrifugal pump was introduced it was in its primitive form, inefficient and only intended for pumping large quantities of water at low heads. Pioneers in the early development spent much time in making various improvements. In later years this continued until to-day the pump is highly developed and made in many types to suit a great variety of service requirements.

Although it was first thought the pump was only adapted to low heads, the limited head at which it was possible to operate the earlier pumps with econony has been overcome by connecting two or more units on one shaft and operating them in series, that is, passing the water through each unit in succession (*in series*) with the result that the total head pumped against is divided between the units. This is multi-stage operation, all of which has been explained in Chapter 4 on Centrifugal Pumps: Principles.

With a sufficient number of stages they may be operated against very heavy pressures.

Although the coupling of the separate units of a multi-stage assembly was bulky, as they consisted of separate units coupled together, now the stages are very compact—all within a separate case or casting. The centrifugal pump gives its best results when intelligently designed for each specific condition of operation.

Construction Details.—Centrifugal pumps are either sing
or double inlet. The preference in most cases is for the doub
inlet type because the end thrusts are equalized whatever vari

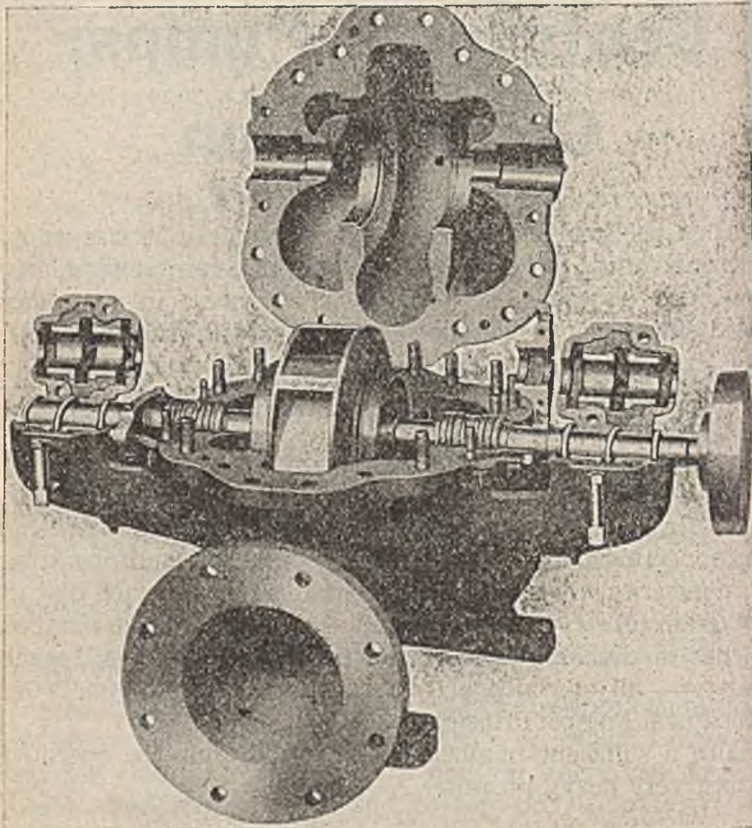


Fig. 1.—View of a split casing, single stage closed double inlet impeller.
Centrifugal pump "opened up" exposing the "works." In this construction
the entire rotating element may be removed without disturbing the pipe
connections or anything.

tions of pressure may occur on either discharge or inlet sides. Various details are given representing late practice.

The Casing.—This is frequently a two part casting split on a horizontal plane with inlet and discharge openings cast

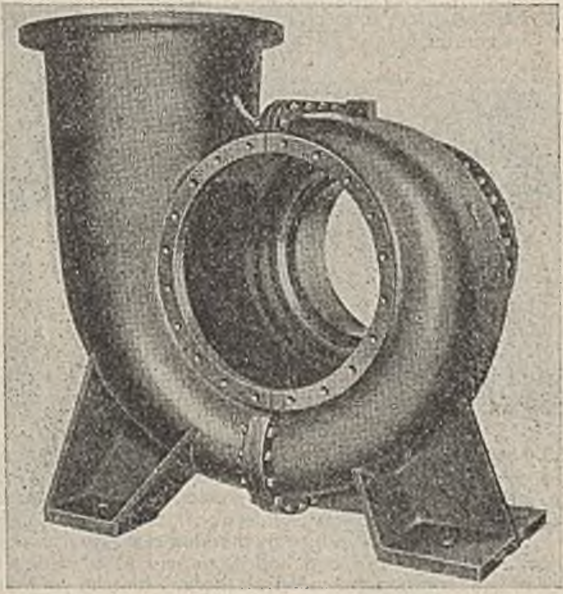


Fig. 2.—Volute casing as made for large pumps (36 in. illustrated here) is divided along the shaft center line. One casing half can thus be removed without disturbing suction or discharge connections.

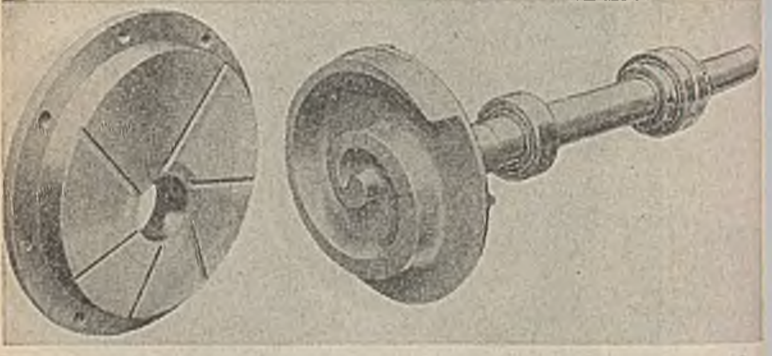
integral with the bottom half. This permits the removal of the top half or cover for inspection of the interior without disturbing piping connections.

An example of this type casing is shown in fig. 1 with the upper part raised. With the casing open various parts of the assembly are visible.

The pump here shown is a single stage double inlet type. As can be seen the bearings are ring oiling of bronze horizontally split. There are thrust collars at each bearing.

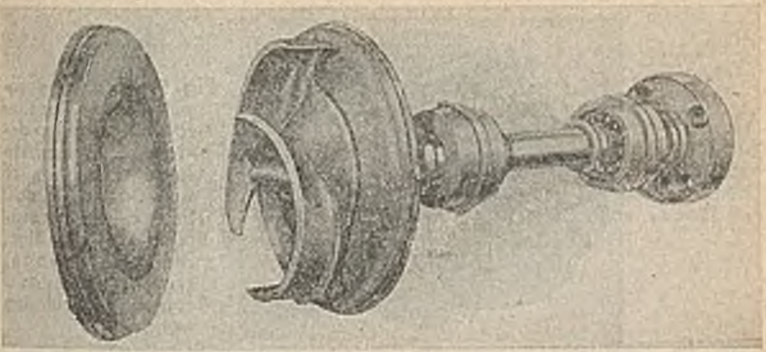
Another type casing is divided in a vertical plane along the shaft center line as in fig. 2.

The illustration shows casing without the side covers. The design is such that one casing half may be removed without disturbing the inlet or discharge connections.



Figs. 3 and 4.—Renewable inlet cover, cutting ring and single vane semi-open impeller for pumping of primary sludge. The single blade impeller rotating at 690 to 1,200 revolutions per minute, shears across the eccentric cutting ring once each revolution. Any solids about to enter the pump impeller are cut into very small particles by this shearing action. The shearing edge of the impeller is "Stellited" and hollow ground to a cutting edge. The impeller when handling abrasive materials that are an inherent part of primary sludge retains its sharpness. Hollow grinding is used in order that the cutting edge will be self-sharpening. Radial vanes protruding from and cast integrally with the back of the impeller shroud insure that no particles of rags, wood, or rubber will pack between the shroud and the stuffing box cover to stop the pump. In operation, these vanes produce a partial vacuum at the stuffing box.

Impellers.—The design of impellers varies greatly to meet the great variety of service conditions. The correct impeller for a given installation is of prime importance in order to secure economical and satisfactory operation.



Figs. 5 and 6.—Two vane open impeller with stationary side plates which seat on recesses in the pump casing and cover.

Fig. 7.—Two vane semi-open impeller with vanes designed especially for handling paper stock of high consistencies.

Fig. 8.—Three vane semi-open non-clogging type impeller. The enclosed side consists of a disc extending from the hub, and with a projection on the back, serving as a wearing ring. The water channels formed by heavy vanes and disc open toward the inlet side. Relief holes through the disc, between wearing ring and hub, lead to the inlet eye of the impeller and equalize the greater part of end thrust.

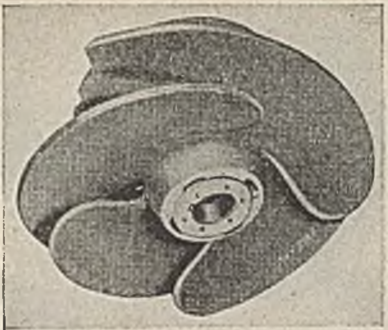
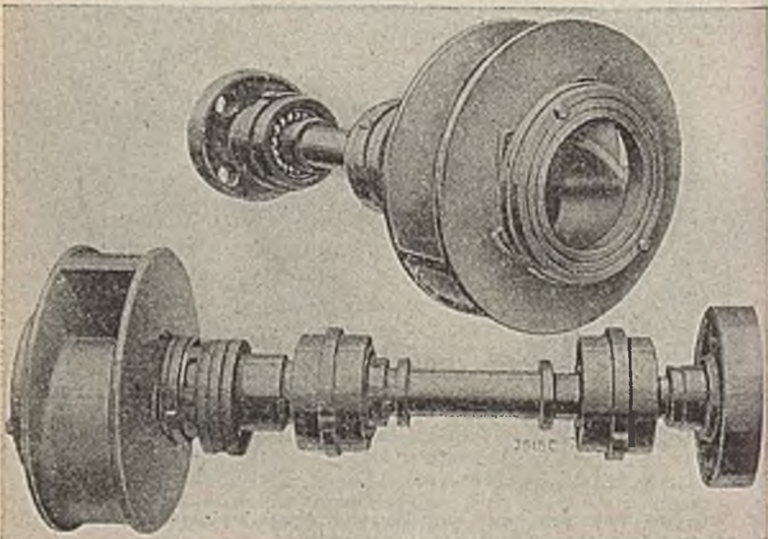


Fig. 9.—Four vane propeller type impelle[r] for vertical pump designed to handle large quantities o[f] water at low heads. Such as are encountered in irrigation, drainage, sewage, storing water and condenser circulating work.



Figs. 10 and 11.—Closed impeller with repelling vanes for pumping sewage. The vanes are cast integral on both sides, and are designed to: 1, prevent packing of fibrous materials between stationary covers and rotating impeller; 2, minimize circulation from discharge side of impeller to inlet side and to packing box side; 3, minimize wear at points of running clearance between stationary covers and rotating impeller; 4, reduce internal pressure on packing box, thus making outside seal to packing box more effective.

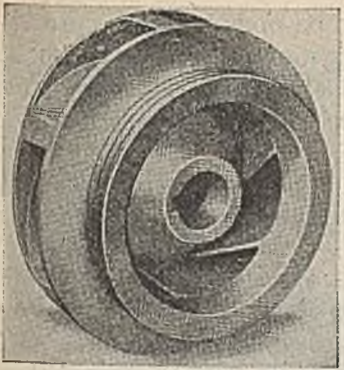
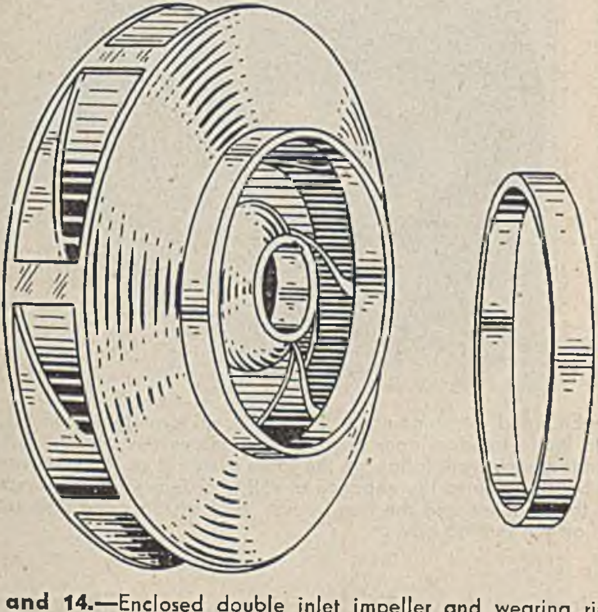


Fig. 12.—Closed double inlet impeller. In machining, the impeller is finished smooth all over, the exterior surfaces being machine finished and polished, the interior being made smooth by chipping and filing.



Figs. 13 and 14.—Enclosed double inlet impeller and wearing ring. The impeller is cast in one piece of bronze except in cases of pumping liquids requiring special metals, such as chrome, monel, nickel or other suitable alloys.

A high degree of efficiency can be obtained with open im
pellers under certain conditions by carefully proportioning th
curvature of the blades and by reducing the side clearances to
minimum with accurate machining of impeller edges and sid
plates.

In general, however, more efficient results can be obtaine
by the use of enclosed or shrouded impellers. Various type

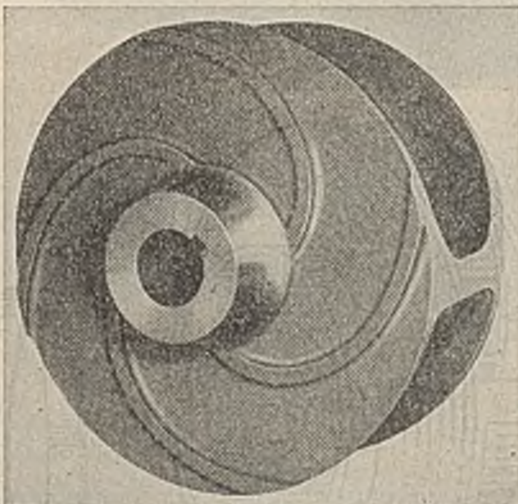


Fig. 15.—Enclosed non-binding type impeller accurately machined and bal anced to insure freedom from vibration. As shown the impeller has pumpin vanes on the back wall, following the same curvative as the main vanes. Thes prevent binding due to the entrance of solids or sem-solids between the bac wall of the impeller and the head cover, and at the same time reduces th pressure on the stuffing box.

of impellers are shown in the accompanying illustrations figs 3 to 14. Their characteristics and adaptation being explained in the text under the illustrations.

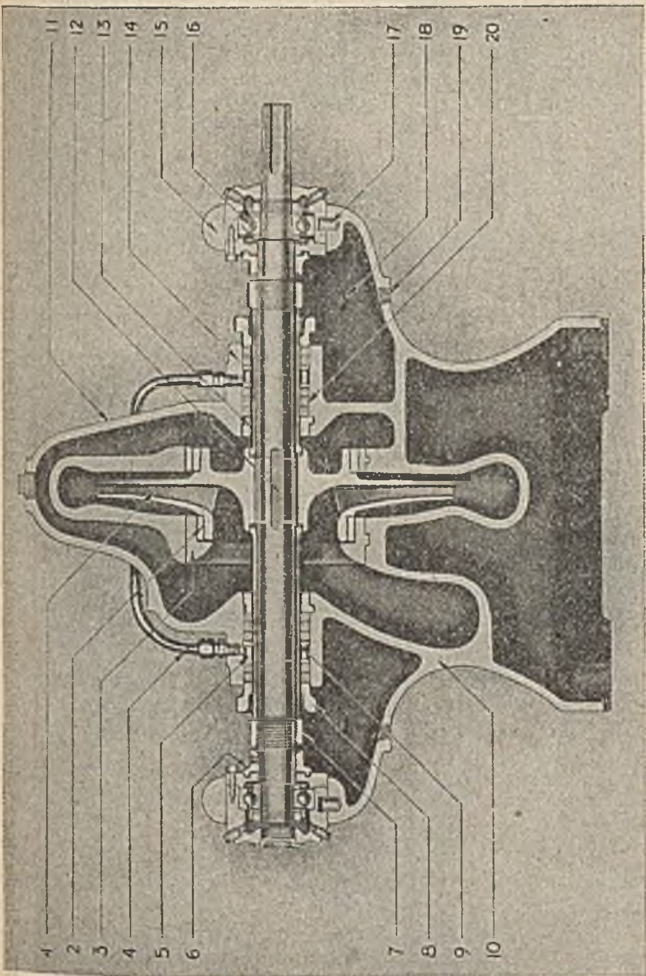


Fig. 16.—Split case, single inlet closed impeller. Sectional view showing parts which are as follows: 1, impeller; 2, impeller wearing rings; 3, casing wearing rings; 4, water seal piping; 5, brass lantern ring; 6, water deflector; 7, bronze sleeve nut; 8, bronze gland; 9, bronze shaft sleeve; 10, lower half casing; 11, hydraulic balance crossover; 12, impeller key; 13, packing retaining ring; 14, deep stuffing box; 15, bearing housing cover; 16, deep groove cartridge type ball bearing; 17, bearing down pin; 18, overflow drain hole; 19, tapped drain hole; 20, packing rings.

SINGLE STAGE—*Double Inlet*

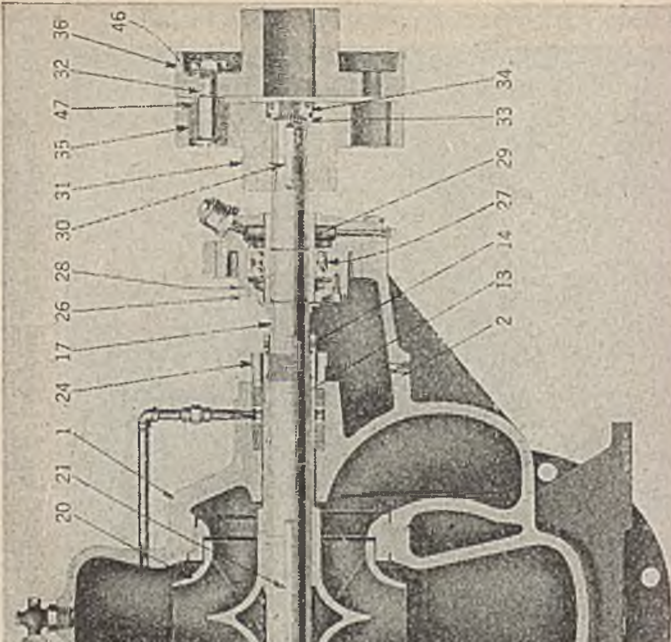


Fig. 17.—Single stage volute pump with parts numbered as listed on page 143

LIST OF PARTS

1. Upper Half of Casing	13. Gland Nut & Bolt	28. Line Bearing Housing
2. Lower Half of Casing	14. Shaft Sleeve Nut	29. Line Bearing Collar
3. Nameplate Cover	15. Shaft Sleeve	30. Coupling Key
4. Bearing Lock Nut	16. Packing	31. Driven Half Coupling
5. Bearing Lock Nut Washer	17. Shaft	32. Coupling Pin & Nut
6. Thrust Bearing Housing	18. Impeller Guide Ring	33. Coupling Lock Washer
7. Grease Cup	20. Impeller	34. Coupling Lock Nut
8. Bearing Strap	21. Impeller Key	35. Rubber Bushing
9. Thrust Bearing	23. Water Seal Cage	36. Driving Half Coupling
10. Thrust Bearing Cover	24. Gland	47. Coupling Pin Collar
11. Thrust Bearing Spacer	26. Line Bearing Cover	48. Water Seal Piping
12. Water Shield	27. Line Bearing	

Bearings and Housings.—Most pumps are fitted with ball bearings. A typical construction consists of a single row deep groove ball bearing of ample size to withstand all axial and radial loads. The bearing housings are of the cartridge type in order that the entire rotating element may be removed from the pump without disturbing the alignment or exposing the bearings to water or dirt.

The bearing housings are positioned by means of dowel pins in the lower half of the casing and are securely clamped by covers split on the same horizontal plane as the pump casing. In this construction the entire bearing may be removed from the shaft without damage by using the sleeve nut as a puller.

It should be noted that single ball bearings are the exception, most pumps being provided with double ball bearings.

Shaft Assembly.—Referring to fig. 18, the shaft is accurately key seated at center and on the driven to receive the impeller and pin and rubber bushing flexible coupling.

The shaft is protected against corrosion or abrasive action of the liquid pumped by means of centrifugally cast bronze shaft sleeves which fit against the impeller hub and are sealed by a thin gasket.

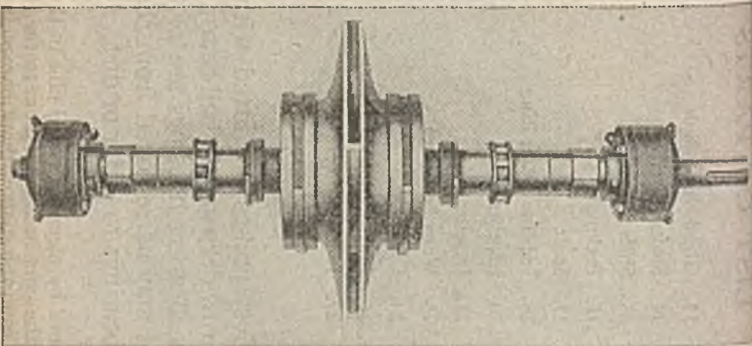


Fig. 18.—Complete rotating element with double inlet closed impeller, sealed ball bearings intact.

The shaft sleeves are locked on the shaft by means of a heavy sleeve nut which is threaded to the shaft outside the stuffing box, and held in place by a special hollow head set screw.

This unit is so designed and placed that in addition to locating the shaft sleeve, it forms a bearing puller whereby the ball bearings may be removed from the shaft in dismantling the pump.

The complete rotating element with sealed ball bearings is plainly shown in the above illustration.

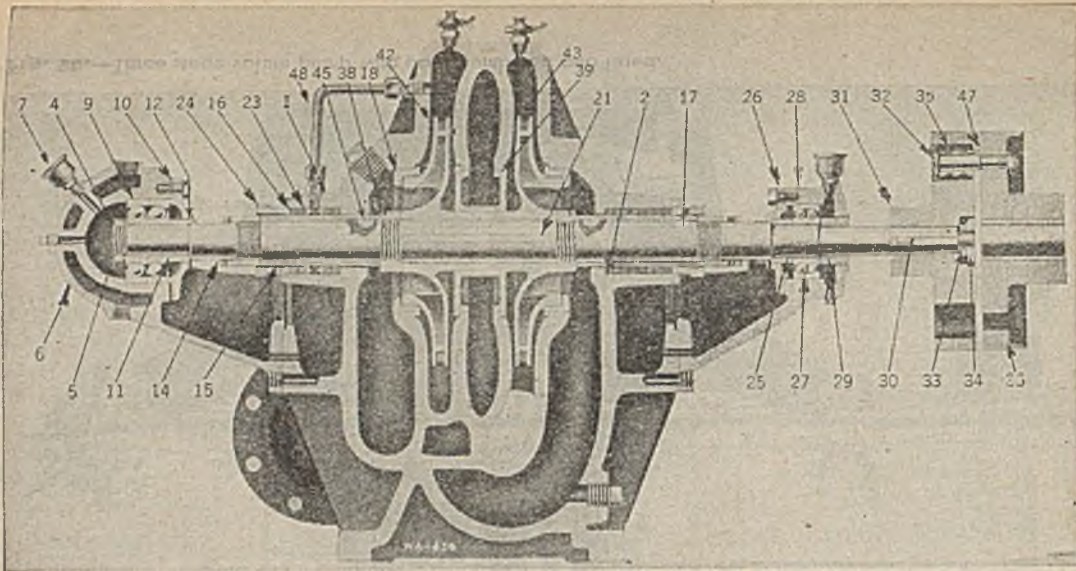


Fig. 19.—Two stage volute pump with parts numbered and listed.

1. Upper Half of Casing
2. Lower Half of Casing
4. Bearing Lock Nut
5. Bearing Lock Nut Washer
6. Thrust Bearing Housing
7. Grease Cup
9. Thrust Bearing
10. Thrust Bearing Cover
11. Thrust Bearing Spacer
12. Water Shield
14. Shaft Sleeve Nut
15. Shaft Sleeve
16. Packing
17. Shaft
18. Impeller Guide Ring
21. Impeller Key
23. Water Seal Cage
24. Gland
25. Line Bearing Spacer
26. Line Bearing Cover
27. Line Bearing
28. Line Bearing Housing
29. Line Bearing Collar
30. Coupling Key
31. Driven Half Coupling
32. Coupling Pin and Nut
33. Coupling Washer
34. Coupling Nut
35. Rubber Bushing
36. Driver Half Coupling
38. Impeller Nut
39. Stage Piece
42. Impeller 1st Stage
43. Impeller 2nd Stage
45. Woodruff Key
46. Coupling Pin Lock Washer
47. Coupling Pin Collar
48. Water Seal Tube.

THREE STAGE—*Single Inlet*

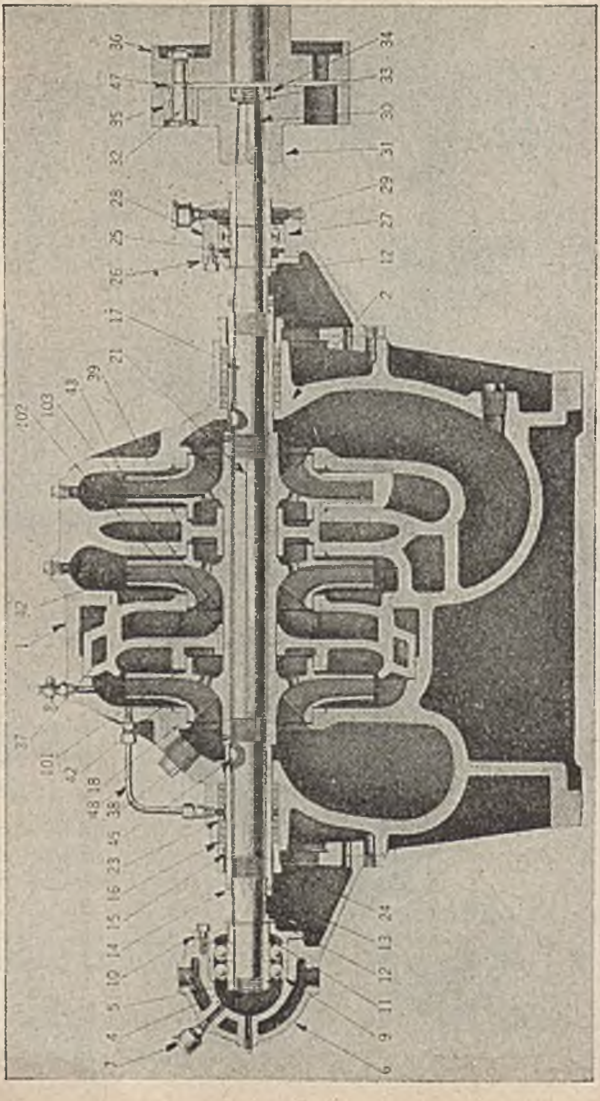


Fig. 20.—Three stage volute pump with parts numbered and listed.

LIST OF PARTS

1.	Upper Half of Casing	17.	Shaft	34.	Coupling Lock Nut
2.	Lower Half of Casing	18.	Impeller Guide Ring	35.	Rubber Bushing
4.	Bearing Lock Nut	21.	Impeller Key	36.	Driving Half Coupling
5.	Bearing Lock Nut Washer	23.	Water Seal Cage	37.	Return Channel Diffuser
6.	Thrust Bearing Housing	24.	Gland	38.	Impeller Nut
7.	Grease Cup	25.	Line Bearing Spacer	39.	Stage Piece
9.	Thrust Bearing	26.	Line Bearing Cover	42.	Impeller 1st & 2nd Stage
10.	Thrust Bearing Cover	27.	Line Bearing	43.	Impeller 3rd Stage
11.	Thrust Bearing Spacer	28.	Line Bearing Housing	45.	Woodruff Key
12.	Water Shield	29.	Line Bearing Collar	47.	Coupling Pin Collar
13.	Gland Nut & Bolt	30.	Coupling Key	48.	Water Seal Tube
14.	Shaft Sleeve Nut	31.	Driven Half of Coupling	101.	Impeller Balancing Ring
15.	Shaft Sleeve	32.	Coupling Pin & Nut	102.	Return Channel Bushing
16.	Packing	33.	Coupling Lock Washer	103.	Impeller Spacer

Water-seal rings are provided with piping either from the pump volute or for outside fresh water connections as preferred. The piping 4, is seen connecting the two water seals.

Single and Multi-Stage Pumps.—The accompanying illustrations, figs. 15 to 18, have all the parts numbered so that by reference to the list the name of any part is obtained.

Wearing Rings.—Removable casing and impeller rings are built into the pump. With respect to material of this construction, casing rings are cast iron and have streamline casing. They are held in place by a semi-circular tongue fitting into a groove in the lower half casing.

Impeller rings are of hard bronze, machined flat and pressed onto the impeller where they are locked in place by threaded keys. Clearances between these rings are such as to insure against excessive leakage and to reduce wear to a minimum.

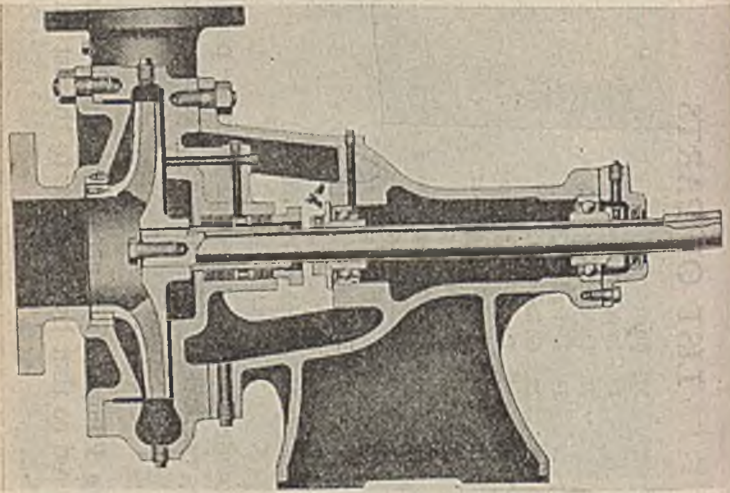


Fig. 21.—Single admission single stage enclosed impeller pump. Sectional view showing stuffing box, ball bearings, wearing rings, etc.

Stuffing Boxes and Glands.—In fig. 17, 24, is one of the glands, 16, packing of one of the stuffing boxes. The assembly of gland and box is usually known as a stuffing box. In the construction shown in fig. 17, the glands are made of hard bronze and are split. The two halves are held together by means of bronze clamps. The glands are splash-proof and so designed that all drippings fall into the drain pocket.

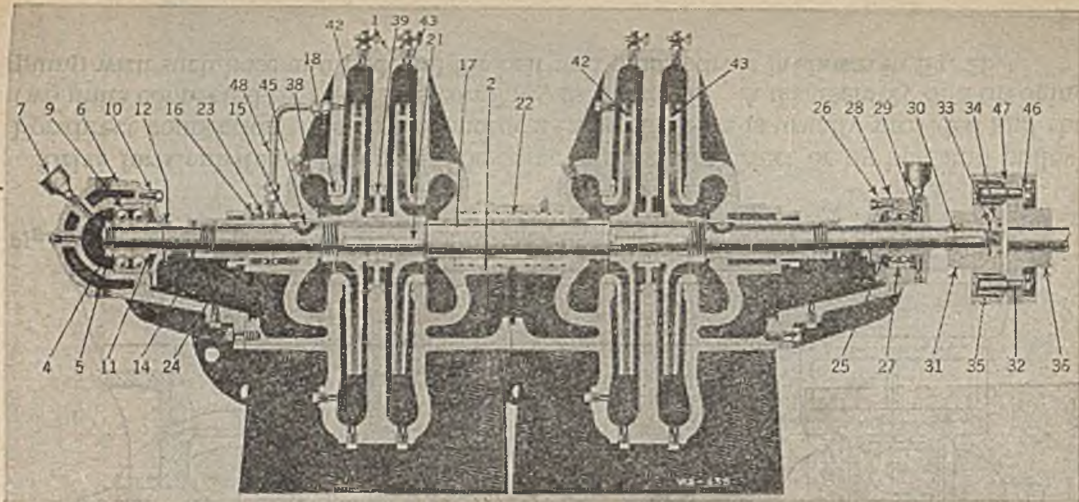


Fig. 22.—Four stage volute pump with parts numbered and listed.

1. Casing Upper Half
2. Casing Lower Half
4. Bearing Lock Nut
5. Bearing Lock Washer
6. Thrust Bearing Housing
7. Grease Cup
9. Thrust Bearing
10. Thrust Bearing Cover
11. Thrust Bearing Spacer
12. Water Shield
14. Shaft Sleeve Nut
15. Shaft Sleeve
16. Packing
17. Shaft
18. Impeller Guide Ring
21. Impeller Key
22. Intermediate Stuffing Box Bushing
23. Water Seal Cage
24. Gland
25. Line Bearing Spacer
26. Line Bearing Cover
27. Line Bearing
28. Line Bearing Housing
29. Line Bearing Collar
30. Coupling Key
31. Coupling Driven Half
32. Coupling Pin and Nut
33. Coupling Lock Washer
34. Coupling Lock Nut
35. Rubber Bushing
36. Driver Half Coupling
38. Impeller Nut
39. Stage Piece
42. Impeller 1st and 3rd Stages
43. Impeller 2nd and 4th Stages
45. Woodruff Key
46. Coupling Pin Lock Washer
47. Coupling Pin Collar
48. Water Seal Tube

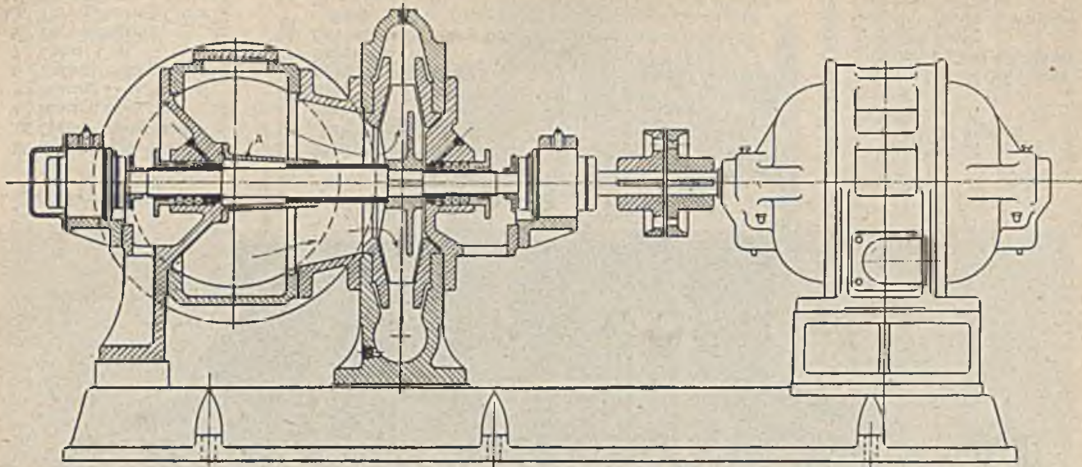


Fig. 23.—Centrifugal pump direct connected to motor, both units being mounted on a large sub-base.

Drive for Centrifugal Pumps.—The drive is generally direct or by belt and pulley. For direct connection to engine or motor a large sub-base is usually provided and the two units connected by a suitable coupling, as in fig. 23. A disassembly of a dredging pump with such base and pedestal for belt and pulley drive is shown in fig. 25.

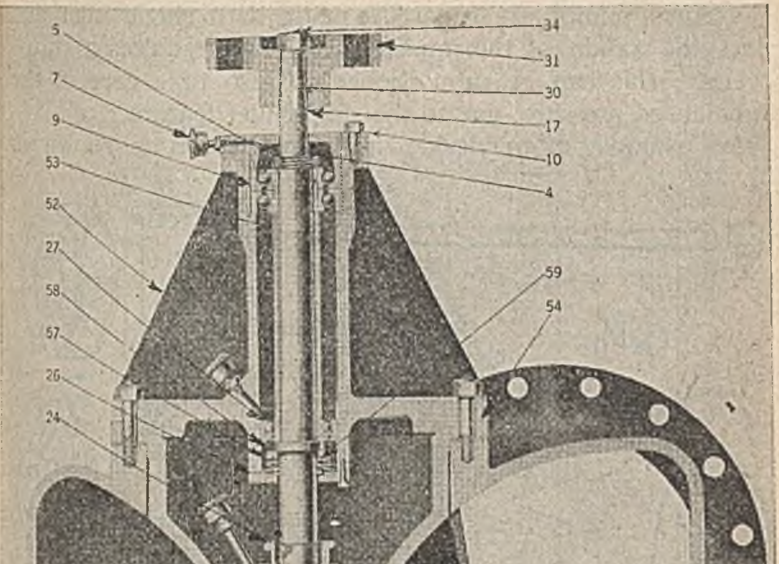


Fig. 24.—Vertical single stage volute pump with parts numbered and list which is as follows: 1, casing; 4, bearing lock nut; 5, bearing lock washer; 7, grease cup; 9, thrust bearing; 10, thrust bearing cover; 15, shaft sleeve; 16, packing; 17, shaft; 20, impeller; 21, impeller key; 23, sealing cage; 24, gland; 26, line bearing cover; 27, line bearing; 30, coupling key; 31, driven

Vertical pumps are generally made with single inlet and since the weight of the impeller and shaft requires a thrust bearing, this can be proportioned to also take care of the unbalanced pressure, due to the single inlet feature.

An example of vertical pump construction is shown in fig. 24.

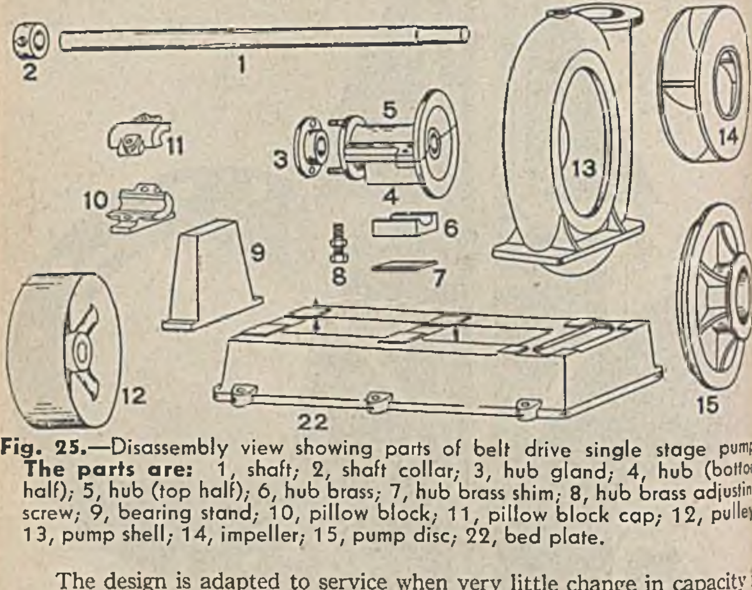


Fig. 25.—Disassembly view showing parts of belt drive single stage pump **The parts are:** 1, shaft; 2, shaft collar; 3, hub gland; 4, hub (bottom half); 5, hub (top half); 6, hub brass; 7, hub brass shim; 8, hub brass adjusting screw; 9, bearing stand; 10, pillow block; 11, pillow block cap; 12, pulley; 13, pump shell; 14, impeller; 15, pump disc; 22, bed plate.

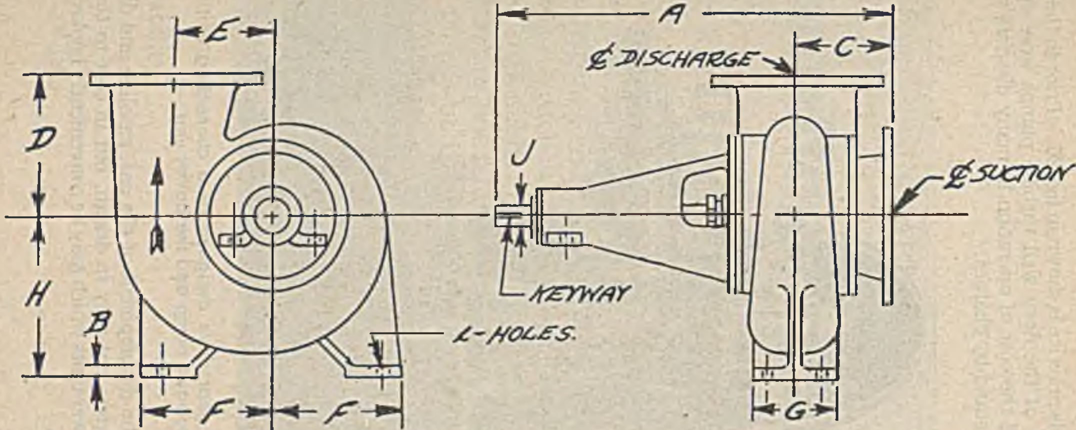
The design is adapted to service when very little change in capacity is designed for varying heads, this feature being possible as a result of the steepness of the characteristic curve.

Gearing is seldom advisable except for vertical pumps where bevel gears may be used for transmitting the necessary power from a horizontal to a vertical shaft.

Fig. 24.—*Text continued.*

half coupling; 34, coupling nut; 38, impeller nut; 51, suction head; 52, bearing bracket; 53, bearing spacer sleeve; 54, discharge head; 55, suction guide ring; 56, impeller bushing ring; 57, sylphon; 58, sylphon ring; 59, sylphon spring.

PRINCIPAL DIMENSIONS



Pump Size	A	B	C	D	E	F	G	H	J	Keyway	L
10"	4'-1½"	1⅛"	16½"	14"	10½"	14"	9"	17½"	2¼"	⅝"×5/16"×4"	4-1⅛"
12"	4'-2"	1⅛"	17"	15"	10¾"	15"	13½"	18"	2¼"	⅝"×5/16"×4"	4-1⅝"
16"	5'-6¼"	1¼"	24"	20"	14½"	20"	18"	24"	3"	¾"×¼"×4½"	4-1⅜"

Figs. 26 and 27.—Dimension drawings of horizontal angle flow centrifugal pump with tabulated dimensions.

An example of this practice is shown in fig. 26. Prior to the introducti[on] of this drive, most of the deep well turbine pumps were operated w[ith] electric motors, and the cost of operation in many districts was excessi[ve] on account of fixed stand-by charges.

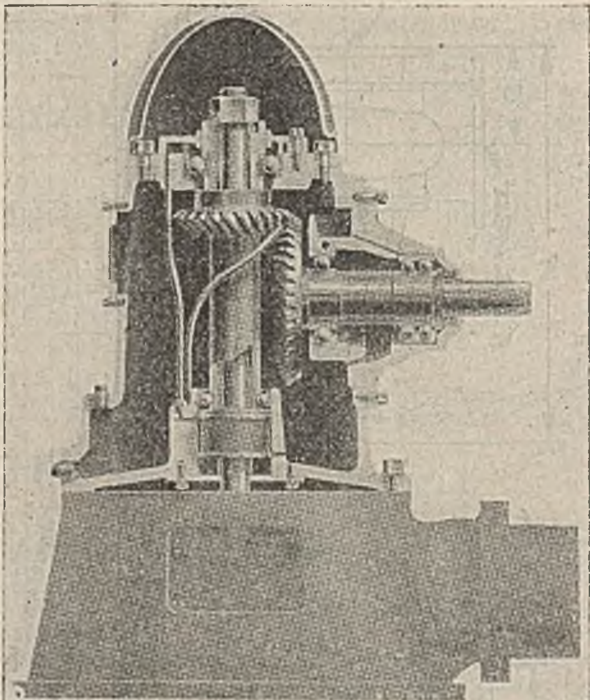


Fig. 28.—Right angle gear drive designed as a connecting unit between the deep well turbine or sewage pump and the power unit.

In recent years the development of the gas, gasoline and diesel engines has made rapid progress not only in design, but in cost to the consumer and these various power units which may be conveniently hooked up through the angle gear drive.

CHAPTER 6

Centrifugal Pumps; Operation

Assuming that the proper pump has been selected for the service requirements, and correctly installed and if a few simple operating rules be followed the pump will give good service and will be reasonably trouble free. There are still a few engineers who are not entirely familiar with the operation of centrifugal pumps and the information given in this chapter will be found helpful.

Location

The several important points in selecting the location for the pump are:

1. Accessibility
2. Light
3. Height of lift
4. Piping

The pump should be located where it is easily accessible; that is, room all around the pump and where there is enough light that the packing and bearings can be readily inspected.

A centrifugal pump does not require much attention, but if it be not conventiently accessible it probably will receive no attention until something happens resulting in a break-down requiring major repairs.

Ques. What is the practical static lift limit of a centrifugal pump pumping cold water?

Ans. About 10 feet.

Pipe friction, foot valve and strainer losses may amount to an additional 4 or 5 ft. If the water be hot the inlet lift must be reduced. For example, water at 212° Fahr. must flow to the pump under a head in amount depending on many conditions involved in the specific installation.

Temperature and height of the pump above sea level affects the lift.

Liquids heavier than water can be lifted but shorter distance than water depending upon their density.

Ques. Mention precautions to be taken in laying out an new installation.

Ans. The elevation of the pump with respect to the level of the liquid to be pumped should be such that the dynamic lift will be within practical limit.

Ques. How should the pump be located with respect to the piping?

Ans. The location should be such that the piping layout will be as simple as possible.

Foundation

The foundation may consist of any structure sufficiently heavy to afford permanent rigid support to the full base area of the bed plate and to absorb any normal amount of strains and shocks that may be encountered in service.

Concrete foundations built up from solid ground are the most satisfactory.

In building the foundation make allowance for grouting between rough surface of concrete and underside of base.

Foundation bolts of the specified size should be located according to drawings submitted prior to shipment of unit and each bolt should be surrounded by a pipe sleeve three or four diameters larger than the bolt.

After the concrete is poured, the pipe is held solidly in place while the bolts may be moved around to conform to the holes in the bed plate.

Ques. How should a unit be placed when mounted on steel work or other structure?

Ans. It should be placed directly over, or as near as possible to the main members, beams and walls and be so supported that the base plate cannot be distorted and alignment disturbed by any yielding or springing of the structure or the base plate.

Ques. What allowance is made for grouting?

Ans. The bottom of the bed plate should be about ¾ in. above the top of the foundation to allow room for grouting.

Installation

Leveling.—Pumps are generally shipped mounted, and it is usually unnecessary with units of moderate size to remove the pump or driver from its base plate when leveling. The unit should be placed over the foundation supported by short strips of steel plate and wedges close to the foundation bolts, allowing for grouting from ¾ to 2 ins. space between the bottom of the base plate and the top of the foundation.

Remove coupling bolts before proceeding with leveling of unit and alignment of coupling halves.

Employing a small spirit level, the projecting edges of pads supporting pump and motor feet when scraped clean can be used for leveling the base plate. Where possible, it is preferable to place the level on some exposed part of the pump shaft, sleeve or planed surface of casing.

Adjust the wedges under base plate till pump shaft is level and flanges of suction and discharge nozzles, vertical or horizontal as required, at same time observe that the pump is at the specified height and location.

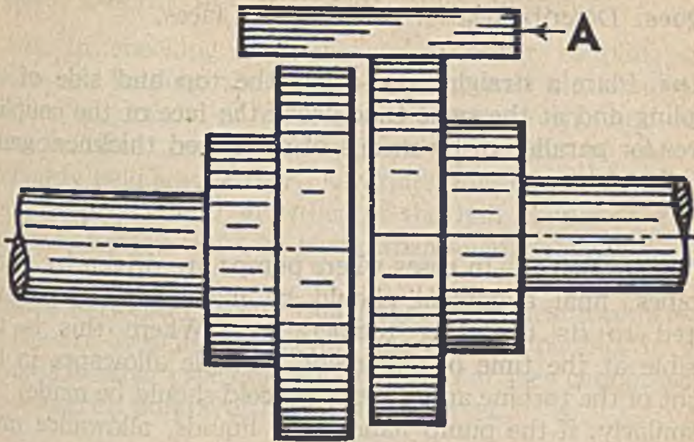
While proceeding with the leveling of pump and base, maintain at the same time accurate alignment of the unbolted coupling halves between pump and driver shafts.

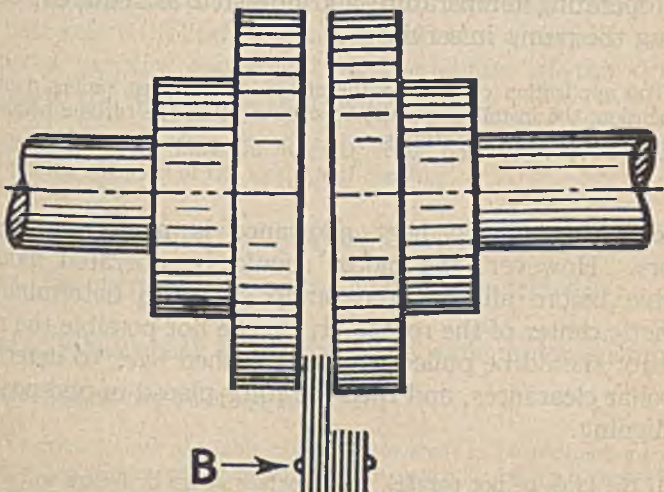
Ques. Describe the procedure followed when checking driver and driven shaft alignment.

Ans. Place a straight edge across the top and side of the coupling, and at the same time check the faces of the coupling halves for parallelism by means of a tapered thickness gauge or feeler gauges, as shown in figs. 1 and 2.

Ques. When does exact alignment exist of coupling halves which are true circles, of same diameter and the faces flat?

Ans. Exact alignment exists when the distance between the faces is the same at all points and a straight-edge will lie squarely across the rims at any point.





Figs. 1 and 2.—Alignment of pump and driver shafts with aid of straight edge and feelers at coupling.

Ques. Describe test for parallelism of faces.

Ans. Place a straight edge across the top and side of the coupling and at the same time check the face of the coupling halves for parallelism by means of a tapered thickness gauge or feeler gauges.

Turbine Drive.—In cases where pumps are driven by steam turbines, final alignment should be made with the driver heated to its operating temperature. Where this is not possible at the time of alignment, suitable allowance in the height of the turbine and shaft when cold should be made.

Similarly, if the pump handle hot liquids, allowance must be made for the shaft being elevated when the pump expands. In any case the alignment should be checked when the unit is at operating temperature and adjusted as required, before placing the pump in service.

The application of heat to the steam and exhaust piping results in expansion; the installation must be so made that the turbine nozzles are not subjected to piping strains.

Motor Drive.—No heat allowance is made for electric motors. However, the motor should be operated alone if possible before aligning the pump so as to determine the magnetic center of the rotor. If this be not possible the rotor of motor should be pulled over and pushed back to determine the collar clearances, and then the rotor placed in mid-position for aligning.

If the faces be not parallel, the thickness gauge or feelers will show a variation at different points. If one coupling be higher than the other, the amount may be determined by the straight-edge and feeler gauges.

Ques. Describe test for manufacturers' tolerances.

Ans. In checking the trueness of either coupling half, revolve it, holding the other coupling half stationary and checking alignment at each quarter turn of the half being rotated. Next revolve and check alignment of the half previously held stationary. A variation within manufacturing limits may be found in either of the half couplings; proper allowance for this must be made when aligning the unit.

Space Between Faces of Couplings.—The clearance between the faces of couplings of the pin and buffer type and the ends of shafts in other types should be set so that they cannot touch, rub or exert a pull on either pump or driver.

Ques. How does the amount of this clearance vary?

Ans. It may vary with the size and type of coupling used. The best rule to follow is to allow sufficient clearance for unhampered endwise movement of the shafts of the driving element to the limit of its bearing clearance.

On motor driven units, the magnetic center of the motor will determine the running position of the motor half coupling.

Ques. How should this position be checked?

Ans. By operating the motor while disconnected.

Ques. What check should be made while motor is running?

Ans. Check direction of rotation.

If current be not available, move motor shaft in both directions as far as bearings will permit then adjust shaft centrally between these limits, thereafter assembling the unit with the correct gap between coupling halves.

Ques. What is the next operation after the unit has been accurately leveled and aligned?

Ans. Grouting.*

Grouting.—By definition grouting is: *The process of pouring a mixture of cement, sand and water into the* **voids** *of stone, brick or concrete work either to give a solid bearing or to fasten anchor bolts, dowels, etc.*

Ques. What is the usual mixture for grouting?

Ans. One part pure cement, two parts sand and enough water to cause the mixture to flow freely under the bed plate.

Ques. Describe the operation of grouting.

Ans. A wooden form is built around the outside of the bed plate to contain the grout and provide sufficient head to assure a flow of the mixture under the entire bed plate.

Ques. How long should the grout be allowed to set?

Ans. Two days (48 hours).

Ques. What is done when the grout becomes hard?

Ans. The holding down bolts should be finally tightened and the halves of coupling re-checked.

Inlet Piping.—In a new installation it is advisable to flush inlet pipe with clear water before connecting same to pump.

---

*NOTE.—The bed plate is ordinarily grouted before the piping connections are made. However, in some special cases, the reverse procedure is permissible.

Ques. What should be noted with respect to the inlet line?

Ans. With the exception of misalignment, most troubles with individual centrifugal pump installations can be traced to faults in the inlet lines.

Accordingly the correct installation technique with respect to the inlet piping is important.

Ques. What should be noted as to size of the inlet piping?

Ans. The inlet pipe should not be smaller than the inlet opening of the pump.

It should be as short and direct as possible. In cases where a long inlet line cannot be avoided, the size of the piping should be increased. Air pockets or high spots in a pump inlet line will invariably cause trouble. The inlet piping may be level, preferably there should be a continual rise without high spots from the source of supply to the pump.

Ques. How deep should the end of the inlet be submerged when the supply liquid is at its lowest level?

Ans. Large pipes are usually submerged four times their diameter while small pipes require from two to three feet submergence.

Ques. After installing inlet piping what should be done?

Ans. It should be blanked off and hydrostatically tested for air leaks before starting up.

Ques. What fitting should be attached to the end of the inlet pipe and why?

Ans. A strainer to prevent lodgment of foreign material in the impellers.

Ques. What should be the net area of the strainer?

Ans. Three to 4 times the area of the inlet pipe.

Ques. What is understood by the net area of a strainer?

Ans. The clear and free opening through the strainer.

If the strainer be likely to become frequently clogged, an accessible place should be selected for the inlet pipe. For large pumps, screens should be placed at the entrance to the inlet well.

Ques. What else besides a strainer should be attached to the end of the inlet pipe and why?

Ans. A foot valve for convenience in priming or where the pump is subject to intermittent service.

Care should be used in the selection of the size and type of foot valve in order to avoid excessive friction loss through the valve.

Do not under any circumstances use an ordinary swing check valve as a foot valve.

Discharge Piping.—The discharge piping like the inlet piping should be as short and free of elbows as possible to reduce friction. A check and gate valve should be placed close to the pump.

Ques. What is the purpose of the check valve?

Ans. To protect the pump casing from breakage due to water hammer.

Ques. What is the idea of the gate valve?

Ans. To shut off the pump from the discharge piping in case of inspection or repairs.

Ques. Mention another function of the check valve.

Ans. On pumps having no foot valve it prevents the pump running backward if the driver should fail to operate.

Ques. After installing the inlet and discharge piping what should be done?

Ans. The alignment should be carefully re-checked.

Pumps Handling Hot Liquids.—Special types of multi-stage pumps for handling very hot liquids are constructed with a key and keyway on lower half casing feet and base. One end of the pump is securely bolted while the other on certain units is bolted with spring washers under the nuts on casing feet, allowing one end to move laterally when casing expands. Standard designs are dowelled at the inboard end, certain special hot liquid multi-stage pumps are dowelled at the thrust bearing end either in the ordinary manner or with the dowels crosswise; dowels at the other end, if used, are fitted in a similar manner to the key and keyway, *i.e.* parallel to pump shaft, to allow the casing to expand when heated.

Ques. What test should be made when handling hot liquids?

Ans. The nozzle flanges after the unit has been in service should be disconnected to check in which direction the expansion of the piping is acting, correct for the effect of the strains as required.

Jacket Piping.—All multi-stage pumps employ jacketed or separately cooled thrust bearings. When the pump handles hot liquid make certain that independent jacket or oil cooler water piping is connected.

Ques. What control should be provided?

Ans. For the purpose of observing whether the water be flowing, and for the regulation of the amount, it is good practice to pipe the discharge from jacket or cooler so that it will have a visible flow into a funnel connected to a drain.

Drain Piping.—All drain and drip connections should be piped to a point where leakage can be disposed of.

Operation

Before starting a centrifugal pump there are a few preliminaries that should be done.

Ques. What should be done first?

Ans. Test the driver for direction of rotation with the coupling halves disconnected. The arrow on pump casing is correct for rotation.

Ques. What attention should the ball bearing receive?

Ans. Supply ball bearings moderately with a good grade of acid free lubricating grease.

Use the grade lubricant recommended by the manufacturer of the pump.

Ques. How much oil should be placed in the oil lubricated bearings?

Ans. Fill level with the overflow.

Ques. What preliminary attention should be given to babbitted bearings?

Ans. Flush thoroughly with kerosene, and fill to operating level with a high grade lubricating oil.

The operating level is generally controlled by an overflow.

Ques. What piping should receive attention?

Ans. The cooling water piping to thrust bearing housing.

Do not use cooling water on bearings which are warm to the hand only; use only sufficient water to keep the lubricant at a safe working temperature.

Ques. What periodic attention should be given?

Ans. Flush the water supply freely to remove particles of scale, etc., which might entirely stop the flow on a throttled valve.

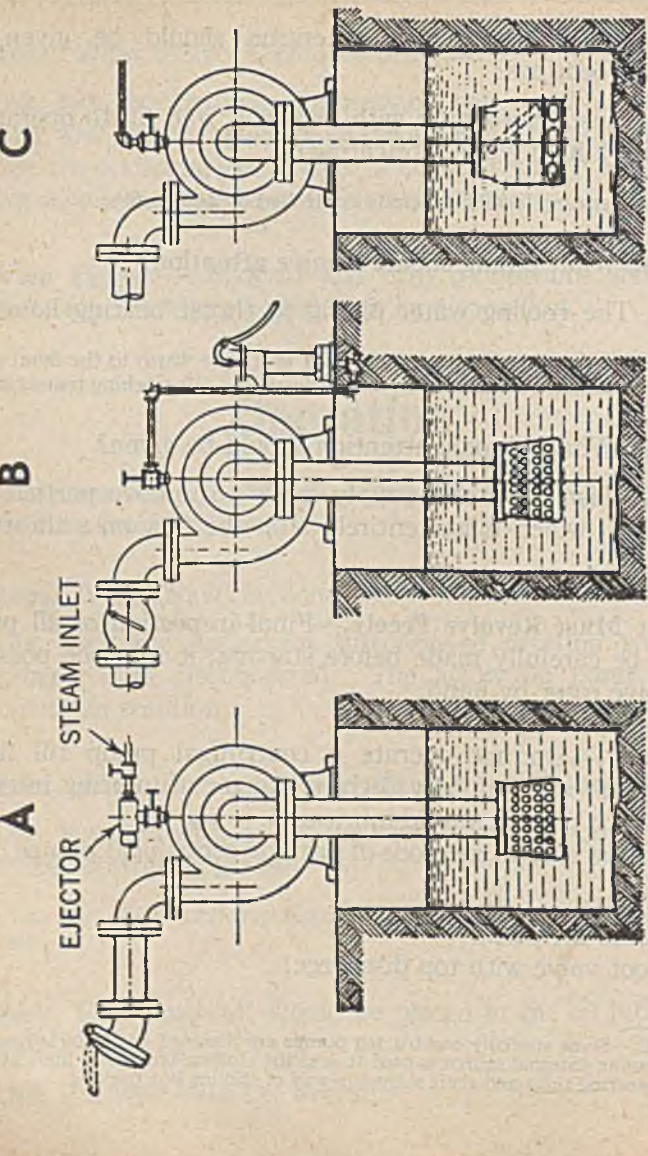
Rotor Must Revolve Freely.—Final inspection of all parts should be carefully made before starting; it must be possible to revolve rotor by hand.

Priming.—Do not operate a centrifugal pump till filled with water* as if run dry there is danger of injuring internal parts which must be liquid lubricated.

There are several methods of priming centrifugal pumps, viz:

1. Ejector
2. Hand lift pump
3. Foot valve with top discharge.

*NOTE.—Some specially constructed pumps are designed to be started empty; liquid from an external source is used to seal the stuffing boxes and lubricate the impeller wearing rings and shaft sleeves in way of stuffing box packing.



Figs. 3 to 5.—Methods of priming centrifugal pumps. A, by ejector; B, by ordinary kitchen lift pump; C, by foot valve with top discharge.

Priming by Ejector.—Fig. 3 shows a horizontal pump fitted with a discharge valve and a steam ejector. The discharge valve being closed, the steam inlet valve to the ejector being opened first, and then the valve between the ejector and the pump opened, the air in the pump and pipes will be exhausted, and the water drawn up into them.

Ques. When the ejector is placed near the pump, how is complete priming indicated?

Ans. By water issuing from the ejector.

Ques. What should be done in shutting off the ejector?

Ans. Close the valve between it and the pump first, and the steam inlet valve last.

Ques. How should the ejector be piped when it is not convenient to place it near the pump?

Ans. The air pipe may be extended, in which case it is necessary that a slightly larger air pipe be used than when the ejector is placed near the pump.

Priming by Hand Lift Pump.—Fig. 4 shows a check valve used in place of the discharge valve and a hand pump (or power air pump used in place of a steam ejector), the priming being accomplished similarly to the method in fig. 3.

Ques. What provision should be made in the air pipe?

Ans. A valve should be placed in the air pipe which should be closed before starting.

Ques. What kind of hand pump is used?

Ans. An ordinary kitchen lift pump.

It should be piped as shown in fig. 4, with the air pipe forming a loop little above the discharge.

It is only necessary to put a little water into this type of pump to water seal it, and make of it a very good air pump.

Ques. What is the object of the loop?

Ans. It prevents the water escaping.

Ques. What is used on large pumps to indicate complete priming?

Ans. A water glass is used similar to those on boilers.

It is placed near the top to the pump casing.

Priming with Foot Valve and Top Discharge.—Fig. 5 shows the method of using a foot valve, in which case the pump and inlet pipe are to be filled with water through the discharge or top of the pump from any convenient source, such as a small tank or hand pump, which can be piped to the top of the centrifugal pump.

Where the inlet pipe is long, there should be at least 5 feet of a discharge head on the pump to prevent the water being thrown out of the runner before the water in the inlet line begins to move, and thus cause failure to start.

Ques. How is the air expelled from the arm?

Ans. Turn the rotor around once or twice by hand.

NOTE.—When a power driven air pump is used for priming a water trap or other means should be placed in the air pipe to prevent water entering and possibly breaking the primer. Where the valves on the air pump are large this may be omitted.

Ques. How is complete priming indicated on a vertical pump?

Ans. When check or discharge valves are used, a vacuum gauge placed on the air priming pipe at the head of the well or pit will show when the pump is primed and avoid climbing down into the pit or well.

Ques. May a steam air ejector (as in fig. 3) be operated by water?

Ans. Yes, where a water pressure of 30 to 40 lbs. is available.

This, however, requires a special ejector.

Ques. Is it possible to have automatic priming?

Ans. Yes.

Ques. Describe the system.

Ans. For automatic priming a pressure regulator may be connected into the discharge line. This regulator starts an air pump automatically if the main pump lose its prime and stops the priming pump when priming is completed.

Figs. 6 and 7 show two other automatic priming arrangements.

Starting.—Preliminary to starting for the first time with oil lubricated bearings, with the oil cold and bearing surfaces comparatively dry, it is important to revolve the rotor a few times either by hand, or with the pump filled with water, by momentarily operating the starting switch (if this procedure do not overload the motor). This starts a flow of oil to the bearing surfaces. Pump may be operated for a few minutes with discharge valve closed without overheating or damage.

NOTE.—A vacuum gauge may be used in the methods of priming shown in figs. 3 and 4, but care must be taken to shut off the gauge before starting the pump, as pressure will ruin a vacuum gauge.

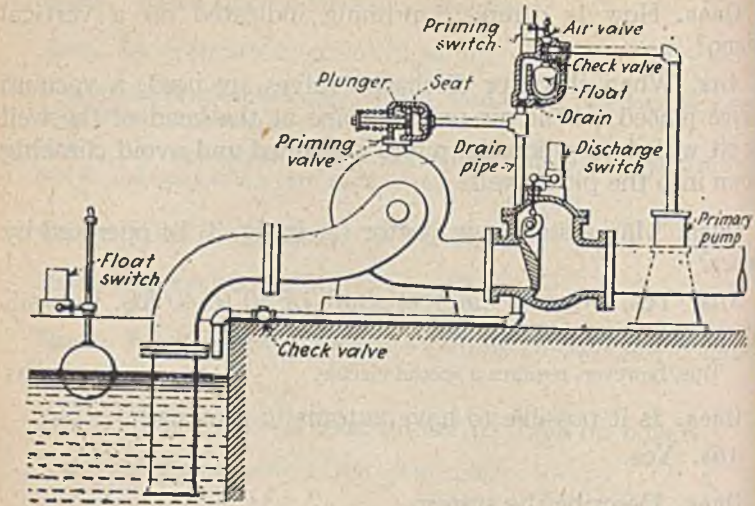


Fig. 6.—Float controlled automatic priming system. **In hook up,** a priming valve is connected between the top of the pump's casing and a float-controlled air valve. The air valve is connected to a priming switch, the contacts of which are in series with the control circuit of the main motor starter. A check valve in the discharge line operates a switch that shunts the priming switch. **In operation,** as long as the pump is not primed, the priming valve and air valve are in the positions indicated. When the float switch closes, it starts the priming pump, which continues in operation until the main pump is primed and water has come up high enough in the float chamber to close the priming switch. Closing the priming switch starts the main pump, and the priming pump is stopped by a contact that is opened when the pump-motor-control circuit is energized. When the main pump is running, the discharge check valve is held open and the contacts on its switch are closed to complete a holding circuit for the pump-motor contactor, around the priming switch. This switch allows the priming switch to open and not shut down the pump. A number of small holes in the priming valve plunger permit air to pass freely during the priming operation. These holes are so proportioned that when priming has been completed and the pump started, the pressure developed by the pump forces the plunger to its seat, thereby cutting off communication to the priming pump. When the priming line is sealed, the water drains from the float chamber and the priming switch opens, but the pump-motor contactor is held closed by a circuit through the discharge valve switch. When the float switch opens, it shuts down the main pump. It will be noted that the float switch starts the priming pump when it closes and stops the centrifugal pump when it opens.

Ques. What should be done now?

Ans. Check various items.

An extended tryout may be necessary on some installations and if this be necessary keep the vent valves open to relieve pocketed air in the pump and system. This circulation of the water prevents the pump becoming unduly heated.

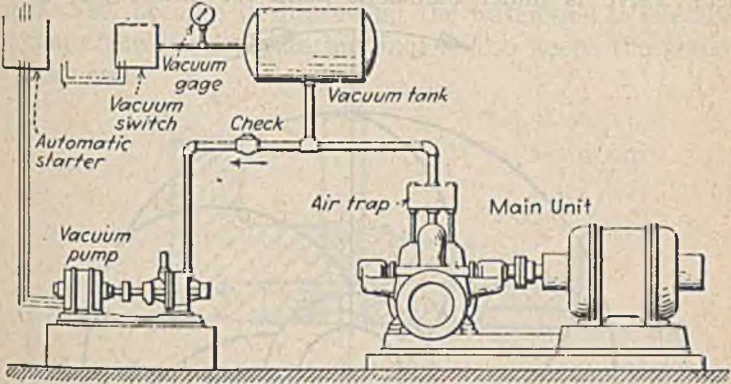


Fig. 7.—Vacuum tank used as reserve to keep main pump primed. The system includes a motor-driven air pump and a vacuum tank connected between the section of the air pump and the priming connections on the centrifugal pump. The air tank serves as a reserve on the system so that the vacuum producer need only be started intermittently. A vacuum switch starts and stops the vacuum producer at the predetermined limits of vacuum. An air trap in the line between the pump to be primed and the vacuum tank prevents water rising into the vacuum system after the pump is primed. Air is drawn from the pump suction chamber whether the pump be idle or under vacuum or in operation. The priming pipes and air-trap valves are under vacuum to insure that the centrifugal pump remains primed at all times. Modifications of this system are also used for priming pumps handling sewage, paper stock, sludge or other liquids carrying solids in suspension.

Ques. When satisfied the pump may be cut into the line, what is done?

Ans. Close vent valves and open discharge valve slowly.

Ques. Why is the pump started with discharge valve closed?

Ans. Because when the discharge valve is closed the pump operates at from only 35 to 50 per cent of full load.

Ques. Under what conditions may the pump be started with discharge valve open?

Ans. In cases where the liquid on upper side of discharge check valve is under sufficient head for starting purposes.

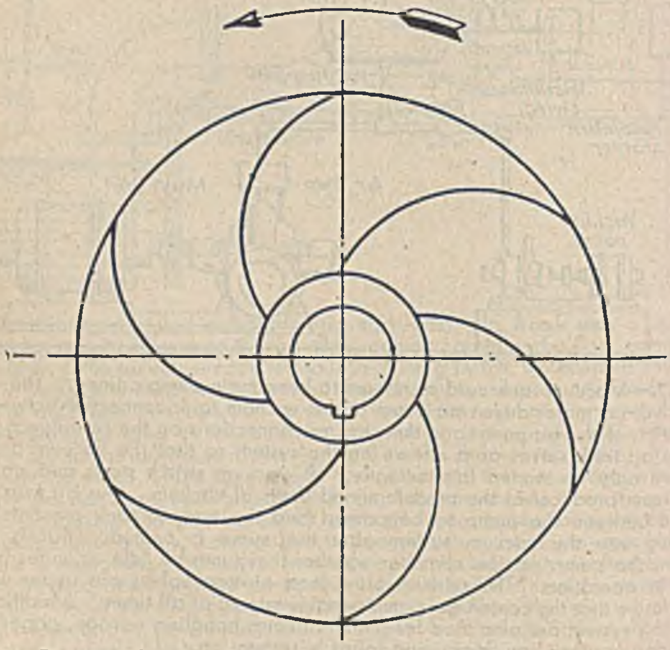


Fig. 8.—Diagram showing direction of rotation of impeller.

Ques. In starting what attention should be given to gland packing?

Ans. It should not be too tight.

Ques. Why?

Ans. As the packing heats up it will expand and may cause burned out packing and scored shafting.

It is best to have the packing leak at first, and then after it becomes warmed up and worn in a bit it may be tightened.

Ques. Is a slight amount of leakage advisable and why?

Ans. Yes, because it shows that the water seal is effective and that there is no undue binding; it also keeps the glands and shaft cool.

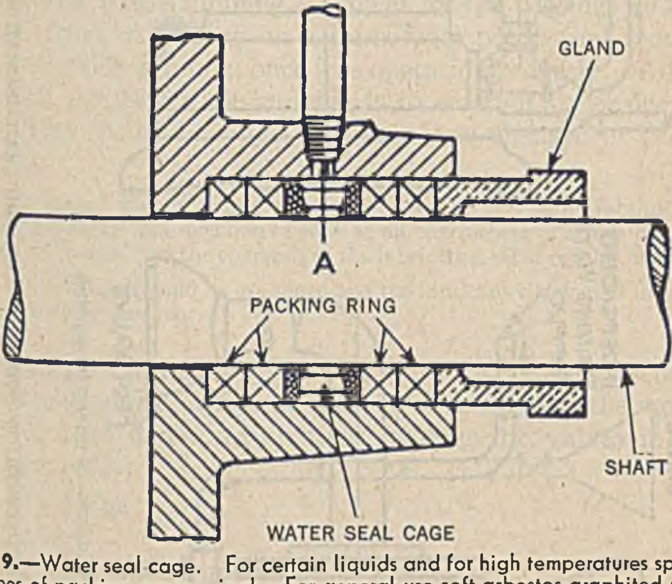
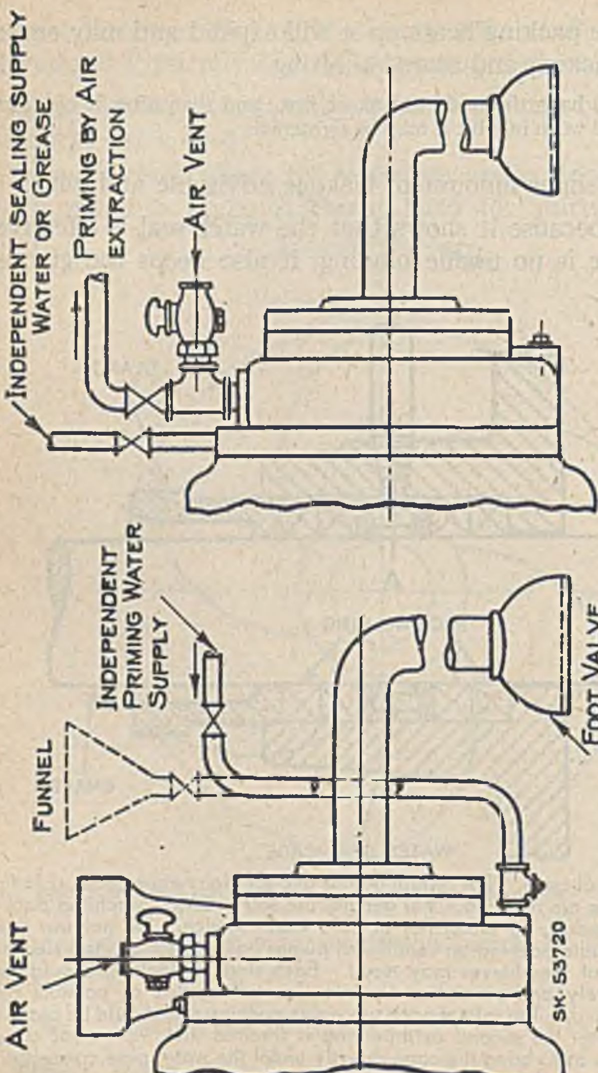


Fig. 9.—Water seal cage. For certain liquids and for high temperatures specific types of packing are required. For general use soft asbestos graphited packing is recommended, for either hot or cold water service. Do not use flax packing or metallic packing on centrifugal pumps having bronze shaft sleeves as rapid wear of the sleeves may result. Each ring of packing should be inserted separately and pushed as far into the stuffing box as possible by means of the gland. The split of each successive packing ring should be placed 90° apart. After the second or third ring is inserted the water seal cage should follow so as to bring the cage directly under the water pipe connection as shown.



Figs. 10 and 11.—Diagrams of piping for priming.

Ques. Should a pump be operated for long periods at low capacity?

Ques. How may a pump be safely operated at low capacity?

Ans. By installing a permanent by-pass from the discharge to the inlet of a size equal to $\frac{1}{5}$ the size of discharge pipe.

Ques. How are hot well pumps started?

Ans. Open inlet and discharge valves and valves on independent stuffing box seals before operating.*

Attention While Running.—Except for the bearings and the glands, there is nothing on a centrifugal pump that requires attention while running, once it is operating properly. All the attention required while running is to see that the proper oil is supplied to the bearings, and that it is changed at proper intervals.

Centrifugal pumps should operate for long periods with practically no attention other than to observe that at all times there is a drip of liquid from the glands, and the changing of the lubricating oil at regular intervals.

Ball bearings should be examined and the lubricant changed at intervals not exceeding one year.

Stopping.—Normally when there is a check valve in discharge line close to pump, it is shut down by first stopping the motor, securing until again required, by closing the valves in the following order:

1. Discharge
2. Inlet
3. Cooling water supply
4. Any other points communicating with the system.

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*NOTE.—Usually the air extraction apparatus is in service before the hot well pump is started, the main turbine is at the same time being heated. This provides an accumulation of water in hot well; if allowed to collect above level of hot well gauge glass, observe that the steam jets or other extracting apparatus do not entrain water; this would render them inoperative. As soon as there is a supply of condensate flowing to hot well, pump may be started and the air valves on top of pump opened as pump comes up to speed. The air valves should then be closed.

Ques. When stopping in this manner must the discharge ga valve be open?

Ans. As a rule no.

In some installations, however, surges in the piping may impose hea shocks on both lines and pump, when the flow of high pressure water arrested. In such cases, it is good practice to first shut the discharge ga valve; this entirely eliminates shock.

Ques. What happens after stopping a pump and it remai idle some time?

Ans. It will gradually lose its priming.

Ques. How does this happen?

Ans. It will partly drain through the glands.

Ques. If pump be required for emergency use and be alwa primed, what should be noted about valves on stopping?

Ans. It is not necessary to close inlet and discharge valve Under this condition, the glands may leak due to sustaine pressure on a stationary shaft. Do not tighten the gland nut unless prepared to loosen them again at starting.

Troubles

Abnormal Conditions.—Pumps, when in good operati condition, run smoothly without vibration. The bearing operate at a constant temperature which is governed some what by the location of the units. This temperature ma be as low as 100° Fahr. but an operating temperature may vary with the pump capacity, being usually maximum temperature at minimum flow.

Ques. If a pump for any reason do not contain liquid or become vapor bound, what happens?

Ans. There will be vibration due to contact between stationary and revolving parts and the pump may become overheated.

Vapor may blow from the glands and in extreme cases thrust bearing may suddenly increase in temperature and be damaged due to rotor being forced hard in one direction.

Ques. What should be done in the case of overheating only, due to a vaporized condition and the rotor has not seized?

Ans. Open all vents and re-prime or flood liquid into pump. Do not admit cold or comparatively low temperature liquid to a heated pump quickly, as this may result in fracture or distortion of parts. The only condition under which an overheated pump may be quickly put on the line is when an emergency exists, such as the need to save a boiler from accident.

Ques. Give another cause for vibration.

Ans. It results from excessive wear on pump rotor or in pump bearings which causes pump and motor shafts to get out of alignment.

Under these conditions defects and errors should be corrected at the first opportunity.

Ques. What should be done when a rotor has seized?

Ans. It is necessary to completely dismantle and rectify the parts by filing, machining or replacement, as required.

Locating Troubles

The difficulties which may be experienced with centrifug pumps, and their causes, are listed here, as follows:

Insufficient Capacity or Pressure and Failure to Deliv Water—

1. Pump not primed.
2. Speed low.
3. Total dynamic head higher than that for which pump is rated.
4. Lift too high (the normal lift is 15′–0″).
5. Foreign material in impeller.
6. Wrong direction of rotation.
7. Excessive amount of air in water.
8. Air leakage in inlet pipe or stuffing boxes.
9. Insufficient inlet pressure for the vapor tension of the liquid.
10. Mechanical defects.
    - *a.* Wearing rings worn
    - *b.* Impeller damaged
    - *c.* Casing gasket defective.
11. Foot valve too small or restricted by trash.
12. Foot valve or inlet pipe not immersed deep enough.

Pump Loses Water After Starting.—1. Air leaks in inle pipe.

2. Water-seal pipe plugged.
3. Lift too high (over 15 ft.).
4. Excessive amounts of air or gases in water.

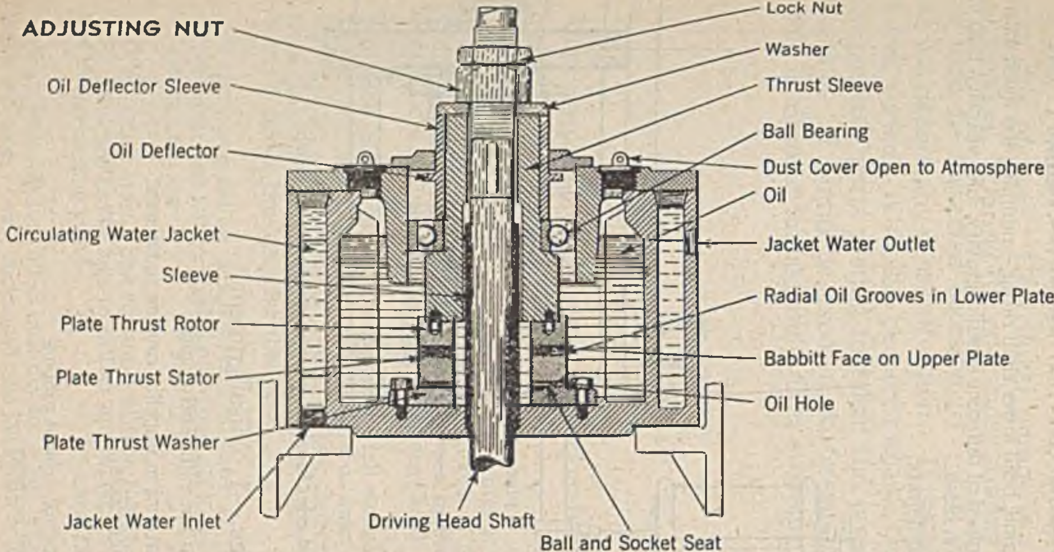

Fig. 12.—Deep well pump thrust bearing and housing. The plate thrust bearing operates in a bath of oil. The oil is taken in at the center of the bearing and forced between the plates. The thrust load is, therefore, carried entirely on a film of oil. There is no metal-to-metal contact to cause wear and loss of efficiency. The lubricating oil is surrounded by a jacket of continually circulated water, to keep the oil at an even temperature for the best results. The thrust bearing housing between the shaft stuffing box and the driving agent is rigidly supported by the driving head frame. Adjusting nuts over the thrust bearing raise or lower the shaft until proper clearance, above and below the impellers, is obtained.

Pump Overloads Driver.—1. Speed too high.

2. Total dynamic head lower than rating—pumping too much water

3. Liquid pumped of different specific gravity and viscosity than t[illegible] for which pump is rated.

4. Mechanical defects.

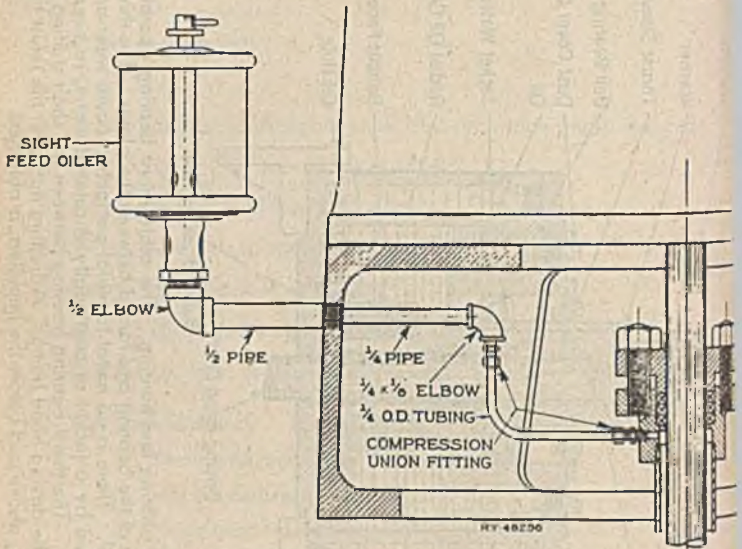

Fig. 13.—Sight feed oiler for shaft bearing of deep well pump.

Pump Vibrates.—1. Misalignment.

2. Foundation not sufficiently rigid.

3. Foreign material in impeller causing unbalance.

4. Mechanical defects.

a. Shaft bent.

b. Rotating element rubbing

c. Worn bearings.

A Few Pointers.—Always run the pump in direction of arrow cast on case. Centrifugal pumps can be run only in one direction. During the operation, stuffing boxes and bearings must be inspected occasionally. The centrifugal pump does not require any other attention. If the pump is to be idle for long periods, it should be taken apart, cleaned and oiled. This prevents parts rusting together and preserves their good condition. If pump be exposed to freezing temperature, it should be drained immediately after stopping.

Ifs

A few pointers are here given for various abnormal conditions as suggested by Union.—

If after starting the pump it throw a little water at the first few revolutions, and then churn and fail to discharge more, it is just evidence that the air was not all out of the pump and pipes, or the lift too great, or a leaky pipe, or a long inlet pipe and insufficient head.

If when first started the pump throw a full stream for a few minutes, and then fail, it is caused by failure of supply or water receding in the well below the lift limit, which in a well is best determined by a vacuum gauge placed on the inlet elbow of the pump. The remedy for this is to lower the pump, thus reducing the lift.

If the pump deliver a full stream of water at the surface, or level of the pump, but fail to pump at a higher discharge point, the speed of the pump is too low.

If the pump start a full stream, and then the discharge decrease very slowly until the pump fails to deliver any water, it is caused by an air leak at the packing gland.

If the pump deliver a full quantity for a few hours and fail, the speed and water supply being unchanged, the inlet pipe or impeller is obstructed.

If when running there be a heavy vibration, the shaft has been sprun the pump is out of alignment, or an obstruction has lodged in one side the impeller.

If the bearings heat unduly, the belt is unnecessarily tight, the bearin lack oil, or there is an end thrust.

If hot liquids are to be pumped the lift should be as small as possib on account of lowering the boiling point under vacuum and conseque loss of priming from the presence of vapor.

If water discharge into a sump or tank near the end of the inlet pip there is danger of entraining air into the inlet pipe.

If a pump be speeded up beyond its maximum rating to increase capacity it will result in a waste of power.

If a pump remain idle for some time, revolve rotor by hand once a week-for an extra long period it should be taken apart, cleaned and oiled as previously instructed.

Maintenance and Repair

In a lateral direction liberal clearances are provided between rotating and stationary parts; this allows for slight machining variations and expansion of casing and rotor when heated.

All stationary diaphragms, wearing rings, return channels, etc., which locate in casing or other stationary part, are made only a few thousandths less in diameter than the bore of casing, which is machined with a 1/32 in. thick gasket between the flanges. The casing must not bind on these stationary parts when the flange nuts are tightened.

All running clearances such as those at wearing rings, according to one manufacturer, are made from one to one and one-half thousandths per in. in diameter, depending on the actual location of the part, the material employed and the bearing span.

Lateral End Movements.—Lateral clearance between rotor and stator parts is necessary for several reasons:

1. Mechanical
2. Hydraulic considerations
3. To permit of variations in expansion between casing and rotor.

It is limited to 1/64 in. in small pumps and pumps of certain types, and is as much as ½ in. on large units and those handling hot liquids.

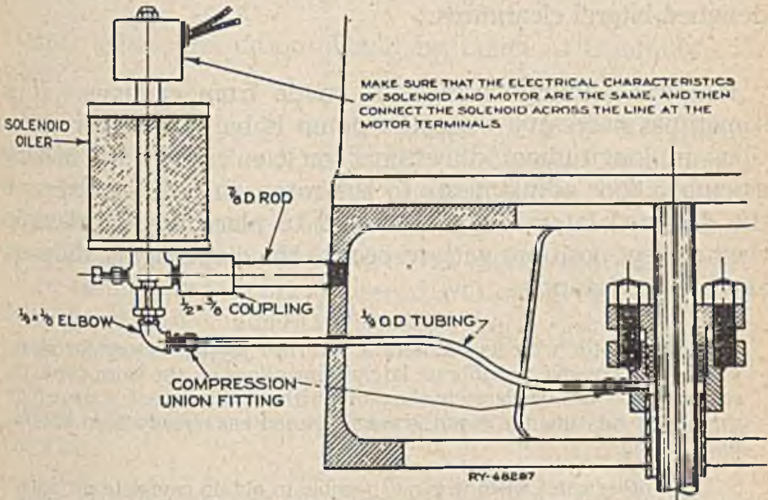

Fig. 14.—Solenoid oiler for shaft bearing of deep well pump.

Ques. How is this clearance divided in the case of cold liquid pumps when the thrust bearing is secured in position and impellers centralized?

Ans. Equally.

Ques. What allowance is made for hot liquid pumps?

Ans. Allowance is made for differences in expansion betwe rotor and casing, noting that the rotor will expand from th fixed thrust bearing end.

Ques. What should be done before proceeding with a important field assembly?

Ans. Determine by examination of the unit or ascertai by asking the manufacturer of the pump, the amounts of th designed lateral clearances.

Renewal.—Pump casings are made from castings. It i sometimes necessary when the pump is built, to favor varia tions in longitudinal dimensions on the casings by makin assembly floor adjustments to the rotor, in order to preserv the designed lateral clearances and to place the impellers in their correct positions with respect to the diaphragms, diffuser and return channels.

When a rotor with its diffusers is returned to the factory for repair if it be still possible to calibrate lateral dimensions on the worn parts, the repaired rotor can be placed in the pump with no adjustment; a spare rotor shipped at the time the pump is manufactured has already been installed in its casing.

In all other cases when it is not possible to obtain complete particulars of the old parts, stock and other replace parts are made to standard dimensions.

Ques. What should be done when making field renewals of rotating or stationary parts?

Ans. Compare all lateral distances with those on the old parts, and duplicate these distances where lateral end movement is affected.

Place the assembled rotor and stage pieces, etc., where so fitted, in the lower half of casing, check total lateral clearance and with thrust bearing assembled and shaft consequently in its proper position, observe that this clearance is suitably divided and impellers centralized in their volutes. Make final adjustments by manipulation of shaft nuts.

Casing Flange Gaskets.—Use the thickness and material originally installed. One-thirty-second inch is the standard thickness. The material is selected to suit the service conditions.

Ques. What precaution should be taken in trimming the gasket?

Ans. The inner edge of the gasket must be accurately trimmed along the edge of the stuffing box bore. At all points where the gasket abuts on the outer diameter and sides of stationary parts between stages the edges must be trimmed squarely and neatly—sufficiently overlapping that the upper half of casing when being tightened will effectively press the edges of the gasket against the stator parts ensuring proper sealing between stages.

Ques. What is the best method of performing the trimming operation?

Ans. First cement the gasket to the lower half of the casing with shellac thereafter cutting all interstage gasket edges square and at the same time overlapping, using a razor blade.

Pointers on Assembly.—Do not use unnecessary force to tighten impeller and shaft sleeve nuts, straining results in bending the shaft and destroying the concentricity of rotor parts operating in close clearances with stator parts, causing rubbing and vibration.

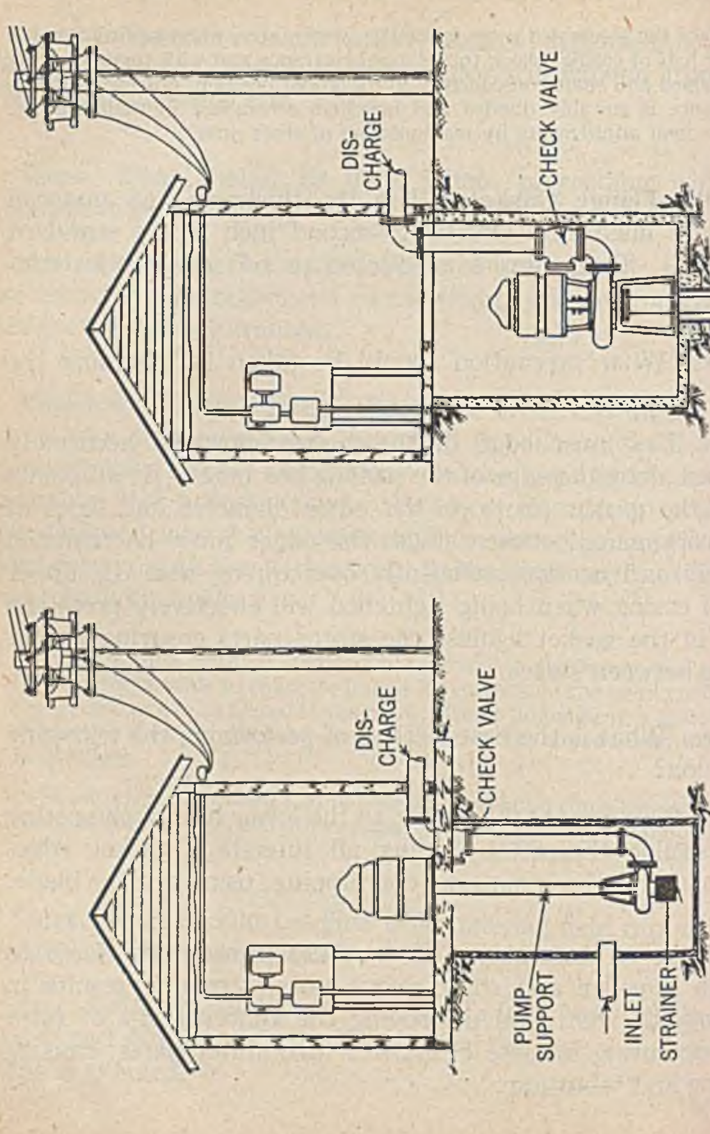

Fig. 15.—Typical installation of a suspended submerged pump on a pit of cheap construction which is not water tight.

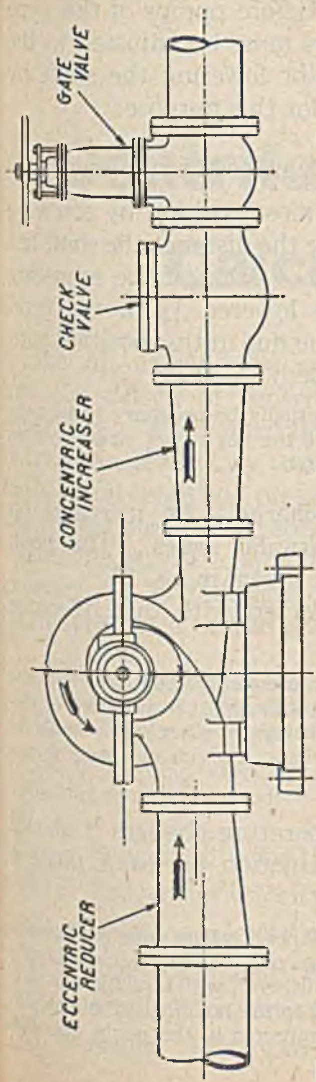

Fig. 17.—Installation of increaser (sometimes called diffuser) on discharge line. Illustration shows hook-up of increaser, check valve and discharge valve.

Locking Screws.—When securing locking screws of the safety type, check using a dial test indicator that the shaft is not bent in the process, manipulate the screws when tightening accordingly. Indent the thread behind the screw slightly to prevent its backing out.

Design considerations require that all parts be mounted on the shaft in the same order as found. Impellers being opposed are both right and left hand in the same casing. Diaphragms, wearing rings, and stage bushings are individually fitted and their sealing flanges between stages tested for their particular locations. Stage bushings with stop pieces are not interchangeable one with another for the reason that the stop pieces locate at varying positions.

All stationary parts assembled on the rotor such as stuffing box bushings, wearing rings, stage bushings, diaphragms, etc., have stops consisting either of individual pins or half flanges in lower casing only to prevent turning.

These parts must be positioned when lowering rotor into casing so that all stops are in their respective recesses in lower half of casing. Otherwise, upper half of casing will foul improperly positioned parts when being mounted.

Deep Well Pump Adjustments.—Before pumps of this type can be run, the impeller or impellers must be adjusted to the correct running position by raising or lowering the shaft by means of the adjuster nut provided for this purpose.

Ques. How is the adjustment made for the axi-flow type?

Ans. Raise the shaft to its uppermost position by screwing down on the adjusting nut. Measure the distance the shaft has been raised above its lowest position. Back off the adjusting nut until after the shaft has been lowered ⅓ of the total distance it has been raised. Lock the nut in this position with the key, set screw or lock nut provided.

With either a key or set screw it will usually be necessary to turn the adjusting nut until it is possible to insert the key or set screw in place through both adjusting nut and motor clutch.

Installation of Increaser on Discharge.—An increaser on the discharge line will reduce hydraulic losses. The hook up with check and discharge valve is shown in fig. 17.

The discharge line should be selected with due reference to friction losses.

It should never be smaller than the pump discharge and preferably one or two sizes larger. Do not use the pump to support heavy inlet or discharge piping, and do not force pipes or fittings in place with the flange bolts, as the pump alignment will be disturbed. Provide independent supports for all piping.

When piping is subjected to temperature changes it should be arranged so expansion and contraction does not place a strain on the pump casing.

In hotels, apartment buildings, hospitals, etc., where noise is objectionable, care must be taken that the discharge pipe be not attached directly to the steel structural work or to hollow walls, etc., with insulation, in such a way that vibration may be transmitted to the building. Preferably in such cases the discharge line should be connected to the pump discharge through a flexible connection.

CHAPTER 7

Centrifugal Pumps; Calculations

Like all engineering work, the various factors entering into the design of centrifugal impellers are determined by experience. The centrifugal pump fundamentally adds energy in the form of velocity to an already flowing liquid; it does not in the usual sense add pressure. All basic considerations of the centrifugal pump must consider kinetic enregy.

The design of centrifugal impellers is determined by experience. The design of a centrifugal pump impeller is ultimately based on the performance of other impellers.

The theory indicates what would be the general effect of altering certain dimensions, hence, successful design consists according to Union, of modifying or changing the design of impellers which have been tested.

In the following theory and accompanying diagram by Union, fig. 1, the following symbols are used.—

V_2 = Tangential velocity, impeller at outer periphery.
V_1 = Tangential velocity, impeller at inner periphery.
Z_2 = Relative velocity of water at outlet.
Z_1 = Relative velocity of water at inlet.
C_2 = Absolute velocity of water at outlet.
J_2 = Radial velocity of water at outlet.

J_1 = Radial velocity of water at inlet.
W = Tangential velocity of water at outlet.
a_2 = Outlet angle of impeller.
a_1 = Inlet angle of impeller.

In fig. 1 the water enters the impeller inlet with a radi velocity J_1, and leaves the impeller with an absolute veloci

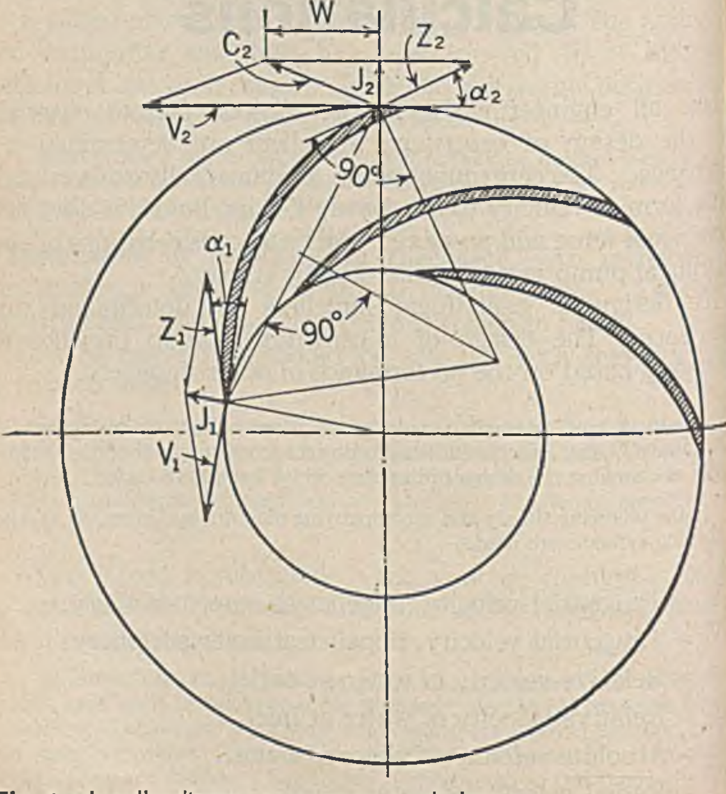

Fig. 1.—Impeller diagram to accompany calculations.

of C_2. The inner peripheral velocity of the impeller is V_1, and the outer peripheral velocity V_2. All velocities are in feet per second

Let H, be the theoretical head in feet against which the pump would deliver water, if there were no losses. Then

$$H = \frac{V_2^2}{2g} \dots\dots\dots\dots\dots\dots (1)$$

In which g = force of attraction of gravity = 32.2 ft. per second.

From formula (1)

$$V_2 = \sqrt{2gH}$$

Having given the head against which the pump must work and the diameter of the impeller, the speed of the pump may be calculated by formula (1).

Example.—A pump is required to pump against a head of 100 ft. and have an impeller 10¾ in. in diameter. What is the required speed of the pump?

By substituting in formula (1),

$$V_2 = \sqrt{2gH}$$

$$V_2 = \sqrt{2 \times 32.2 \times 100} = 80.4 \text{ ft. per sec.}$$

This is equal to $80.4 \times 60 = 4{,}824$ ft. per minute. The circumference of the impeller 10¾ in. in diameter equals

$$10\tfrac{3}{4} \times 3.14 = 33.8 \text{ in. or } 2.8 \text{ ft.}$$

As the impeller has to revolve 4,824 ft. per minute, it will have to run $\frac{4{,}824}{2.8}$ = 1,722 revolutions per minute.

The capacity of a pump depends upon the size of the inlet and discharge openings, the size of the casing, and width and diameter of the impeller. These factors are determined by the designer from experience.

How to Figure the Total Hydraulic Load or Lift + Head of a Centrifugal Pump.—The usual expression "total head" is here carefully avoided as being a misnomer and ridiculous—this goes also for such expressions as "suction lift."

Ques. With respect to centrifugal pumps, what is the lift?

Ans. The vertical distance from the level of the water to be pumped to the center line of the pump.

Ques. What is the condition of operation if the water level be above the center line of the pump?

Ans. The pump is operating under inlet head, sometimes called negative lift.

Ques. How does an inlet head affect the calculation?

Ans. It must be subtracted from the sum of the remaining factors.

Ques. What is the discharge head (as distinguished from an inlet head)?

Ans. The vertical distance between the center line of the pump and the level to which the water is elevated.

Ques. How is the friction head for pipes and elbows of different sizes and capacities found?

Ans. From the accompanying tables.

Pipe Friction

Loss of Head in Feet Due to Friction, per 100 Feet of *15-Year-Old* Ordinary Iron Pipe

For *New and Smooth* Iron Pipe, the Loss of Head will be .71 of that Shown in Table

Gallons per Minute	½" Pipe		¾" Pipe		1" Pipe		1¼" Pipe		1½" Pipe		2" Pipe		2½" Pipe		3" Pipe	
	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.
1	1.05	2.1	....	....	....	....	....	....	....	....	....	....	....	....	....	....
2	2.10	7.4	1.20	1.9	....	....	....	....	....	....	....	....	....	....	....	....
3	3.16	15.8	1.80	4.1	1.12	1.26	....	....	....	....	....	....	....	....	....	....
4	4.21	27.0	2.41	7.0	1.49	2.14	0.86	0.57	0.63	0.26	....	....	....	....	....	....
5	5.26	41.0	3.01	10.5	1.86	3.25	1.07	0.84	0.79	0.39	....	....	....	....	....	....
10	10.52	147.0	6.02	38.0	3.72	11.7	2.14	3.05	1.57	1.43	1.02	0.5	0.65	0.17	0.45	0.07
15	....	....	9.02	80.0	5.60	25.0	3.2	6.5	2.36	3.0	1.53	1.0	0.98	0.36	0.68	0.15
20	....	....	12.03	135.0	7.44	42.0	4.20	11.1	3.15	5.2	2.04	1.82	1.31	0.61	0.91	0.25
25	....	....	....	....	9.30	64.0	5.36	16.6	3.94	7.8	2.55	2.73	1.63	0.92	1.13	0.38
30	....	....	....	....	11.15	89.0	6.43	23.5	4.72	11.0	3.06	3.84	1.96	1.29	1.36	0.54
35	....	....	....	....	13.02	119.0	7.51	31.2	5.51	14.7	3.57	5.1	2.29	1.72	1.59	0.71
40	....	....	....	....	14.88	152.0	8.58	40.0	6.3	18.8	4.08	6.6	2.61	2.20	1.82	0.91
45	....	....	....	....	....	....	9.65	50.	7.08	23.2	4.60	8.2	2.94	2.80	2.05	1.15
50	....	....	....	....	....	....	10.72	60.	7.87	28.4	5.11	9.9	3.27	3.32	2.27	1.38
70	....	....	....	....	....	....	15.01	113.	11.02	53.0	7.15	18.4	4.58	6.2	3.18	2.57
90	....	....	....	....	....	....	....	....	14.17	84.0	9.19	29.4	5.88	9.8	4.09	4.08
100	....	....	....	....	....	....	....	....	15.74	102.0	10.21	35.8	6.54	12.0	4.54	4.96
120	....	....	....	....	....	....	....	....	18.89	143.0	12.25	50.0	7.84	16.8	5.45	7.0
140	....	....	....	....	....	....	....	....	22.04	190.0	14.30	67.0	9.15	22.3	6.35	9.2
160	....	....	....	....	....	....	....	....	....	....	16.34	86.0	10.46	29.0	7.26	11.8
180	....	....	....	....	....	....	....	....	....	....	18.38	107.0	11.76	35.7	8.17	14.8
200	....	....	....	....	....	....	....	....	....	....	20.42	129.0	13.07	43.1	9.08	17.8
220	....	....	....	....	....	....	....	....	....	....	22.47	154.0	14.38	52.0	9.99	21.3
240	....	....	....	....	....	....	....	....	....	....	24.51	182.0	15.69	61.0	10.89	25.1
260	....	....	....	....	....	....	....	....	....	....	26.55	211.0	16.99	70.0	11.80	29.1
280	....	....	....	....	....	....	....	....	....	....	....	....	18.30	81.0	12.71	33.4
300	....	....	....	....	....	....	....	....	....	....	....	....	19.61	92.0	13.62	38.0

Gallons per Minute	4" Pipe		5" Pipe		6" Pipe		8" Pipe		10" Pipe		12" Pipe		14" Pipe		15" Pipe		16" Pipe		20" Pipe	
	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.
40	1.02	0.22	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
45	1.17	0.28	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
50	1.28	0.34	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
70	1.79	0.63	1.14	0.21	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
75	1.92	0.73	1.22	0.24	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
100	2.55	1.23	1.63	0.39	1.14	0.14	....	....	....	....	....	....	....	....	....	....	....	....	....	....
120	3.06	1.71	1.96	0.57	1.42	0.25	....	....	....	....	....	....	....	....	....	....	....	....	....	....
125	3.19	1.86	2.04	0.64	1.48	0.28	....	....	....	....	....	....	....	....	....	....	....	....	....	....
150	3.84	2.55	2.45	0.88	1.71	0.32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
175	4.45	3.36	2.86	1.18	2.00	0.48	....	....	....	....	....	....	....	....	....	....	....	....	....	....
200	5.11	4.37	3.27	1.48	2.28	0.62	....	....	....	....	....	....	....	....	....	....	....	....	....	....
225	5.32	6.61	3.67	1.86	2.57	0.74	....	....	....	....	....	....	....	....	....	....	....	....	....	....
250	6.40	6.72	4.08	2.24	2.80	0.92	1.60	0.22	....	....	....	....	....	....	....	....	....	....	....	....
275	7.03	7.99	4.50	2.72	3.06	1.15	1.73	0.27	....	....	....	....	....	....	....	....	....	....	....	....
300	7.66	9.38	4.90	3.15	3.40	1.29	1.90	0.36	....	....	....	....	....	....	....	....	....	....	....	....
350	8.90	12.32	5.72	4.19	3.98	1.69	2.20	0.41	....	....	....	....	....	....	....	....	....	....	....	....
400	10.20	15.82	6.54	5.33	4.54	2.21	2.60	0.56	....	....	....	....	....	....	....	....	....	....	....	....
450	11.50	19.74	7.35	6.65	5.12	2.74	2.92	0.64	1.80	0.21	....	....	....	....	....	....	....	....	....	....
475	12.30	22.96	7.88	7.22	5.55	3.21	3.10	0.79	1.94	0.25	....	....	....	....	....	....	....	....	....	....
500	12.77	24.08	8.17	8.12	5.60	3.26	3.20	0.81	2.04	0.28	1.42	0.11	....	....	....	....	....	....	....	....
550	....	....	8.99	9.66	6.16	3.93	3.52	0.98	2.25	0.33	1.57	0.14	....	....	....	....	....	....	....	....
600	....	....	9.80	11.34	6.72	4.70	3.94	1.16	2.46	0.39	1.71	0.15	....	....	....	....	....	....	....	....
650	....	....	10.62	13.16	7.28	5.50	4.16	1.34	2.66	0.46	1.85	0.19	1.37	0.09	....	....	....	....	....	....
700	....	....	11.44	15.12	7.84	6.38	4.46	1.54	2.86	0.52	2.00	0.22	1.47	0.10	....	....	....	....	....	....
750	....	....	12.26	17.22	8.50	7.00	4.80	1.74	3.06	0.59	2.13	0.24	1.58	0.11	....	....	....	....	....	....
800	....	....	....	....	9.08	7.90	5.12	1.97	3.28	0.67	2.27	0.27	1.68	0.13	....	....	....	....	....	....
850	....	....	....	....	9.58	8.75	5.48	2.28	3.48	0.75	2.41	0.31	1.79	0.14	....	....	....	....	....	....
900	....	....	....	....	10.30	10.11	5.75	2.46	3.68	0.83	2.56	0.34	1.89	0.16	....	....	....	....	....	....
950	....	....	....	....	10.72	10.71	6.06	2.87	3.88	0.91	2.70	0.35	2.00	0.16	1.73	0.12	....	....	....	....
1000	....	....	....	....	11.32	12.04	6.40	3.02	4.08	1.01	2.84	0.41	2.10	0.19	1.82	0.14	....	....	....	....
1100	....	....	....	....	12.50	14.31	7.03	3.51	4.50	1.20	3.13	0.49	2.31	0.23	2.00	0.16	....	....	....	....
1200	....	....	....	....	13.52	16.69	7.67	4.26	4.91	1.46	3.41	0.57	2.52	0.26	2.18	0.19	....	....	....	....
1500	....	....	....	....	....	....	9.60	6.27	6.10	2.09	4.20	0.85	3.15	0.39	2.73	0.28	2.39	0.24	....	....
2000	....	....	....	....	....	....	12.70	10.71	8.10	3.50	5.60	1.43	4.20	0.66	3.64	0.47	3.19	0.39	....	....
2500	....	....	....	....	....	....	....	....	10.10	5.33	7.00	2.18	5.25	1.01	4.55	0.72	3.99	0.56	....	....
3000	....	....	....	....	....	....	....	....	12.10	7.42	8.40	3.39	6.30	1.57	5.46	1.12	4.79	0.80	3.08	0.27

Vel.—Velocity feet per second. Fric.—Friction head in feet.

Friction of Water in 90° Elbows

Equivalent Number of Feet Straight Pipe

Size of Elbow, Inches	½	¾	1	1¼	1½	2	2½	3	4	5	6	8	10	12	14	15	16	20
Friction Equivalent Feet Straight Pipe	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Consumption of Electric Current For Pumping.

Total Efficiency of Pump, Motor and Transformer.	Consumption of Electric Current in Kilowatt Hours per 24 Hours Run, Pumping 100 Gallons per Minute 100 Feet High.	Total Efficiency of Pump, Motor and Transformer.	Consumption of Electric Current in Kilowatt Hours per 24 Hours Run, pumping 100 Gallons per Minute 100 Feet High.
Per Cent.	Kilowatt Hours.	Per Cent.	Kilowatt Hours.
30	149.2	45	100.
32	142.	50	89.
34	132.	55	81.
36	124.	60	75.
38	118.	65	69.
40	112.	70	64
42	107.	75	59.

WEIGHT AND CAPACITY OF DIFFERENT STANDARD GALLONS OF WATER.

	Cubic Inches in a Gallon.	Weight of a Gallon in Pounds.	Gallons in a Cubic Foot.	Weight of a Cubic Foot of Water. Eng. Standard, Lbs. Avoirdupois.
Imperial, or English . . .	277.274	10.00	6.232102	62.321
United States	231	8.33111	7.470519	. . . . . .

Water Required per Minute to Feed Boilers

Using the "Centennial Standard"—30 pounds or 3.6 gallons of water per horsepower per hour, evaporated from 100° F. to 70 pounds steam pressure per square inch.

H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons
20	1.2	60	3.6	110	6.6	190	11.4	400	21.0
25	1.5	65	3.9	120	7.2	200	12.0	450	27.0
30	1.8	70	4.2	130	7.8	225	13.5	500	30.0
35	2.1	75	4.5	140	8.4	250	15.0	600	36.0
40	2.4	80	4.8	150	9.0	275	16.5	700	42.0
45	2.7	85	5.1	160	9.6	300	18.0	800	48.0
50	3.0	90	5.4	170	10.2	325	19.5	900	54.0
55	3.3	100	6.0	180	10.8	350	21.0	1000	60.0

Contents of Round Tanks in U. S. Gallons for Each Foot in Depth

Inside Diameter Ft.	In.	Gallons One Foot In Depth	Inside Diameter Ft.	In.	Gallons One Foot In Depth	Inside Diameter Ft.	In.	Gallons One Foot In Depth	Inside Diameter Ft.	In.	Gallons One Foot In Depth
1	0	5.87	5	9	194.19	10	6	653.69	15	3	1365.96
1	3	9.17	6	0	211.44	10	9	678.88	15	6	1407.51
1	6	13.21	6	3	229.43	11	0	710.69	15	9	1457.00
1	9	17.98	6	6	248.15	11	3	743.36	16	0	1503.62
2	0	23.49	6	9	267.61	11	6	776.77	16	3	1550.97
2	3	29.73	7	0	287.80	11	9	810.91	16	6	1599.06
2	6	36.70	7	3	308.72	12	0	848.18	16	9	1647.89
2	9	44.41	7	6	330.38	12	3	881.39	17	0	1697.45
3	0	52.86	7	9	352.76	12	6	917.73	17	3	1747.74
3	3	62.03	8	0	375.90	12	9	954.81	17	6	1798.76
3	6	73.15	8	3	399.76	13	0	992.62	17	9	1850.53
3	9	82.59	8	6	424.36	13	3	1031.17	18	0	1903.02
4	0	93.97	8	9	449.21	13	6	1070.45	18	3	1956.25
4	3	103.03	9	0	475.80	13	9	1108.06	18	6	2010.21
4	6	118.93	9	3	502.65	14	0	1151.21	18	9	2064.91
4	9	132.52	9	6	530.18	14	3	1192.69	19	0	2121.58
5	0	146.83	9	9	558.45	14	6	1234.91	19	3	2176.68
5	3	161.88	10	0	587.47	14	9	1277.86	19	6	2233.52
5	6	177.67	10	3	617.17	15	0	1321.54	20	0	2340.46

Ques. What is the velocity head?

Ans. The equivalent distance in feet through which a liquid would have to fall to acquire the same velocity.

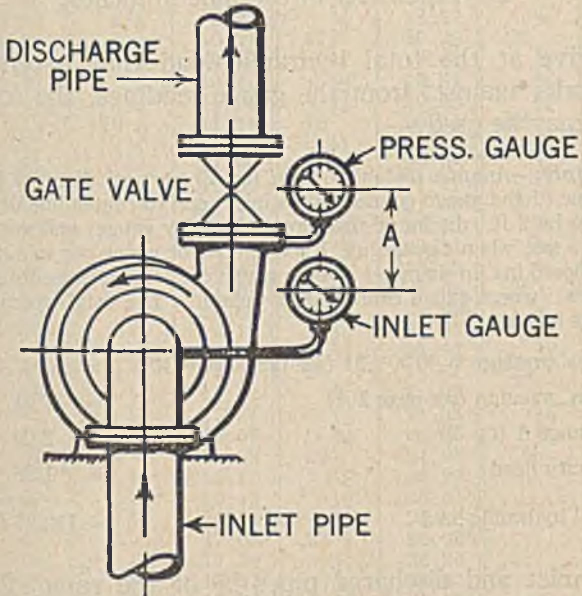

Fig. 2.—Hook up for centrifugal pump showing location of gauges in taking readings in calculating total hydraulic load.

Ques. How is the velocity head determined?

Ans. From the following formula

$$H_v = \frac{V_2^2}{2g} = \frac{V_2^2}{64.4} \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

in which

$$V = \frac{.408 \times \text{gallons per minute}}{D^2} \quad \ldots\ldots\ldots\ldots (3)$$

in which

D = diameter of the pipe in inches.

To arrive at the total hydraulic load that a centrifugal pump works against, from the gauge readings, the following example may be used.—

Example.—Assume distance A (in fig. 2) vertical distance from the center line of the gauge connection in inlet pipe to center line of pressure gauge) to be 2 ft., discharge pressure 40 lbs. (by gauge) and vacuum (by gauge) 15 ins. when discharging 1,000 gallons of water per minute. Discharge pipe 6 ins. in diameter (where gauge connection is made) and inlet pipe 8 ins. (where gauge connection is made). The total hydraulic load then is as follows:

40 lbs. pressure = 40 × 2.31 (see table page 248)	=	92.4 ft.
15 ins. vacuum (see page 201)	=	17.01 "
Distance A (fig. 2)	=	2.00 "
*Velocity head	=	1.36 "
Total hydraulic load	=	112.77 ft.

If the inlet and discharge pipes be of the same diameter where the gauge connections are made, the velocity head will be the same in both, and no correction need be made for same, as the inlet gauge readings include the velocity head in the inlet pipe, which in this instance is the same as the velocity head on the discharge pipe.

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*NOTE.—The velocity head in the 6 in. discharge pipe by formula (2) equals 1.99 ft. The velocity head in the 8 in. inlet pipe by formula (2) equals .63 ft. The total velocity head to be added therefore equals the difference between these two figures or 1.36 ft. Formula (3) according to Ordway.

INCHES VACUUM TO FEET LIFT

Inch Vac.	Feet	Inch Vac.	Feet	Inch Vac.	Feet	Inch Vac.	Feet
¼	0.28	8¼	9.35	16¼	18.42	24¼	27.50
½	0.56	8½	9.64	½	18.71	½	27.78
¾	0.85	8¾	9.92	¾	18.99	¾	28.07
1	1.13	9	10.21	17	19.28	25	28.35
1¼	1.41	¼	10.49	¼	19.56	¼	28.63
1½	1.70	½	10.77	½	19.84	½	28.91
1¾	1.98	¾	11.06	¾	20.13	¾	29.20
2	2.27	10	11.34	18	20.41	26	29.48
2¼	2.55	¼	11.62	¼	20.70	¼	29.76
2½	2.84	½	11.90	½	20.98	½	30.05
2¼	3.12	¾	12.19	¾	21.27	¾	30.33
3	3.41	11	12.47	19	21.55	27	30.62
3¼	3.69	¼	12.75	¼	21.83	¼	30.90
3½	3.98	½	13.04	½	22.11	½	31.19
3¾	4.26	¾	13.32	¾	22.40	¾	31.47
4	4.54	12	13.61	20	22.68	28	31.75
4¼	4.82	¼	13.89	¼	22.96	¼	32.03
4½	5.11	½	14.18	½	23.24	½	32.32
4¾	5.39	¾	14.46	¾	23.53	¾	32.60
5	5.67	13	14.74	21	23.81	29	32.89
5¼	5 95	¼	15.02	¼	24.09	¼	33.17
5½	6 23	½	15.31	½	24.38	½	33.46
5¾	6.52	¾	15.59	¾	24.66	¾	33.74
6	6.80	14	15.88	22	24.95	30	....
6¼	7.08	¼	16.16	¼	25.23	....	....
6½	7.37	½	16.45	½	25.51	....	....
6¾	7.65	¾	16.73	¾	25.80	....	....
7	7.94	15	17.01	23	26.08	....	....
7¼	8.22	¼	17.29	¼	26.36	....	....
7½	8.50	½	17.57	½	26.65	....	....
7¾	8.79	¾	17.86	¾	26.93	....	....
8	9.07	16	18.14	24	27.22	....	....

To convert inches vacuum into feet, multiply by 1.13.

Where the discharge pipe is smaller in diameter than the inlet pipe, the difference between the velocity heads in both pipes should be added to the other readings given above in order to arrive at the total hydraulic load.

The difference in velocity heads in the inlet and discharge pipes should be subtracted from the sum of the other readings given, if the inlet pipe be smaller than the discharge pipe where the gauges are connected. In the example just given, the friction head in the inlet and discharge pipes is included in the gauge readings.

CHAPTER 8

Rotary Pumps; Principles

By definition a rotary pump is *a positive displacement pump with a circular motion;* a pump whose piston or pistons partake of the nature of cams rotating upon an axis and being in contact at one or more points with the walls of the enclosing chamber.

Ques. How does a rotary pump differ from a centrifugal pump?

Ans. The rotary pump continuously scoops the fluid from out of the pump chamber whereas a centrifugal pump imparts velocity to a stream of fluid.

The action of a rotary pump approaches a positive nature whereas the centrifugal pump does not.

Classification.—There are two general kinds of rotary pump classed with respect to the impelling element as:

1. Gear

2. Blade or bucket

The gear pump may be classed as:

1. Spur
   a. External
   b. Internal

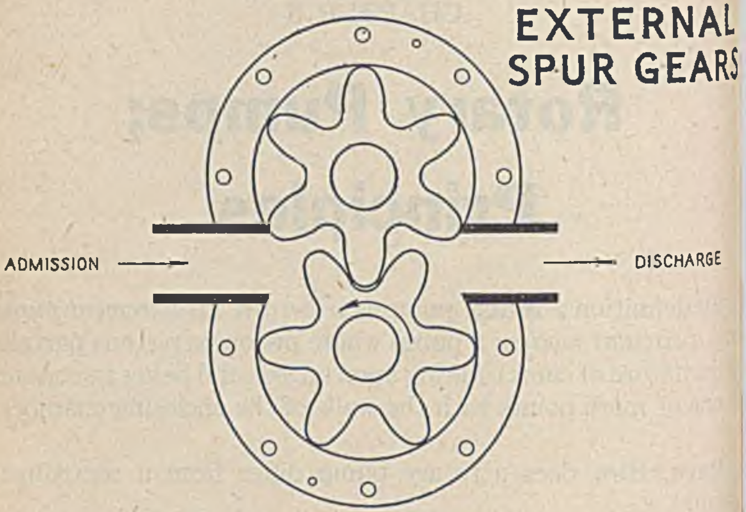


Fig. 1.—External spur gear pump.

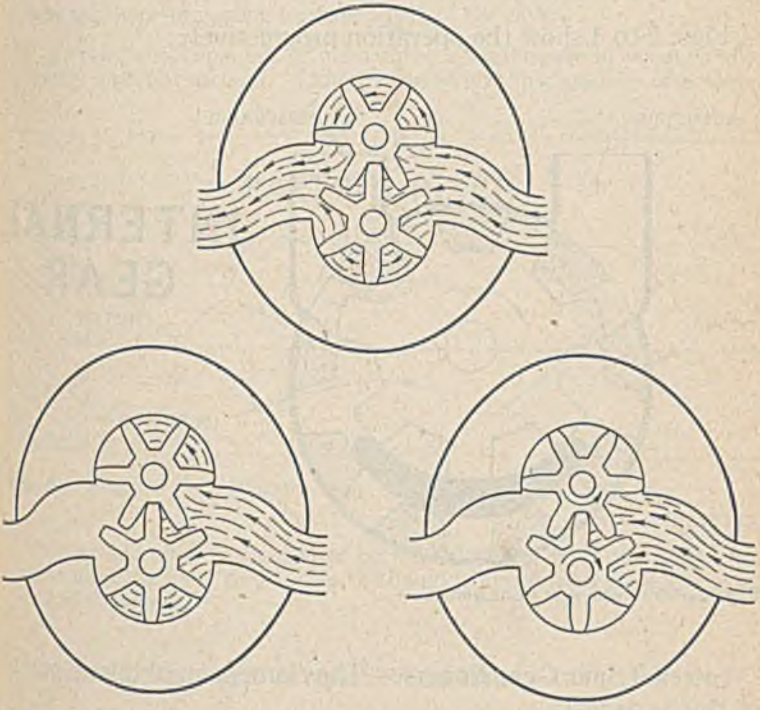
2. Helical
   a. Partial pitch
   b. Screw

With respect to the blade or bucket class rotary pumps various types are included as:

1. Rotating-reciprocating blades
2. Rotating-oscillating buckets

Spur Gear Rotary Pumps.—There are two types, the external gear and the internal gear.

Fig. 1 shows the essential elements of the external gear



Figs. 2 to 4.—Operation of external spur gear pump. Fig. 2, liquid entering through admission opening; fig. 3, liquid being carried between the gear teeth toward discharge side; fig. 4, liquid being forced out of pump.

pump. There are only three parts, two external gears and a casing.

In operation, as each pair of meshing teeth separate, a space forms with vacuum and atmospheric pressure forces in the liquid to fill these spaces.

Any liquid which fills the space between any two adjacent teeth must follow along with them as they revolve and be forced out of the discharge opening since the meshing of the teeth during rotation forms a seal separating the admission and discharge parts of the secondary chamber.

Figs. 2 to 4 show the operation progressively.

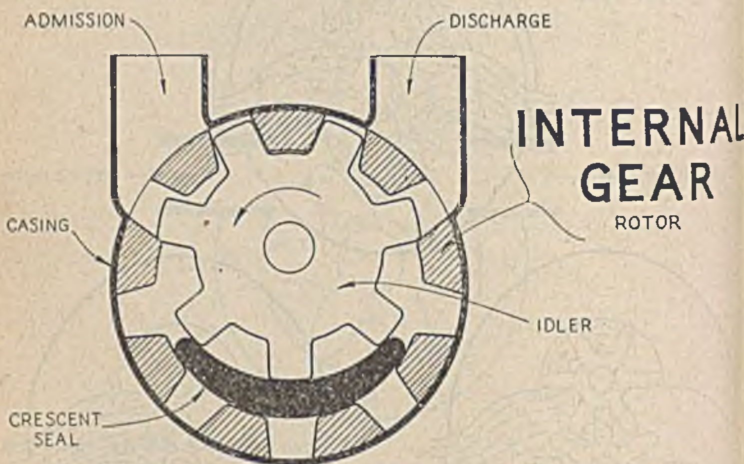


Fig. 5.—Internal spur gear pump.

Internal Spur Gear Pumps.—The elements making up pumps of this type are:

1. Internal gear (rotor).
2. Idler
3. Crescent seal
4. Casing

Fig. 5 shows the assembly of these elements. The pumps work on the "*internal gear principle.*"

In operation (see fig. 5) power is applied to the rotor and transmitted to the idler gear with which it meshes. The space between the outside diameter of the idler and the inside diameter of the rotor is *sealed* by a crescent shape projection forming a part of the cover.

As the teeth come out of mesh there is an increase in volume, which creates a partial vacuum. Liquid is forced into this space by atmospheric

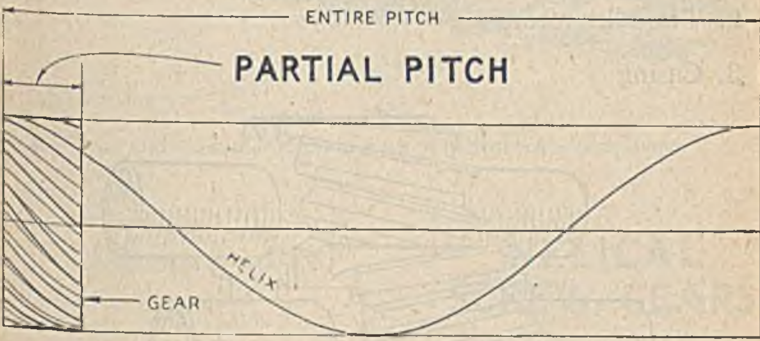


Fig. 6.—Diagram illustrating the term "partial pitch."

pressure and stays in the spaces between the teeth both of the rotor and idler until the teeth mesh when the liquid is forced from these spaces and out of the pump.

Ques. Is this pump reversible and how?

Ans. Yes, when provision is made for swinging the crescent through 180°.

Partial Pitch Helical Gear Pumps.—A variation of this type is the external partial pitch helical gear pump whose operation follows the same principles as just described.

Fig. 6 illustrates what the author means by the term "partial pitch" and needs no further explanation.

The general appearance of a pair of these partial pitch gears in mesh is shown in fig. 7.

Helical Screw Gear Pumps.—The elements of a pump of this type are:

1. Power rotor
2. Two idler rotors
3. Casing

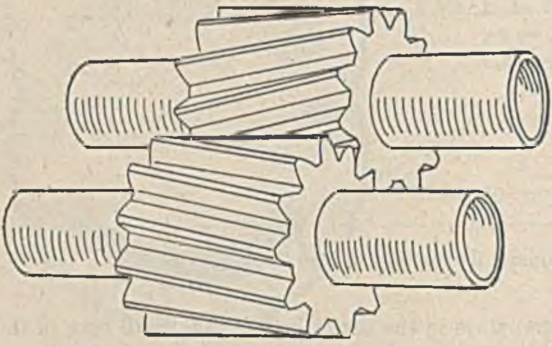


Fig. 7.—Appearance of a pair of partial pitch helical gears in mesh.

The three rotors are shown in fig. 8 before assembling in the case, and assembled complete in fig. 9.

As shown in fig. 8, convex surfaces of the power rotor mesh with concave surfaces of the sealing rotors in such a way that practically fluid tight closures between the rotors are obtained.

In these pumps the standard direction of rotation is clockwise when the observer is standing at the driver and looking toward the coupling end of the pump.

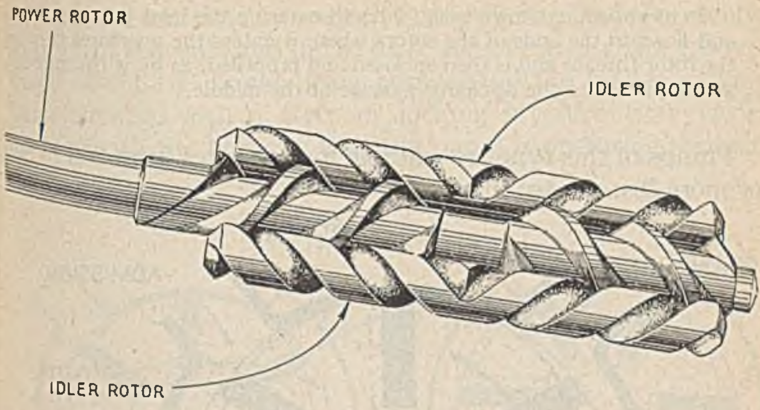


Fig. 8.—Power rotor and idler rotors in mesh of helical screw gear pump.

HELICAL SCREW GEARS

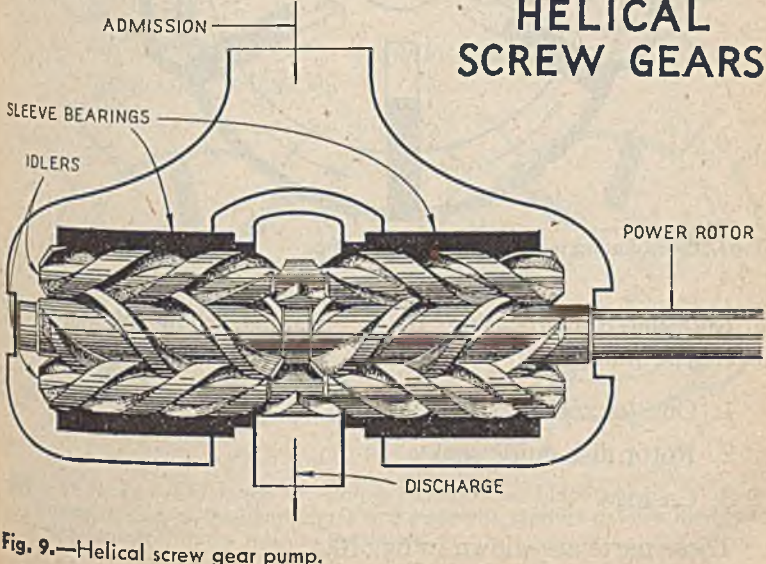


Fig. 9.—Helical screw gear pump.

In operation, as shown in fig. 9 liquid entering the inlet passage divi and flows to the ends of the rotors where it enters the apertures betw the rotor threads and is then enclosed and propelled, as by a continuo acting piston, to the discharge passage at the middle.

Pumps of this type are suitable for pressures from 200 to 5 or more lbs. per sq. in.

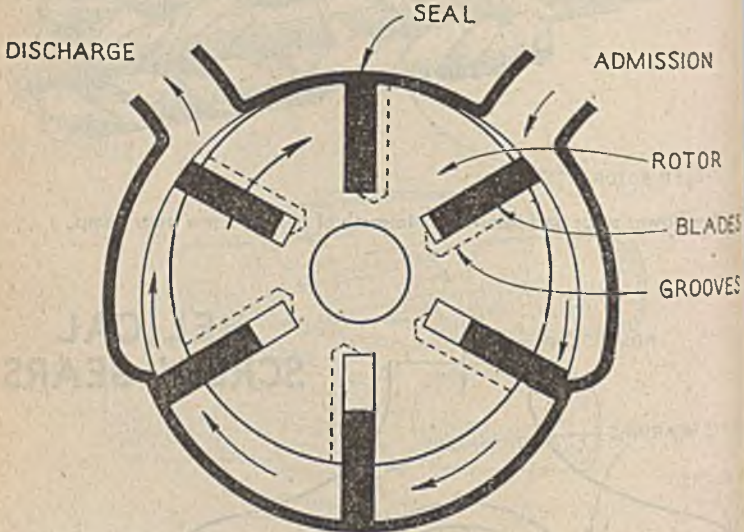


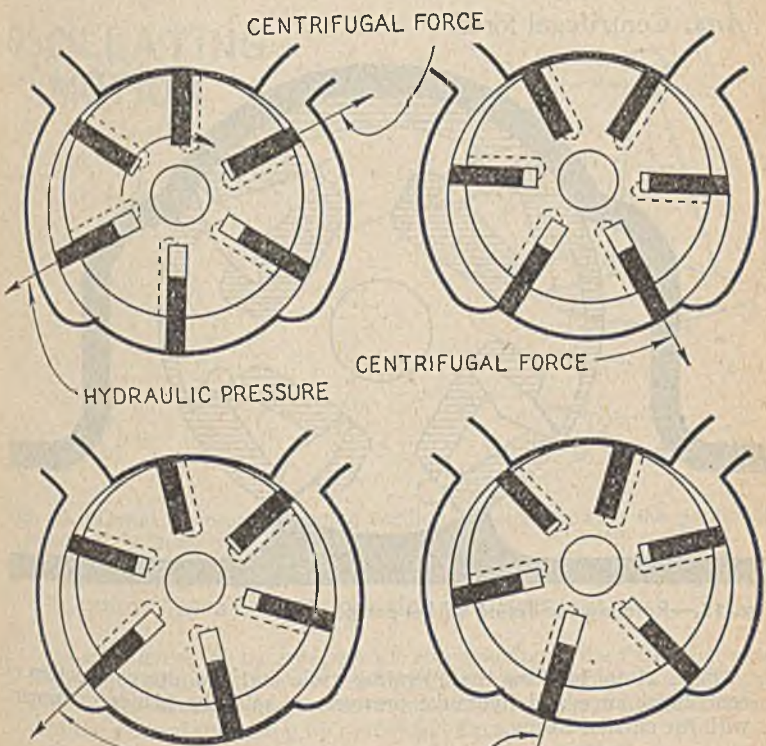
Fig. 10.—Rotating reciprocating blade pump.

Rotating-Reciprocating Blade Pumps.—The elements o this type pump are:

1. One (or more) pairs of blades
2. Rotor disc guide seal
3. Casing

These parts are shown in fig. 10.

The rotor has slots into which fit the sliding blades or vanes. In front of the slots and in the direction of rotation are grooves which admit the liquid being pumped by the blades moving them outward with a force or locking pressure that varies directly with the pressure that the pump is operating against.



Figs. 11 to 14.—Operation of rotating-reciprocating blade pump. Note the alternate action of centrifugal force and hydraulic pressure to keep the blades in firm contact with the walls of the casing.

Ques. What other operation takes place on account of the grooves?

Ans. They serve to break the vacuum on the admission side.

Ques. What causes the blades to move outward while traversing the admission zone?

Ans. Centrifugal force.

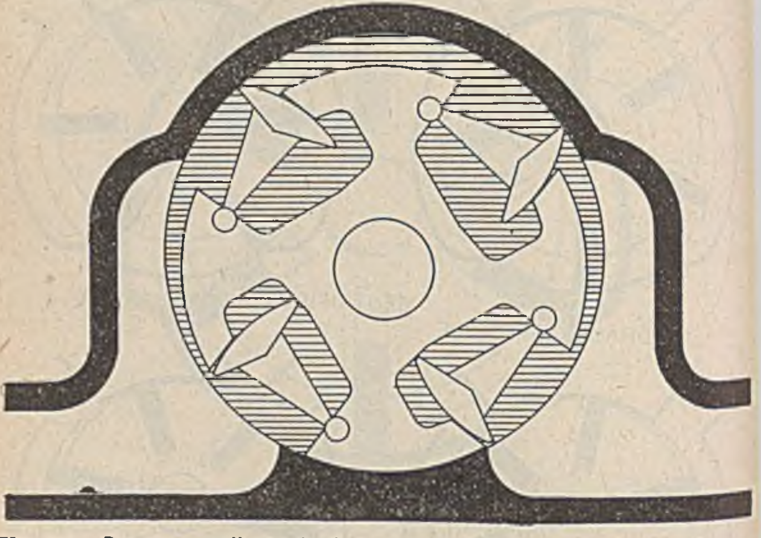


Fig. 15.—Rotating-oscillating bucket pump.

Figs. 11 to 14 show the operating cycle and the alternate action of centrifugal force and hydraulic pressure to hold the blades in contact with the circular casing.

Rotating-Oscillating Bucket Pumps.—These pumps operate virtually on the same principle as the rotating-reciprocating blade pump just described. The only differences are the use of buckets instead of blades and of centrifugal force alone to keep them in firm contact with the casing wall instead of a combination of alternating acting hydraulic pressure and centrifugal force.

The essentials of the rotating-oscillating bucket pump are shown in fig. 15.

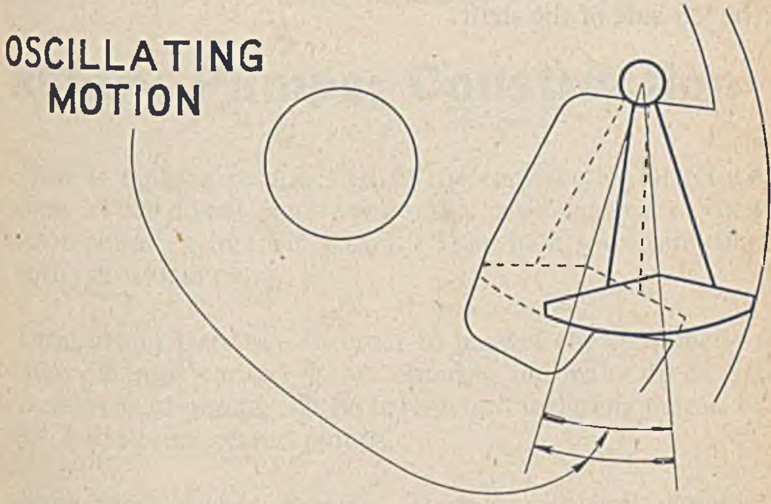


Fig. 16.—Detail of rotor of rotating oscillating pump showing the oscillating motion of the buckets.

The pump operates with positive displacement.

In operation as the buckets pass the admission port, the vacuum created fills the space behind them and the liquid is carried around to the discharge outlet. The buckets are pivoted as shown and are kept in firm contact with the walls of the casing by centrifugal force.

The pumps are usually equipped with tight and loose pulleys. Built in relief valves are used on some types; on others an external relief valve is provided. In every case the by passing is done outside the pumping chamber to eliminate end thust on the working parts of the pump.

These pumps are not reversible. Direction of rotation is given from the drive end of the shaft. For clockwise, or right hand rotation, the intake connection is on the right side of the shaft. For counter-clockwise, or left hand rotation, the intake is on the left side of the shaft.

CHAPTER 9

Rotary Pumps; Construction

There is nothing complicated in the construction of rotary pumps. They do not require any piston reversing gear as they are non-pulsating in their action. They look something like a small centrifugal pump.

Construction Details.—In order to present the construction of rotary pumps various details entering the make up of the several types of pumps will be here given, including details of blade, bucket and geared pumps.

Blade Type Rotary Pumps.—There are several varieties of the blade pump depending upon the method employed to keep the blade ends in contact with the cylinder walls during rotation.

Various means are used to hold the blades in contact with the walls as by:

1. Hydraulic pressure
2. Centrifugal force
3. Springs
4. Cylinder Bore.

Hydraulic Pressure Type.—In the first pump presented, the pressure of the liquid being pumped holds the blades against the cylinder walls or lining.

A view of the casing or "cylinder" is shown in fig. 1. It is a heavy casting machined for alignment and tested at 500 lbs. hydrostatic pressure.

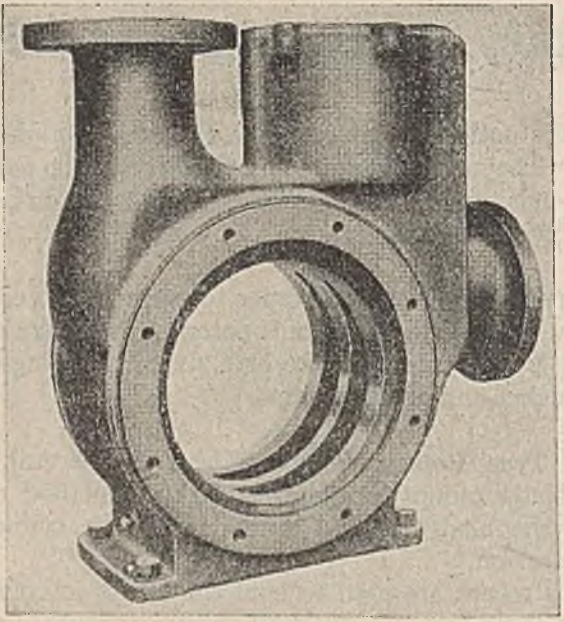


Fig. 1.—Hydraulic pressure blade type pumps 1: Casing or cylinder construction.

Liner.—The cylinder is usually fitted with a cast iron or bronze liner held in place by a key in the body which locates the admission and discharge ports.

Rotor.—This also is either of cast iron or bronze and has six machined slots which contain the sliding blades as shown in fig. 2.

In front of the slots, and in the direction of rotation, are grooves which admit the liquid being pumped under the vanes, moving them outward with a force or locking pressure that varies directly with the pressure that the pump is operating against.

These grooves not only regulate the locking pressure of the sliding blades against the liner but also break the vacuum on the inlet side, allowing the blades to move out freely by centrifugal force.

Blades.—These are of composition and weighted with metal to increase centrifugal force on the inlet side and make them drop more readily on the discharge side.

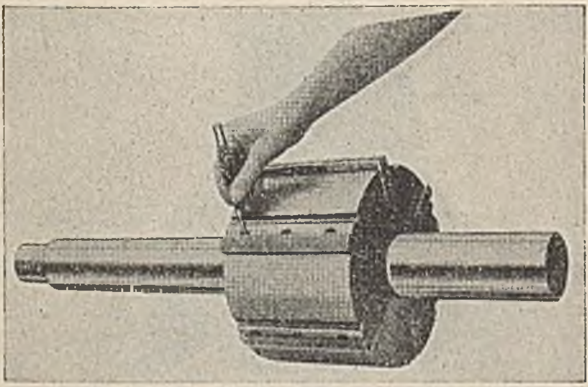


Fig. 2.—Hydraulic pressure blade type pumps 2: Rotor construction.

End Plates.—The appearance of an end plate is shown in fig. 3. They are of cast iron or bronze and bear against the ends of the liner being machined to produce the proper clearance between the rotor and end plates.

Heads.—These are also shown in fig. 3. They are of cast iron and are located by a dowelled flange on the body proper; they are held in place by cap screws.

Bearings.—These are located in extension on the heads and are of the sleeve type made of a so called anti-friction metal.

Sufficient clearance is allowed between the shaft and the bearing to permit the liquid being pumped to circulate through the bearings into the inlet side of the pump.

The object of this is not only to cool the bearings, but the inlet pressure on the outer ends of bearings created by the vacuum eliminates the un-balancing of the rotating element and permits packing the pump against vacuum only, instead of pressure.

On some pumps ball or roller bearings are provided, one on the outboard end of each head instead of the sleeve type just described.

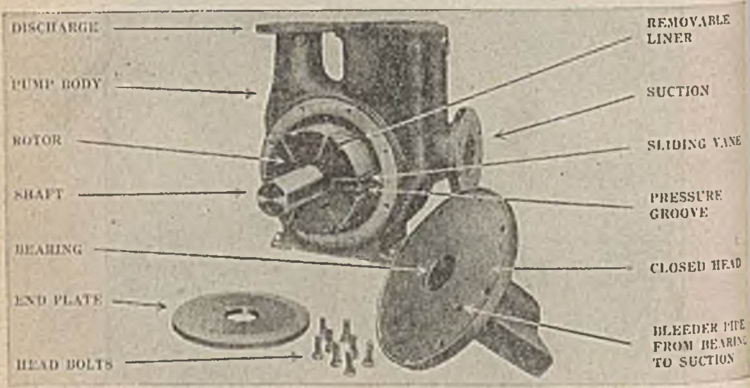


Fig. 3.—Hydraulic pressure blade type pumps 3: Disassembly view showing various details with names of parts.

This construction is desirable where it is required to keep the liquid being pumped from contact with the bearings. Two adjustable stuffing boxes are provided on each—one on the inboard end and one on the out-board end of each head between the pump body and the bearing.

By-Pass Relief Valve.—On some pumps it is desirable to have a relief valve. In order to prevent chatter, the relief valve is of the pop type and pressure adjustment is made by a set screw and lock nut.

A sectional view of a relief valve as built into the pump is shown in fig. 4. The various parts are numbered and their names given in the text underneath the cut.

Bucket Type Pumps.—The particular pump here described depends upon centrifugal force to swing out the buckets and cause them to ride lightly against the cylinder walls during the revolution. A pump of this kind designed for hand operation is shown in fig. 5.

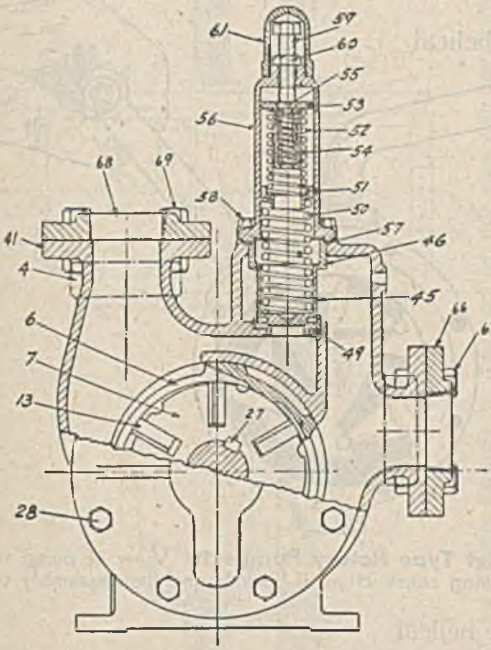


Fig. 4.—Hydraulic pressure blade type pumps 4: Sectional view showing general construction and detail of built in relief valve. The parts are: 45, by-pass valve; 46, valve cage; 49, valve cage gasket; 50, high pressure spring; 51, high pressure spring retainer; 52, intermediate pressure spring; 53, intermediate pressure spring retainer; 54, low pressure spring; 55, low pressure spring button; 56, valve bonnet; 57, valve bonnet gasket; 58, valve bonnet cap screw; 59, adjusting screw; 60, jam nut; 61, cover nut.

Another example of this pump is one designed as a truck pump as shown in fig. 6, with accompanying list of parts. A detail of the built in relief valve is shown in fig. 7.

Gear Pumps.—There are numerous types of pumps in which the rotor element consists of two gears which mesh with each other and work within a close fitting casing. Among these are the:

1. Spur
2. Single helical

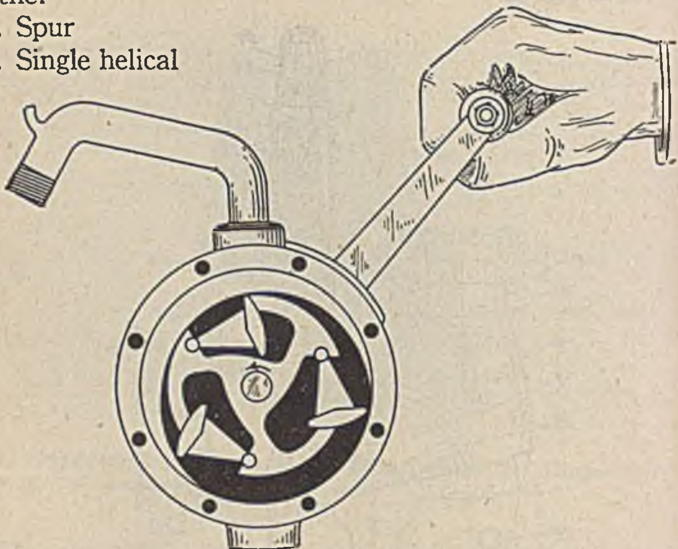


Fig. 5.—Bucket Type Rotary Pumps 1: View of pump with face plate removed showing construction of buckets and their assembly on rotor.

3. Double helical

   a. Short *b*. Long

4. Internal

Spur and Single Helical Gear Pumps.—These two types are practically the same in construction with exception of the

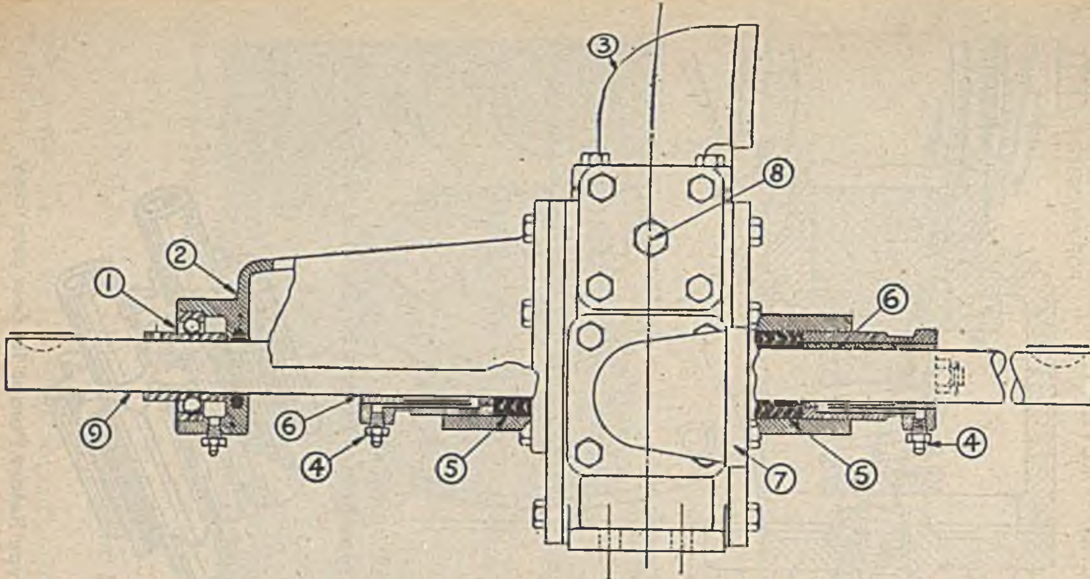


Fig. 6.—Bucket Type Rotary Pumps 2: Truck pattern side view with parts numbered. **The parts are:** 1, ball bearing; 2, outboard bracket; 3, outlet, flanged ell; 4, alemite fittings; 5, chevron packing which seals bearings from pumpage; 6, sleeve bearings; 7, inlet flanged ell; 8, relief valve; 9, shaft.

gears and in the case of the single helical gear pump the provision for absorbing end thrust by means of a thrust collar.

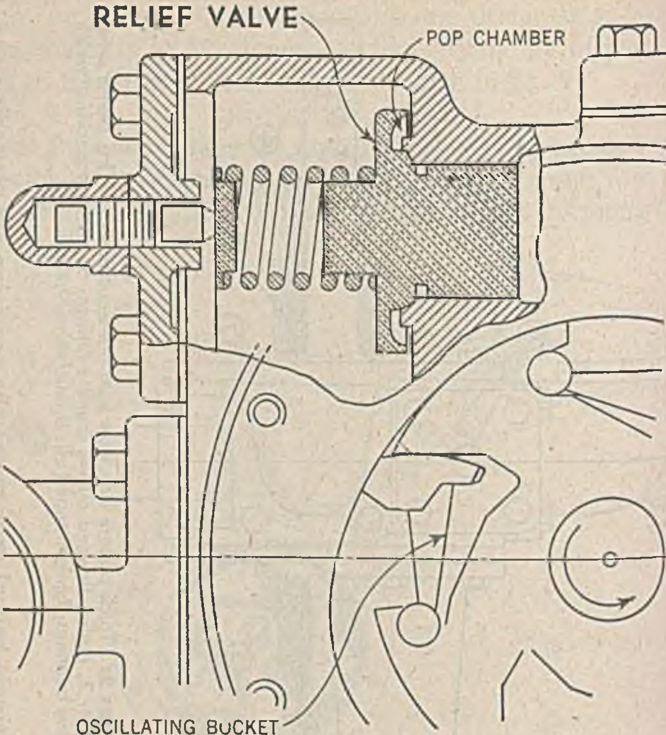


Fig. 7.—Bucket Type Rotary Pumps 3: Detail of relief valve.

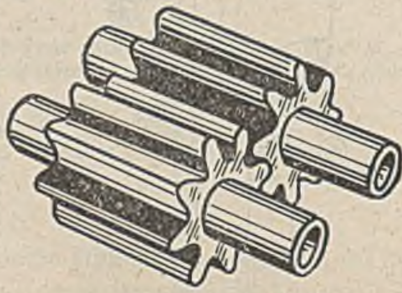


Fig. 8.—Gear Type Rotary Pumps 1: Spur gear rotary.

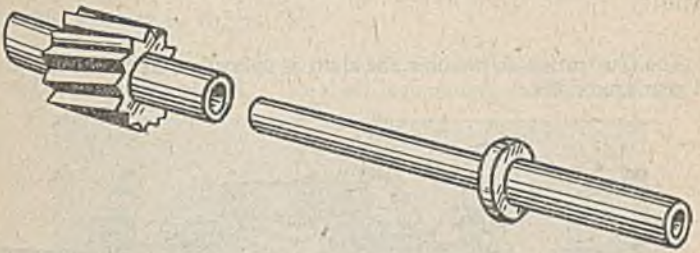


Fig. 9.—Gear Type Rotary Pumps 2: Single helical gear rotor, shaft and thrust collar.

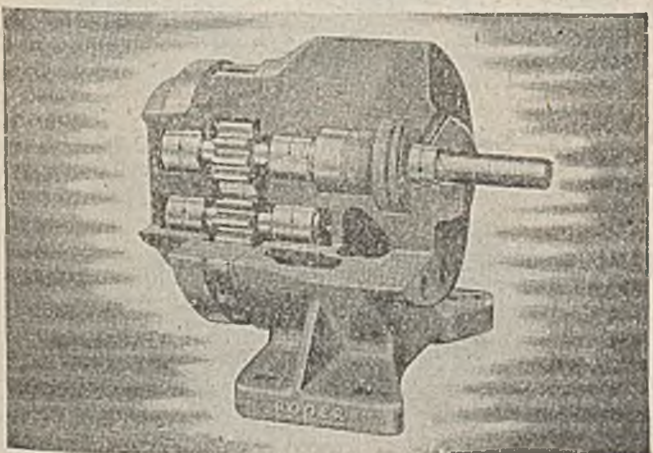


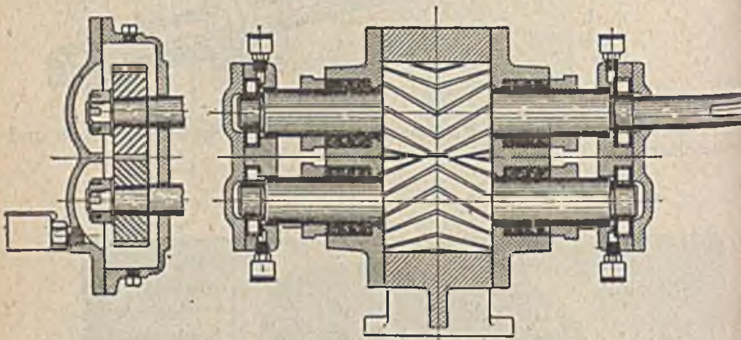
Fig. 10.—Gear Type Rotary Pumps 3: Spur gear rotor assembly in casing.

Figs. 8 and 9 show rotor elements of the two pumps.

The spur gears (fig. 8) are best where maximum volumetric efficiency is required and where noise is not a serious factor.

Fig. 9 shows one single helical gear, shaft and thrust colla
on shaft.

The illustration shows how the shaft is splined to fit broached openin
in gear trunnion.

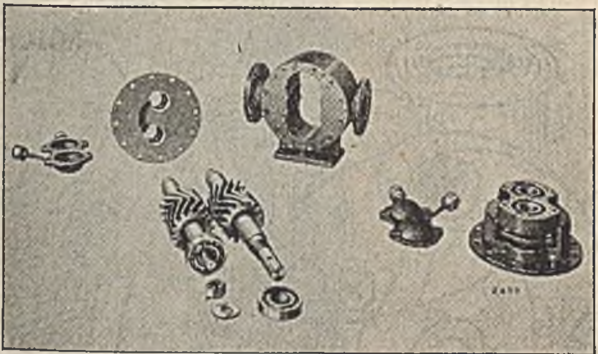


Figs. 11 and 12.—Gear Type Rotary Pumps 4: Doible helical ge
rotor assembly in casing. Shafts are mounted on single ball bearings.

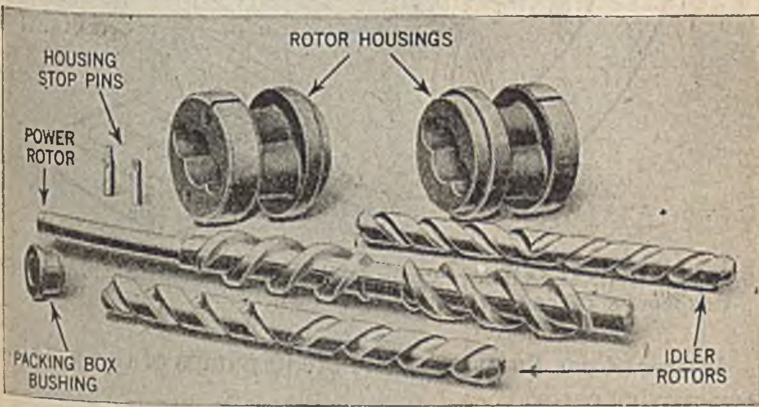
Figs. 13 to 18.—Gear Type Rotary Pumps 5: Disassembly view of doub
helical gear pump with bronze bearings.

The assembly of the two gears of the spur gear pump is shown in the sectional drawing, fig. 10. Various details of construction are here visible.

The case face plate and back plate are made of cast iron. The packing box has a mechanical seal. The shaft is mounted on roller bearings.



Figs. 19 to 24.—Gear Type Rotary Pumps 6: Disassembly view of double helical gear pump with external ball bearings.



Figs. 25 to 32.—Gear Type Rotary Pumps 7: Rotor details, double helical gear screw pattern. These internal parts consist of power and idler rotors, rotor housings and packing box breakdown bushings.

Figs. 11 and 12 show various details of construction of th double helical gear pump.

These pumps are constructed with plain or ball bearing. Figs. 13 to 1 show disassembly view of the plain bearing construction and figs. 19 to 2 the ball bearing type.

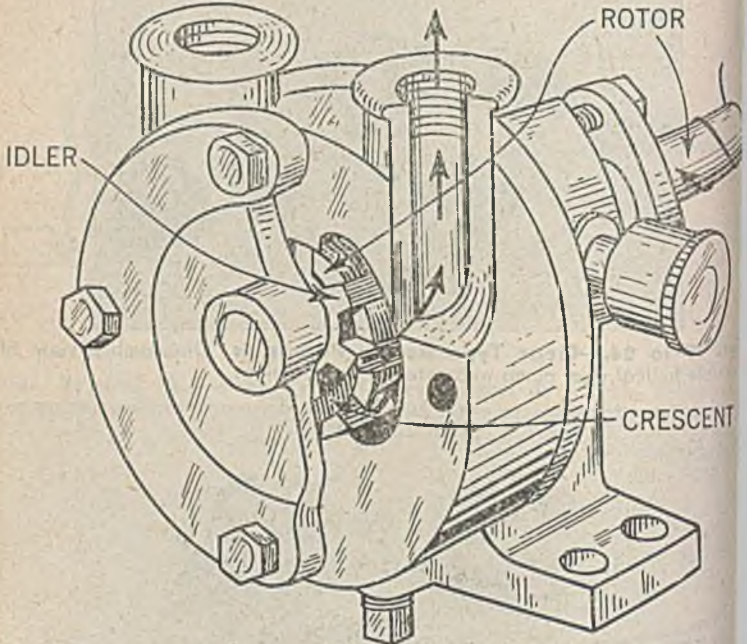
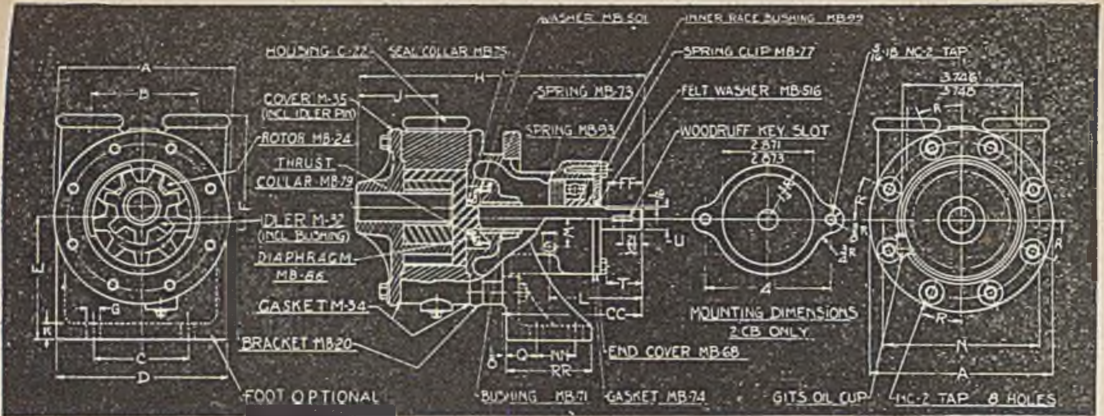


Fig. 33.—Gear Type Rotor Pumps 8: Internal spur gear pattern. Cutaw view showing casing rotor gears and crescent in part.

Internal Gear Pumps.—There are in pumps of this type b two moving parts:

1. Outside annular gear
2. Inside spur gear



Pump	Pipe Tap	A	B	C	D	E	F	G	H	J	K	L	M	N	Q	R	T	U	CC	FF	NN	RR	NC2 Tap	Woodruff Key	Ball Bearing	Ship. Wt. (Lbs.) Pump Only	With Foot
2CB	½	4⅝	2¾	2	4½	2.997 3.000	5¾	11/16	7 5/16	1½	⅝	3 5/16	11/16	—	11/16	—	1½	⅝	4 11/16	1¼	15/16	2⅜	—	#7	204-P	20	26
3CB	¾	6	3½	3	5½	3.872 3.875	7½	11/16	8 11/16	1 11/16	½	3 5/16	11/16	5½	1	22½°	1½	¾	4 11/16	1¼	1¼	2¾	5/16-18	#9	206-P	28	34
4CB	1	6	3½	3	5½	3.872 3.875	7½	11/16	9 1/16	2⅛	½	3 1/16	11/16	5½	1	22½°	1½	¾	4 11/16	1¼	1¼	2¾	5/16-18	#9	206-P	31	37
5CB	1¼	7½	4¾	3	5½	4.750 4.753	9⅛	11/16	11 11/16	3⅛	⅝	4⅛	1⅛	6½	1	17°	2	1	6⅛	1⅝	1¼	2¾	⅜-16	#15	206-P	52	62
6CB	1½	7½	4¾	3	5½	4.750 4.753	9⅛	11/16	12⅛	3⅛	⅝	4⅛	1⅛	6½	1	17°	2	1	6⅛	1⅝	1¼	2¾	⅜-16	#15	206-P	56	66

Figs. 34 to 37.—Gear Type Rotary Pumps 9: Working drawing of internal gear pump showing details of construction and various data required in building. These working drawings and the table illustrate the method of making one set of drawings answer for numerous sizes. The letters on the drawings refer to letters in the dimension tabulation herewith given.

The large gear has a shaft at the back which extends throug the pump casing and carries the driving pulley. The spur i mounted on a stud extending from the opposite side of th casing, which also has a crescent-shaped projection on one side dividing the space between the two gears.

An example of this construction is shown in fig. 33.

The pump will operate in either direction. The illustration shows th path of the liquid for clockwise rotation (facing the shaft end). Th details of construction are shown more clearly in figs. 34 to 37.

CHAPTER 10

Rotary Pumps; Operation

In the installation, operation and maintenance of rotary pumps there are many operations in common with centrifugal pumps. Accordingly in order to avoid useless repetition the reader in numerous instances will be referred to similar operations for centrifugal pumps.

Location

A rotary pump like any other piece of machinery needing occasional attention should be put in an accessible place where there is plenty of light.

Foundation

Since rotary pumps are as a rule much smaller units than the centrifugal type, the foundations are correspondingly smaller. The instructions for centrifugal pump foundations (page 157) will apply here.

Installation

Alignment.—Correct alignment is absolutely essential to successful operation. *A flexible coupling will* **not** *compensate alignment.* Rotary pumping units should be aligned as accurately as if the coupling were solid.

The flexible coupling will then serve its purpose, that is, to prevent the transmission of end thrust from one machine to the other and to compensate for slight changes in alignment which may occur during normal operation.

There are two separate alignments:

1. Factory
2. Field

Every pump is accurately aligned at the factory before being shipped.

Ques. Describe the factory alignment.

Ans. The unit after assembly is accurately aligned by placing the base plate on a surface plate and then leveling the machined pads.

Shims are inserted under the feet of the pump and driver when necessary to obtain perfect alignment.

Ques. What should be noted about base plates?

Ans. All base plates are elastic, and for this reason the manufacturer cannot assume responsibility for the proper mechanical operation of a unit unless the shop alignment is reproduced when the unit is erected on its foundation.

Ques. How are rotary pumps usually shipped?

Ans. On their bed plates.

Ques. In field aligning is it necessary to remove pump and driver from the bed plate while leveling?

Ans. This is very seldom necessary.

Ques. How is the unit placed on the foundation?

Ans. It is supported on the foundation by wedges placed near the foundation bolts.

Ques. What should be removed?

Ans. The paint from the projections of the base plate pads upon which the pump feet are supported.

Ques. How are the wedges under the base plate adjusted?

Ans. Place a spirit level on the pads and adjust the wedges to bring the pump shaft level.

Checking the Alignment.—The alignment should now be checked and corrected so as to bring the driver half coupling in perfect alignment with the pump half coupling.

Ques. How is the alignment checked?

Ans. It is accomplished by the use of a straight edge across the top and sides of the coupling. (See fig. 1, page 159.)

Ques. What is the procedure if the coupling's flanges be not perfectly true, or not of the same diameter?

Ans. Check the alignment at each quarter turn.

If any variation be found, proper allowance must be made in aligning the unit.

Ques. What important adjustment must be made?

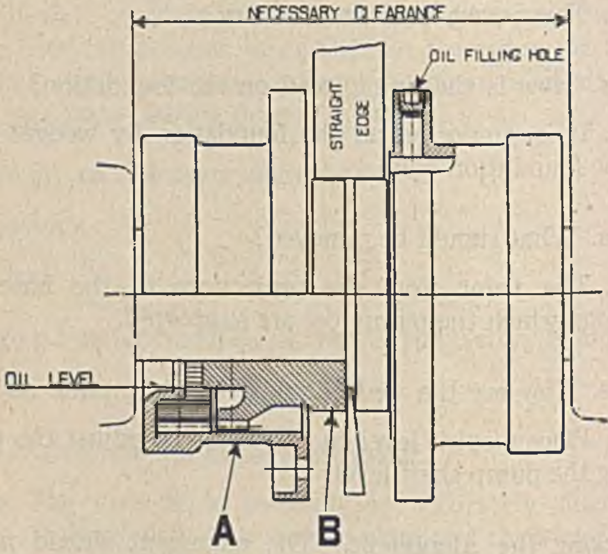


Fig. 1.—Method of lining up coupling with straight edge and wedge.

Ans. The clearances between the coupling halves should be set so that they cannot strike, rub, or exert end thrust on either pump or driver.

When Fast's couplings are provided with rotary pumps, the same care should be exercised in alignment.

Ques. How far must the hubs be separated?

Ans. They must be separated the distance stamped on their alignment faces.

Shafts should then be lined up by using a straight edge and thickness gauge on the alignment faces, as shown in fig. 1.

The flanges of the sleeve should be cleaned and gaskets examined to insure oil seal and then bolted solidly together.

Ques. Mention an important thing to be done before the unit is put into operation.

Ans. Oil should be put in the coupling.

Ques. Is there any more alignment checking to be done and why?

Ans. Yes. Alignment must be checked after the pump has been completely piped up because pumps are frequently sprung and pulled out of position by drawing up flange bolts when the flanges are not squared up before tightening.

Ques. What particular attention should be given to the piping?

Ans. The inlet and discharge piping should be properly supported to prevent a strain or pull on the pump.

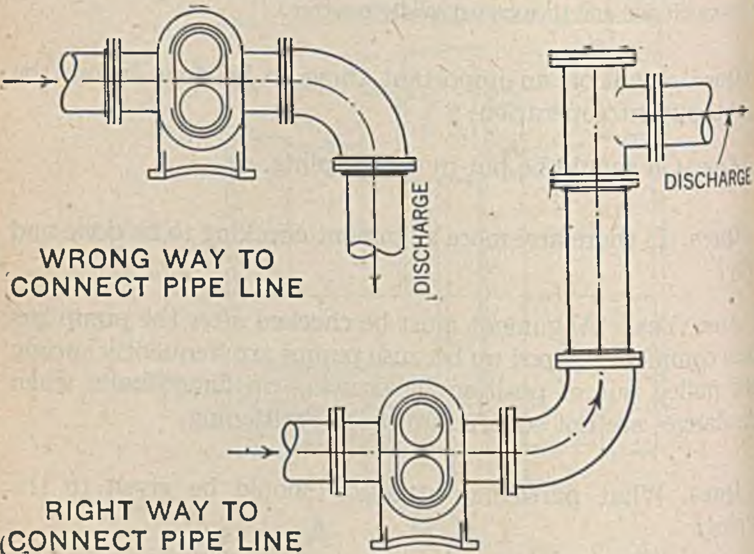
Ques. What frequently causes misalignment, hot bearings, wear and vibration?

Ans. A strain or pull on the pump due to piping which is not supported.

Grouting.—See page 164.

Piping; Inlet and Discharge.—The general requirements for first class installations are similar to those for centrifugal pumps. (See pages 164 to 166).

Ques. How should the inlet piping be installed for pumping highly volatile liquids such as butane, propane, hot oils, etc.?



Figs. 2 and 3.—Wrong and **right** way to install discharge piping.

Ans. There should be sufficient static negative lift (static head) on the inlet line in addition to the vapor pressure to prevent vaporization of the liquid within the pump.

Ques. What should be noted with respect to the discharge piping?

Ans. Always carry the discharge piping up through a riser approximately five times the diameter.

Note right and wrong way as shown in figs. 2 and 3.

Ques. What is the object of the hook up shown in fig. 3.?

Ans. It prevents gas or air pockets in the pump and will act as a seal in high vacuum service.

A valve on the top of the riser may be used as a vent when starting the pump.

Ques. How is the pump protected against excessive pressures caused by increased pipe friction in cold weather or accidental closing of valve in discharge line?

Ans. A by pass line with a relief valve may be installed.

Ques. How is this relief valve adjusted?

Ans. It is set slightly higher than the maximum pump discharge pressure, but not more than 10%.

Jacket Piping.—If steam jackets be supplied, the inlet is located on the top and outlet on the bottom. On water jackets the inlet is at the bottom and the outlet at the top. Valves should be installed in the inlet lines to regulate the quantity of jacket supply.

Stuffing Boxes.—See instructions for Centrifugal Pumps.

Ques. What kind of lubricant should be used to seal the lantern gland when pumping gasoline, propane and similar liquids?

Ans. Non-soluble grease.

Priming.—Before starting up for the first time, prime the discharge side of the pump thoroughly through the opening provided for the purpose and oil.

Ques. How are rotary pumps run and tested at the factory?

Ans. The regular practice is to leave the oil in the pump to protect the internal mechanism against corrosion.

If this oil will be detrimental to the system, it will be necessary to disassemble the pump, clean all parts thoroughly and fill the pump with the liquid to be pumped.

Direction of Rotation.—This varies according to type. For instance taking the double helical design, the direction of rotation is as follows: Standard direction of rotation is counter-clockwise when standing at and facing the shaft extension end.

This is indicated by an arrow on the pump body.

Rotation of internal roller bearing pumps may be reversed by removing the outside bearing cover and stuffing box and transferring the small plug in the side plate casting to the opposite side. These plugs (one in each side plate) should be on the discharge side to induce circulation through the bearings to the inlet, and to maintain inlet pressure on the stuffing box and ends of the drive shafts.

Another pump working on the internal gear principle makes note of the following directions:

1. In determining desired rotation, the observer stands at the shaft end of the pump.
2. Note that the balancing groove in the shoe must be on the inlet side.
3. If change in direction of rotation be desired, it is necessary only to remove the cover, slip out both the top and bottom shoes, turn them end for end so that the grooves will be on the new inlet side, and reassemble the pump.

Another model of the same make is listed as *automatic reversing pump*. It has a unidirectional flow regardless of direction of shaft rotation and without the use of check valves.

Finally a rotary pump of the helical gear type, the instructions for determining the direction of rotation are as follows: To determine direction of rotation, stand at the driving end facing the pump.

If the shaft revolve from left to right, the rotation is clockwise. If the shaft revolve from right to left, the rotation is counter-clockwise.

The diagram fig. 4, is fundamental of flow of liquid in this pump.

Changing direction of rotation of pump drive shaft reverses

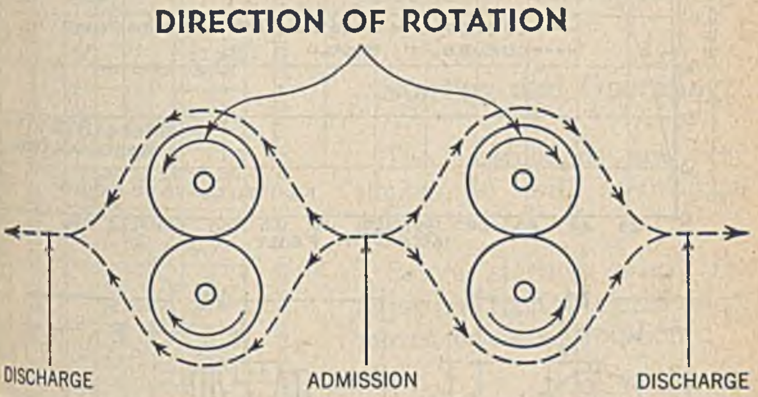
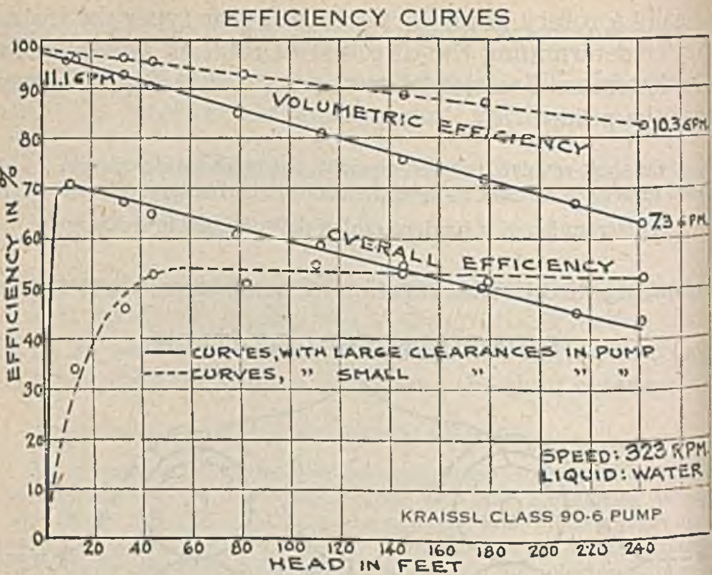


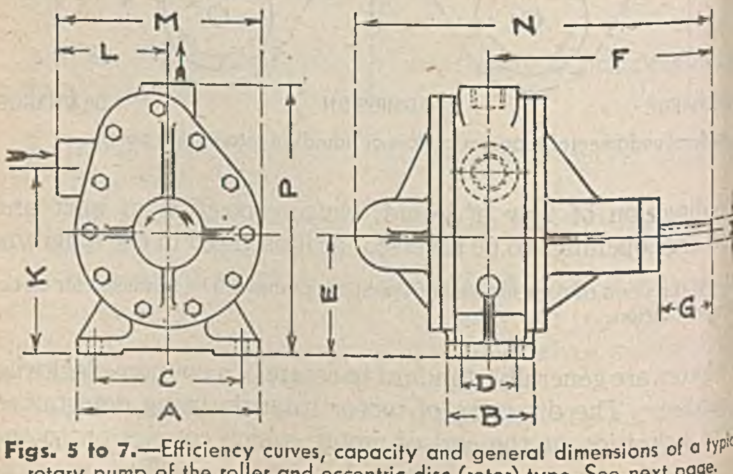
Fig. 4.—Fundamental diagram of flow of liquids in rotary gear pumps.

the direction of flow of liquid, causing position of inlet and discharge openings to be reversed, as illustrated in the diagram.

If direction of rotation be not specified, pump will be furnished for clockwise rotation.

Motors are generally standard to rotate in a counter-clockwise direction. The direction of motor rotation being determined from a position at the end of motor, which couples on to the pump.





Figs. 5 to 7.—Efficiency curves, capacity and general dimensions of a typical rotary pump of the roller and eccentric disc (rotor) type. See next page.

TABLES TO ACCOMPANY FIGS 6 AND 7

CAPACITY AND GENERAL DIMENSIONS

PUMP No.	Pipe Size	Capacity G.P.M	A	B	C	D	E	F	G	H	K	L	M	N	P	Mounting Holes
90-4* 90-5	1/2	5	5	2 3/8	3 [illegible]	1 1/2	3 1/4	7 3/16	1 3/4	1	5	3	5 9/16	11 7/16	7 3/8	11/32
90-6* 90-7	3/4	10	5	4 1/4	3 [illegible]	3 1/4	3 1/4	8 1/8	1 3/4	1	5	3	5 9/16	13 7/16	7 3/8	11/32
90-8* 90-9	1	15	5	6 1/4	3 [illegible]	5	3 1/4	9	1 3/4	1	5	3	5 9/16	15 7/16	7 3/8	11/32
90-10* 90-11	1 1/4	20	6 1/4	4 3/4	5	3 1/2	4 1/4	9	2 1/2	1	6 3/4	4	7 3/8	14 1/4	9 5/8	3/8
90-12* 90-13	1 1/2	35	6 1/4	8	5	6 1/2	4 1/4	11 1/2	2 3/4	1 1/4	6 3/4	4	7 3/8	18 7/8	9 5/8	3/8

* EVEN NUMBERS - BRONZE CONSTRUCTION   ODD NUMBERS - IRON CONSTRUCTION

From the foregoing it is seen that for the various types of pumps direction of rotation should be ascertained from manufacturer's instructions. It will be seen that some rotary pumps are reversible and some are not. An example of a non-reversible pump is the "bucket" pump.

Starting and Operating

The instructions are very similar as with centrifugal pumps.

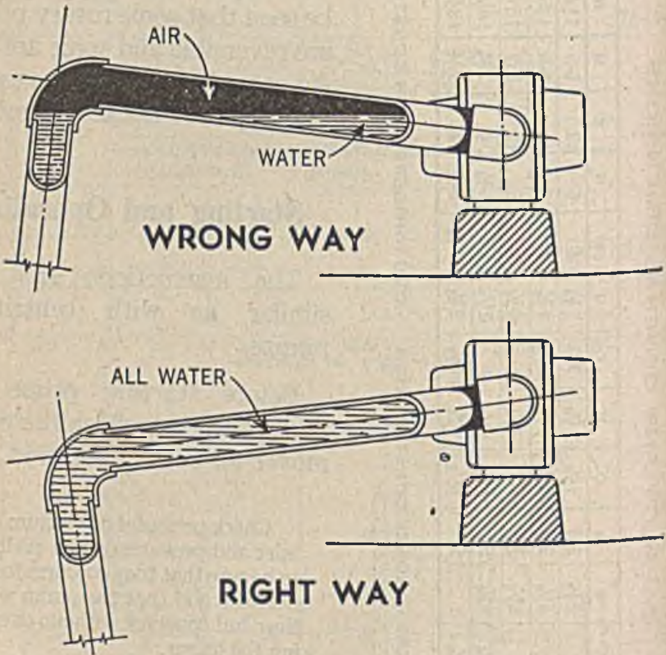
Before starting prime the pump and then check the prime mover for correct rotation.

Check pressures or vacuum on the inlet and pressure on the outlet side to be sure that they conform to specifications and that the pump will deliver full capacity without overloading the driver.

It is advisable to start operation at reduced load gradually increasing to maximum service conditions.

Ques. What attention is required while operating?

Ans. External bearing pumps require occasional lubricatio of soft grease in the bearings. If no grease fittings be furnishe on internal bearing pumps, no attention for lubrication i necessary.



Figs. 8 and 9.—Wrong and **right** way to install inlet line.

Troubles

Rotary pumps like centrifugal pumps require little attentio while running, but if what little attention were given instea of (as usual) none at all, a lot of the troubles tabulated here would not be encountered.

Some of the most frequent causes of trouble are here given. The operator can often avoid unnecessary expense by careful consideration of the points outlined:

No liquid delivered.

1. Stop pump immediately.

2. Pump not primed.

   Prime according to instructions.

3. Lift too high.

   Check with vacuum gauge on inlet. If too great lower position of pump and increase size of inlet pipe. Check inlet line for air leaks.

4. Wrong direction of rotation.

Not enough liquid delivered.

1. Air leaks in inlet line or through stuffing box.

   Oil and tighten stuffing box gland. Paint inlet pipe joints with shellac.

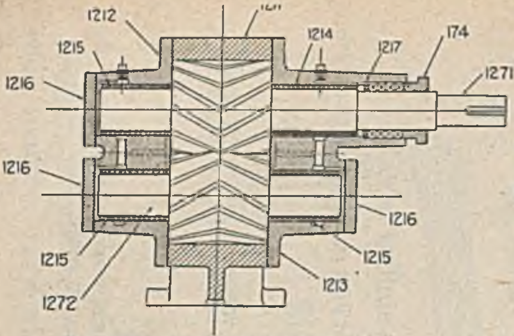
2. Speed too low.

   Check *r.p.m.* Cause may be due to low voltage or low steam pressure. Driver may be overloaded.

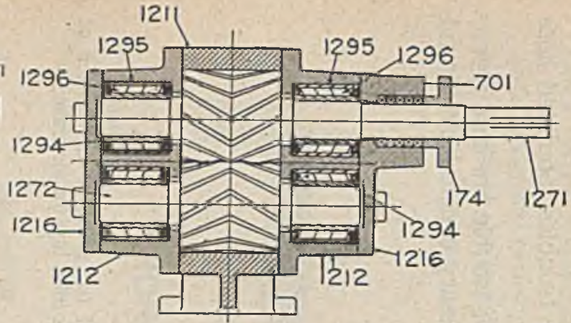
3. Lift too high.

   Light fractions in some liquids vaporize easily and take up part of the displacement of the pump. Check with vacuum gauge.

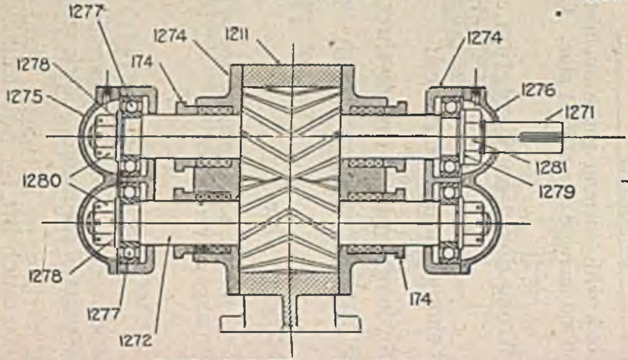
4. Too much lift for hot liquids.



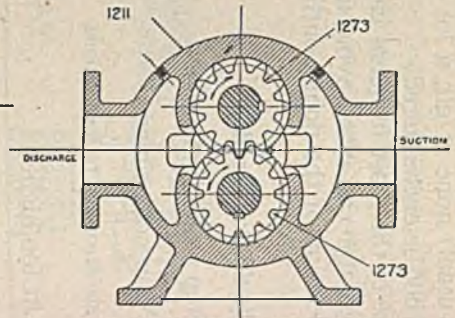
PUMP WITH INTERNAL BRONZE BEARINGS



PUMP WITH INTERNAL ROLLER BEARINGS



PUMP WITH EXTERNAL BALL BEARINGS



SECTION THROUGH IMPELLER GEARS

Figs. 10 to 13.—Sectional views of a double helical gear rotary pump with parts numbered to correspond

NAMES OF PARTS FOR FIGS 10 TO 13

No.	Name of Part	No.	Name of Part	No.	Name of Part	No.	Name of Part
1211	Pump body	1217	Stuffing box ring	1275	Bearing cover (outer)	1281	Nut for driving shaft
1212	Outer side plate	174	Stuffing box gland	1276	Bearing cover (drive side)	174	Stuffing box gland
1213	Driving side plate	1271	Driving shaft	1277	Ball bearings	701	Stuffing box and end plate
1214	Long bearing bushing	1272	Driven shaft	1278	Small shaft washer	1294	Inner race
1215	Short bearing bushing	1273	Impeller gear	1279	Large shaft washer	1295	Outer race
1216	Bearing cap	1274	Ball bearing side plate	1280	Castellated nuts	1296	Roller assembly

SPARES OR REPAIRS may be ordered directly from the Holyoke Works, or through Sales Offices. In ordering, the following information should be given:

Size and serial number of pump
Name and number of each part wanted
Quantity of each part wanted
Diagram number (116228)

SERIAL NUMBER OF PUMP is stamped on name plate; also on upper rim of driving side plate, 1213.

5. Pump may be worn.

6. Foot valve not immersed deeply enough.

7. Foot valve too small or obstructed.

8. Piping improperly installed permitting air or gas to pocket in pump.

9. Mechanical defects:

Pump damaged or packing defective.

Pump delivers for a while, then quits.

1. Leaky inlet line.

2. End of inlet valve not immersed deeply enough.

3. Air or gases in liquid.

4. Supply exhausted.

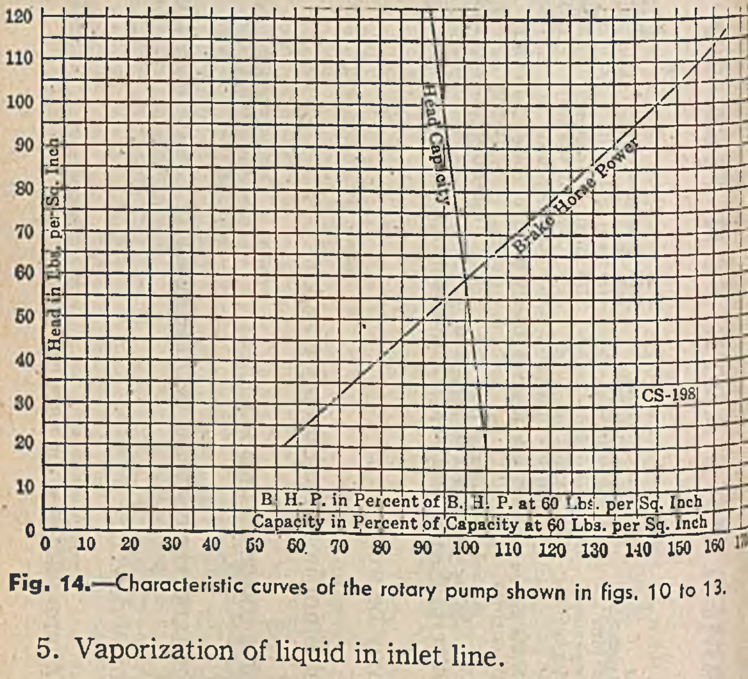


Fig. 14.—Characteristic curves of the rotary pump shown in figs. 10 to 13.

5. Vaporization of liquid in inlet line.

Check with vacuum gauge to be sure that pressure at the pump is greater than vapor pressure of liquid.

6. Air or gas pockets in inlet line.

7. Pump cut by sand or other abrasives in liquid.

Rapid wear.

1. Grit or dirt in liquid pumped.

Install a fine mesh strainer or filter in inlet.

2. Pipe strain on pump casing.

Causes working parts to bind. Release pipe connections and check alignment.

3. Pump operating against excessive pressure.

4. Corrosion roughens surfaces.

5. Pump running dry or with insufficient liquid.

Pump takes too much power.

1. Speed too high.

2. Liquid heavier or more viscous than water.

4. Mechanical defects.

Shaft bent.

Rotating element binds.

Stuffing boxes too tight.

Misalignment due to improper connections to pipe lines or installing on foundation causing spring in base.

5. Misalignment of coupling (direct connected units).

Noisy operation.

1. Insufficient supply.

Liquid vaporizing in pump. Lower position of pump and increase inlet pipe size.

2. Air leaks in inlet.

   Air will cause a crackling noise in pump.

3. Air or gas pockets in inlet.
4. Pump out of alignment.

   Causes metallic contact between rotors and casing.

5. Operating against excessive pressure.
6. Coupling out of balance.

CHAPTER 11

Rotary Pumps; Calculations

A few calculations are here given illustrating how to figure size, horse power, lift, head, total load, etc.

To Determine Proper Size Pump.—In any installation it is important to calculate the proper size pump, in order to avoid putting in a pump too small or too large for the job.

The following example will illustrate the method:

Example.—In a certain installation an 8,000 gallon tank must be filled in two hours to meet the full withdrawal demand. What size pump is required?

Since pumps are rated in gallons per minute, reduce 8,000 gallons in two hours to gallons per minute, thus

$$\text{gals. per min.} = \frac{8{,}000}{2 \times 60} = 66\tfrac{2}{3}\text{, say 70 gals. per min.}$$

Since the capacity of a pump is roughly proportional to the speed and assuming it is rated at 450 *r.p.m.* to deliver 70 *g.p.m.* The next problem is to find the speed required to deliver 66⅔ *g.p.m.* This is obtained by the simple proportion

$$\text{required speed} = 450 \times \frac{66\frac{2}{3}}{70} = 428.6$$

That is, if the pump be of such size as to deliver 70 *g.p.m.* at 450 *r.p.m.* it will deliver 66⅔ *g.p.m.* at 428.4 *r.p.m.*

CONVERTING FEET HEAD OF WATER INTO PRESSURE

Feet Head	Pounds Per Square Inch	Feet Head	Pounds Per Square Inch	Feet Head	Pounds Per Square Inch
1	.43	60	25.99	200	86.63
2	.87	70	30.32	225	97.45
3	1.30	80	34.65	250	108.27
4	1.73	90	38.98	275	119.10
5	2.17	100	43.31	300	129.93
6	2.60	110	47.64	325	140.75
7	3.03	120	51.97	350	151.58
8	3.40	130	56.30	400	173.24
9	3.90	140	60.63	500	216.55
10	4.33	150	64.96	600	259.85
20	8.66	160	69.29	700	303.16
30	12.99	170	73.63	800	346.47
40	17.32	180	77.96	900	389.78
50	21.65	190	83.29	1,000	433.09

CONVERTING PRESSURE INTO FEET HEAD OF WATER

Pounds Per Square Inch	Feet Head	Pounds Per Square Inch	Feet Head	Pounds Per Square Inch	Feet Head
1	2.31	40	92.36	170	392.52
2	4.62	50	115.45	180	415.61
3	6.93	60	138.54	190	438.90
4	9.24	70	161.63	200	461.78
5	11.54	80	184.72	225	519.51
6	13.85	90	207.81	250	577.24
7	16.16	100	230.90	275	643.03
8	18.47	110	253.98	300	692.69
9	20.78	120	277.07	325	750.41
10	23.09	125	288.62	330	808.13
15	34.63	130	300.16	375	865.89
20	46.18	140	323.25	400	922.58
25	57.72	150	346.34	500	1154.48
30	69.27	160	369.43	1,000	2308

Friction of Water in Pipes.—In order to obtain some standard to use as a basis for figuring friction of water in pipes, the table on pages 195 and 196, was prepared using 15 year old wrought iron or cast iron pipe when pumping soft clear water:

To determine the friction in pipes of different ages of service, multiply the figure under the size pipe to be used opposite the capacity in the following coefficients:

New, smooth pipe....................	coefficient	.71
10 year old pipe....................	"	.84
15 " " "	"	1.00
20 " " "	"	1.22

Figuring the Load.—Before the horse power required to drive the pump can be obtained, it is necessary to determine the total load, that is, the *dynamic column*, or as is ***ignorantly called***, the "total head." This is made up of

1. The dynamic lift, plus
2. The dynamic head

The Dynamic Lift.—In any installation the dynamic lift is made up of

1. Static lift
2. Frictional resistance in entire length of inlet piping from water level to entrance of pump.

To determine the static lift measure vertical distance from water level in well to the center of the inlet opening of the pump, similarly as L_S in fig. 1, page 434.

To this static lift must be added the frictional resistanc that is, frictional loss in the entire length of the pipe from wate level in the well to the entrance to pump.

Example.—Required 70 *g.p.m.* when pump is located 10 feet abo level of the source in the well and a 40 foot horizontal distance assum new 2 in. pipe is to be used containing two 2 in. elbows.

Static lift from well level to pump............ 10 feet

Friction loss for 70 *g.p.m.* through 100 feet of 15 year old 2 in. pipe from table, 18.4 feet. Since new pipe is being used, multiply 18.4 (l in 15 year old pipe) by the coefficient (under the table) by .71 which giv for new pipe

$$18.4 \times .71 = 13.064 \text{ feet}$$

Now add total length of the inlet line including the elbows which ha been converted into equivalent feet of straight pipe as follows:

Vertical pipe................................	10 feet
Horizontal pipe..............................	40 "
Elbows equivalent to.........................	16 " (see page 1
Total..................................	66 feet

Now multiply 13.06 (loss in new pipe) by 66 (feet in inlet line) mar off in hundredths because table is based on 100 feet, as follows:

$$13.06 \times .66 = 8.6196, \text{ say } 8.62 \text{ feet}$$

From which dynamic lift is

$$10 + 8.62 = 18.62 \text{ feet}$$

It should be noted that the dynamic lift should (as in this case) be than the limit 25 feet.

Ques. What should be done if the dynamic lift figured more than 25 feet?

Ans. Either the pump would have to be lowered or a lar pipe should be used to reduce the frictional resistance to f and thus bring the dynamic lift within the limit.

The Dynamic Head.—In any installation the dynamic head is made up of

1. The static head
2. Frictional resistance in entire discharge line from pump outlet to point of discharge.

To determine the static head measure the vertical distance from center of pump outlet to discharge water level similarly as H_S, fig. 1, page 434.

To this static head must be added the frictional resistance, that is, frictional loss in the entire length of the discharge line from pump outlet to point of discharge.

Example.—The pump of the previous example is to force water through a vertical pipe 30 feet long and a horizontal pipe 108 feet long; 2 in. pipe and 3 elbows. Capacity 70 *g.p.m.*

Static lift.. 30 feet

Friction loss for 70 *g.p.m.* through 100 feet of 15 year old 2 in. pipe is (from table page 195) 18.4 feet.

For new pipe (the coefficient being .71), friction loss = 18.4 × .71 = 13.064 feet.

Total length of discharge line = 30 + 108 =	138
Equivalent for 3 elbows	24
Total	162

Multiply 13.06 (loss in new pipe) by 1.62 (ft. in discharge line, marked off in hundredths because table is based on 100 ft.) that is 13.06 × 1.62.. 21.16

Dynamic head 51.16

From the two examples the total load on the pump (dynamic column)

= dynamic lift + dynamic head

= 13 + 51 = 64 ft.

which is equivalent to

$$\left.\begin{matrix} 64 \times .433 \\ \text{or } 64 \div 2.31 \end{matrix}\right\} = 27.7 \text{ lbs. per sq. in.}$$

Horse Power Required.—The power required is the actual po necessary to raise a given quantity of water a given elevation and not theoretical. That is, it is the theoretical power plus the extra power to frictional resistance and the low efficiency of the pump.

The theoretical power is determined by the following formulae

$$t.h.p. = \frac{g.p.m. \times 8\frac{1}{3} \times d.c.}{33{,}000} \quad \ldots\ldots\ldots\ldots$$

$$\text{or } \frac{c.f. \times 62.4 \times d.c.}{33{,}000} \quad \ldots\ldots\ldots\ldots$$

in which

$g.p.m.$ = gallons per minute

$8\frac{1}{3}$ = approximate weight one gallon of water in lbs.

$d.c.$ = dynamic column

$c.f.$ = cu. ft. of water

62.4 = weight of 1 cu. ft. of water at ordinary temperature

$t.h.p.$ = theoretical horse power

The result determined in this way must then be corrected for the po loss in the pumping equipment. This is accomplished by dividing horse power obtained by the efficiency of the pumping outfit, expressed a decimal.

With these corrections formula (1) becomes

$$\text{Actual horse power} = \frac{g.p.m. \times 8\frac{1}{3} \times d.c.}{33{,}000 \times E} \quad \ldots\ldots\ldots$$

in which

E = efficiency of the pump

Ques. What is the actual horse power required to drive pump running under conditions of the foregoing example a having an efficiency of 57%?

Ans. As calculated the $d.c. = 64$ ft., $g.p.m. = 70$. Substit ing in formula(3)

$$\text{Actual horse power} = \frac{70 \times 8\frac{1}{3} \times 64}{33{,}000 \times .57} = 1.82.$$

Another Example.—Determine horse power required to pump 200 *g.p.m.* against combined static lift and static head of 50 ft.; pump efficiency 57%. Pipe line 4 in. diameter, 200 ft. long; 3, 90° elbows.

Friction loss in 100 ft. of 4 in. pipe discharging 200 *g.p.m.* = 4.4 ft. For 200 ft. = 2 × 4.4 = 8.8 ft.

Friction loss in one 4 in. 90° elbow = 16 ft. For 3 elbows = 3 × 16 = 48 ft.

$$d.c. = 50 + 8.8 + 48 = 106.8 \text{ ft.}$$

Substituting in formula (3)

$$\text{horse power} = \frac{200 \times 8\frac{1}{3} \times 106.8}{33{,}000 \times .57} = 9.46, \text{ say } 10$$

Ques. What size electric motor or gas engine should be selected to drive the above pump?

Ans. One having a power rating about 25% in excess of the power required to drive the pump.

Useful Information

Area of a circle = diameter squared × .7854.

Circumference of a circle = diameter × 3.1416.

Pressure in pounds per square inch of a column of water = head in feet × .434.

Head in feet of a column of water = pressure in pounds per square inch × 2.30947.

A U. S. gallon = 231 cubic inches.

A U. S. gallon of fresh water weighs 8.33 pounds.

A U. S. gallon of sea water weighs 8.547 pounds.

An Imperial gallon = 277.274 cubic inches.

An Imperial gallon of fresh water weighs 10.005 pounds.

An Imperial gallon of sea water weighs 10.266 pounds.

A cubic foot of water (1728 cubic inches) contains 7.481 U. S. gallons and weighs 62.355 pounds.

Doubling pipe diameter quadruples the capacity.

Friction of liquids in pipes increases as the square of its velocity.

A Miner's inch of water is approximately equal to 11½ U. S. gallons per minute.

Areas of circles are to each other as the squares of their diameters.

Atmospheric pressure at sea level is usually estimated at 14.7 pounds per square inch, and this pressure will maintain a column of water 33.9 feet high when the normal pressure in the column is relieved by the creation of a vacuum. This is the theoretical distance that water may be drawn by suction. In practice, however, pumps should not be placed over 20 to 25 feet above the water supply, and nearer if possible.

Power Units

1 Horsepower =	33,000 foot-pounds per minute. 746 watts.
1 Watt (unit of elec. power) =	.00134 horsepower. 44.24 foot-pounds per minute.
1 Kilowatt =	1000 watts. 1.34 horsepower. 44,240 foot-pounds per minute.

CHAPTER 12

Reciprocating Pumps; Principles

Before considering pumps "in the flesh" that is to say in the metal where some parts are not visible and where confusing details are shown, it is better to first study pumps in skeletonized form, that is stripped of all non-essential details with views showing the inside. These are **elementary** *pumps* because they show only the essential elements, the purpose being to show how the various basic types *work* rather than how they are constructed.

In general and with respect to operating principles, all pumps may be classed as:

1. Reciprocating.
2. Centrifugal
3. Turbine
4. Rotary

To fully present these pumps they will be treated in separate chapters.

The word *reciprocating* is defined as:

Having a to and fro motion; moving backwards and forwards or up and down, as distinguished from a circular motion. From this, evidently a reciprocating pump is: *A* ***piston, plunger,*** *or **bucket pump** as distinguished from a **centrifugal** or rota pump*. The difference between reciprocating and circu motion is illustrated in fig. 1, with the familiar example show how the *reciprocating motion* of the *wrist pin* of an engine converted into *circular motion* by means of a connecting li called the *connecting rod*.

In a reciprocating pump there are three moving elemen necessary for operation:

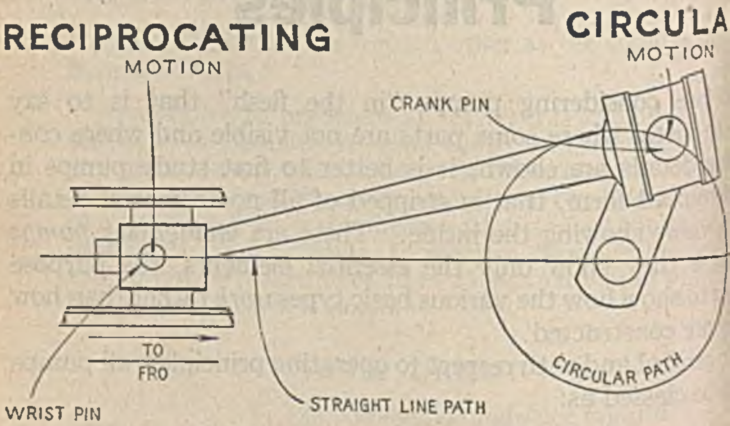


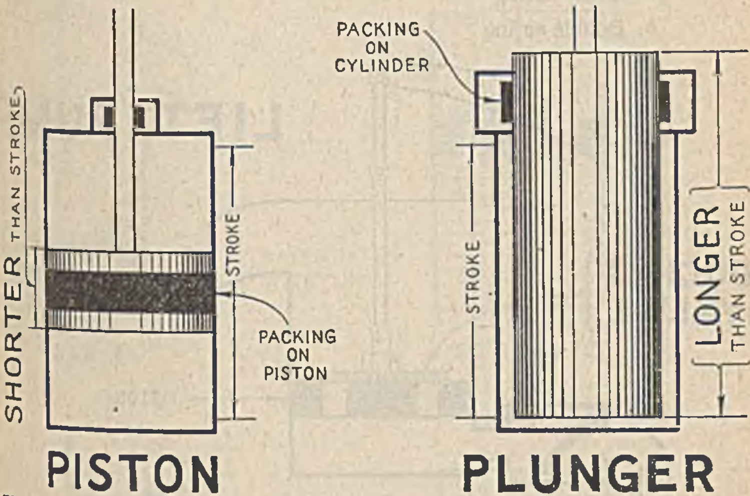
Fig. 1.—The difference between **reciprocating** and **circular** motion illus ing conversion of circular motion into reciprocating motion by means of connecting rod.

1. Inlet or admission valve
2. Piston or plunger
3. Outlet or discharge valve.

The piston or plunger works within a water-tight cylinder.

Before proceeding further the reader should thorough know the difference between a *piston* and a *plunger* as t word plunger is very frequently used erroneously for pisto even by those who ought to know better. In fact there are a lot of greenhorns who think that anything that works inside a pump barrel is a plunger.

Ques. What is the difference between a piston and a plunger?



Figs. 2 and 3.—The difference between a piston and a plunger.

Ans. A piston is shorter than the stroke, whereas a plunger is longer.

This distinguishing feature is clearly shown in figs. 2 and 3.

Ques. Describe another feature which distinguishes a piston from a plunger.

Ans. A piston must have packing inlaid on its rim to provide a tight joint. With a plunger the packing is carried in a stuffing box at the end of the cylinder. See figs. 2 and 3.

Classification of Reciprocating Pumps.—In general and wi respect to how the water is handled, reciprocating pum may be classed as:

1. Lift
2. Force
   a. Single acting
   b. Double acting

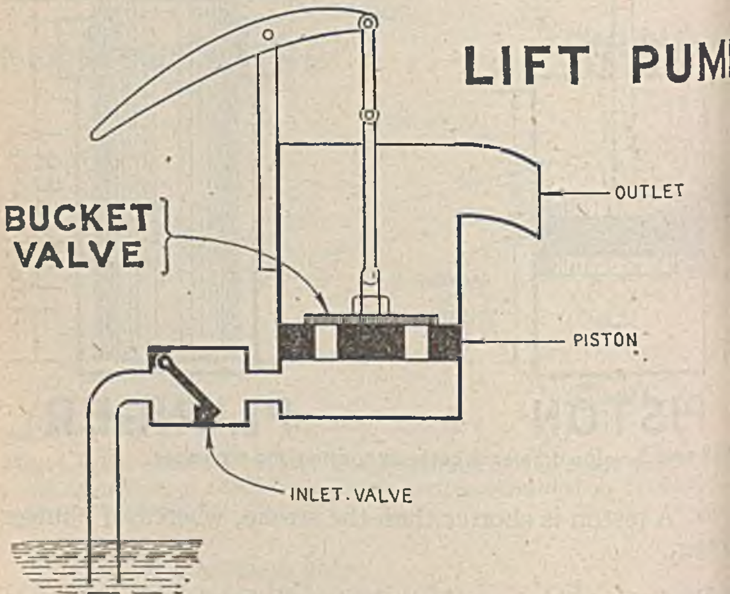
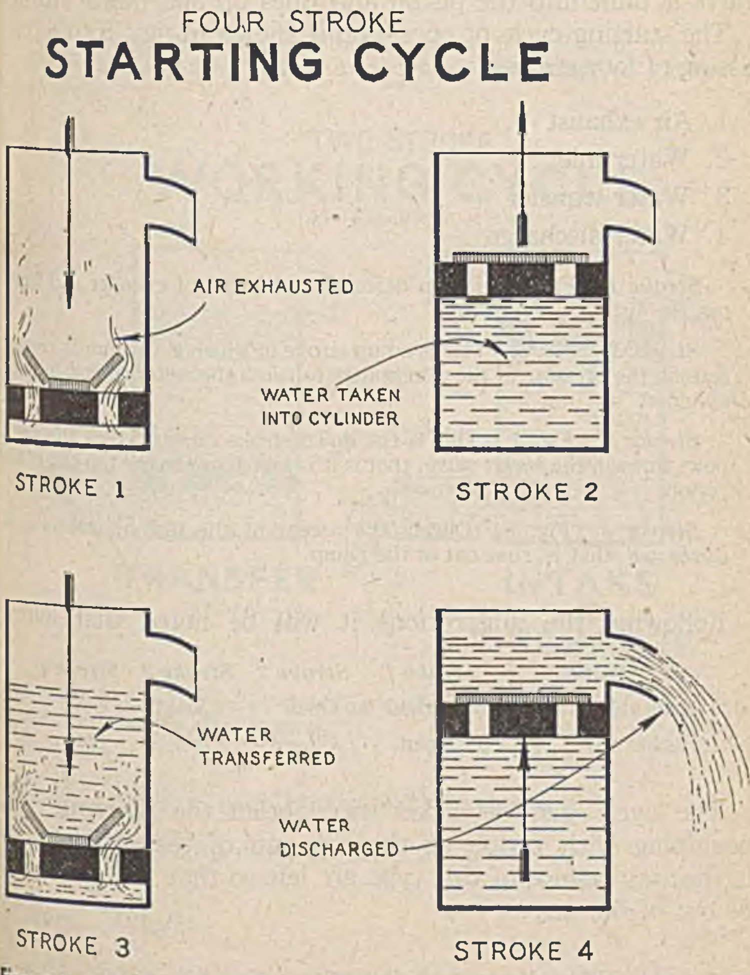


Fig. 4.—Single acting lift pump. Note **inlet** and **bucket** valves.

Lift Pumps.—By definition, a lift pump is *a single acti pump with open cylinder and discharge valve in the piston.* T discharge valve is called a *bucket valve.* The combination open cylinder and bucket valve makes it a lift pump, that in operation it lifts the water and does not force it. Fig



Figs. 5 to 8.—Four stroke **starting cycle** of single acting lift pump. In these diagrams the inlet valve is not shown, but is in figs. 9 and 10.

shows the essential parts of a lift pump. Note that the bucke valve is built into the piston and rides up and down with it.

The starting cycle of operation is shown in figs. 5 to 8 con-sisting of four strokes:

1. Air exhaust
2. Water inlet
3. Water transfer
4. Water discharge

Stroke 1. Fig. 5. Piston descends to bottom of cylinder and forc out the air.

Stroke 2. Fig. 6. This is an up stroke in which a vacuum is creat causing the pressure of the atmosphere to cause the water to flow into th cylinder.

Stroke 3. Fig. 7. This is the down stroke during which the wat flows through the *bucket* valve, that is it is *transferred* to the top side of th piston.

Stroke 4. Fig. 8. During the ascent of the piston, the water *discharged*, that is, runs out of the pump.

Following the illustrations it will be noted that durin

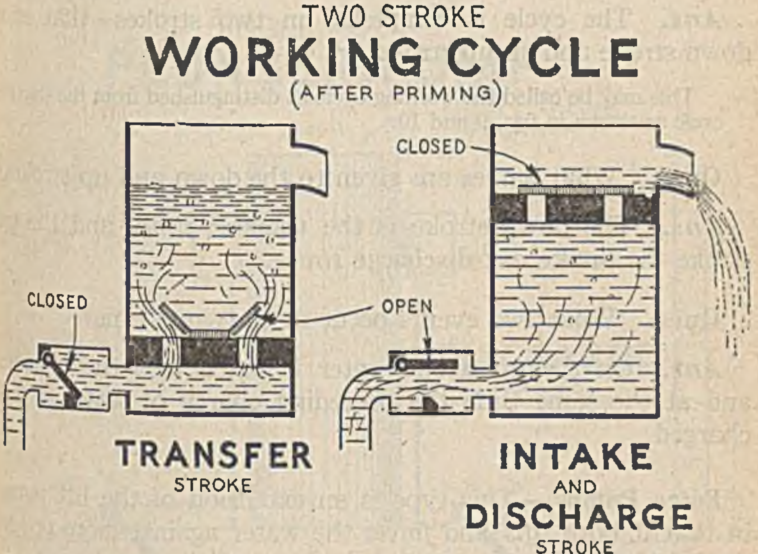
Valve	*Stroke 1*	*Stroke 2*	*Stroke 3*	*Stroke 4*
Inlet valve..........	Closed	Open	Closed	Open
Bucket valve........	Open	Closed	Open	Closed

The cycle just described was labeled the ***starting*** cyc beginning with piston at the top end of the stroke. Th is the way pumps of this type are left so that the handle w be out of the way.

Ques. What happens when the pump is out of use for period of time?

Ans. The water leaks out so that there is nothing but air in the cylinder.

Accordingly in starting the first down stroke is an attempt to get rid of the air in the cylinder so that a vacuum will be created on the next or up stroke.



Figs. 9 and 10.—Two stroke **working cycle** of single acting lift pump which shows cycle events after priming.

Ques. What is usually necessary in starting?

Ans. The pump must be primed.

Ques. Why?

Ans. Some of the water passes down below the piston, thus reducing the *clearance* and some sealing the bucket valve.

This results in a better vacuum being obtained on the up stroke, mak more pressure due to atmosphere available to force the water past t inlet valve and into the pump.

Ques. What happens after the pump has been primed a is in operation?

Ans. The cycle is completed in two strokes—that is down stroke and an up stroke.

This may be called the working cycle as distinguished from the start cycle as shown in figs. 9 and 10.

Ques. What names are given to the down and up stroke

Ans. The down stroke is the **transfer** *stroke* and the stroke the **intake** *and* **discharge** *stroke.*

Ques. What two events occur at the same time?

Ans. On the up stroke water is taken into the cylind and at the same time the preceding charge of water is di charged.

Force Pumps.—This type is an extension of the lift pun in that it both *lifts* and *forces* the water against more or le external pressure.

By definition, a ***force** pump* is: *A pump employed to for water above the range of atmospheric pressure*, distinguished fro a *lift pump*, in which the water as previously explained, elevated to run out of a spout.

Ques. How does a force pump act?

Ans. In a force pump the water is ***forced** out by a pis or plunger working against a pressure corresponding to **head** or elevation above the inlet valve to which the water is pump*

Ques. What are the distinguishing features of a *force* pump as compared with a *lift* pump?

Ans. Instead of inlet and bucket valves as in the case of a lift pump, a force pump has *inlet* and *discharge* valves and a *closed* cylinder.

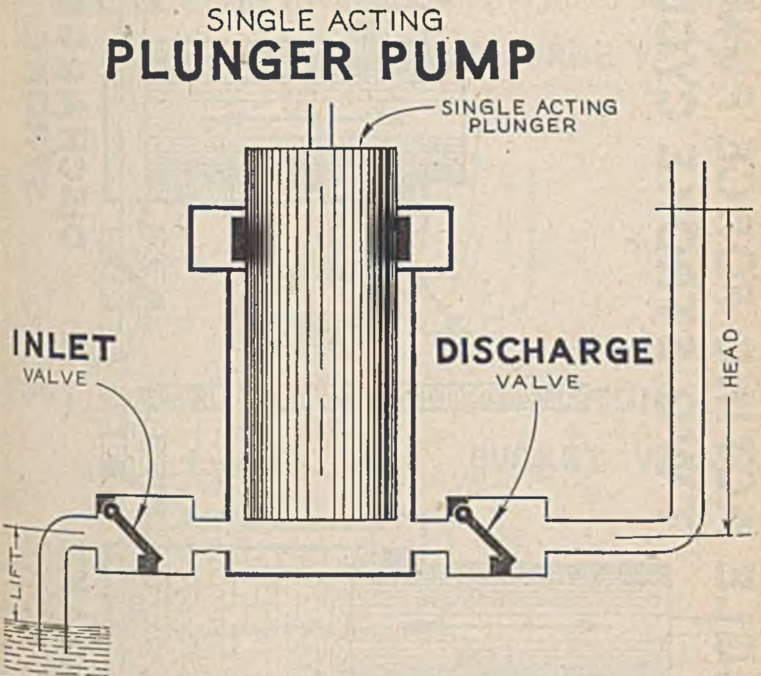
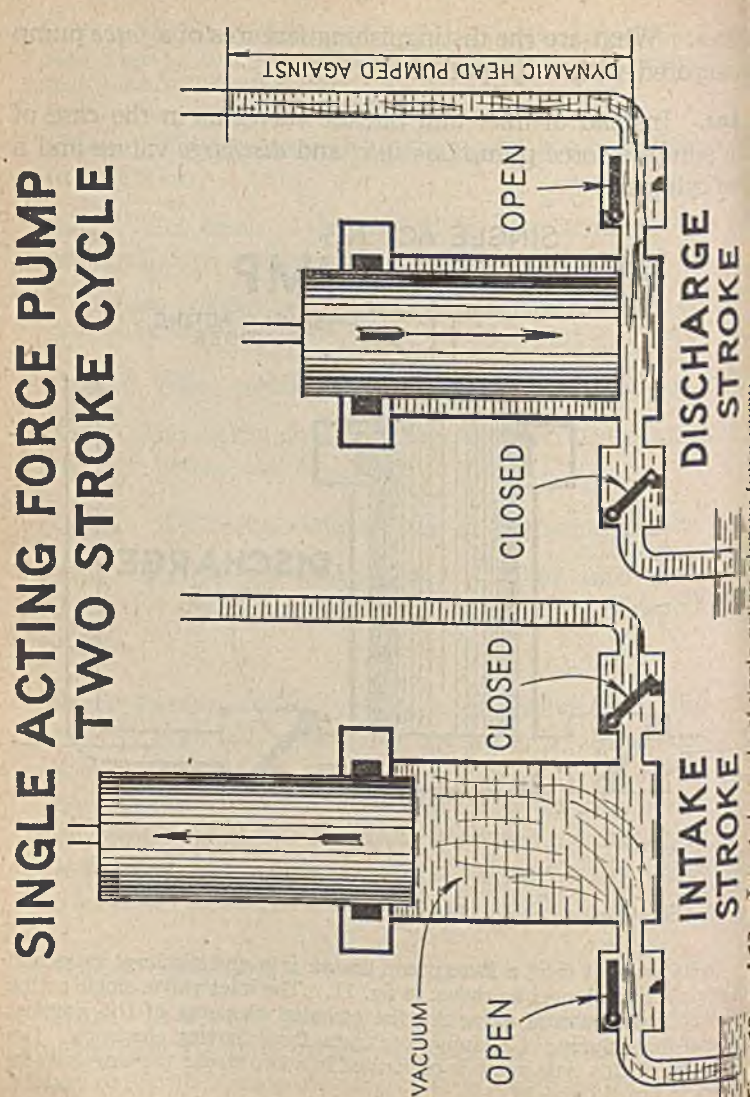


Fig. 11.—Single acting plunger force pump.

In its simplest form a force pump has an *inlet* and *discharge* valve and a single acting plunger as shown in fig. 11. The inlet valve, single acting plunger and discharge valve are the essential elements of the simplest type of force pump. Carefully note these three moving elements. The operation of this type pump is performed in a two stroke working cycle as shown in figs. 12 and 13.


SINGLE ACTING FORCE PUMP
TWO STROKE CYCLE
DYNAMIC HEAD PUMPED AGAINST
OPEN
CLOSED
DISCHARGE
STROKE
CLOSED
VACUUM
OPEN
INTAKE
STROKE

Ques. Describe the two stroke working cycle of the simple single acting plunger pump, shown in fig. 11.

Ans. During the up stroke (fig. 12) the vacuum created in the cylinder causes the available atmospheric pressure to force the water into the cylinder. During this event the inlet valve is open and the discharge valve closed as shown.

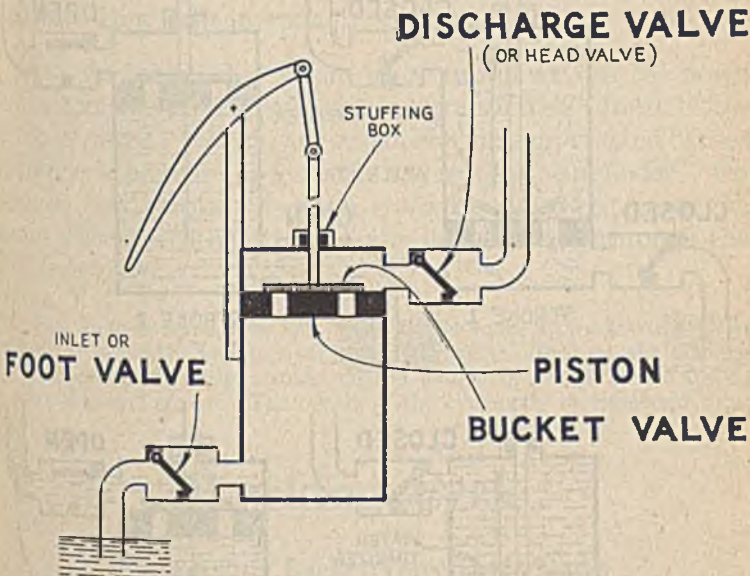


Fig. 14.—Single acting bucket valve force pump.

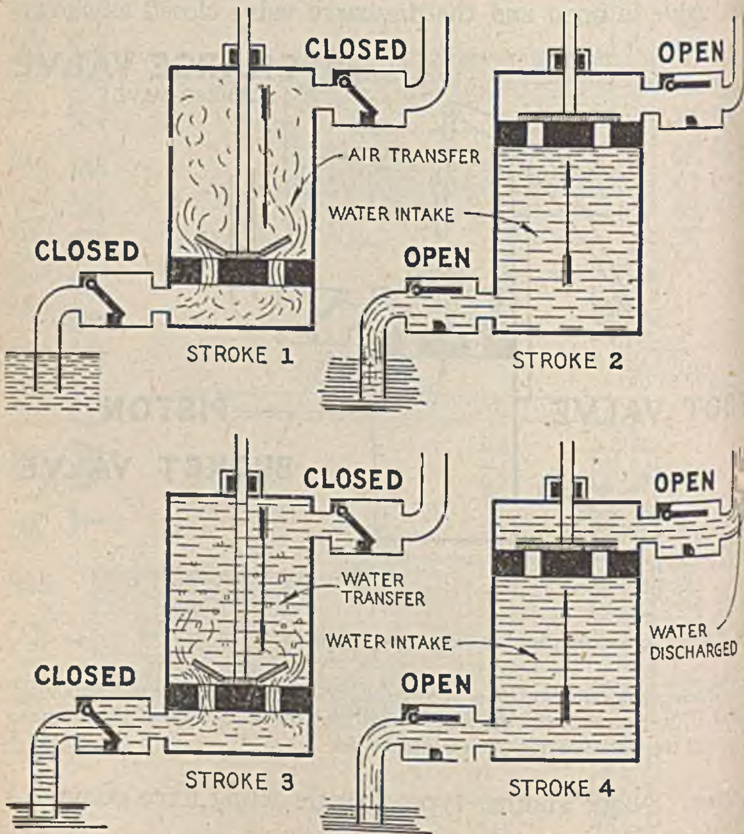
During the down or discharge stroke (fig. 13) the plunger "displaces" that is, forces open the discharge valve and the water out of the cylinder against the pressure due to dynamic head.

Ques. Name another type of single acting force pump.

Ans. The piston pump having inlet, bucket and discharge valves.

STARTING CYCLE

SINGLE ACTING BUCKET VALVE PUMP



Figs. 15 to 18.—Four stroke starting cycle of single acting bucket valve force pump.

Ques. Describe its essential features.

Ans. The piston works within a closed cylinder as in fig. 14 (note stuffing box at top for the piston rod). The water in traversing the pump passes progressively through the inlet or *foot* valve, the *bucket* valve and the *discharge* valve.

Ques. How does it work in starting?

Ans. Assuming the system to be full of air, on the down stroke the air in the cylinder is "transferred" from below to top of piston. On the up stroke the vacuum created causes atmospheric pressure to force the water into the cylinder. On the next (3rd) stroke this water is *transferred* to the top of the piston whence on the 4th stroke it is discharged through the discharge valve.

Figs. 15 to 18 show these four strokes which make up the starting cycle. However, when the system is cleared of air and in operation, the working cycle is completed in two strokes, that is, a down or *transfer* stroke and an up or *discharge* stroke. The events of the cycle may be tabulated thus:

Working Cycle

Single Acting Bucket Valve Force Pump

Down or *transfer stroke.*	Water transferred through bucket valve.
Up or *discharge stroke.*	*Above piston:* Water discharged *Below piston:* Water admitted

The positions of the valves during the cycle are tabulated as follows:

Single Acting Bucket Valve Force Pump

Stroke	Foot valve	Bucket valve	Discharge valve
Transfer—down.....	Closed	Open	Closed
Discharge—up......	Open	Closed	Open

Double Acting Piston Force Pump.—By definition: *A ty of pump in which the piston discharges the water on one si while drawing it into the cylinder on the other side, witho any transfer stroke.*

That is to say, water is discharged *every* stroke instead of every ot stroke, as with single acting pumps. Accordingly the double acting pu has double the capacity of a single acting pump of the same cylind displacement.

The essentials of the double acting piston pump are show in fig. 19, and the two stroke working cycle in figs. 20 and 2

In these two figures note positions of the valves for the two strok It will be seen that diagonally opposite valves work in unison, that they are either open or closed at the same time.

Double Acting Plunger Force Pump.—This type operates t same as the one just described with the exception that it has plunger instead of a piston. There are two types class with respect to the location of the packing, as:

1. Inside packed.
2. Outside packed.

Fig. 22 shows essentials of the inside packed pump.

Note that this inside packing virtually divides the long cylinder two separate chambers. The operation of this pump is shown in

23 and 24. The plunger in its up and down movements alternately displaces water in the two chambers.

Ques. What troubles are experienced with the inside packed pump?

Ans. In order to adjust or renew the packing, it is necessary to remove the cylinder head. Moreover in operation it cannot be ascertained whether or not there is leakage through the packing.

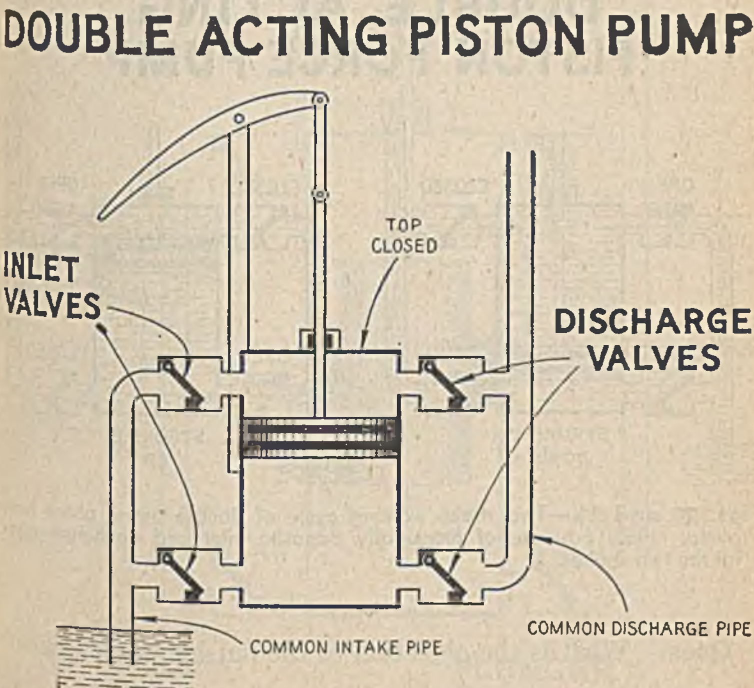
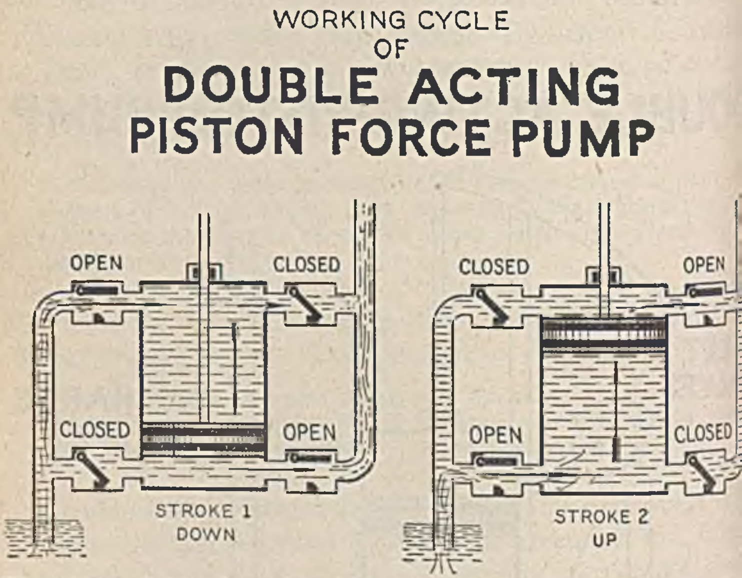


Fig. 19.—Double acting piston force pump showing two sets of inlet and discharge valves with common intake and discharge pipe connections.

Ques. How were both of these troubles overcome?

Ans. By the outside packed pattern.

Fig. 25 shows the elements of this design. It will be noted that tw plungers are required which must be rigidly connected together by yoke and rods. The packing being on the outside, it is easily serviced and i condition can be seen.



Figs. 20 and 21.—Two stroke working cycle of double acting piston for pump. Note positions of diagonally opposite inlet and discharge valv for the two strokes.

Ques. What is the objection to the outside packed pump

Ans. The construction is more complicated and accordingl more expensive than the inside packed pump.

Figs. 26 and 27 show the two stroke working cycle. Note that the plungers move in unison with result that water is discharged at one end while the other plunger is receding to fill the other end.

DOUBLE ACTING PLUNGER PUMP INSIDE PACKED

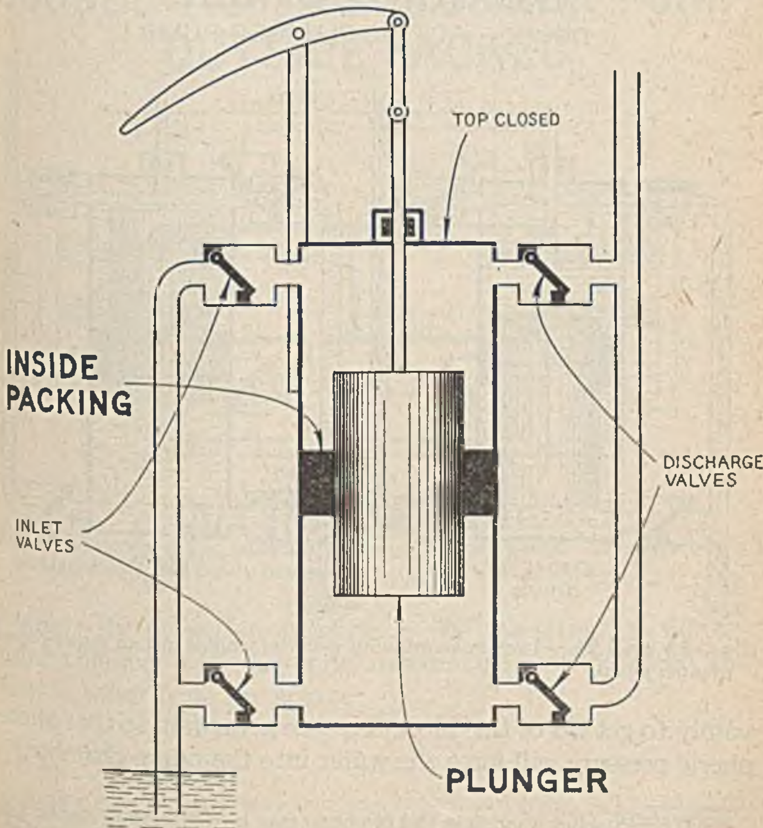
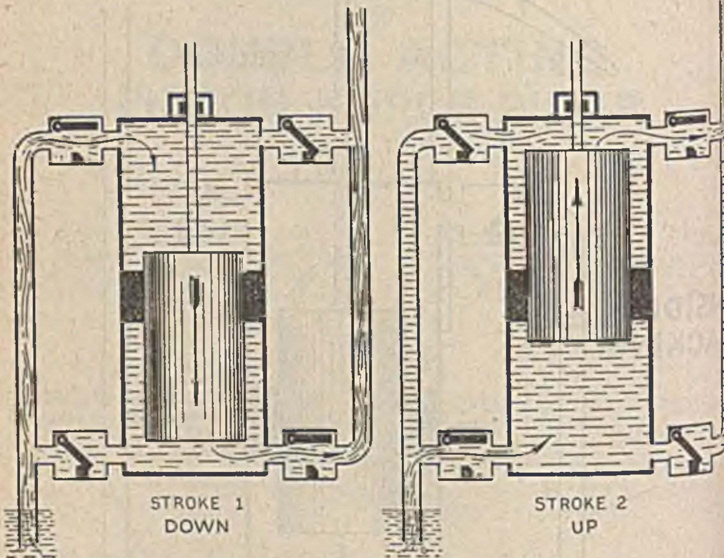


Fig. 22.—Double acting **inside packed** plunger pump.

Self Priming or So-Called Siphon Pumps.—The troub with most pumps is in starting when the whole system is fill with air. This is on account of the considerable clearance space enclosed between the inlet and discharge valves wi result that the piston or plunger in its initial movements tri

WORKING CYCLE

DOUBLE ACTING PLUNGER PUMP



Figs. 23 and 24.—Two stroke working cycle of double acting inside pack plunger pump.

vainly to get rid of this air and create a vacuum so that atmo pheric pressure will force the water into the pump chamber.*

*NOTE.—Starting a pump in this condition may be compared to starting a engine rather than a steam engine. On opening a steam throttle the engine sta However, after a considerable number of *B.t.u.* and profanity have been expend

Ques. What is done to get rid of the air in a pump chamber?

Ans. It must be primed.

Ques. Define priming.

DOUBLE ACTING PLUNGER PUMP OUTSIDE PACKED

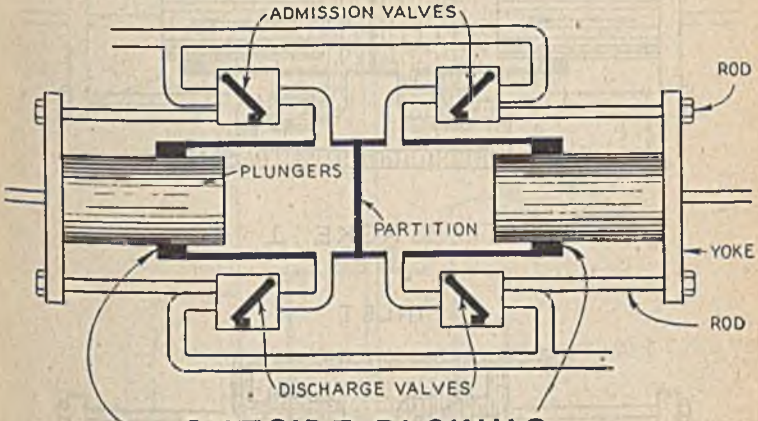
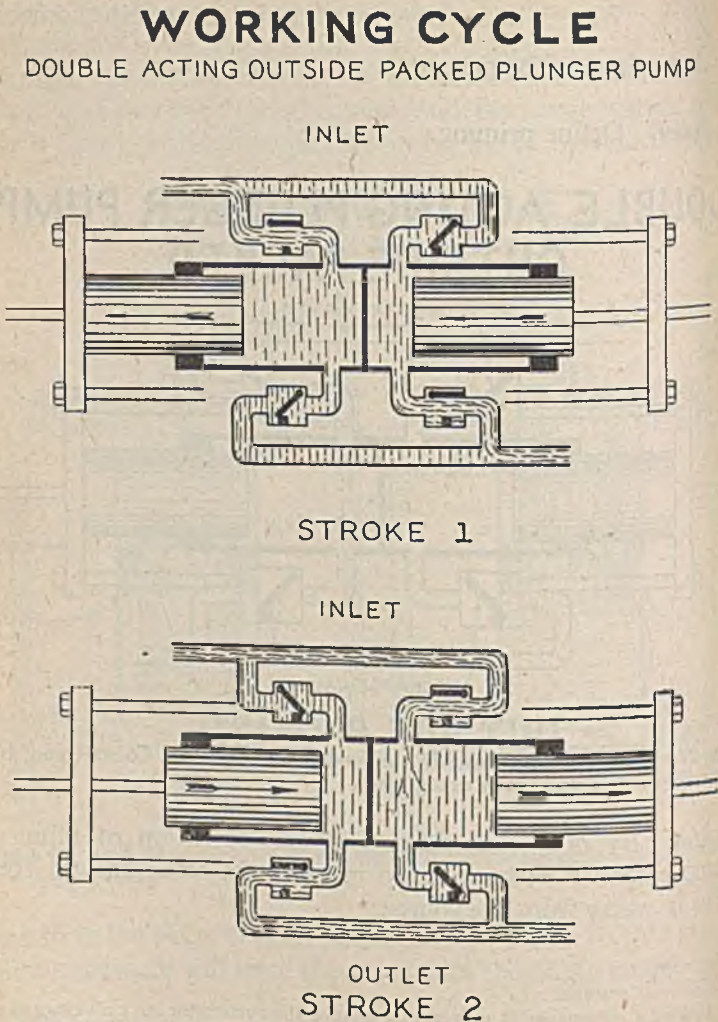


Fig. 25.—Double acting **outside packed** plunger pump. Complicated, but packing visible and easily accessible.

Ans. By definition priming is the operation of filling a pump chamber with water to increase the vacuum and thus draw in water from the source.

NOTE.—*Continued.*
in cranking a gas engine, it may start provided the carburetor isn't all clogged up and the battery and ignition system are in first class condition. With the usual greenhorn makeshift attendance these requisites for gas engine starting are conspicuous by their absence.



Figs. 26 and 27.—Two stroke working cycle of double acting **outside packed** plunger pump.

Ques. What trouble is experienced in priming?

Ans. Unless the pump be provided with a vent, and inlet opening, some part must be removed to get the water into the cylinder.

Ques. Describe the essentials of the so-called siphon pump.

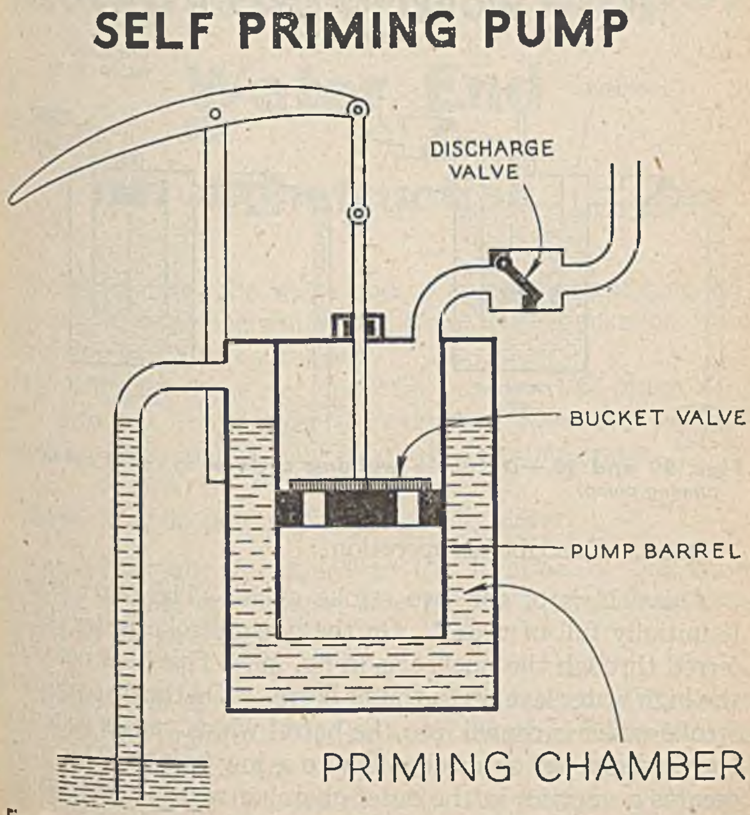
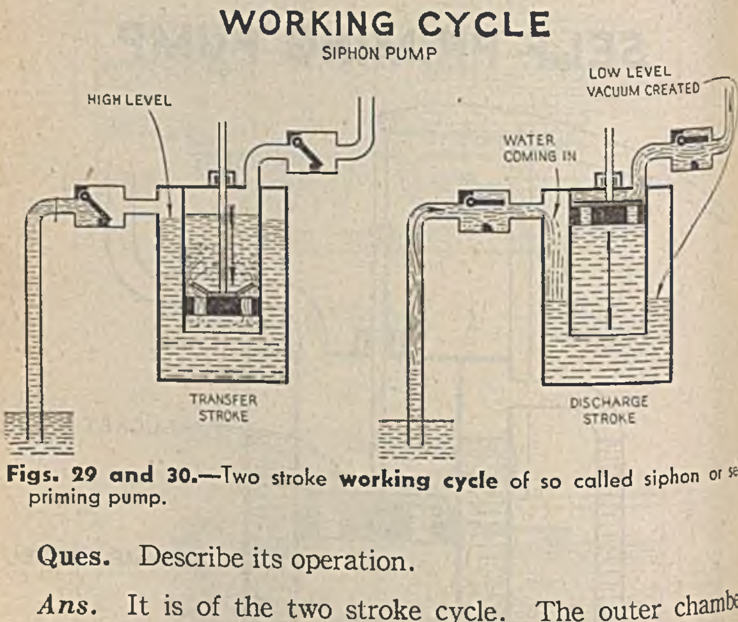


Fig. 28.—So called siphon or self priming pump.

Ans. These are shown in fig. 28. The pump casting consists of a barrel and outer concentric chamber, the lower end of the barrel opening into the outer chamber. As shown the piston is of the bucket valve type with discharge valve at the top. The inlet is located also high up so that water will be trapped in the outer chamber.



Figs. 29 and 30.—Two stroke **working cycle** of so called siphon or self priming pump.

Ques. Describe its operation.

Ans. It is of the two stroke cycle. The outer chamber is initially full of water. On the down stroke water is transferred through the bucket as in fig. 29. This does not change the high water level in the outer barrel. On the up or discharge stroke water is drawn into the barrel which causes water level to recede in the outer chamber to a low point. This in turn creates a vacuum in the outer chamber which causes water to flow in from the source to fill the outer chamber.

CHAPTER 13

Reciprocating Pumps; Water End

Construction

In construction, the water end of any pump must be very substantial to permanently withstand the shocks or water hammer to which it is subjected.

By definition the water end of a pump is: *the pump itself*, that is to say, *the cylinder* (*or cylinders*) *valves and piston or plunger*. These parts being necessary to pump water.

Ques. How do pump cylinders chiefly differ?

Ans. Principally in regard to the location of the valves and the piston or plunger construction, particularly with respect to the packing design.

Classification of Water Ends.—A classification of the water end would be, in part, a duplication of the general classification given in Chapter 12; however, there are a few types of water end that should be here mentioned. Accordingly water ends of pumps may be classified:

1. With respect to cylinder construction, as:

   a. Cast iron
   b. Steel
   c. Composition
   d. Lined
   e. Close clearance
   f. Removable

2. With respect to the number of cylinders, as:

   a. Simplex
   b. Duplex
   c. Triplex

3. With respect to the pumping element, as:

   a. Piston
   b. Plunger
   c. Single acting
   d. Double acting

4. With respect to design of stuffing box, as:

   a. Inside packed
   b. Outside packed $\begin{cases}\text{center}\\ \text{end}\end{cases}$

5. With respect to the valves, as:

   a. Single
   b. Multi
   c. No inlet valves (Magma)

6. With respect to valve construction, as:

   a. Rubber

   b. Rubber with metal backing

   c. Composition,

   etc.

7. With respect to the valve chest, as:

   a. Single deck

   b. Double deck

   c. Turret

   d. Pot

8. With respect to pressure pumped against, as:

   a. Low pressure

   b. Medium pressure (tank)

   c. High pressure

   d. Hydraulic (extra high pressure)

To this list could be added numerous other classifications, but the variations here listed will indicate the considerable diversity of designs necessary to provide water ends suitable for the many different service conditions encountered.

Cylinder Construction.—The principal differences in cylinder design are the form of piston or plunger and the location of the valves.

Fig. 1 shows the ordinary construction, being a cylinder in which the piston fits tightly in the bore, having a seal of either snap rings or fibrous packing.

The valve chest or "water box" as it is sometimes called is of the double deck plate type in which the discharge valves are directly over the inlet valves.

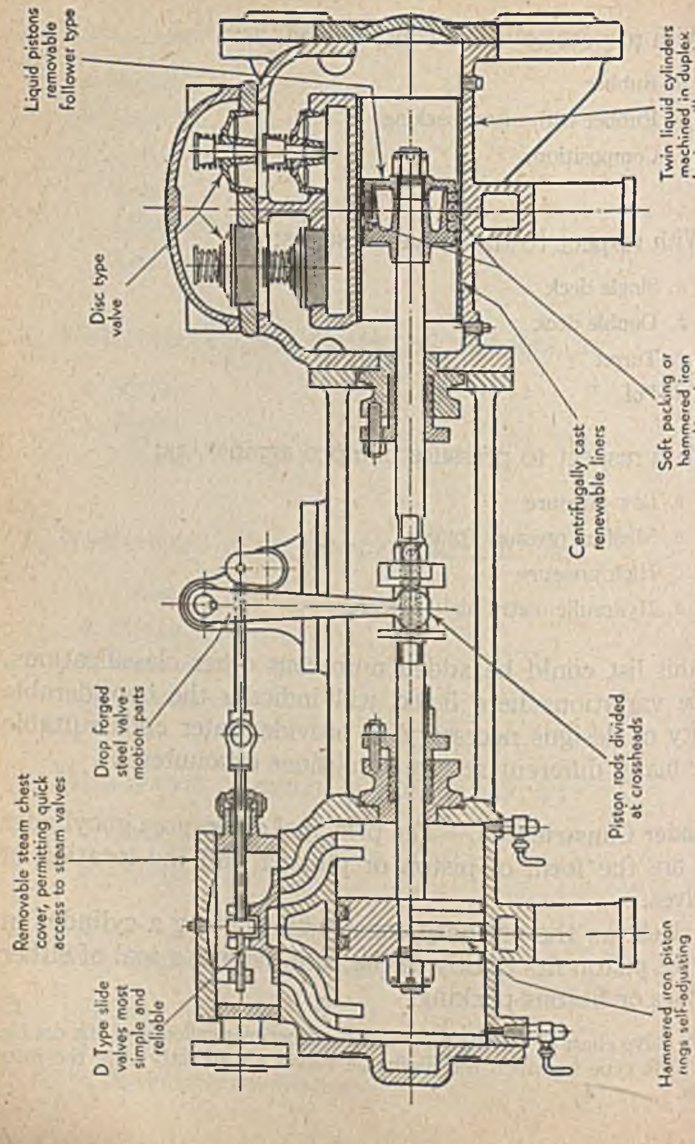


Fig. 1.—Double deck, plate type with piston fitting tightly in the bore.

The discharge valves are mounted on a *plate*, the removal of which gives easy access to the inlet valves.

That is by removing the chest cover and the plate the inlet valves are accessible. The double deck construction is sometimes called the "submerged type" as the inlet valves are above the piston or plunger and the cylinder normally remains full of water, which renders priming unnecessary on starting.

Ques. What should be noted about the submerged type?

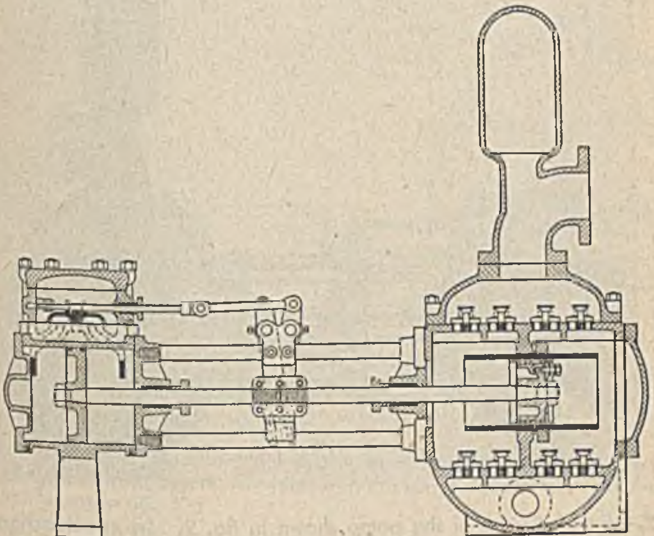


Fig. 2.—Single deck, piston pump. Sectional view showing entire mechanism.

Ans. While it has the advantage of being very compact the flow of water must be reversed since it first flows into the cylinder through the lower inlet valves and is then discharged through the upper discharge valves.

Ques. What type of pump is used to avoid this reversal of water flow?

Ans. The single deck type.

In the single deck type of pump the inlet valves are placed in a deck below the plunger and the discharge valves in a deck above as shown in fig. 2.

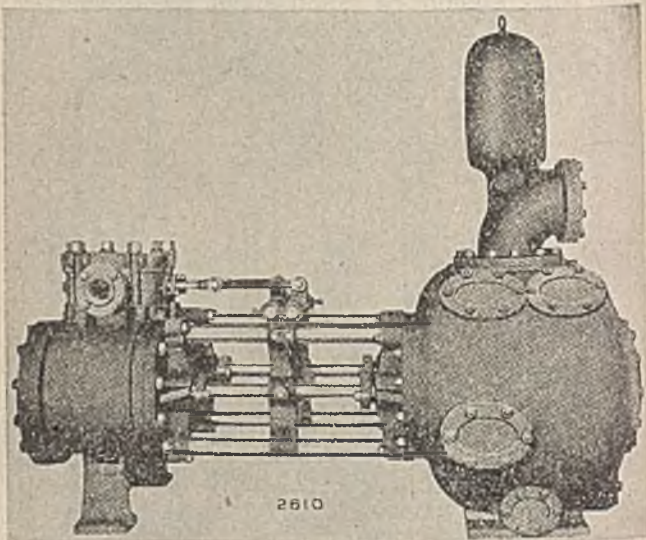


Fig. 3.—Exterior view of the pump shown in fig. 2. **In construction** (refer to both figures) both inlet and discharge valve decks are cast integral with the cylinders, the inlet valves being on the lower deck below the piston. In this design the valve port area being very liberal, the pump is adapted particularly to installations where at times a greater supply of water is needed for short periods.

Ques. What are the characteristics of the single deck arrangement?

Ans. The design gives direct passages for the water through the pump, the uniflow of water permitting operation at high piston speed without excessive shock or water hammer.

Ques. For what size pump is the single deck type adapted?

Ans. For large size.

This type is not as compact as the double deck, and is not adapted to small sizes because the casting must be large enough to provide room for hand holes through which the valves are reached.

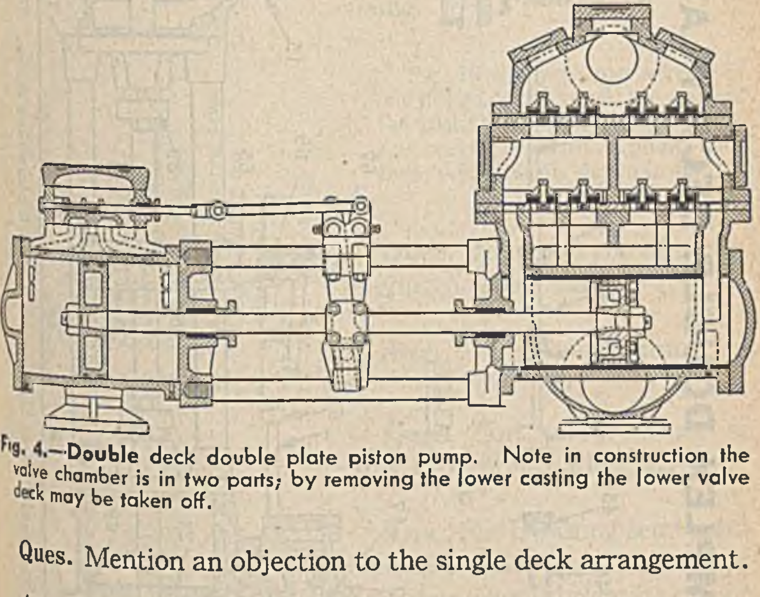


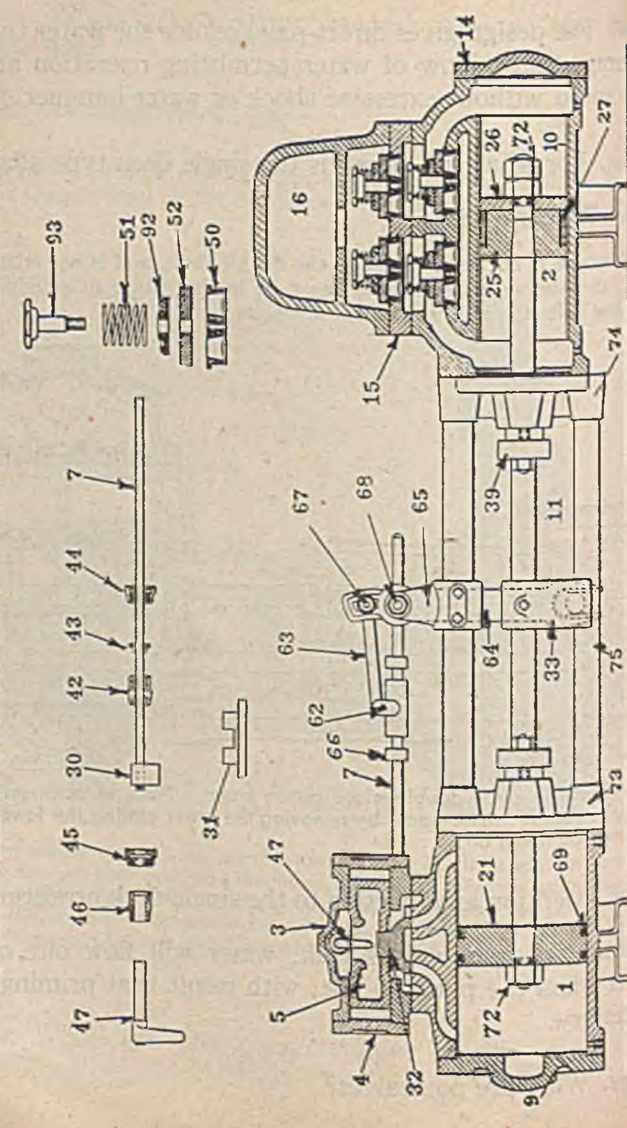
Fig. 4.—**Double** deck double plate piston pump. Note in construction the valve chamber is in two parts; by removing the lower casting the lower valve deck may be taken off.

Ques. Mention an objection to the single deck arrangement.

Ans. If the inlet valves leak, water will flow out of the cylinder when the pump is idle, with result that priming may be necessary.

Ques. What are pot valves?

SIMPLEX DOUBLE DECK PLATE PUMP



Figs. 5 to 15.—Sectional view of Simplex double deck plate type pump. The parts are given on page 285.

Sectional View and Parts List.

1. Steam Cylinder	27. Fibrous Packing for Pump Piston	62. Valve Rod Dog
2. Pump Cylinder	30. Clamp for Small Slide Valve	63. Link
3. Steam Chest	31. Small Slide Valve	64. Lever
4. Steam Chest Head	32. Main Slide Valve	65. Stand
5. Chest Piston	33. Cross Head	66. Valve Rod Dog Collar
7. Valve Rod	39. Piston Rod Gland	67. Link Stud
9. Steam Cylinder Head	42. Bushing } For Valve Rod	68. Lever Stud
10. Pump Cylinder Lining	43. Gland } Stuffing Box	69. Steam Piston Ring
11. Piston Rod	44. Nut	72. Piston Rod Nut
14. Pump Cylinder Head	45. Nut } For Starting Lever	73. Steam Cradle Head
15. Discharge Valve Plate	46. Bushing } Stuffing Box	74. Pump Cradle Head
16. Water Chest Cover	47. Starting Lever	75. Cradle Bar
21. Steam Piston	50. Water Valve Seat	92. Plates for Valve
25. Pump Piston Head	51. Water Valve Spring	93. Valve Stem
26. Pump Piston Follower	52. Water Valve	94. Valve Guard

Ans. The term relates more to the valve chest than the valves, that is, pot valves are valves having a separate or bolted on valve chest shaped something like a pot, hence the name.

Fig. 16 shows a cross section of one design of pot valve, the walls of the casting being of circular form. An exterior view of a pump with these pot valves is shown in fig. 17.

Ques. Name a type of pump having no inlet valves.

Ans. The magma pump.

Ques. For what service is this type pump adapted?

Ans. For handling semi-solid materials, such as acid sludge or settlings from lubricating oils or other very heavy residuums.

Ques. What is the distinguishing feature of the magma pump?

Ans. It has no inlet valves.

Ques. How does it work?

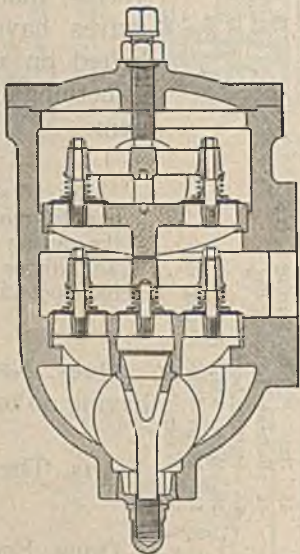


Fig. 16.—Section through a double deck pot valve. **In construction,** the valve decks are contained in a circular pot shaped chamber, hence the name. Often there are separate individual castings, their shape giving great strength and when separate from the main casting avoids the extra expense of a large complicated casting.

Ans. The residuum to be pumped gravitates into a hopper or funnel inlet entering the cylinder on top at mid stroke. In operation, the piston which is of the solid plug type, shears off a slug of this residuum each stroke and forces through large discharge "block" valves at each end of the cylinder.

The construction is shown in the chapter on Special Service Pumps. The piston is shown in mid position, hopper closed. As it approaches either end of the stroke it opens the hopper to the cylinder.

Packing.—With respect to the method of packing plunger pumps to secure a water tight working joint, they may be be classed as:

1. Inside packed.
2. Outside center packed.
3. Outside end packed.

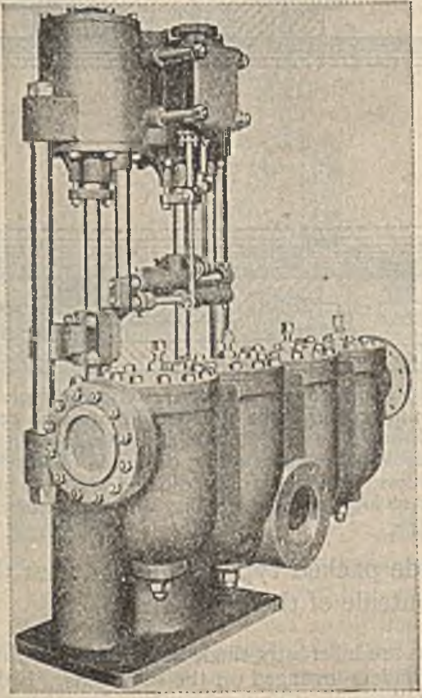


Fig. 17.—Exterior view of vertical duplex pot valve pump.

Fig. 18 shows design of inside packing.

In this arrangement the plunger passes through a stuffing box in the center of the pump cylinder.

In comparison with the outside packed type there are two strong objections to the inside packed pump: 1, the cylinder head must be removed either for adjusting or renewing the packing; 2, the condition of the packing as to leakage cannot be ascertained when the pump is running. Evidently the inside packed pump is only suited to pumping water free of foreign matter.

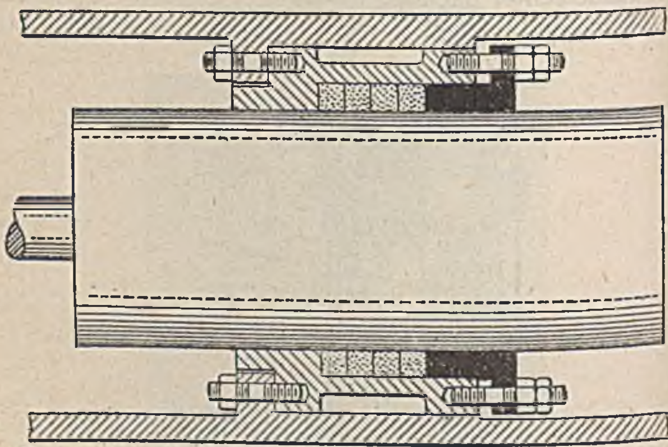


Fig. 18.—Inside packed plunger. **In construction,** the plunger passes through a stuffing box in the center of the cylinder. Not much can be said in favor of this type and should be used only with perfectly clean liquids—no foreign matter as nobody knows what is going on inside during operation, that is to say, leakage, unless excessive, is not indicated while the pump is in operation. Totally unsuited to high pressure working with dirty water.

In the outside packed types, plungers pass through stuffing boxes on the outside of the cylinder.

These pumps are inherently single acting, and for that reason there are usually two cylinders arranged on the same axis. Either a long plunger (which is virtually equivalent to two plungers) or two separate plungers are used.

Figs. 19 and 20 show respectively single deck and double deck *outside* center *packed* pumps.

In each the stuffing box ends of the cylinders face each other, there being one long plunger projecting into each single acting cylinder.

The outside packed plunger pattern is designed for rough and heavy services for which the regular piston pump is not suited.

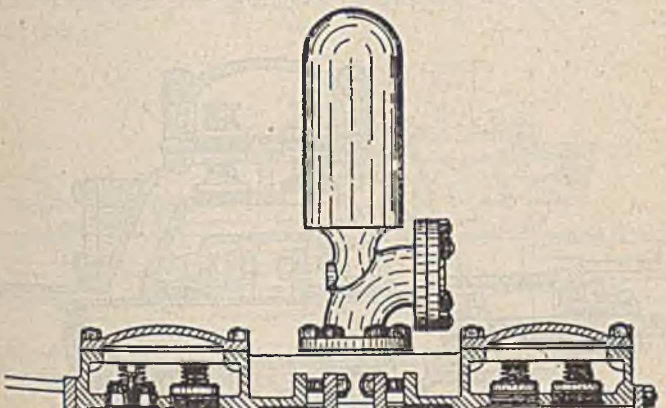


Fig. 19.—Single deck turret type outside center packed pump.

The outside packed type should be used on all work where the water to be handled contains considerable quantities of sand and grit, where the working pressure is over 140 lbs. per sq. in. and where it is important that the moving parts are repacked quickly. This arrangement makes it possible to pack the plungers from the outside without opening up the pump. Also any leakage is at once apparent, it being easy to keep the machine always at its highest pumping efficiency.

Ques. Mention some inherent characteristics of outside packed pumps.

Ans. Being single acting it requires two cylinders instead of one double acting cylinder for equal capacity and accordingly outside packed pumps are more bulky than the inside packed type.

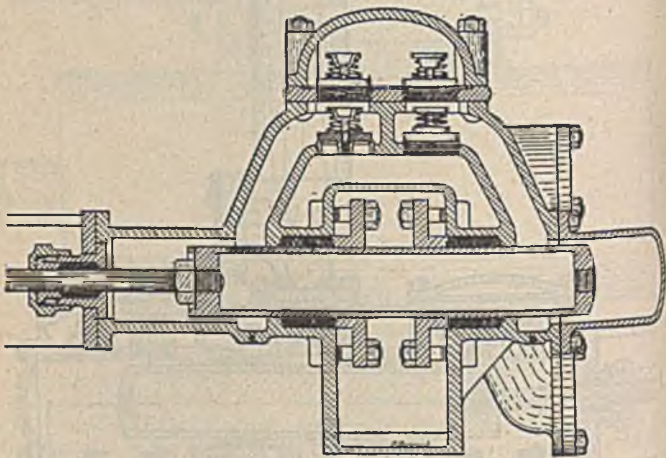


Fig. 20.—Double deck plate outside center packed pump showing one long plunger projecting into each single acting cylinder.

Ques. Mention one difference in construction between outside center packed and outside end packed pumps.

Ans. The outside center packed type has one long plunger but the outside end packed type requires two plungers because the stuffing boxes are on the far ends.

Ques. What is the object of the outside end packed arrangement?

Ans. It is the most accessible type made.

Fig. 21 shows an outside end packed pump of the double deck (side by side) pot valve pattern. The arrangement is here called double deck to distinguish it from the single deck pump. As designed each pump valve can be reached through its own hand hole cover.

Ques. How are the plungers connected in the outside end packed pump?

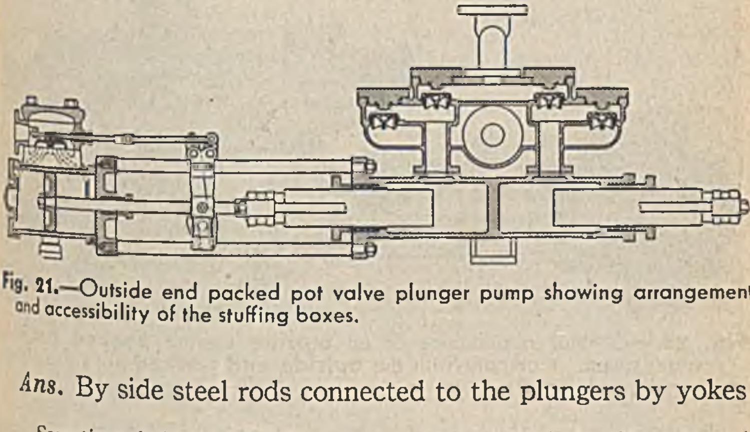


Fig. 21.—Outside end packed pot valve plunger pump showing arrangement and accessibility of the stuffing boxes.

Ans. By side steel rods connected to the plungers by yokes.

Sometimes babbitted bearings are provided in which the side rods work and which support the weight of the moving parts, reducing wear on plungers and stuffing box throats to a minimum.

Turret Type Water End.—In the design of water ends a casting of circular form is stronger than a straight or box-like shape because the metal is subjected to tension instead of transverse stress. Accordingly for a given pressure a lighter casting is permissible. This is the underlying reason for the turret pattern.

The general appearance of the turret type pump is shown in fig. 24.

In fig. 24 note that the upper part of the valve chamber is circular or turret shaped, hence the name. In the absence of the "plate" construction, the valves on the lower deck are accessible by removing the large hand cover seen in fig. 24.

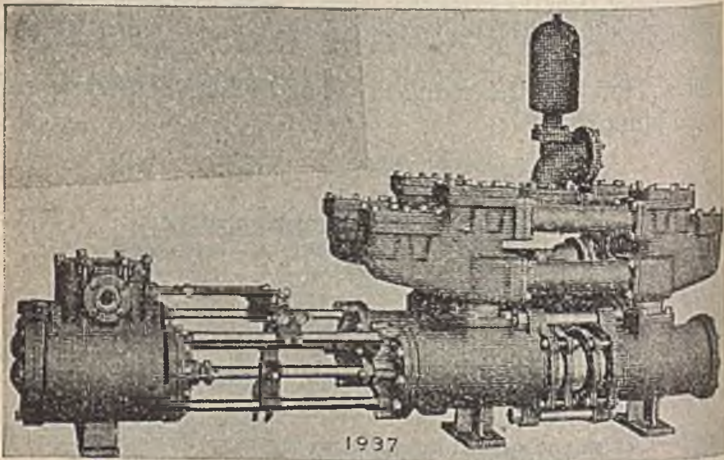


Fig. 22.—General appearance of an **outside center packed** pot valve plunger pump. Compare with the **outside end packed** pot valve plunger pump shown in fig. 21.

Cylinder Linings.—On piston pumps there is usually provided a brass or bronze cylinder lining. The lining may be either a force or sliding fit. For a sliding fit the lining is held in place by bolts or studs passing through a flange at one end.

The reason for this construction is to render it easy to remove the lining. The force fit liners are brass tube and the removable liners are of cast bronze.

Pistons.—As usually constructed, pistons are made of cast iron of the body and follower type, as shown in fig. 26. The

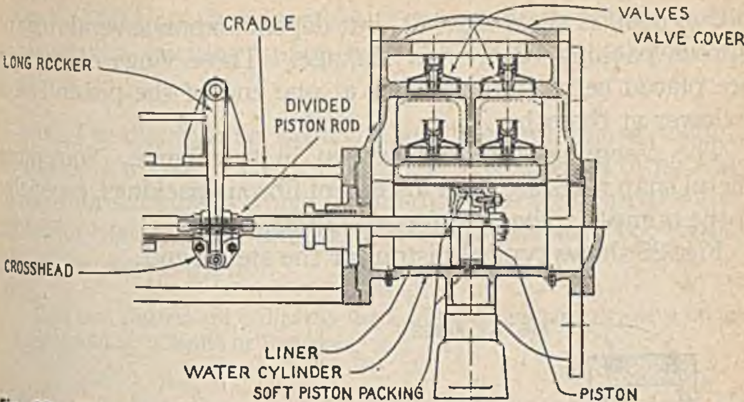


Fig. 23.—Water end showing one cylinder of the horizontal duplex double deck **turret** type pump, shown in fig. 22.

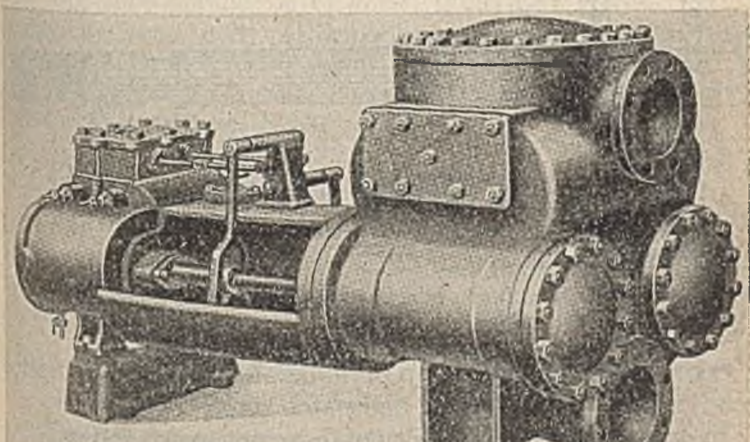


Fig. 24.—External appearance of a horizontal duplex double deck **turret** type pump for general services; for handling liquids at pressures up to 200 lbs. per sq. in. Note the large hand hole for gaining access to the valve on the lower deck.

piston itself is not a tight fit, but depends upon several rings of fibrous packing to prevent leakage. These rings of packing are placed between a shoulder at one end of the piston and a follower at the other.

The assembly is plainly shown in the figure. Sometimes metal snap rings are used instead of fibrous packings, especially if the pump is to be used for hot liquids.

Fig. 25 shows typical piston for the steam end.

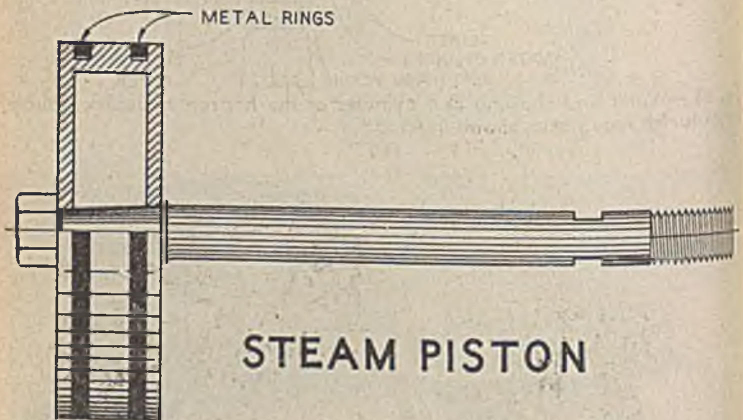


Fig. 25.—Pump steam piston of the box type with divided piston rod.

Some designs are made up of numerous parts such as: piston head, follower, ring, spring and patch ring.

In order to avoid possibility of disarrangement and resulting trouble with these numerous parts, a simplex arrangement has been introduced consisting of only three parts: Solid head and two rings. This makes it easy to place in the cylinder or change the rings.

Some manufacturers rivet the piston onto the rod.

Ques. Mention an important requirement in regard to the width of the packing space between shoulder and follower of piston.

Ans. The distance between the shoulder and follower should be a little in excess of the width of the several rings of packing to allow for expansion which occurs due to absorption of water; otherwise the packing would grip the cylinder walls too tightly causing excessive friction.

Cast iron pistons are ordinarily used, but for pumping corrosive liquids they should be of brass or bronze.

Piston Rods.—For ordinary construction piston rods are made of various grades of steel including stainless, but on first class jobs they should be of *brass*, preferably *bronze*. With composition rods, such performance as water squirting out of the stuffing box of a pitted rod is not encountered.

Piston rods may be classed as

1. Whole
2. Divided

Rods are attached to the pistons or piston and plunger by

1. Shoulder and nut
2. Tapered end and nut
3. Threaded end

These types of end construction are shown in figs. 27 to 29.

Whole or through rods are used on small pumps, but on larger sizes divided rods are preferable.

For instance, one leading manufacturer uses whole or one piece rods on sizes up to 5¼ × 3½ × 5. Larger sizes are fitted with divided rods.

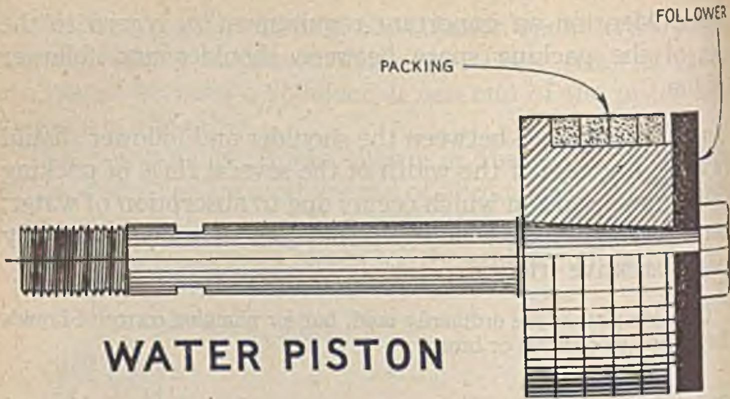
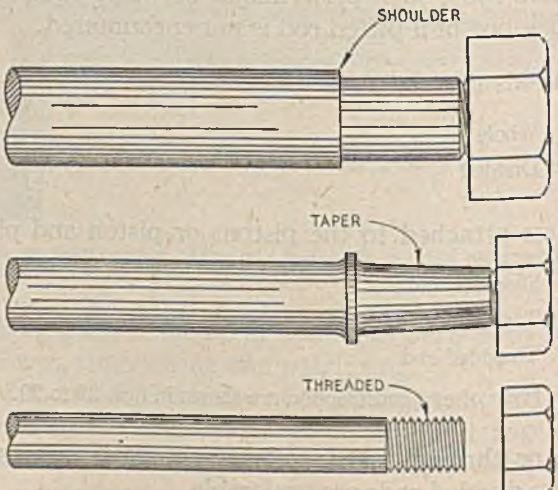


Fig. 26.—Pump water piston with divided piston rod. A water tight joint with the cylinder walls is obtained usually with several rings of fibrous packing as shown, which are held in place by a shoulder at one end of the piston and the follower or follower plate at the other.



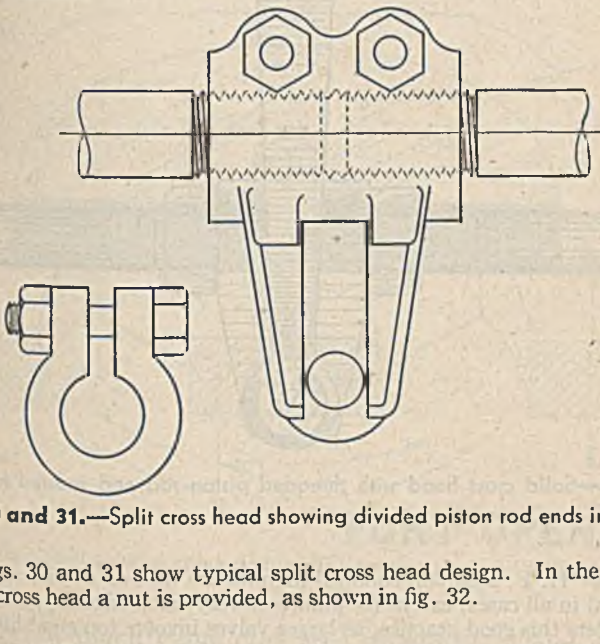
Figs. 27 to 29.—Various types of piston rods classed with respect to the method of attachment to the piston or plunger. Fig. 27, shoulder and nut; fig. 28 taper and nut; fig. 29, thread and nut.

Ques. How are the cross head ends of rods attached?

Ans. The ends are threaded and screwed into a cross head usually of the split type.

Ques. In approved construction what provision is made for turning the rods in attaching to the cross head?

Ans. The ends are milled square so they may be turned with a wrench.



Figs. 30 and 31.—Split cross head showing divided piston rod ends in position.

Figs. 30 and 31 show typical split cross head design. In the case of a solid cross head a nut is provided, as shown in fig. 32.

Valves.—It has been proven by practice after long and costly experiments in years past, that a number of small valves, instead of one or two larger ones, are more durable and tend to quieter operation, that is to say, the "slamming of the valves," especially in high speed pumps, as power pumps, is not so pronounced. In other words, the lower the *lift* the quieter the operation.

Pioneers such as Worthington, Dunham, Leavitt, Holly and others had occasion to find the truth of these statements early in their careers—and so did the author who at one time designed power pumps for city water

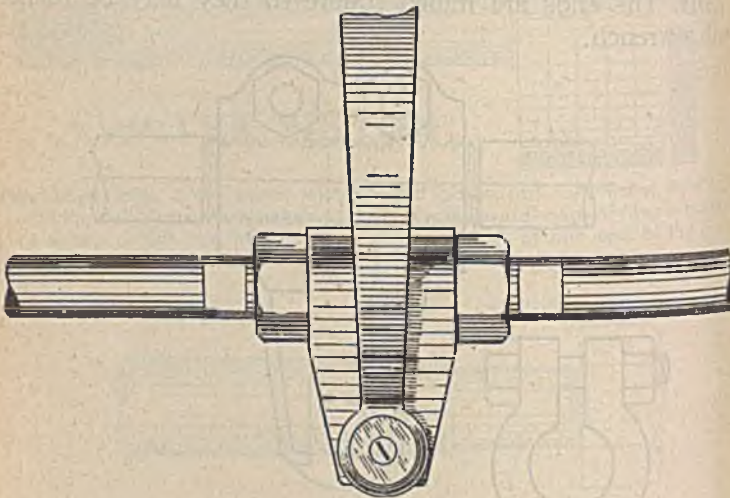


Fig. 32.—Solid cross head with threaded piston rod end secured in position by nut.

works. H. F. Dunham confined his practice to 4 or 4½ in. valves (disc valves) in all cases, except for pumps of very small capacity. The author considers this good practice, as larger valves involve too great lift, and the smaller sizes necessitate an undue multiplicity of valve units—both expensive in manufacture and entailing a greater degree of servicing.

Note the *flats* in each rod to prevent the rod turning when applying a wrench to tighten the nuts. Without the flats, a sloppy mechanic would probably use a Stillson wrench which would leave its teeth marks on the rod, disfiguring same.

The "slamming" of large valves under moderate speeds (to say nothing of the higher speed of power pump practice) proved itself a difficulty hard to overcome, until the principle of keeping the valve port area as low as possible within reasonable limits had been fully demonstrated.

It should be understood that the expression "keeping the valve port area as low as possible" refers to the area per unit and not to the total area, as in pumps of any size there are numerous valve units.

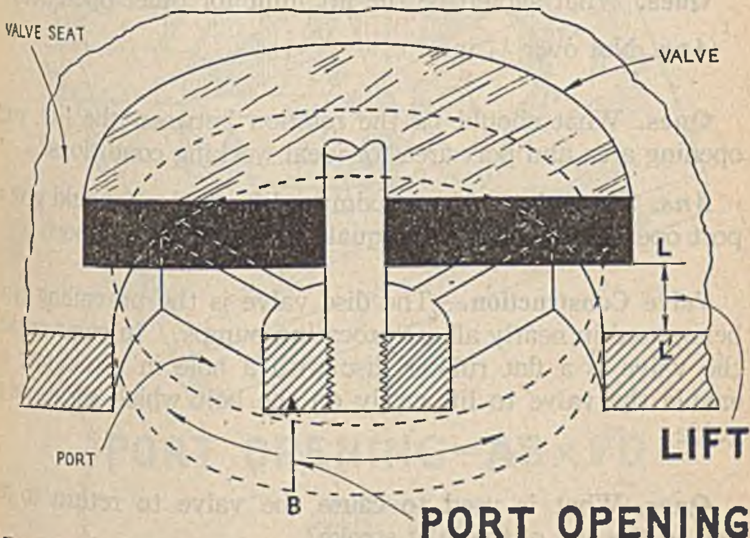


Fig. 33.—Sectional view of valve and seal illustrating terms lift and port openings.

Ques. How much valve port area should be provided?

Ans. There should be an ample number of valves so that the flow through the valve seat and out of the valve shall not exceed 200 ft. per min.—250 ft. at most.

Ques. What is the valve port opening?

Ans. The opening when the valve lifts off its seat for the water to flow out of the ports and from under the valve.

Thus in fig. 33 the valve is shown off its seat at a distance LL' which is called the *lift*, that is, it is "lifted" off its seat that distance by the force of the incoming water. When the valve is in this position the port opening is that opening represented in fig. 33 by the dotted ring whose width is AB and touching the outer sides of the ports.

Ques. What should be the lift limit for quiet operation?

Ans. Not over ¼ in.

Ques. What should be the relation between the lift, port opening area and port area for ideal working conditions?

Ans. The valve at its maximum lift (¼ in.) should give a port opening whose area is equal to the area of the port.

Valve Construction.—The disc valve is the prevailing type being used in nearly all reciprocating pumps. In construction the valve is a flat rubber disc with a hole in the center to enable the valve to lift easily on the bolt which serves as a guide.

Ques. What is used to cause the valve to return to its seat at the end of the inlet stroke?

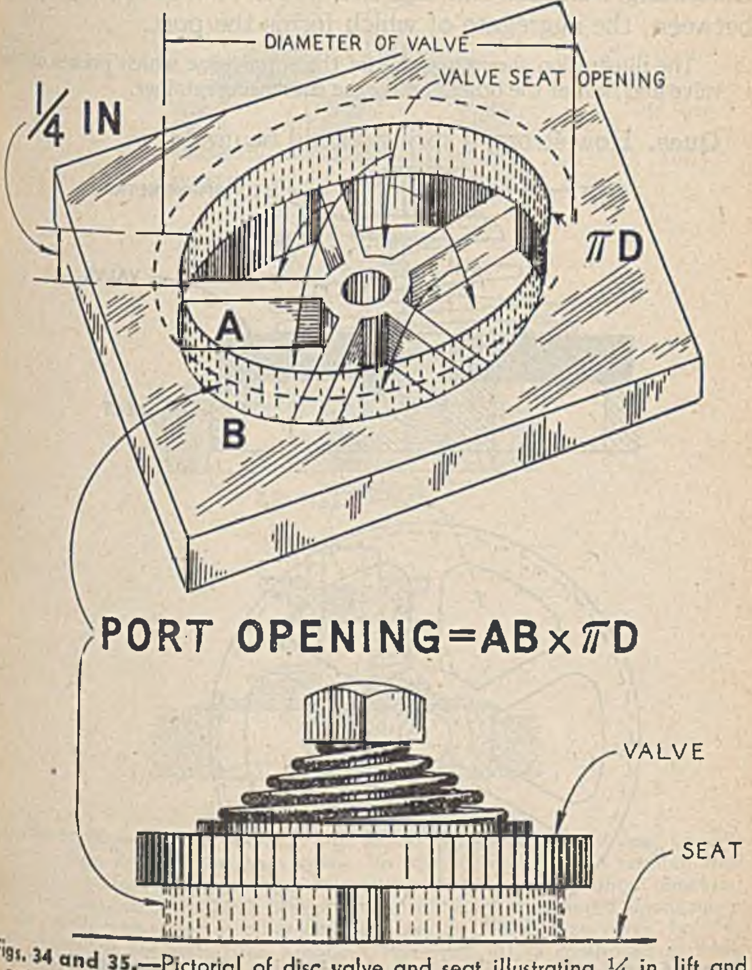
Ans. A spring.

Ques. What spring shapes are used?

Ans. Usually spiral springs, but in some cases helical springs.

Ques. Describe a typical disc valve unit.

Ans. As shown in figs. 34 and 35 the valve consists of a disc of rubber having a hole through its center, being a loose

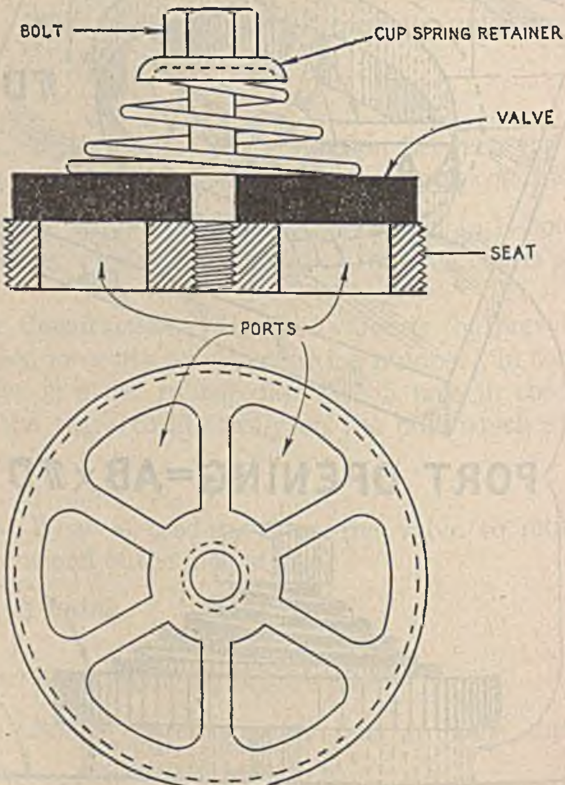


Figs. 34 and 35.—Pictorial of disc valve and seat illustrating ¼ in. lift and port opening.

fit so that it can slide vertically or lift upon a guide bolt which is screwed into a hub formed in the center of the valve seat. Extending from the hub are a number of ribs leaving spaces between, the aggregate of which forms the port.

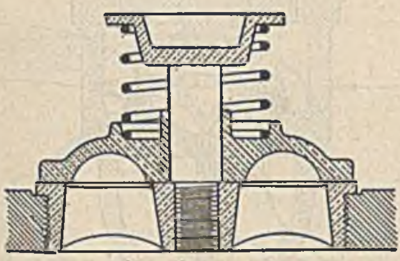
The illustration shows a spring of the spiral type which presses on the valve and held at the other end by the cup spring retainer.

Ques. How strong a spring should be used?

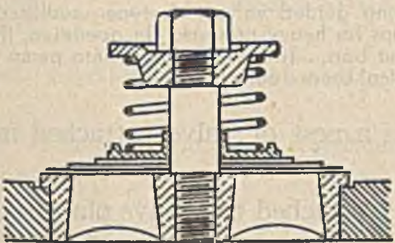


Figs. 36 and 37.—Details of typical rubber disc valve and seat. In the plan fig. 37, the port design is clearly shown.

RUBBER VALVE MARINE TYPE



BRONZE VALVE



KINGHORN VALVE

Figs. 38 to 40.—Various valves. Fig. 38, rubber valve marine type; fig. 39, metal valve; fig. 40, Kinghorn valve. In fig. 38 note built-in retainer with clamp nut on top. This construction permits changing valves without removing stud from its seat. The force fit seat shown in fig. 38 is preferred where pump must handle salt water or other corrosive liquids. The bronze valve, fig. 39, is suitable for hot water, the Kinghorn valve, fig. 40, is composed of three or more thin bronze plates. This is often preferred over the bronze valve because of its light weight.

Ans. Sufficient to firmly hold the valve on its seat, but not too firmly, especially when pumping under long lift.

Ques. Name one advantage of the disc valve.

Ans. When worn on one side it may be reversed and finally both sides may be resurfaced.

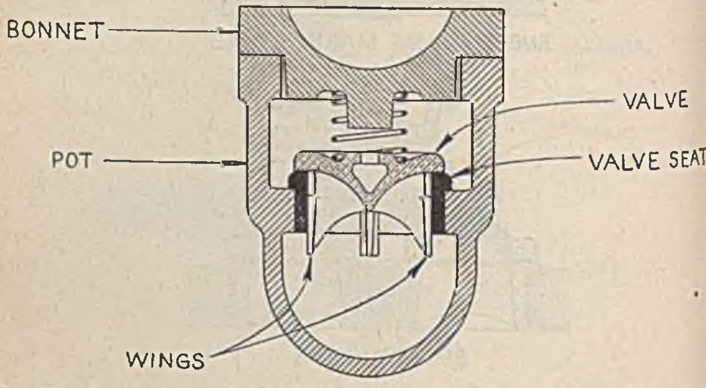


Fig. 41.—Bronze wing guided valve. A type usually standard equipment for pot valve pumps for heavy pressure. In operation, the valve works in a bronze seat without bars. The seat is forced into pump cylinder deck on a slight taper. Helical bronze coils are used.

Ques. How is a nest of valves attached indirectly to the valve deck?

Ans. They are attached to a valve plate out of which is cut the ports, the plate being bolted direct to the valve deck?

This was the method employed by the author in designing power pumps for City Water Works.

Ques. What valve material is used for pumping hot water?

Ans. A composition that will not be injured by the heat; frequently metal is used.

Ques. How is the valve seat attached to the valve deck?

Ans. It may be either screwed or forced into the deck.

Ques. What kind of rubber is used in valve construction?

Ans. It is varied to suit the requirements of the service.

The following tabulation shows general practice.

Air Pumps.................................Soft rubber

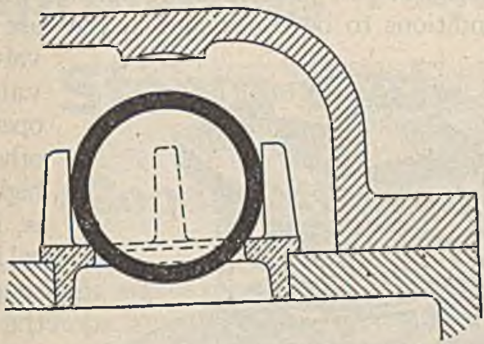


Fig. 42.—Ball valve. It is a hollow bronze ball working in a bronze seat.

Pumping up to 75 lbs. per sq. in..........Medium soft rubber
" " " 150 " " " " ....   " hard "
" " " 300 " " " " ....Hard vulcanized rubber

Ques. What reinforcement is sometimes provided for extra heavy service?

Ans. The valve is encased in a metal cap.

Such practice in the opinion of the author is not necessary as if the valve be sufficiently thick there will be no need of such reinforcement and the weight will be reduced.

Ques. What construction is suitable for pumping hot water?

Ans. The disc should be made of a composition that will not be injured by the heat, metal discs being frequently used.

Ques. What type valve is sometimes used for pumping thick liquids?

Ans. Ball valves.

The balls are usually hollow and made of bronze although a lead or iron core covered with rubber is sometimes used.

The valve is returned to its seat by gravity instead of spring pressure.

Spring Pressure.—In proportioning springs, there are two opposing conditions to be considered in the case of the inlet valves. These valves should open easily as otherwise on a high lift, that is, if the vertical height of the supply level to the valves were considerable, the lift of the valves should be small to prevent excessive *slip* or back flow, while the valves are closing.*

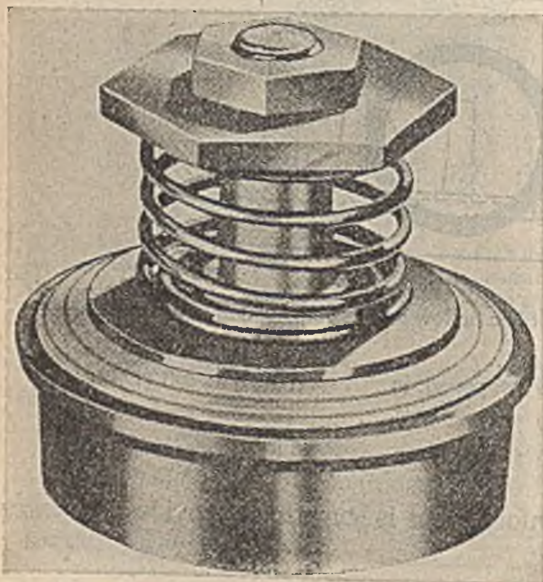


Fig. 43.—General external appearance of a bronze multi-disc valve and seat.

*NOTE.—The inlet valves under such conditions should close quickly to reduce the slip; however, too stiff a spring with very high lift may interfere with the proper working of the pump.

Various types of valves are shown in the accompanying illustrations.

CHAPTER 14

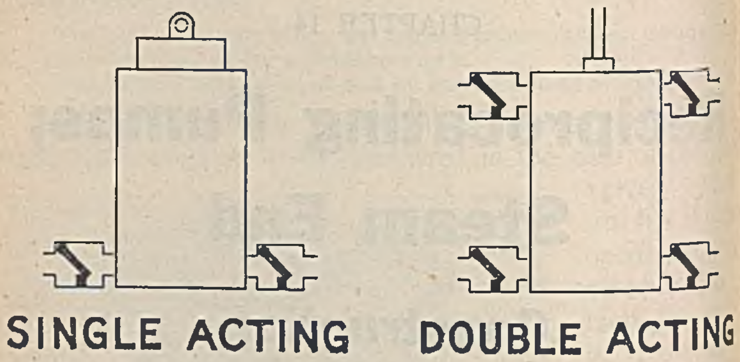
Reciprocating Pumps; Steam End Construction

Before the coming and development of the centrifugal pump, reciprocating pumps were used for practically every service. The smallest units up to the largest City Water Works pumps were of the reciprocating type. The latter, although they were highly efficient, were very bulky and expensive.

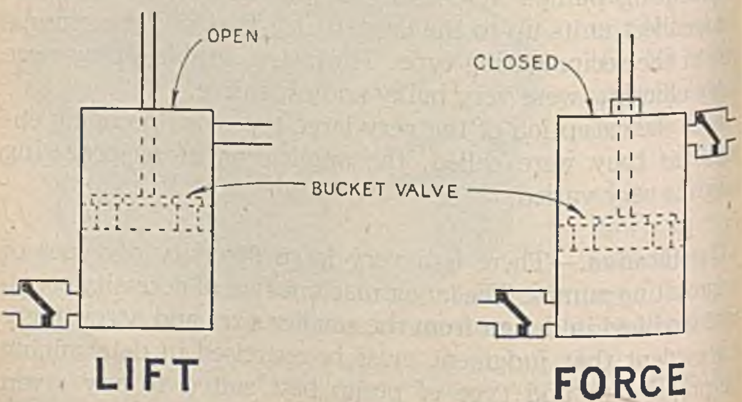
With the exception of the very large types or "pumping engines" as they were called, the application of reciprocating pumps is very varied.

Classification.—There is a very large diversity of types of reciprocating pumps. The larger machines are of necessity somewhat modified in design from the smaller sizes and accordingly it is evident that judgment must be exercised in determining the proper size and type of pump best suited to any given requirements.

A classification to be fully comprehensive would fill many pages but in the listing which follows the various types will be classified in sufficient detail for the reader to get an idea of the great variety of reciprocating pumps being made.



Figs. 1 and 2.—Reciprocating pumps 1; Single acting and double acting

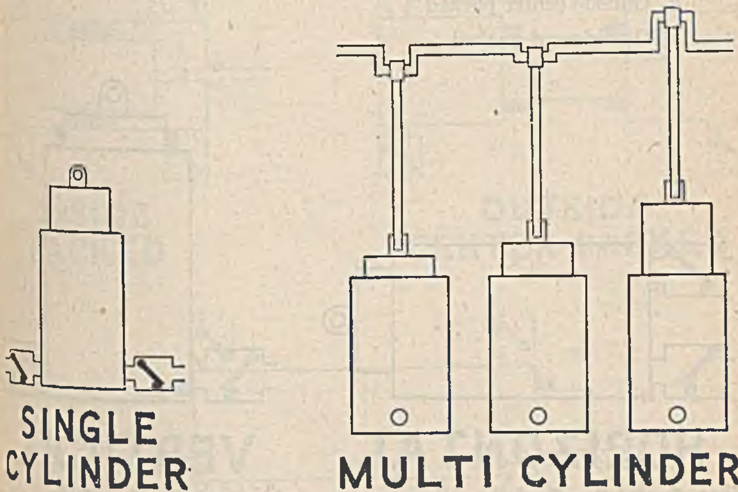


Figs. 3 and 4.—Reciprocating pumps 2; Lift and force single acting.

Reciprocating pumps may be classed:

1. With respect to the cycle of operation, as

   a. Single acting
   b. Double acting
   c. Lift
   d. Force



Figs. 5 and 6.—Reciprocating pumps 3; Single cylinder and multi-cylinder.

2. With respect to the number of cylinders, as

   a. Single cylinder
   b. Multi-cylinder

3. With respect to the position of the cylinders, as

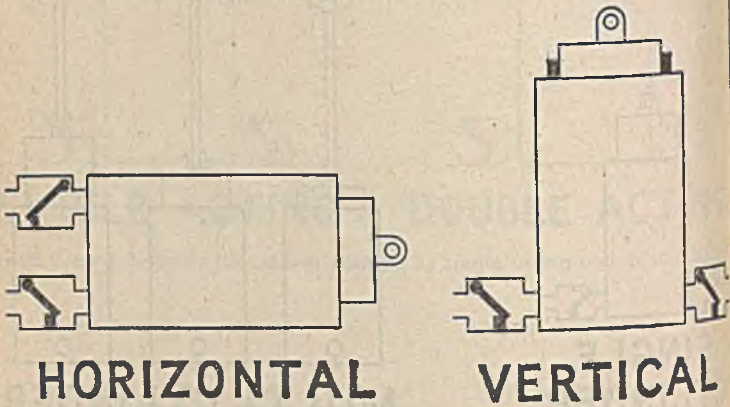
   a. Horizontal
   b. Vertical

4. With respect to the pumping element, as

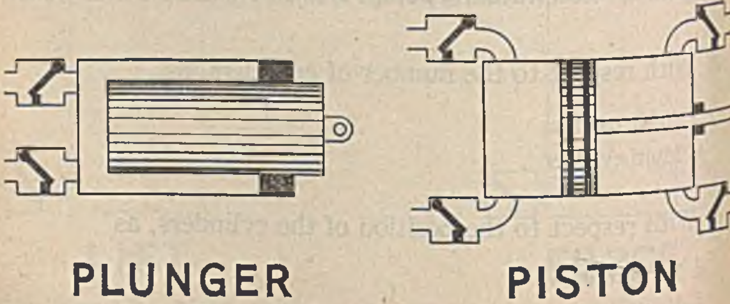
   a. Plunger
   b. Piston

5. With respect to the stuffing box, as

   a. Inside packed
   b. Outside center packed
   c. Outside end packed.



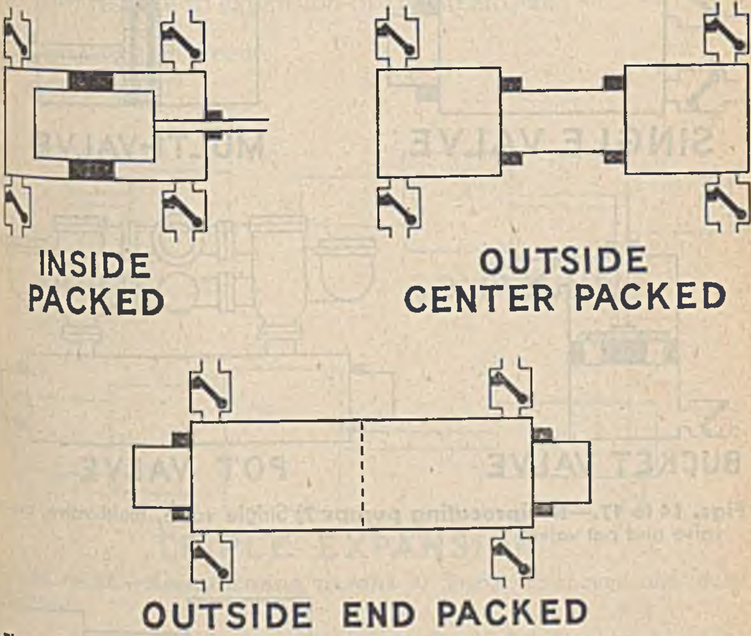
Figs, 7 and 8.—Reciprocating pumps 4; Horizontal and vertical.



Figs. 9 and 10.—Reciprocating pumps 5; Plunger and piston.

6. With respect to the valve arrangement, as

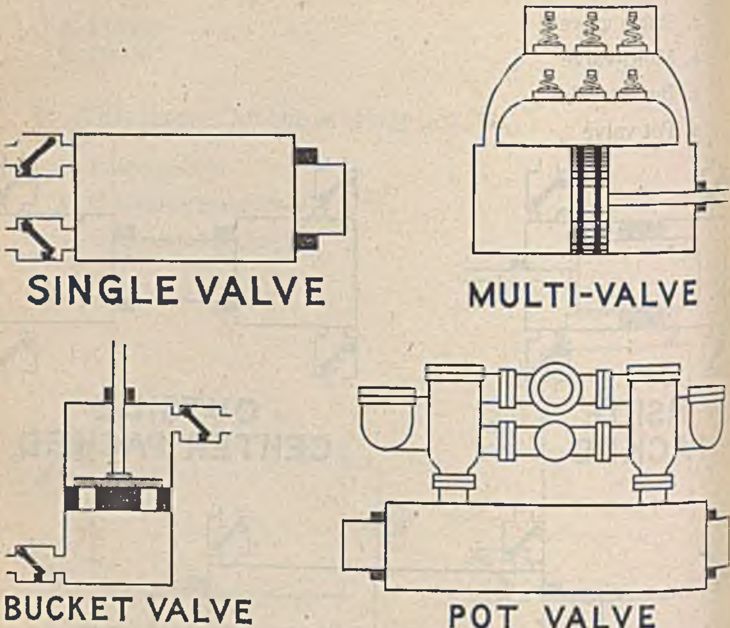
   a. Single valve
   b. Multi-valve
   c. Bucket valve
   d. Pot valve



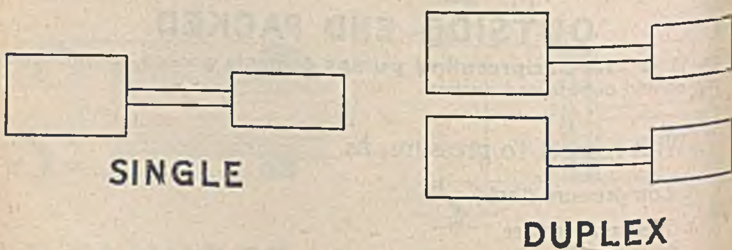
Figs. 11 to 13.—Reciprocating pumps 6; Inside packed, outside center packed and outside end packed.

7. With respect to pressure, as

   a. Low pressure (tank)
   b. Medium pressure
   c. High pressure
   d. Extra high pressure (hydraulic)



Figs. 14 to 17.—Reciprocating pumps 7; Single valve, multi-valve, bucket valve and pot valve.



Figs. 18 and 19.—Reciprocating pumps 8; Single or "Simplex" and duplex.

8. With respect to cylinder arrangement, as

   a. Single (simple)

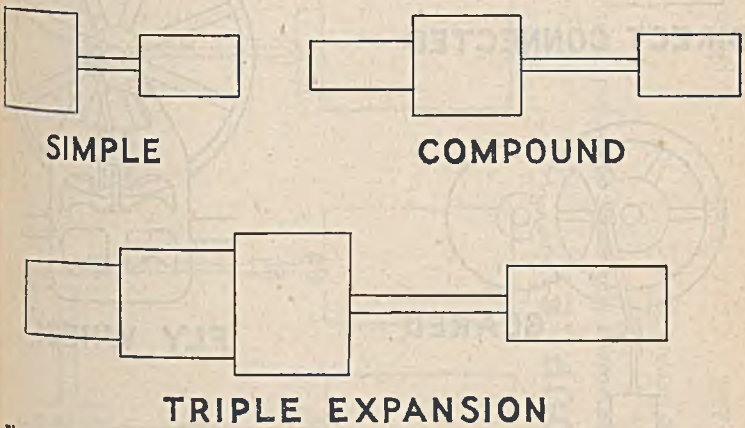
   b. Duplex

   c. Twin

9. With respect to expansion of the steam, as

   a. Simple or high pressure

   b. Compound



Figs. 20 to 22.—Reciprocating pumps 9; Simple, compound and triple expansion.

   c. Triple-expansion

   d. Quadruple expansion

10. With respect to the application of the power, as

   a. Direct connected

   b. Power {fly wheel, geared}

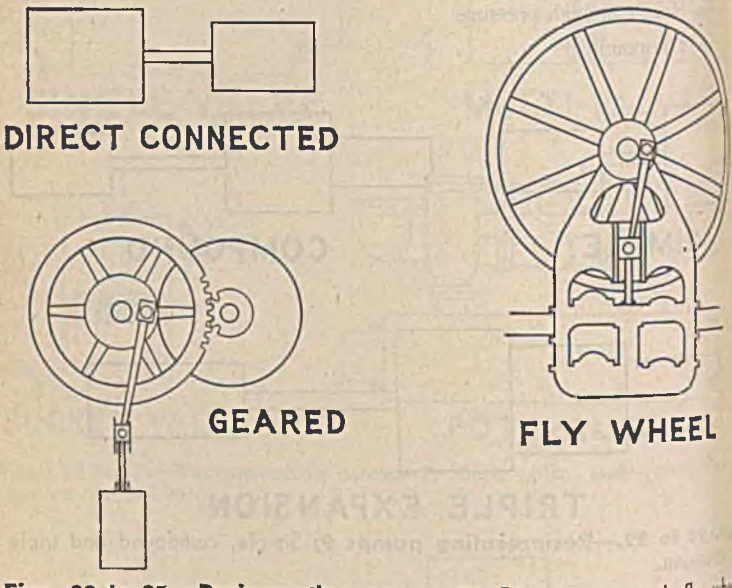
11. With respect to the kind of power, as

a. Steam

b. Internal combustion engine

c. Electric

d. Hydraulic



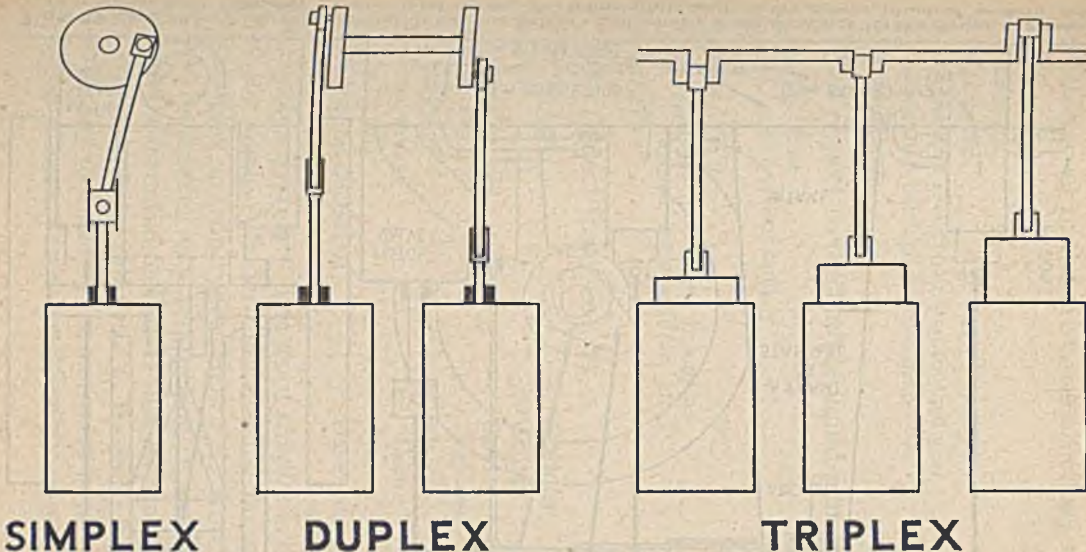
Figs. 23 to 25.—Reciprocating pumps 10; Direct connected, fly wheel and geared.

12. With respect to the number of cylinders of power pumps, as

a. Simplex

b. Duplex

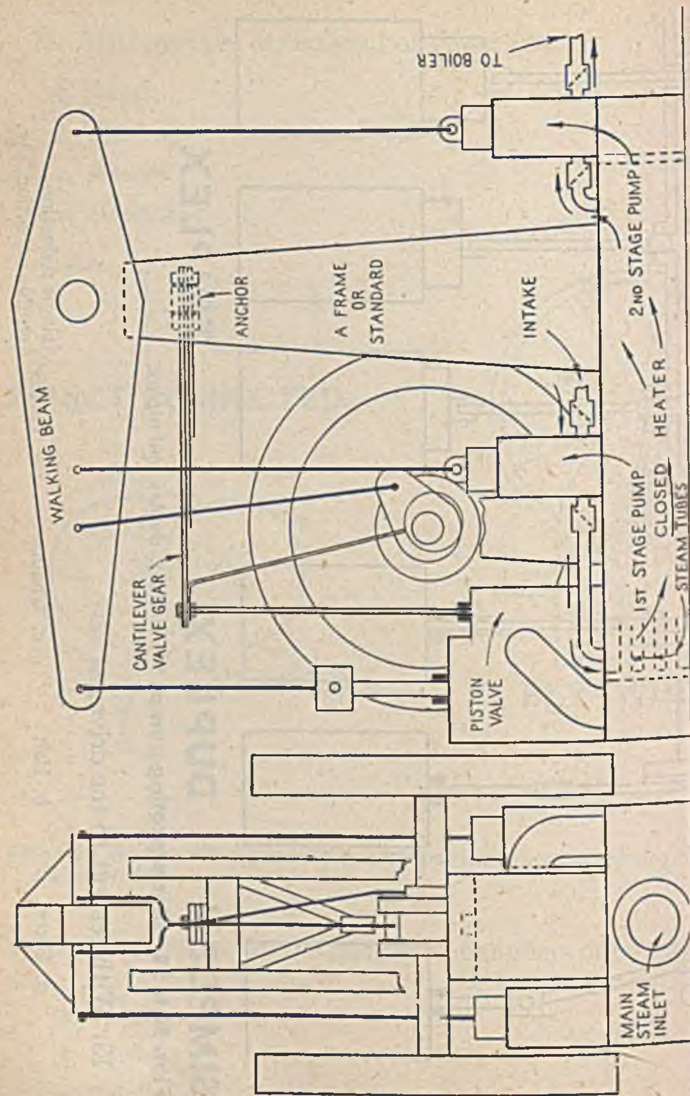
c. Triplex



Figs. 26 to 28.—Reciprocating pumps 11; Simplex, duplex and triplex.

13. With respect to the drive, as

a. Direct *b*. Belt *c*. Chain *d*. Geared {single reduction / double reduction, etc.



Figs. 29 and 30.—Authentic reproduction of the author's preliminary study drawings for the design of western river steam boat "**Doctor**" boiler feed pumps. A characteristic unit of the power plant of western river steamers.

14. With respect to the drive construction, as

a. Spur gear
b. Helical gear
c. Worm gear
d. Combination silent chain and toothed gear
e. Combination belt and toothed gear
f. Walking beam

15. With respect to the kind of liquid pumped

a. Water {clean / with grit or sand}
b. Oil
c. Sugar
d. Magma {thick and heavy liquids as syrups, molasses, tar, etc.}
e. Corrosive.

16. With respect to service, as

a. Boiler feed
b. Tank
c. General service
d. Fire
e. Hydraulic
f. Water works
g. Circulating
h. Ballast (on shipboard)
i. Wrecking
j. Mining
k. Sinking
l. Deep well
m. Air
n. "Doctor"
o. Test
p. Diesel, etc.

The Simplex System.—The word *simplex* is here used to distinguish the single cylinder direct acting pump from the two cylinder duplex or twin type. Henry R. Worthington in 1840 invented the simplex or direct acting reciprocating pump. In this pump the essential feature is that *the movement of the steam piston (which is direct connected to the water piston or plunger) is automatically operated by the movement of the piston.*

The arrangement varies considerably on pumps of different makes.

Ques. What is the difficulty in obtaining automatic action at the steam end?

Ans. The necessity of providing a special valve gear, somewhat complicated, is because of the absence of a rotating part which prevents the use of an eccentric.

Simplex Pump Valve Gears.—In most cases the necessary movements of the valve gear are obtained from three sources:

1. The movement of the piston
2. The movement of the piston rod
3. The steam pressure

A valve gear thus operated usually consists of:

1. A main valve

Which admits and exhausts steam from the cylinder.

2. An auxiliary piston

Connected to the main valve and moving in a cylinder formed on the valve chest.

3. An auxiliary valve

Controlling the steam distribution to the auxiliary piston cylinder and operated with suitable gear by the main piston or piston rod.

Ques. How does the gear just mentioned work?

Ans. As the main piston approaches the end of the stroke, it moves the auxiliary valve.

Ques. What results from this movement?

Ans. It causes steam to be admitted to one end of the auxiliary piston and exhausted from the other resulting in a movement of the auxiliary piston.

Ques. What happens when the auxiliary piston moves?

Ans. The movement of the auxiliary piston moves the main valve.

The steam distributed to the main cylinder, thus affected, reverses the motion of the main piston, and the return stroke takes place, completing the cycle.

Ques. What detail varies mostly in pumps of different makes?

Ans. The auxiliary valve and the method by which it is operated.

With respect to these features the majority of pumps may be divided into two classes as these having:

1. A separate auxiliary valve.
2. Auxiliary valve and auxiliary piston combined.

Simplex Gears with Separate Auxiliary Valve.—In pumps of this type the **auxiliary valves** *usually have* **stems** *or* **tappets** *which project into the cylinder at the ends and are moved by contact with the main piston as it nears the end of the stroke.*

An example of this class is shown in fig. 31.

Each auxiliary valve I has a short stem which projects into the cylinder.

Ques. What happens when the piston strikes one of the auxiliary valves I?

Ans. The valve is driven back and opens an exhaust passage E, from the corresponding end of the auxiliary piston F, which immediately shifts under pressure of live steam on the opposite side of the auxiliary piston head.

Ques. What is the object of the little hole in each end of the auxiliary piston?

Ans. When both auxiliary valves are closed the steam passing through these holes leaves the auxiliary piston entirely surrounded by live steam and accordingly in perfect balance endwise.

Ques. How long does this balance last?

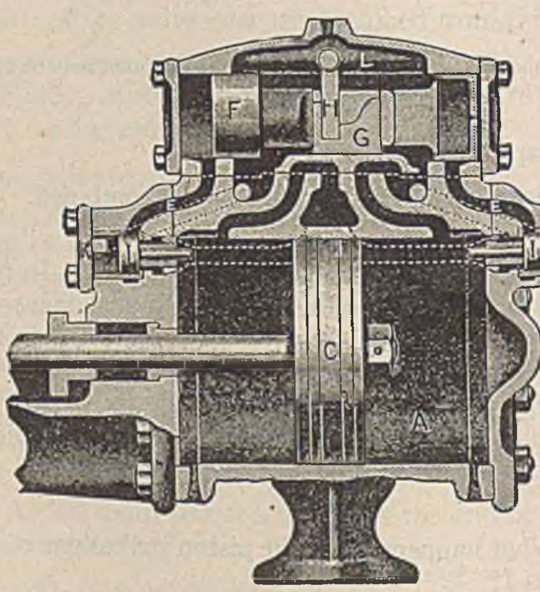


Fig. 31.—Simplex pump of the separate auxiliary valve type. In operation the piston as it nears the end of each stroke strikes the stem and lifts the valve (?) off its seat. This allows the exhaust steam behind the piston valve to escape. The live steam pushes the piston toward the exhausted end carrying the main slide valve along with it.

Ans. Until the main piston strikes the stem in the opposite cylinder head, at which time the valve moving operations are repeated in the opposite direction.

Ques. With what does the space back of the auxiliary valves communicate?

Ans. The steam chest.

The connecting passages are shown in dotted lines. The valve is therefore closed by steam pressure as soon as the piston moves back from the stem.

Ques. What should be noted about the piston?

Ans. It closes the exhaust passage before the end of the stroke.

Ques. What is the reason for this?

Ans. To trap the steam so as to form a cushion between the piston and the cylinder head.

Ques. How is the piston started on the return stroke?

Ans. Sufficient steam is admitted through a little passage cut in the cylinder wall to start the piston.

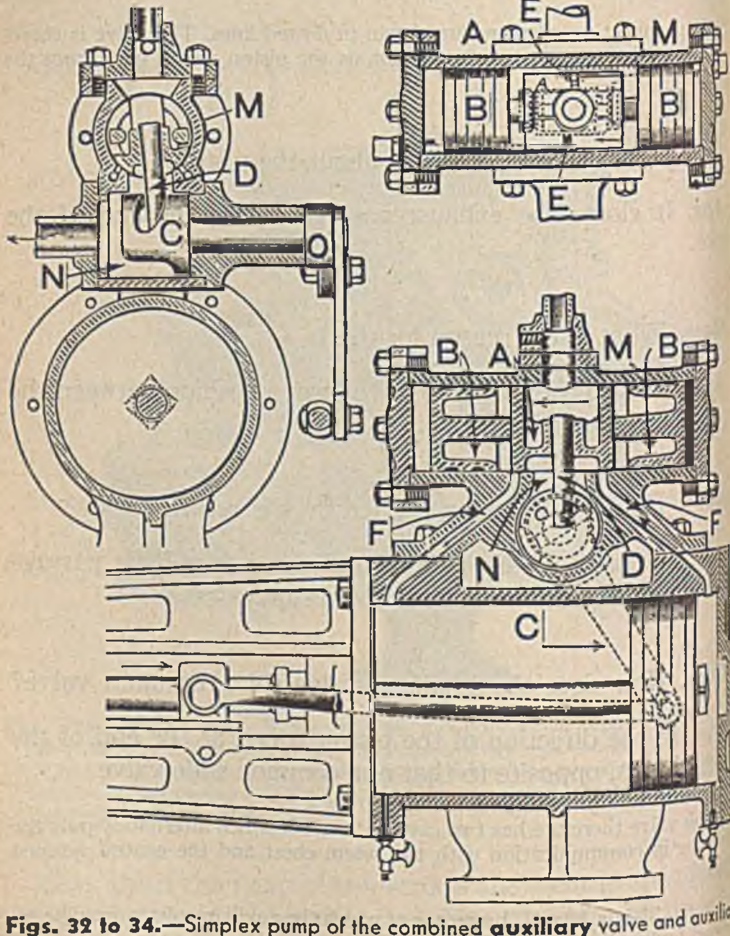
Ques. How does the auxiliary valve shift the main valve?

Ans. In the direction of the piston travel at the end of the stroke, that is, opposite to that of a common slide valve.

This valve therefore has two cavities, each of which alternately puts the cylinder in communication with the steam chest and the central exhaust port.

In fig. 31 is a lever L by means of which the auxiliary piston may be reversed by hand when expedient.

Simplex Gears with Auxiliary Valve and Auxiliary Piston Combined.—With this arrangement *an initial rotary motion is given the* **auxiliary piston** *by the external gear causing it to*



Figs. 32 to 34.—Simplex pump of the combined **auxiliary** valve and auxiliary piston type as described in the text.

uncover ports which give the proper steam distribution for its linear movement.

An example of this class is shown in figs. 32 to 34.

The main valve is operated by a positive mechanical connection between it and the main piston rod, also by the action of the steam on the valve pistons.

Fig. 32 shows the details of the valve gear and steam cylinder.

Ques. Describe the construction referring to figs. 32 to 34.

Ans. The steam end consists of the cylinder M, valve A, and valve pistons B and B. These pistons are connected with sufficient space between them for the valve A, to cover the steam ports F and F, as in fig. 34.

Ques. How is the valve operated?

Ans. By the steel cam C, (fig. 32) acting on a steel pin D, which passes through the valve into the exhaust port N, in which the cam is located.

Ques. What is provided in addition to this positive motion?

Ans. Steam is alternately admitted to and exhausted from the ends of the valve piston through the ports E and E, which moves the pistons B and B, fig. 33.

Ques. Describe the operation of the pump.

Ans. Assuming the pump to be at rest with valve A, covering the main steam ports F and F_1, in which position the cam C, holds the main steam valve by means of the valve pin D, so that ports E and E, admit steam to one end of the valve piston at the same time connects the other end with the exhaust port.

The steam acting on the valve pistons moves both, opening the main ports F and F, admitting steam to one end of the steam cylinder and opening the other end to the exhaust.

Ques. What happens if the valve occupy any other position than the one described?

Ans. The main ports F and F, will be opened for the admission and exhaust of steam.

It is accordingly clear that the pump will start from any point of the stroke.

Ques. What happens on the admission of steam to the cylinder?

Ans. The main port F, the main piston, cam and valve will move in the direction indicated by the arrows in fig. 34.

Ques. What does the first movement of the cam do?

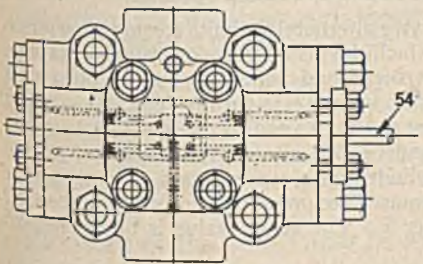
Ans. It oscillates the valve preparatory to bringing it into proper position for the opening of one of the auxiliary steam ports E, to live steam, and the other to exhaust, also to close the valve mechanically just before the main piston reaches the end of its stroke.

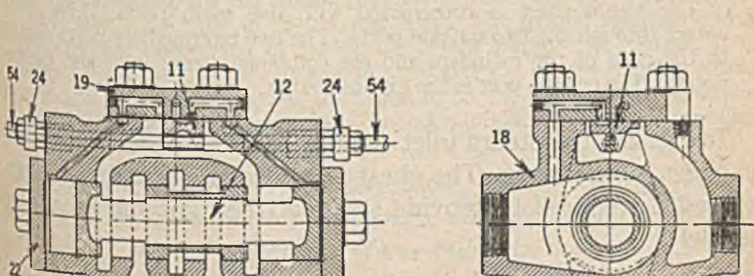
This causes a slight compression and fully opens one of the ports E to steam and the other to exhaust. By the admission of steam to one end, the other being open to the exhaust, the valve pistons move the valve to allow the admission and exhaust of steam from the cylinder for the return stroke.

Simplex Gears Piston Steam Valve Type.—The single direct acting pump steam valve mechanism here described is a recent development. Changing conditions of service, increasing steam pressures, and high temperatures have necessitated the use of piston steam valves in replacement of the conventional slide valve heretofore employed, for the reason that the movement of large, unbalanced slide valves may be sluggish, resulting in uneven and noisy pump operation, and in rapid wear on the steam valve and on the cylinder face on which the valve operates.

This gear is designed not only for high pressure high temperature steam, but also for service where the pump must operate without steam cylinder lubrication, as in marine installations.

In the valve operating mechanism shown in figs. 35 to 37 the reciprocating motion of the piston rod is transmitted to the





Figs. 35 to 37.—Simplex pump gears, piston steam valve type.

pilot valve 11, through the piston rod spool 34, the lever 39, the lost motion block 47, adjustable valve rod link 49, with tappet collars, and the steam valve rod 54. The small pilot valve is of the D slide type.

The slide type pilot valve is used in this new gear, as such a valve is more easily tightened originally and reconditioned when worn, than is a piston valve of the very small size which would be required in these pumps.

The pilot valve is inverted and operated on a separate valve plate with both the plate and the valve readily removable, without disturbing any other parts.

Steam and exhaust ports drilled in the pilot valve plate register with ports drilled in the steam chest 18, which ports lead to the ends of the piston valve chamber in the chest and to exhaust opening.

Reciprocation of the pilot valve alternately admits steam to and permits exhaust from the chamber in which the piston valve operates. This piston valve chamber in the chest is bored and honed for the reception of the ground piston valve. This piston valve 12, in turn reciprocates, admitting steam alternately from the chest to the two ends of the main steam cylinder and from the cylinder to exhaust. This alternate admission of steam to the ends of the main steam cylinder with synchronized exhaust from the opposite end results in the required reciprocating movement of the steam piston 7, in the steam cylinder 1. The steam valve is of the balanced piston type.

The provision for externally changing the length of travel of the pilot valve permits easy adjustment of the valve gear to care for wide variations in speed and in relative pressures in the steam and liquid ends of the pump.

The steam chest is constructed with five main ports. Live steam enters through the two outside ports. The two intermediate ports connect to the ends of the cylinders and are combined steam inlet and exhaust ports. The central port is the exhaust outlet.

In fig. 35, the steam inlet to the chest is at the left and the exhaust at the right. The chest is symmetrical and can be reversed if required to provide steam at the right and exhaust at the left.

In the piston valve 12, three ports are provided, the two outer ports for steam inlet and the center for exhaust. The valve is hollow and the steam ports intercommunicate. There is consequently a flow of steam from two sources when the ports are uncovered. There is but one main port to each end of the main steam cylinder. Two communicating openings are provided between each of these ports and the cylinder bore.

The small opening at the end of the cylinder has two functions. Through it, a small volume of steam is admitted to the cylinder at the beginning of the stroke, while the piston covers the inner and larger ports, preventing admission of steam at that point.

By retarding the admission of steam at this beginning of the stroke, the moving parts of the pump and water column are gradually actuated up to the point where the main steam port is uncovered by the steam piston, thus insuring that the pump will operate without shock or water hammer.

When the piston advances so that the inner, larger steam port is uncovered, steam enters the cylinder through both openings. Near the end of the stroke, the larger inner port through which the spent steam has been passing to exhaust is closed by the piston.

The steam remaining in that end of the cylinder is, therefore, trapped and compressed, providing cushion to absorb the inertia of the moving parts and of the water column and giving smooth, quiet deceleration. As the trapped steam then passes out through the starting port, the piston continues its movement slowly until reversal occurs.

The stroke of the pump may be lengthened by an increase in the distance between the nuts on the outside adjustable valve link.

This increase in the lost motion retards the movement of the pilot valve. Conversely, a decrease in the amount of the lost motion will result in a corresponding shortening of the stroke of the pump. In general, it is desirable to so adjust the lost motion as to give the longest possible stroke attainable, without permitting the piston to hit the heads, as a long stroke results in most satisfactory operation and minimum steam consumption.

The Duplex System.—By definition, a duplex direct acting pump is *a combination of two pumps arranged side by side and so connected that the piston rod of one pump in making its stroke, acts through a simple mechanism to move the valve which admits steam to the cylinder of the other after which it finishes its stroke* *and waits for its own steam valve to be acted upon by the movement of the piston of the other side before it can make its own return stroke.*

Ques. Name a desirable characteristic of the duplex pump.

Ans. There is no dead point at any stage of the stroke.

Ques. Why?

Ans. Because one or the other of the steam ports is always open.

Ques. What is the comparison between duplex valves and simplex valves?

Ans. The valves of a duplex pump are mechanically operated and not the steam thrown valves necessary with the simplex pumps.

Ques. Why does a duplex pump have five ports for each cylinder?

Ans. In addition to the three ports necessary for operation two other ports are required to provide cushioning.

It will be seen from fig. 38, that the valve seat has five ports, giving separate steam and exhaust passages and a central exhaust cavity as shown.

The passages Q and K, nearest the ends are *steam passages*, and the inner passages O and R, are for exhaust. These inner passages are covered or closed by the piston just before the end of the stroke whereby a portion of the exhaust steam is compressed and made to act as a cushion between the piston and cylinder head, thus preventing the piston striking the cylinder heads when operating at high speed.

Ques. Describe the steam distribution with the piston approaching the end of the stroke.

Ans. In the position shown in fig. 39, the valve covers four ports. Steam is here admitted through the passage at the right end, causing the piston to move as indicated by the arrow. During this the steam port at the other end is closed and the exhaust port at that end is open, but the exhaust passage is closed by the piston, which traps some of the steam and cushions the piston.

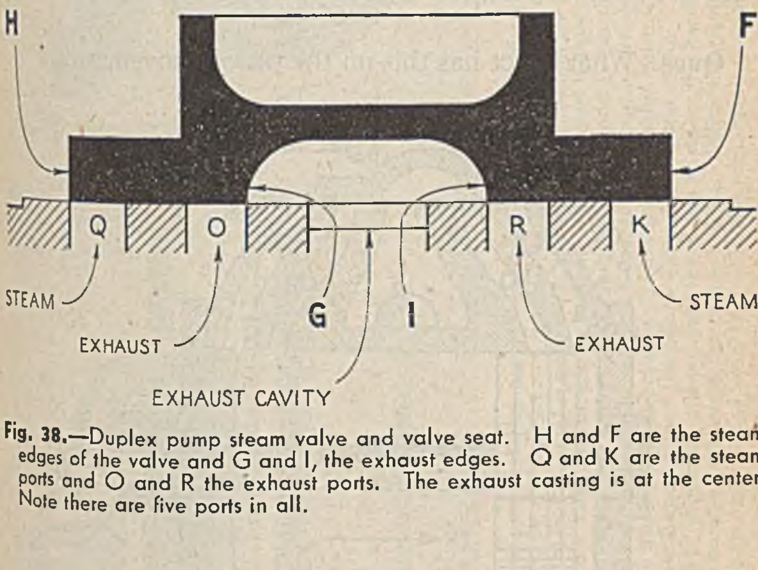


Fig. 38.—Duplex pump steam valve and valve seat. H and F are the steam edges of the valve and G and I, the exhaust edges. Q and K are the steam ports and O and R the exhaust ports. The exhaust casting is at the center. Note there are five ports in all.

Ques. How is the degree of cushioning regulated in some pumps?

Ans. By cushion valves.

Ques. Where are they located?

Ans. These valves are placed in a passage leading from the steam port to the exhaust port at each end as indicated in fig. 39.

Ques. Explain their operation.

Ans. When a cushion valve is partly open, some of the steam compressed in the clearance space and its steam port escapes into the exhaust port, thus reducing the cushioning effect.

Ques. What effect has this on the piston movement?

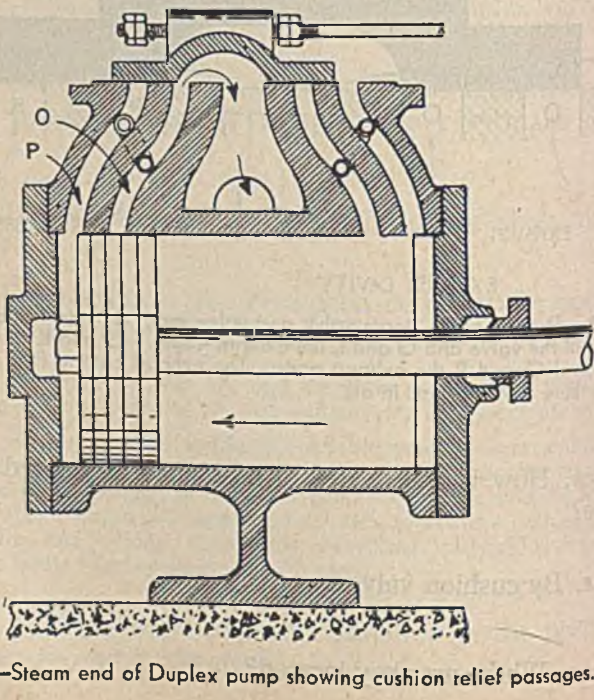
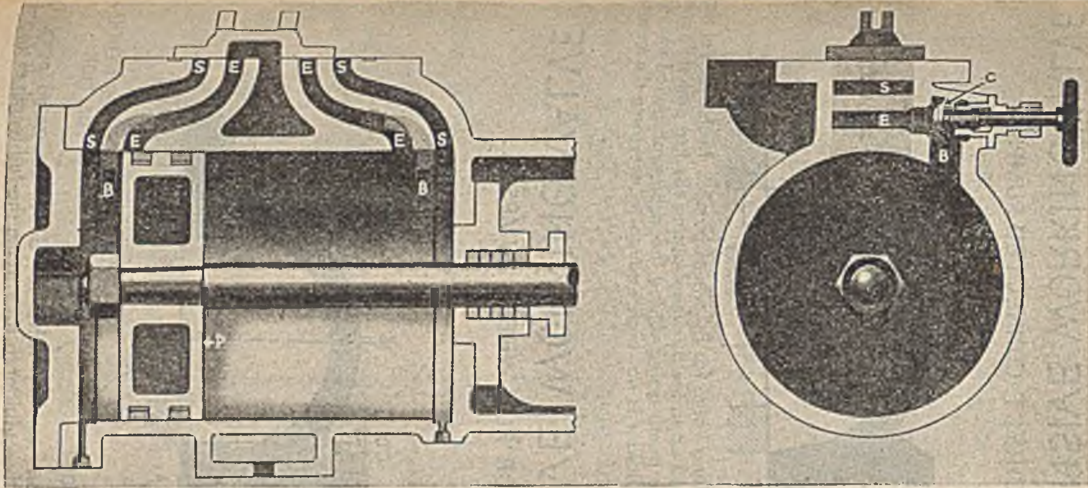


Fig. 39.—Steam end of Duplex pump showing cushion relief passages.



Figs. 40 and 41.—Steam end of duplex pump illustrating cushion valve. In construction, passage B, connects with the exhaust port E, by a drilled hole covered by cushion valve C. When the piston P, nears the end of its stroke, it runs over the port E, and traps a portion of the exhaust steam in the space between the end of the cylinder and the piston. This forms an elastic cushion, which prevents the piston striking the end of the cylinder or head. The cushion steam cannot escape through exhaust port E, because the port is covered by the piston P, and it cannot escape through steam port S, because this port is covered by the slide valve, which is held down by the live steam in the steam chest. However, if cushion valve C, be opened, it will pass out through B, into the exhaust port E, and the piston will make a longer stroke. Therefore, when pump is running the cushion valve C, should be opened until the piston P, strikes the heads, and then the cushion valve should be closed just enough to prevent the piston striking. When the pump is running slow or working hard, the cushion valve may be opened wider than when it is running fast or under light load.

Fig. 42.—Expansive working valve for fly wheel pump. In this class of p... the dynamic inertia stored up in the fly wheel during the first part of the st... (up to cut off) keeps the pump going during the expansion period where ... pressure in the steam cylinder is not high enough to overcome the resist... at the water end.

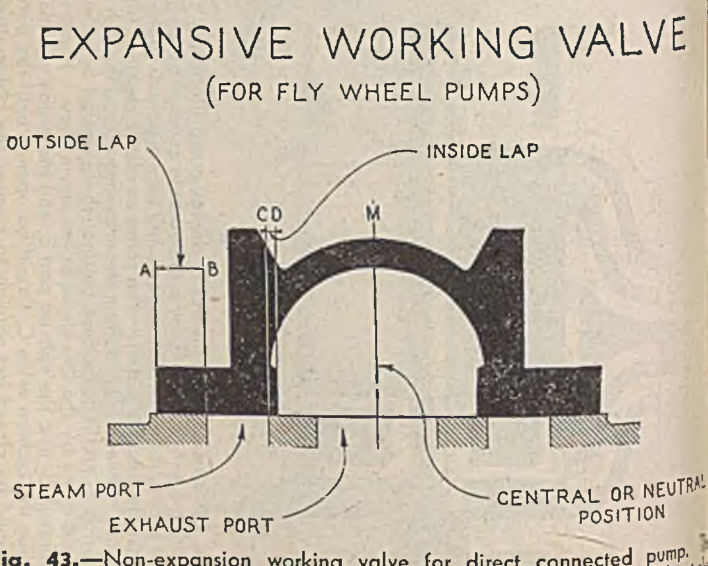


Fig. 43.—Non-expansion working valve for direct connected pump. ... absence of lap allows the cylinder to take steam the full length of the stroke

Ans. The piston moves closer to the cylinder head before it stops.

The need for cushioning increases with the speed of the pump. Where the speed is increased it is necessary to increase the cushioning effect by partly closing the cushion valves. These valves provide a simple means of obtaining a full stroke of the piston without danger to the pump whether its speed be fast or slow.

Ques. How are the valves proportioned?

Ans. They have no **outside lap** nor **inside lap.**

Ques. What is lap?

Ans. It is that portion of the valve face which overlaps the ports when the valve is in its central or neutral position, as in fig. 43.

Ques. What is the difference between outside lap and inside lap?

Ans. Outside lap A B, is lap referred to the steam port and inside lap C D, is lap referred to the exhaust port, as in fig. 43.

Ques. Why do the valves have no lap?

Ans. Because a direct acting pump must take steam the whole length of the stroke and have no compression except that needed for cushioning. See fig. 42.

According in neutral positions the valves just cover the steam ports leading to opposite ends of the cylinders. Note the relation as shown in fig. 38.

The Duplex Pump Valve Gear.—The mechanism or gear which operates the valves of a duplex pump consists (for each cylinder) of a:

1. Cross head
2. Long rocker arm
3. Rocker shaft
4. Short rocker arm
5. Connecting link
6. Valve stem

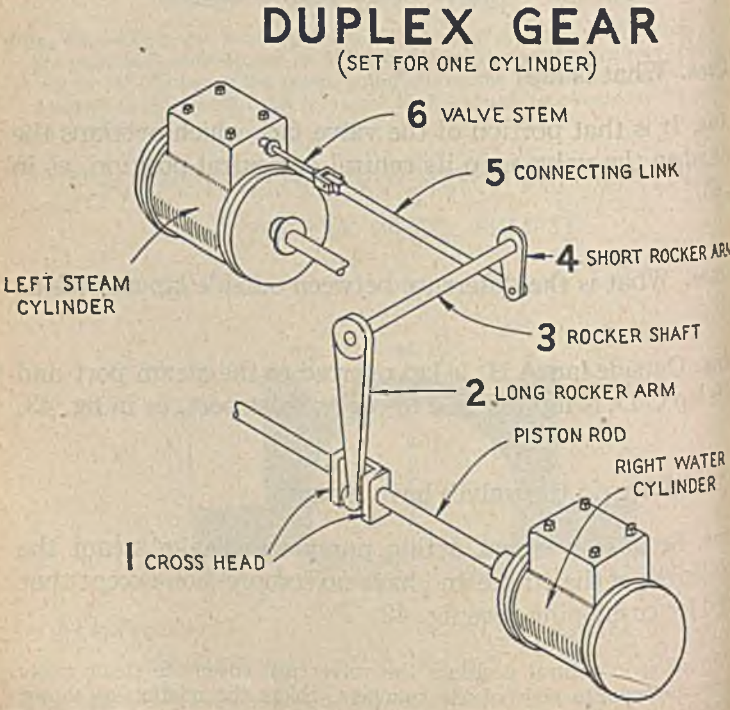


Fig. 44.—Duplex valve gear shown for one side of the pump.

On the pump there are two sets of these parts—one set for each cylinder. However, in order to clearly show the assembly one set only is shown in fig. 44.

Referring to the illustration, the source of motion for operating the valve of the left steam cylinder is obtained from the piston rod of the cylinder on the other side. As shown there is a cross head **1**, attached to the rod in which works a long rocker arm **2**. The reciprocating motion of the rod imparts an oscillating motion to the long rocker arm. This rocker arm is attached to a rocker shaft **3**.

The rocker shaft works in a long bearing which is not shown in order to make working parts visible.

At the other end of the rocker shaft is attached a short rocker arm **4**, which in operation rocks or oscillates in unison with the long rocker arm.

The oscillating motion of the short rocker arm imparts a reciprocating motion to the valve stem **6**, through the connecting link **5**.

Ques. Name one detail not shown in fig. 44.

Ans. Means for introducing "lost motion" in the operation of the gear.

Duplex Valve Gear "Lost Motion."—*There is always a lost motion between the slide valve and the valve rod operating it so that the valve does not move until the piston on the one side which operates it, has travelled some distance.*

This affords a short *pause* in the flow of water at the end of a stroke and gives the water valve an opportunity to seat quietly before the reverse stroke takes place. The piston on the other side in the meantime having renewed its movement, tends to lessen pressure and flow fluctuations.

Ques. How long is the lost motion pause in general practice?

Ans. From about one-quarter to possibly one-half the whole stroke of the piston, depending upon the amount of lost motion in the valve gear.

Ques. Is the lost motion adjustable?

Ans. On some pumps yes; on others no. The lost motion should be adjustable on all pumps.

"Lost Motion" Mechanism.—On some pumps this detail is inside the valve chest where it can't be reached without stopping, cooling the pump and taking off the valve chest head. Such designing is stupid.

On other designs it is placed outside forming a part of the connecting link (5, fig. 44) where it ought to be—easily accessible.

With this arrangement the lost motion may be adjusted while the pump is in operation. In fact the pump should be in operation to properly "tune up" the lost motion.

The lost motion may be either

1. Fixed or
2. Adjustable

and the mechanism may also be classed with respect to location as,

1. Inside
2. Outside

Fig. 46 shows an *inside fixed* or *non-adjustable* mechanism.

As shown, it consists of a block threaded on the valve rod, the block being a loose fit between two lugs on the steam valve, the amount of space between the block and the lugs constituting the lost motion.

This arrangement might go on very small or cheap pumps, but in a first class design such makeshift would not be tolerated.

This inside fixed lost motion is shown in more detail in fig. 45. This illustration also shows the short rocker arm in its two extreme positions

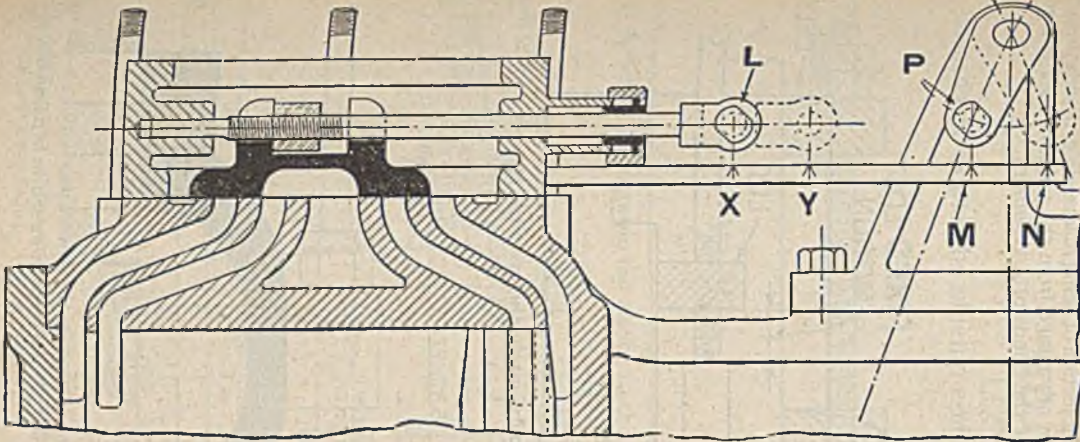


Fig. 45.—Valve and short rocker arm detail of duplex pump showing short rocker arm in its extreme positions M and N, and the corresponding position of the valve stem end X and Y.

M and N, and the corresponding position of the valve stem end X and Y, also the standard which carries the sleeve bearing for the rocker shaft.

Another inside design is shown in fig. 48, which is adjustable.

Here two nuts are provided screwed on the valve rod at end of lug so that the space between the nuts and the lug can be increased or decreased as may be desired.

The arrangement has the objection of being inaccessible, despite the often advanced claim that it is an advantage from the point of view of a protection against tampering at the hands of a greenhorn.

Fig. 49 shows a first class outside adjustable mechanism.

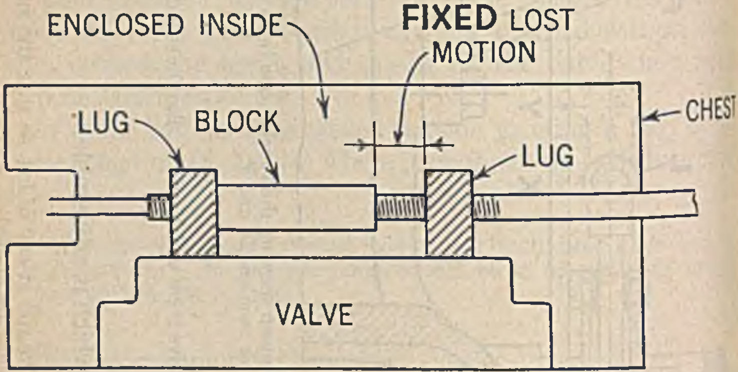


Fig. 46.—Inside **fixed** lost motion mechanism as made for very small or chea pumps.

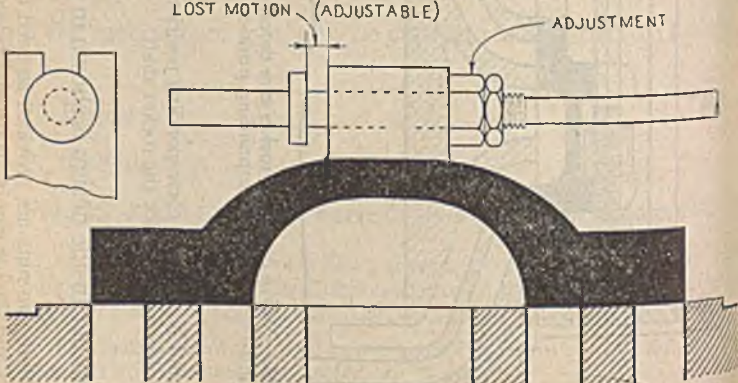


Fig. 47.—Inside **adjustable** lost motion mechanism shown in detail also placement of short rocker arm.

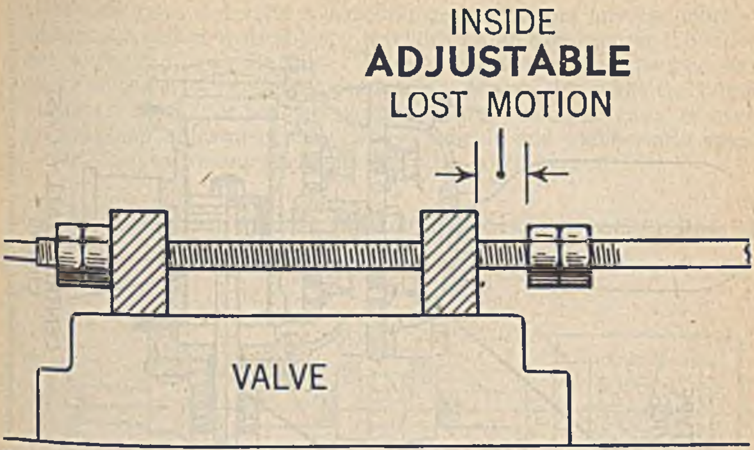


Fig. 48.—Inside **adjustable** lost motion mechanism.

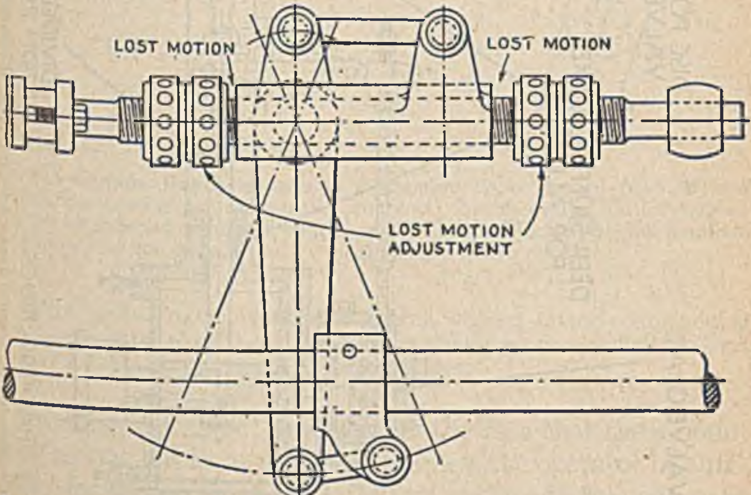


Fig. 49.—Outside **adjustable** lost motion mechanism as usually designed for large pumps. In this arrangement the lost motion can be adjusted while the pump is in operation.

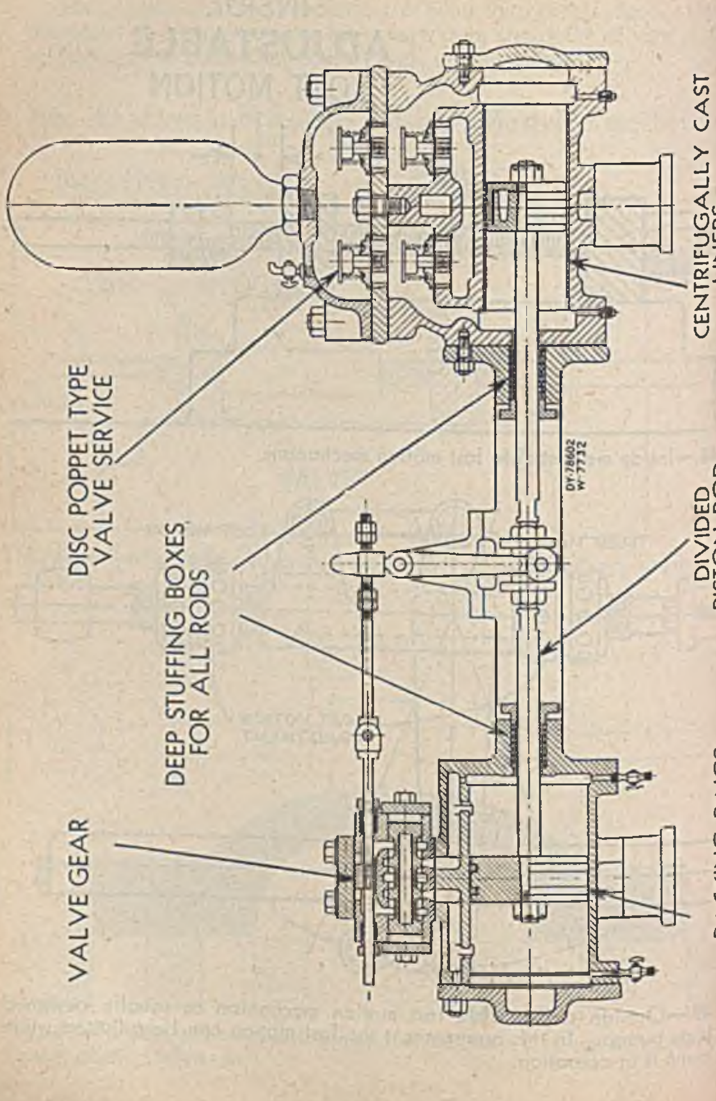


FIG. 50.—Simplex horizontal pump showing general construction. See page 280 for duplex pump construction.

The slide valve and valve rod move together without any lost motion, the clearance or lost motion being provided on an extension of the valve stem (corresponding to the link). In this construction the collars or nuts may be adjusted to increase or decrease the lost motion while the pump is running. Since it is not necessary to remove the chest cover, or even stop the pump, adjustment may be very quickly and conveniently made "tuning" the gear to any change in operating conditions.

Fig. 51 shows an *outside fixed* lost motion arrangement, a type designed mostly for large and expensive pumps.

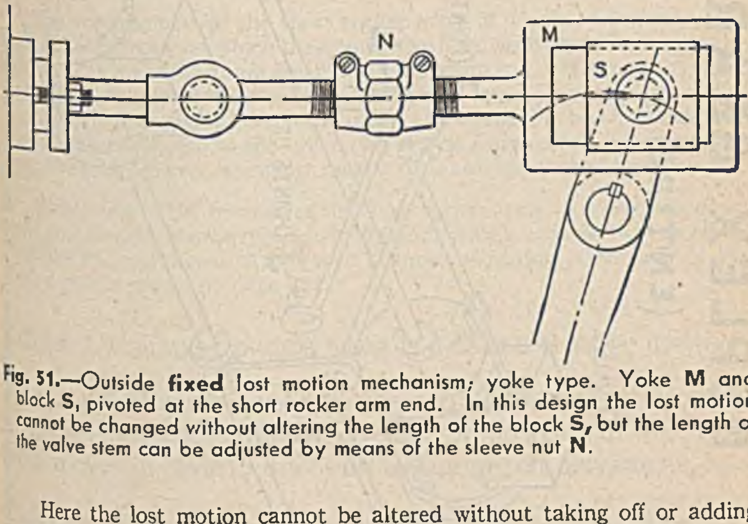


Fig. 51.—Outside **fixed** lost motion mechanism; yoke type. Yoke **M** and block **S**, pivoted at the short rocker arm end. In this design the lost motion cannot be changed without altering the length of the block **S**, but the length of the valve stem can be adjusted by means of the sleeve nut **N**.

Here the lost motion cannot be altered without taking off or adding to the ends of the block S, but the sleeve nut in the connecting link is a good device for altering the length of the valve stem.

According to Raabe: "It is seldom the case that the amount of lost motion has to be altered and unless the operator be thoroughly familiar with the details and design of the pump he should not undertake such alterations, as the designer knows best the requirements."

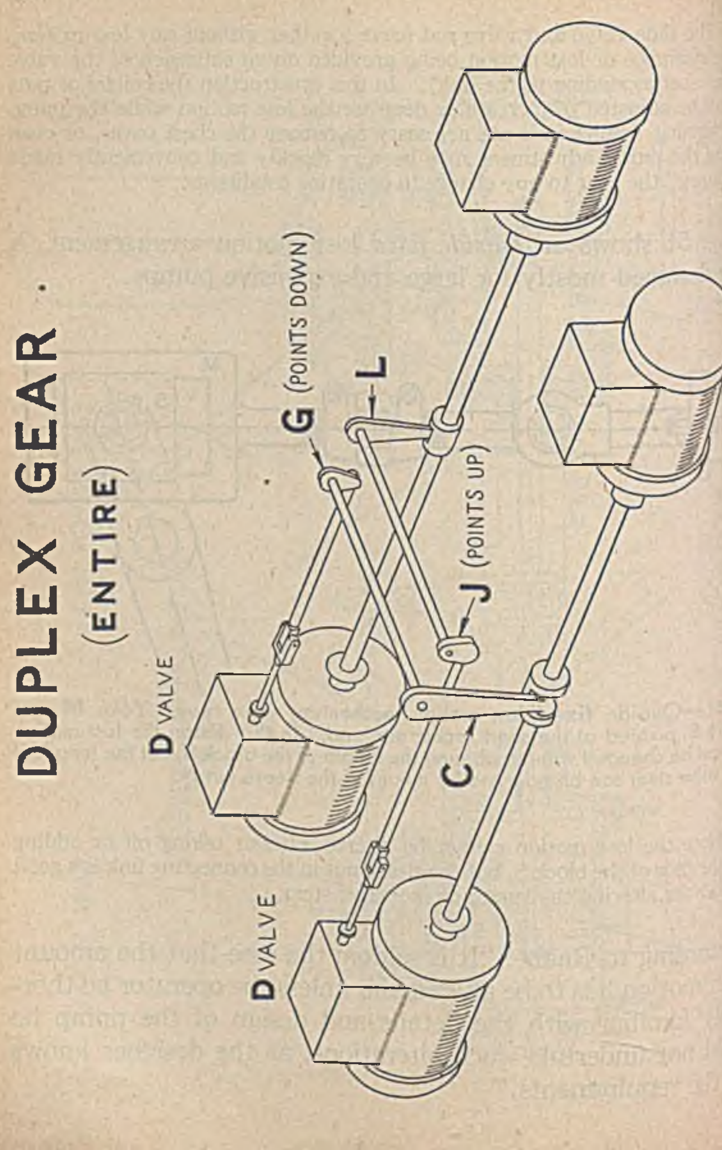


Fig. 58.—Duplex horizontal pump showing valve gear for D valves. See page 334 for single unit of this valve gear.

Duplex Valve Gear, Entire.—The complete valve gear, that is, the two separate gears (one for each pump), is shown in fig. 52. It will be seen that although there are two "sets" of gears, the parts making up the mechanism are the same for both sides excepting long rocker arms C and L, which are of different lengths.

It will also be noted that short rocker arm G (which is moved by long rocker arm C) *points* **down**, while short rocker arm J (moved by long rocker arm L) *points* **up**.

This arrangement of the short rocker arms is necessary in order that one of the slide valves which they move (both valves being alike and of the plain "*D*" type) shall admit steam at the opposite end of the steam cylinder so as to reverse the piston stroke. With regard to the long and short rocker arms, while they are of different lengths the two moving together have the same ratio of lengths as the other two and as each rocker arms have the same length of travel, the short rocker arms also move the same distance.

On account of the reversal of the short rocker arms (one pointing down and the other up) the difference in long rocker arms is a mechanical necessity in order to bring the valve rods to a common elevation. One unit of the valve gear is shown on page 334.

Ques. On simplex pumps when is a **D** and **B** valve used?

Ans. The **D** valve is used on gears in which the valve follows the piston movement; the **B** valve is used on gears in which the valve moves in reverse direction to the piston movement.

Ques. What names are usually objectionably used for short rockers and for long rockers?

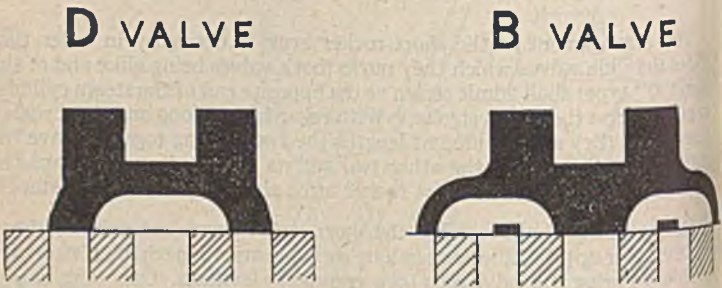
Ans. Cranks for short rockers and levers for long rockers.

Duplex "D and B" Type Valve Gear.—*In order to avoid the separate passage at each end of the cylinder*, a type of gear has been devised which employs a **D**, valve on one cylinder and a

B, valve on the other cylinder. The difference between these valves is shown in figs. 53 and 54.

Using a **D** valve and a **B** valve permits a unique arrangement of steam passages.

The prime reason for this design as just stated is to get rid of the separate passage at each end of the cylinder used for cushioning.



Figs. 53 and 54.—Difference between **D** and **B** valves. The reason for these two types of valves is explained in the text. **B** valves are used on pumps having separate auxiliary valve in which the piston contacts with stems or tappets which project into the cylinder at the ends.

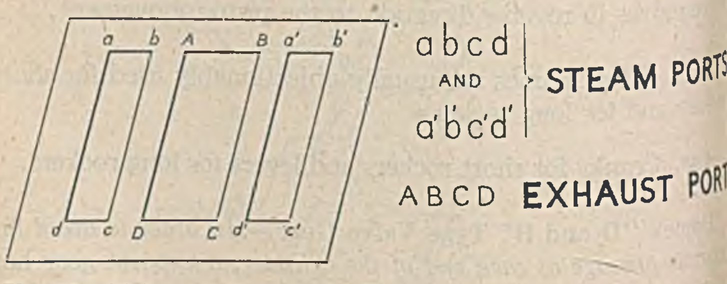


Fig. 55.—Valve seat illustrating ports.

Ques. Why are the extra cushioning passages objectionable?

Ans. It increases the *clearance* considerably which reduces the efficiency of the pump, that is, it results in a waste of steam.

All direct acting pumps, since they must take steam full stroke, do not have the advantage of the saving due to expansion and are therefore notoriously large consumers of steam.

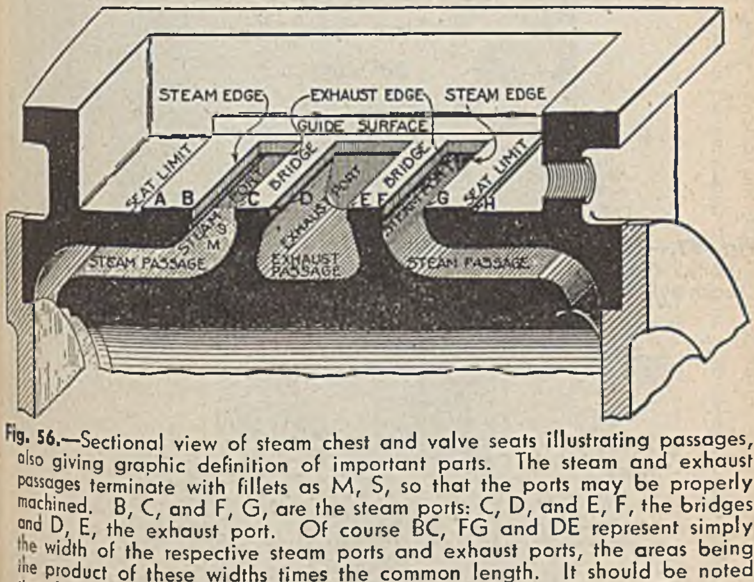


Fig. 56.—Sectional view of steam chest and valve seats illustrating passages, also giving graphic definition of important parts. The steam and exhaust passages terminate with fillets as M, S, so that the ports may be properly machined. B, C, and F, G, are the steam ports: C, D, and E, F, the bridges and D, E, the exhaust port. Of course BC, FG and DE represent simply the width of the respective steam ports and exhaust ports, the areas being the product of these widths times the common length. It should be noted that the valve seat extends from A 'to H, giving the seat limits. These seat limits are provided expressly so that the valve will overtravel each in its "extreme positions" in order not to wear a shoulder in the metal.

Ques. What is clearance?

Ans. The volume between the cylinder head and piston when the piston is beginning the stroke plus the volume of steam passage.

Clearance is expressed as a percentage of the volume displaced by the piston in one stroke.

Ques. What mistake is usually made in using the word *port?*

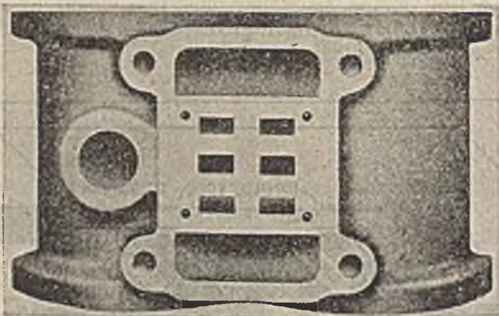


Fig. 57.—Top view of steam cylinders of a duplex **D-B** pump showing the ports. Note only three ports to each cylinder. The two outside ports of each cylinder are the steam ports and the center port the exhaust port.

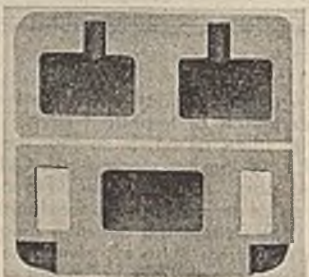


Fig. 58.—Bottom of one **B** and one **D** valve forming a part of the **D-B** system.

Ans. Calling a passage a port.

Ques. What is the difference between a port and a passage?

Ans. A port is the entrance at the valve seat to either a

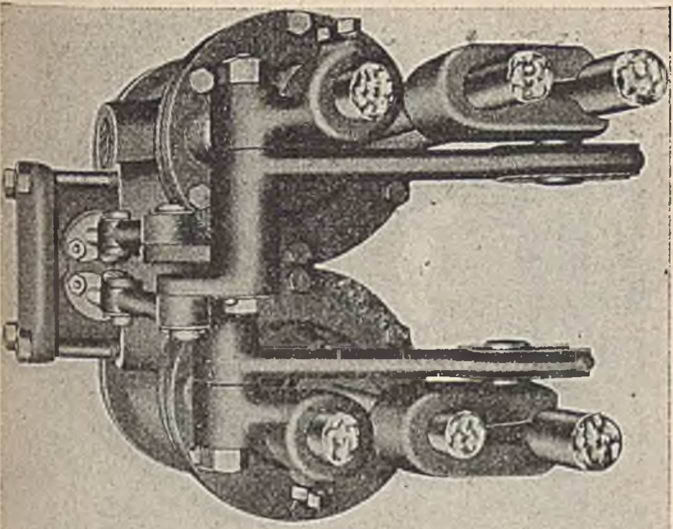


Fig. 59.—Common type of duplex pump valve gear with pump end removed showing section through cradle.

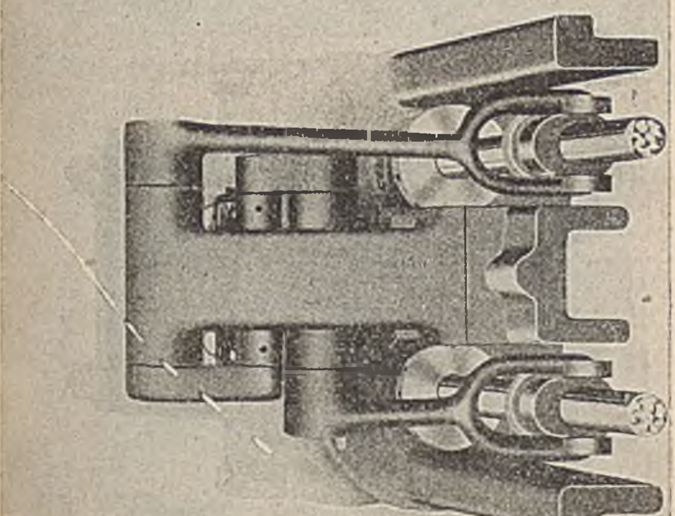


Fig. 60.—"Durable" type of duplex pump valve gear with pump end removed showing section through cradle.

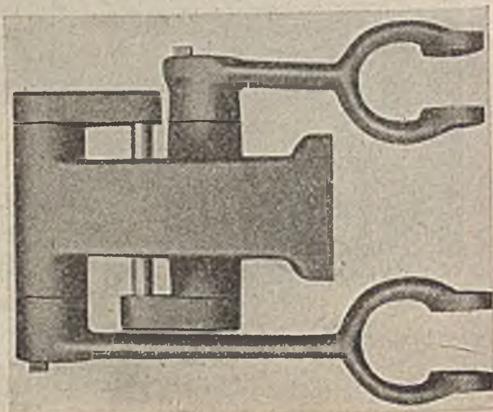


Fig. 61.—Valve gear details of ordinary duplex pump.

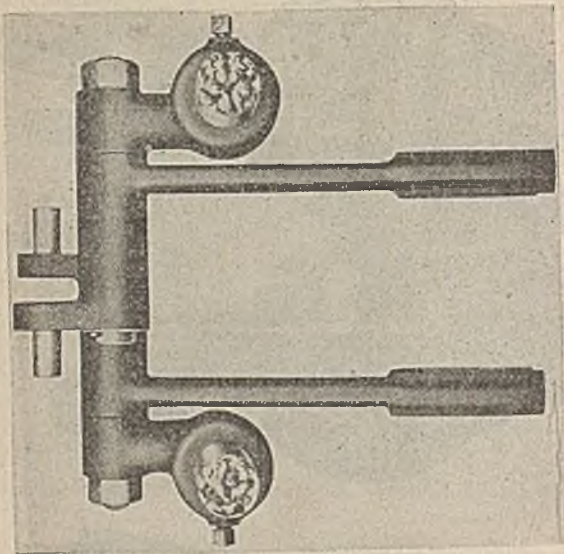
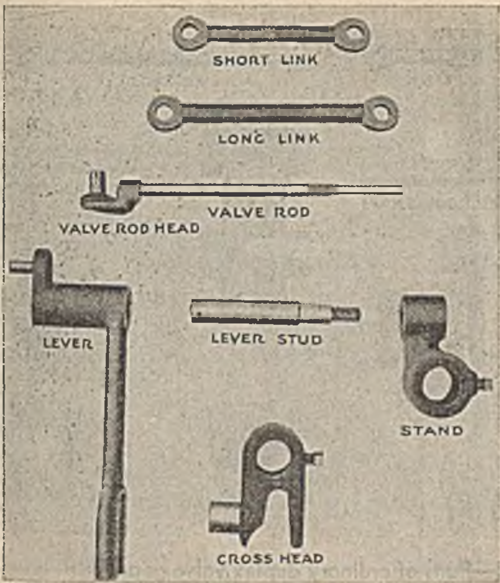


Fig. 62.—Valve gear details of **D-B** type. Note comparison with fig. 61.



Figs. 63 to 69.—Parts of duplex **B-D** valve gear.

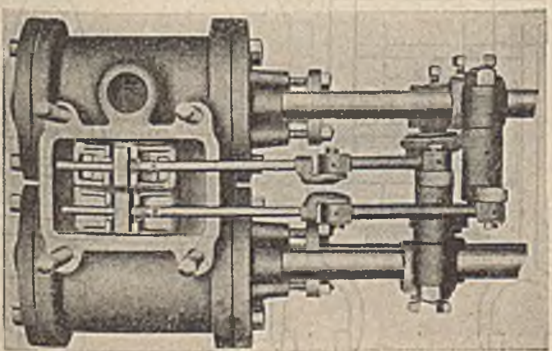
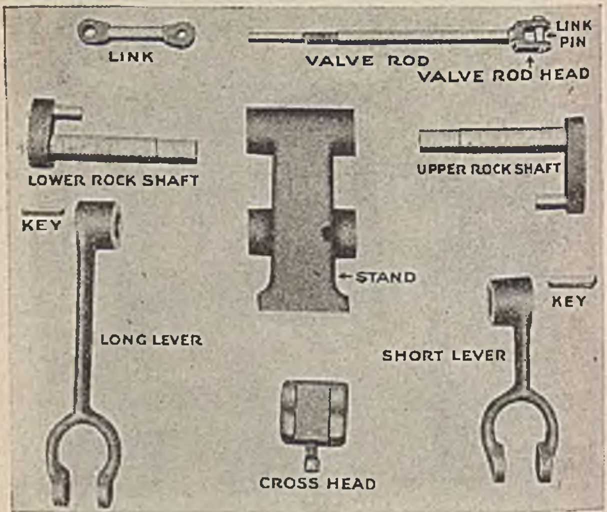
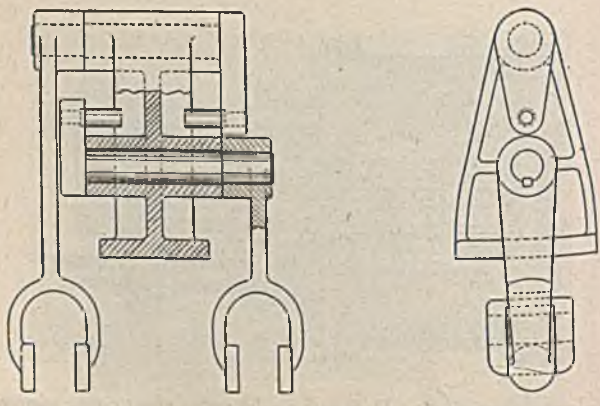


Fig. 70.—Top view of duplex **D-B** pump with valve chest cover removed showing the **D-B** valves side by side, also other details of the valve gear.



Figs. 71 to 80.—Parts of ordinary duplex valve gear.



Figs. 81 and 82.—Views of ordinary duplex rockers and standard.

steam paassge leading to the cylinder, or 2, the exhaust passage leading to the exhaust pipe.

It must be evident from the answer that a port has only 2 dimensions and a passage 3 dimensions; in other words a port has *area* and a passage volume. The difference is shown in figs. 55 and 56.

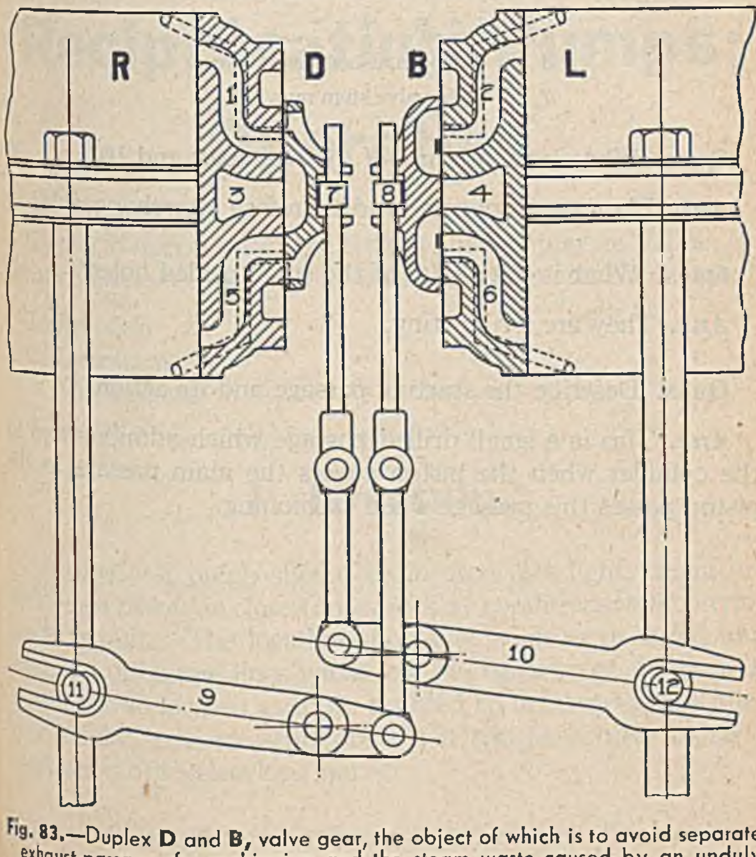


Fig. 83.—Duplex **D** and **B**, valve gear, the object of which is to avoid separate exhaust passages for cushioning and the steam waste caused by an unduly large clearance.

Ques. Describe the D-B valve gear.

Ans. The gear is shown in fig. 83, with the cylinders turned each 90° for convenience in showing the rockers and steam valves. All the parts are here plainly shown.

1	and	5	are	D	valve steam passages
2	"	6	"	B	" " "
3	"	4	"	exhaust passages	
7	"	8	"	valve stem nuts	

Ques. What is the feature of the rockers 9 and 10?

Ans. They are duplicates and therefore interchangeable.

Ques. What is the object of the small drilled holes?

Ans. They are for starting.

Ques. Describe the starting passage and its action.

Ans. This is a small drilled passage which admits steam to the cylinder when the piston covers the main passage as the piston passes this passage when cushioning.

CHAPTER 15

Reciprocating Pumps; Operation

The term operation in the title of this chapter as well as the other chapters on "operation" is extended broadly to include:

1. Erection
2. Operation
3. Maintenance
4. Repair

1. Erection

Location.—A pump should be located in a light, clean, dry and warm place, as close to its work as conditions will conveniently permit. The location should be such as to avoid long inlet and discharge lines without a multiplicity of elbows. If a pump be so located as to be exposed to an atmosphere filled with smoke, grit, moisture or dirt, it will be subject to rapid deteriation of the working parts.

Ques. What precaution should be taken when it is necessary to place a pump in a pit?

Ans. Provision should be made to safeguard against floods.

Ques. How should electric drive pumps be located?

Ans. They should not be placed in damp or moist places.

Unless the oufit was especially designed for such service.

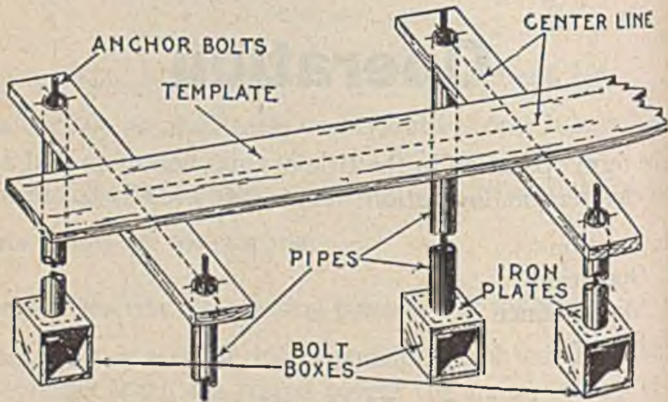


Fig. 1.—Template (in part) for locating anchor bolt centers, pipes through which the bolts pass and bolt boxes at lower end of bolts. The complete foundation is shown in fig. 2, with template removed. The template is made of plain boards upon which the center lines are drawn, and bolt center located. Holes are bored at the bolt centers to permit insertion of the pipes as shown.

Ques. What is especially important in location?

Ans. A proper space should be provided all around the pump so that all parts requiring inspection, adjustment or repair are conveniently accessible.

Ques. Mention another important point.

Ans. The pump should be located and connected so as to secure a full and uniform supply of water or other liquid to be handled regardless of fluctuation of the liquid level at the source.

Ques. How should the pump be located with respect to the water supply level?

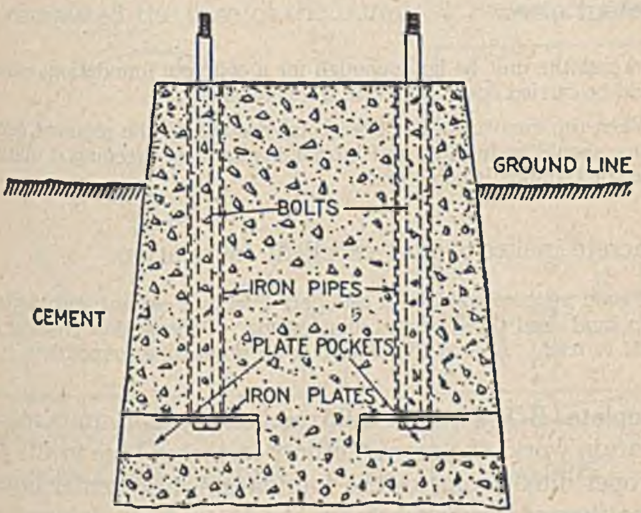


Fig. 2.—Concrete foundation showing method of installing the anchor bolts.

Ans. It should be placed low enough to work with a practical
even under unfavorable conditions.

The extreme theoretical height to which water can be lifted by atmospheric pressure alone is 34 ft. In practice 25 ft. is considered the limit for satisfactory operation.

In locating the pump with respect to the water supply level, due allowance should be made for the frictional resistance of the inlet pipe line so that the *dynamic* (not the *static*) lift will not exceed 25 ft.

Foundation.—Having determined the exact location of the pump, the next thing to be considered is the foundation. An ordinary wooden floor is sufficient foundation for small pumps but larger units should be placed upon a well constructed concrete foundation.

Erection including construction of foundation and anchoring pump thereon should be done only by a thoroughly skilled and competent man.

In case the unit be large enough for a concrete foundation, excavation should be carried down until firm soil is reached.

When the excavation has been carried down to the required depth, its surface should be levelled and thoroughly tamped, keeping it quite damp while the tamping is being done.

Concrete makes a most excellent foundation.

A good mixture consists of one part Portland cement, two parts clean sharp sand, and three parts broken stone. *Only the best grade of cement should be used.* A good foundation is an unyielding foundation.

Template, Bolt Pipes and Boxes.—For pump units requiring foundation work, the manufacturer furnishes blue prints giving the proper dimensions for the foundation, also center lines giving locations of the bolt holes in the pump base casting.

A wooden template should be made with holes corresponding to those in the pump base, as directed by the blue print. Detail of part of such a template is shown in fig. 1.

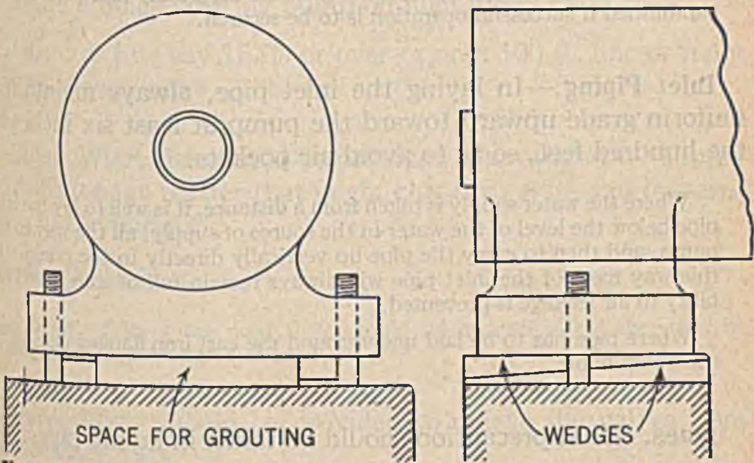
The template is placed in position, such as it would be if resting on the completed foundation, being carefully levelled and anchored in position.

Next, tin pipes, at least 2 ins. larger in diameter than the bolts, are suspended centrally from each hole, reaching to the anchor space, so that when the concrete is poured, there will be a margin of space around each bolt permitting lateral adjustment to allow for any minute errors in measurements, and to facilitate the removal of a bolt in case of breakage.

After these pipes and the necessary forms are in place, the concrete should be prepared in sufficient quantity that the volume of the foundation may be filled with one pouring. The foundation should be completed at least fifteen days before the pump is placed upon it, by which time it should have become sufficiently hard to resist the weight of the engine.

Of course, the time required for hardening of the cement will depend upon the size of the foundation.

Placing Pump on Foundation.—When the foundation has been completed the base of the pump (or the entire pump de-



Figs. 3 and 4.—End and side view of pump cylinder showing placement of wedges and space for grouting.

pending upon size) should be placed on the foundation in position so that the bolt holes in the base will register with those in the foundation.

Thread in the foundation bolts, assembling pocket plates and nuts as shown in fig. 2.

Suitable levelling wedges as shown in figs. 3 and 4, of iron or steel, are placed at proper intervals to support the load solidly without springing.

Adjust the wedges until the baseplate is level as indicated by an accurate level. Keep the level and all surfaces very clean when levelling.

After the grout is poured and fully set, the foundation bolts can be tightened up, using care not to distort the base or bed plate by the pull of the foundation bolts.

Check alignment after pulling up on foundation bolts, as every base plate is elastic, no matter how heavy it is, and will spring to a certain extent. The alignment of all pumping units must be accurately and permanently established if successful operation is to be secured.

Inlet Piping.—In laying the inlet pipe, always maintain a uniform grade upward toward the pump at least six inches to the hundred feet, so as to avoid air pockets.

Where the water supply is taken from a distance, it is well to lay the inlet pipe below the level of the water in the source of supply, all the way to the pump, and then to carry the pipe up vertically directly to the pump. In this way most of the inlet pipe will always remain full of water, and liability to air leakage is prevented.

Where pipe has to be laid underground use cast iron flanged pipe for all sizes obtainable.

Ques. What precaution should be taken in laying pipes?

Ans. Care should be taken to prevent foreign matter, such as sand, sticks, or metal chips from tapped pipes, entering the line.

Material of this character will quickly cut out the lining, pistons and valves of a pump and cause serious injury. The same remarks apply with even greater emphasis to the steam pipe of the pump.

Ques. In laying inlet pipes what should be guarded against?

Ans. Air leaks.

A very small air leak will prevent the proper working of the pump and impair its efficiency greatly.

Ques. What should be done before covering the inlet pipe?

Ans. The inlet pipe should be tested at a pressure of about 25 lbs. per sq. in. to determine if there be any air leaks.

Ques. What provision should be attached to the inlet line to insure a quick starting pump on high lifts?

Ans. For lifts say 15 ft. or over or on a 100 ft. line or more provide a foot valve.

Ques. What else should be provided when the water supply contains foreign matter that might clog the valves and passageways of the pump?

Ans. A strainer.

When strainers are used they should be located where they can be frequently inspected and cleaned.

Ques. What should be provided to relieve the valves from unnecessary duty?

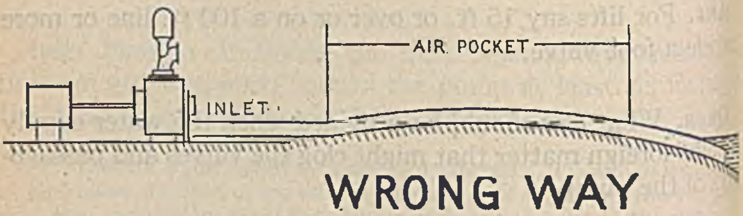
Ans. A check valve should be placed in the discharge line close to the pump so that the weight of the column of water will not rest on the valves in the water cylinder when the pump is stopped. This makes it easier to start the pump. It also prevents water entering when pump is opened for examination and repairs.

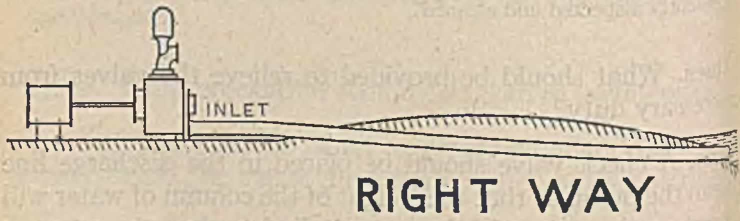
Ques. What should be provided where the pump works with lift?

Ans. A priming pipe leading to the cylinder of the pump. If the inlet pipe be provided with a foot valve, this priming pipe may be led to the inlet box of the pump.

The priming pipe may take its supply from a tank or from the discharge pipe of the pump, being connected to the discharge pipe at a point beyond the main discharge shut off valve.

An air cock is provided on the water cylinder cap, and on opening this air cock the water from the priming pipe will fill the cylinder, the air which it contains passing out through the air cock.





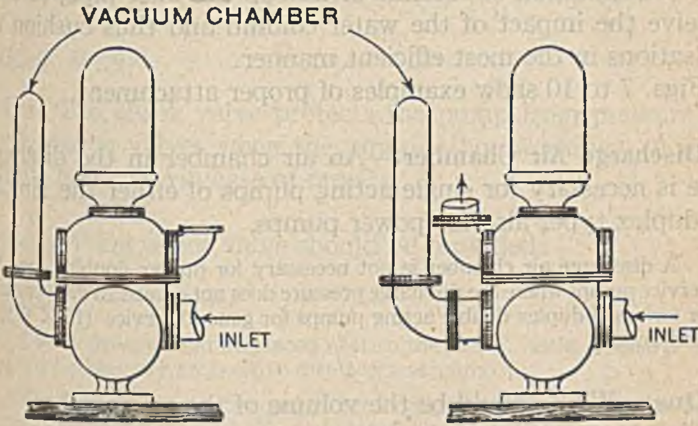
Figs. 5 and 6.—Right and **wrong** method of installing inlet piping to pump.

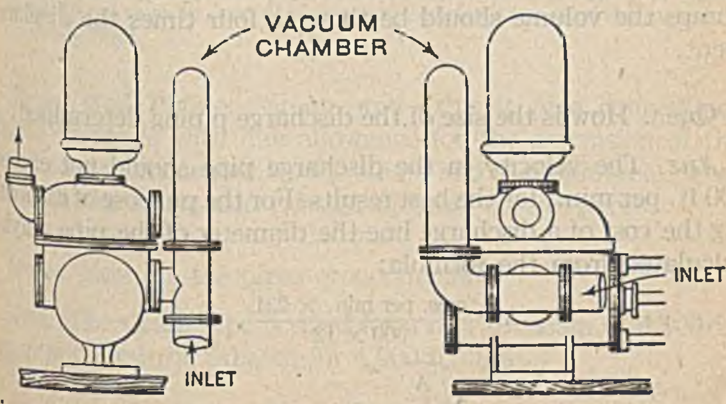
Ques. How deep should the inlet pipe project into the well or source of water supply?

Ans. Deep enough to insure the pipe being submerged when the water is at its lowest level.

It should not extend too near to the bottom of the well where there is a possibility of the pipe becoming clogged up with foreign matter. Large pipes are usually submerged four times the diameter and small pipes two or three feet.

Figs. 5 and 6 show wrong and right way to install inlet pipe line.





Figs. 7 to 10.—Various locations of vacuum chamber on pumps.

Inlet Air Chamber.—An inlet air chamber placed on the inlet pipe close to the pump is desirable for fire pumps, pumps with high lift, short stroke pumps and pumps running at high speed. Care should be taken to locate this inlet air chamber in a continuation of the line of flow in the inlet pipe, so as to receive the impact of the water column and thus cushion the pulsations in the most efficient manner.

Figs. 7 to 10 show examples of proper attachment.

Discharge Air Chamber.—An air chamber in the discharge side is necessary for single acting pumps of either the simplex or duplex type, also for power pumps.

A discharge air chamber is not necessary for duplex double-acting low service pumps where the discharge pressure does not exceed 75 lbs. per sq. in. or for small duplex double acting pumps for general service (10 × 6 × 1? and under).

Ques. What should be the volume of the air chamber?

Ans. It should be six to eight times the displacement for single direct acting or for crank and flywheel pumps. For duplex pumps the volume should be three to four times the displacement.

Ques. How is the size of the discharge piping determined?

Ans. The velocity in the discharge pipe should not exceed 300 ft. per min., for the best results. For the purpose of estimating the cost of a discharge line the diameter of the pipe may be calculated from the formula:

$$A = \frac{\text{gals. per min.} \times 231}{300 \times 12}$$

$$D = \sqrt{\frac{A}{\frac{1}{4}\pi}}$$

in which A = Cross area of pipe in sq. ins.

D = Diameter of pipe in ins.

Piping Accessories.—The discharge pipe should be installed with a check valve and a gate valve near the pump outlet.

Ques. Why?

Ans. The check valve protects the pump from pressure on the discharge valves when the pump is not working. A gate valve is necessary in case of repairs.

Ques. What other valve should be provided?

Ans. A relief valve.

A relief valve should be placed next to the pump, in the discharge pipe of every power driven positive displacement pump.

This valve is for the purpose of protecting the pump against breakage caused by the closing of the main discharge valve, thus increasing the pressure above the maximum for which the pump is designed.

The spring should be frequently tested to prevent sticking. The spring should be set so as to open at a pressure slightly in excess of the maximum operating head.

Steam End Piping.—Steam and exhaust pipe connections should be made with due allowance for the expansion of the steam pipe when heated by the steam and of ample size, never less than the steam and exhaust opening on the pump.

Ques. How are the pipes proportioned?

Ans. The steam pipe is sized for a flow of steam of 6,000 ft. per minute and the exhaust for 4,000 ft. flow.

Ques. What valves should be put in the steam line?

Ans. A throttle valve, as close to the pump as possible, and a drip cock or bleeder valve for draining the main steam pipe before starting.

It is desirable to place a drip cock or small valve between the throttle and pump to protect the pump when not running from an accumulation of condensate in case there be any leakage past the throttle—this is very important in case of units liable to stand idle for long periods.

2. Operation

In operating a reciprocating pump there are some preliminary things to be done before starting the pump.

In general the lubrication system should be filled with the proper lubricant.

The cylinder relief cocks should be opened and if there be a by pass, the cylinder may be warmed by slightly opening the by pass valve. In the case of large compound condensing pumps with independent air pump, the air pump should be started before warming the cylinders.

Ques. In starting a pump or steam engine of any type, what is the main thing to guard against and why?

Ans. Excessive condensation in the cylinder because of the danger of injuring the cylinder head and moving parts as the piston approaches the end of the stroke.*

*NOTE.—Water is an unyielding substance when it fills the clearance space and is acted upon by an approaching piston, especially in the case of fly wheel pumps. It causes a shock to the cylinder head and moving parts as well, and in extreme cases even though the relief cocks be open, there is danger of injuring the pump. Accordingly in starting, steam should be admitted very gradually and the pump slowly brought up to speed.

Ques. In the progressive opening of the throttle if a knock be heard at the end of each stroke, what does this indicate and why?

Ans. Water hammer due to excessive condensation.

Ques. What should be done in such case?

Ans. Gradually close the throttle, or if the knock be not severe do not open throttle any wider until the knocking ceases.

Ques. After pump has been brought up to speed what should be done?

Ans. The relief cocks should be gradually closed.

In closing them if a knock be heard it indicates that they have been closed too soon and must be reopened and not closed until all parts have reached their normal working temperature, under which condition there will be no further water hammer due to excessive condensation.

Navy Instructions on Operating Reciprocating Pumps.—The instructions of the Bureau of Engineering, United States Navy Department, on operating reciprocating steam pumps are, in condensed form, as follows:

To start a reciprocating pump, proceed as follows:

Oil pins of steam valve operating gear and set up on all grease cups. Open water end valves, first inlet, then discharge.

Open cut-out valve, first in exhaust line, then in steam line. Open steam cylinder drains, first top, then bottom.

Open exhaust valve at pump.

Crack throttle valve, and open it slowly so as to admit steam and warm up gradually.

Close steam cylinder drains, after pump makes a few strokes and steam cylinder is clear of water.

Bring pump up to proper speed by sufficiently opening throttle valve.

Close cushioning valves until an adjustment is obtained that permits silent and smooth working of pump, without knocking at end of stroke and without at the same time reducing speed of pump too much at end of stroke.

To stop and secure a reciprocating pump, proceed as follows:

Close throttle valve.

Close exhaust.

Open cylinder drains, first top, then bottom.

Close water end inlet valve.

Close water end discharge valve.

Close steam and exhaust cut-out valves, or root valves.

After steam cylinder is drained, close steam cylinder drains.

Instructions on starting condensing with various types of condensers are given in Chapter 24, "Condensers: Operation."

Ques. In starting what attention should be given to valves?

Ans. When starting a pump, make sure that valves in exhaust, inlet and discharge pipe lines are open, and also all drain valves on steam end of pump before opening steam valve.

Ques. What other attention should be given before and during running?

Ans. All bearings, joints of moving parts and piston rods should be lubricated before starting pumps, and then at intervals when in operation.

Lubrication should be frequent but not so abundant as to gum up or clog joints.

Use only a good quality of mineral oil in the steam cylinder and a lighter grade of good mineral or animal oil on the valve gear.

Ques. What should be done if the pump is to remain idle for some time?

Ans. Fill the lubricator with oil, and open the lubricator cock so that this oil can flow into the steam chest. Then let the pump make half a dozen quick strokes to distribute the oil well over the inside of the steam end of the pump, and so prevent danger of rusting while the pump is standing still.

Operating a Power Pump.—The type pump here selected is horizontal duplex with enclosed crank case.

Its construction is shown in figs. 11 to 13, with accompanying list of parts. It is recommended that a check valve be placed in the discharge line adjacent to the pump, opening away from the pump, to make it possible to inspect the interior of the liquid end, without draining the discharge line.

Ques. What provision should be made to insure safe operation?

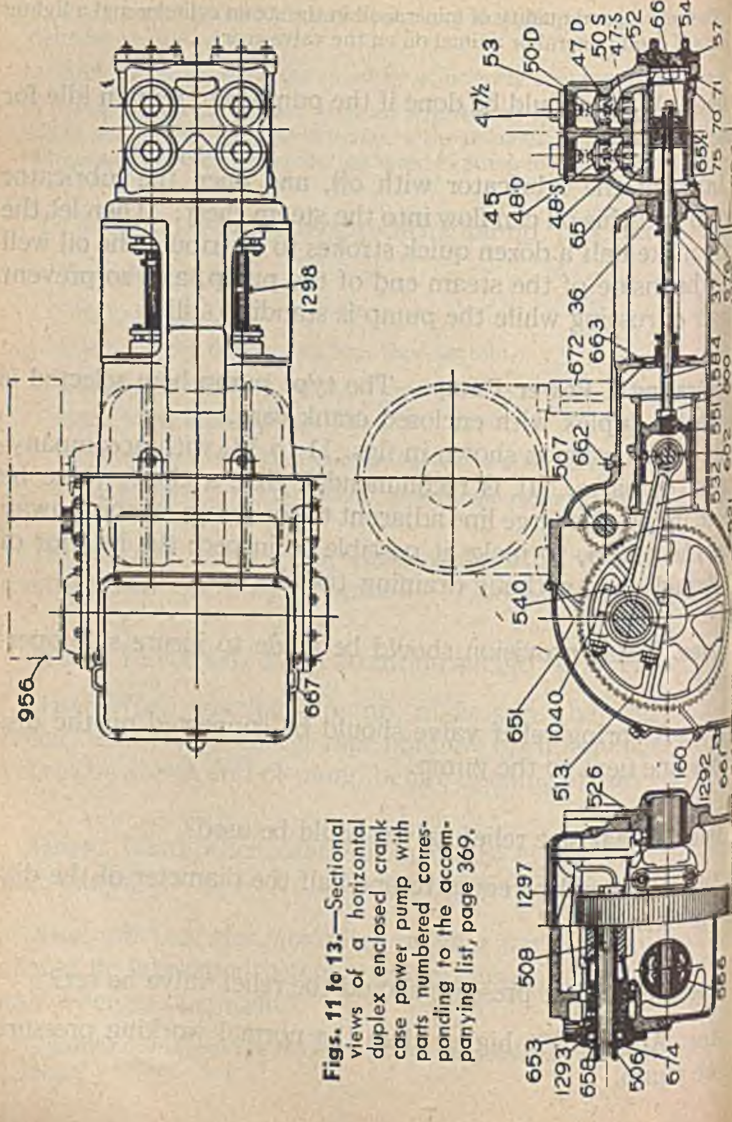
Ans. A spring relief valve should be connected on the discharge line next to the pump.

Ques. What size relief valve should be used?

Ans. It should be equal to one half the diameter of the discharge opening.

Ques. For what pressure should the relief valve be set?

Ans. About 10% higher than the normal working pressure of the pump.



Figs. 11 to 13.—Sectional views of a horizontal duplex enclosed crank case power pump with parts numbered corresponding to the accompanying list, page 369.

LIST OF PARTS

HORIZONTAL DUPLEX POWER PUMP

FIGS. 11 TO 13

36. Piston Rod Stuffing Box.
37. Piston Rod Gland.
45. Pump Cylinder.
47D. Valve Seat (discharge).
47S. Valve Seat (suction).
47½. Valve Chamber Plug. (Sizes 4, 4½ and 5 x 10 pumps take bolted on cover instead of plug.)
48D. Pump Valve (discharge).
48S. Pump Valve (suction).
50D. Valve Spring (discharge).
50S. Valve Spring (suction).
52. Pump Cylinder Head.
53. Chamber Plug Gasket.
54. Cap Nuts.
57. Lining Stud.
65. Pump Piston Body.
65½. Pump Piston Packing.
66. Pump Piston Follower.
70. Lining Packing.
71. Lining Spacer.
75. Pump Cylinder Lining.
160. Cover to Frame Studs.
446. Drain Plug.
506. Pinion Shaft.
507. Main Pinion.
508. Pinion Key.
513. Crankshaft and Main Gear.
521. Frame.
526. Main Bearing Cover.
532. Connecting Rod.
540. Connecting Rod Bolt.
551. Crosshead.
556. Crosshead Pin.
576. Piston Rod Gland Lining.
584. Piston Rod.
602. Crosshead Bushing.
651. Frame Cover.
653. Pinion Shaft Bearing.
658. Felt Oil Wiper-P. S. Bearing.
661. Oil Gauge.
662. Oil Hole Cover.
663. Diaphragm Stuffing Box.
667. Cover Handle.
672. Motor Support.
674. Retaining Plate-P. S. Bearing.
800. Piston Rod Check Nut.
804. Pump Cylinder Foot.
956. Motor Gear or Pulley.
1040. Connecting Rod Shim.
1291. Pump Cylinder Throat Lining.
1292. Main Bearing Shell.
1293. Pinion Shaft Bearing Shell.
1298. Piston Rod Collar.

Ques. What kind of valves should be used on the inlet and discharge lines?

Ans. Gate valves.

Ques. What precaution should be taken in case of freezing?

Ans. Both the inlet and discharge lines, and the pump as well, must be provided with drains of ample size, which drains must be opened on shutting down.

The pump cylinder is provided with four drain openings at the ends of the cylinder, with one drain at the inlet chamber between the cylinders and one drain in the discharge chamber.

Ques. What should be done before starting the pump?*

Ans. Inspect the interior of the crank case. Clean out by washing with kerosene and wiping dry. See that all oil holes are free and clean. After replacing the crank case cover, fill the case to the required level with a good grade of lubricating oil having a viscosity of 500 to 600 S.S.* Universal. Inspect the oil level in the crank case occasionally, and maintain it at the level as indicated on the gauge at the side of the crank case.

Before starting, see that the motor gear reduction is properly lubricated with a good quality of light gear grease sufficiently viscous to remain on the gears when they are in operation.

Do not run the pump without lubrication. Even a few minutes of operating dry will cause cutting of the wearing surfaces with consequent trouble.

Ques. What provision should be made to insure continuous operation?

*NOTE.—S.S. = Seconds Saybolt.

Ans. It is well to carry on hand as spares, a set of valve details, and packings for the pistons in the liquid end. These parts are subject to wear, more or less rapidly, dependent upon the liquid pumped.

Ques. What should be avoided?

Ans. Avoid taking the machine apart, except when necessary for the replacement of working parts.

Ques. What should be done in starting?

Ans. Use the four relief valves to relieve the cylinder of air.

If pump fail to start properly:

1. Make sure of an ample supply of liquid to be pumped and within reasonable inlet lift.

2. See that the air valves are on the cylinders and are operating properly to relieve the cylinders of air.

3. Go over the inlet pipe and connections to make sure that there be no air leaks.

4. See that the inlet pipe and strainer are not clogged and that there are no summits or air pockets in the inlet line.

5. Inspect the pump valves to see that they are not hung up off their seats by foreign substances.

6. Check the stuffing box packings to see that they are in proper condition.

3. Maintenance

No piece of machinery can be expected to continue in satisfactory operation unless it receive proper and periodic attention to correct any faults or derangements that may arise. This comes under the heading of ordinary servicing rather than repairs.

Troubles.—In the operation of reciprocating pumps, various troubles are encountered from time to time and the following troubles with their causes and remedies will be found helpful.

Pump Fails to Start.

Secure it. Do not attempt to adjust tappet collars. Examine discharge exhaust lines for closed valves or for a valve disc possibly detached from stem.

Ques. If no valve trouble be found, what might be the trouble?

Ans. The plunger or steam piston may be frozen, especially if the pump were out of service for some time.

Ques. What should be done in such case?

Ans. Jack pump with a bar to determine if there be excessive friction.

If so, this is probably the source of trouble.

Ques. What precaution should be taken in use of bar?

Ans. Never use a bar to start pump with throttle open.

Ques. Give direction for servicing valve gear when pump refuses to start.

Ans. Disconnect auxiliary valve stem from the operating gear without disarranging adjustment of tappet collars. Open exhaust, inlet and discharge valves and then crack throttle. Work auxiliary valve by hand; it should work freely by hand.

Should the pump still refuse to start, secure it. Remove valve chest cover and examine main valve to see if it has over-ridden or stuck.

If pump cannot now be started a complete overhaul of working parts of the steam end is necessary to stop steam leakage either in steam piston or valves, which is the most probable cause of pump not starting.

Ques. What does jerky operation in starting indicate?

Ans. Failure of the water supply to follow water piston.

Ques. How is this trouble corrected?

Ans. See that all inlet line stop or check valves are open and that the line is clean of obstructions.

Ques. When a feed pump is vapor bound what should be done?

Ans. Turn a hose on water end.

Ques. If a pump race without increasing its output what is the cause?

Ans. A leaky plunger, leaky, broken or stuck water valve or by air leakage.

Ques. What should be done in such case?

Ans. Stop pump as soon as practicable in order to ascertain and correct trouble. Should the pump after it has been running properly, suddenly lose pressure on one stroke, look for a broken valve at once.

Ques. What should be done if pump be found working with considerable negative lift?

Ans. Throttle inlet or install heavier springs on inlet valves.

Ques. On pumps not fitted with a vacuum chamber how may pounding be stopped?

Ans. Install a snifting valve on inlet side.

Ques. What does groaning in water end usually indicate?

Ans. Packing too tight or a broken part.

Ques. Give an indication of trouble in the steam end.

Ans. This is indicated by erratic action, as by sticking in any part of the stroke, or stops frequently with throttle valve opened proper amount.

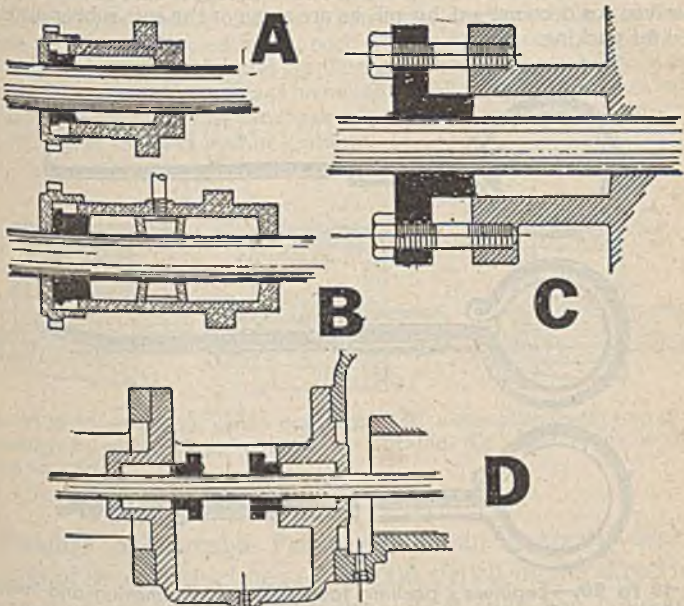
Stuffing Boxes.—Small size pumps are generally furnished with stuffing boxes as shown in figs. 14 and 15. The customary type of stuffing box on large sizes are of the type shown.

Ques. What kind of stuffing box does a pump operating on high vacuum require?

Ans. One with a water seal on the water end.

Fig. 16 shows one type and the open pot type seal in fig. 17.

Packing.—The object of a packing is to prevent a leak of any fluid, which may be either a liquid or a gas. This may be a comparatively simple case as when gaskets are used to prevent steam escaping at a joint, or it may be quite complicated, as by the use of packing on piston rods.

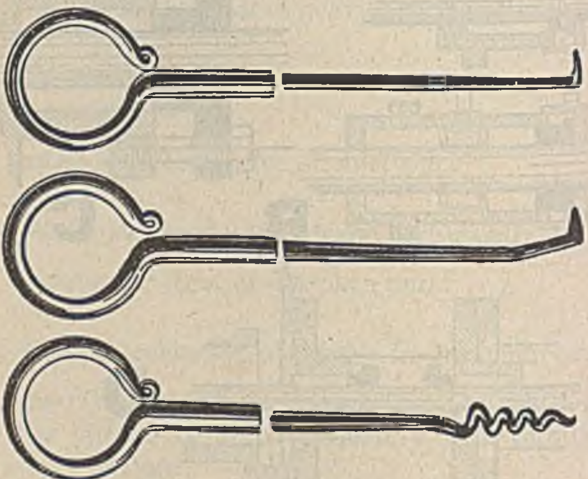


Figs. 14 to 17.—Various stuffing box designs. **A,** plain screwed box; **B,** screwed type with lantern gland; **C,** bolted type; **D,** open pot water seal stuffing box.

They should not be decomposed or rotted by soaking, but should retain their elasticity, or softness, even under frequent making or breaking of the joint.

Air, Ammonia, and Oil Packings.—The requirements for these purposes are limited, and the regular steam and water packings generally can be applied to these needs. The air pipes of forced ash pit or induced draught systems, require some packing of fibrous or rubber material for cold air piping, and of asbestos or metallic material for hot air piping or ducts.

The main requirement in packing for oil piping is that it should not be dissolved, or decomposed by oil, as are some of the soft rubber materials used for packing.



Figs. 18 to 20.—Engineer's packing tools for use in removing and inserting packing.

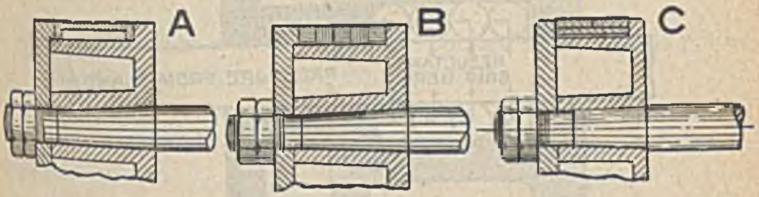
Packings for Stationary Parts.—For these parts packings are generally applied in the forms of sheets, called gaskets, thin and well spread out between flat faces or flanges of valves, cylinders, valve chests, etc.

Occasionally engineers prefer to use a round metal or fibrous ring in a recess, instead of a flat sheet, as it is more easily made and kept tight, while bearing only on a narrow line or ridge, all around the opening.

The choice of the packing is greatly influenced by the consideration of whether the joint is to be broken often, or will remain comparatively undisturbed for a longer period.

In the latter case materials may be employed that cannot be used over again, such as red lead or iron rust cement, with or without fibrous gaskets, while, in the first case, quickness of making and preservation of the packing for continued used will be of importance.

The packings used for stationary parts may be divided into two classes: those that cannot be used again, such as iron rust cement, red lead, with or without chopped hemp, and graphite; and those that can be more or less frequently used again, such as the metallic gaskets, asbestos, pure rubber, mixed rubber and canvas, canvas or paper soaked in linseed oil, or applied in connection with red lead or graphite.



Figs. 21 to 23.—Various piston packings. **A,** water piston with ring grooved packing; **B,** water piston with fibrous packing; **C,** water piston with three ring packing.

Packings for Movable Parts.—These are generally arranged in coils of several thicknesses to gain depth in the direction of the motion, or in the longitudinal axis, like the coils in the stuffing boxes of piston rods, valve stems, condenser tubes, or in the packing of air pump pistons.

Packings for movable parts should be employed in good depth, with relatively slight pressure upon them, to retain elasticity and secure good durability and long life with little attendance and adjustment.

The renewing of stuffing box or piston packing is generally a troublesome experience, attended with great expenditure of time and labor, like that of the condenser tubes, where the water chests must be removed, in order to get at the stuffing boxes, or that of the air pump piston, where the top covers, valves and moving parts must be removed for access to the interior.

Oils, red leads and other substances that harden or bake should not be applied to packing for movable parts; also the packings themselves should be of substances that do not burn. Softness, pliability and elasticity should be their chief properties, and these should be retained as long as possible.

The packing used for moving parts is generally employed in round or square rings or strands of soft metal, asbestos, rubber, combinations, or hemp made up into braids.

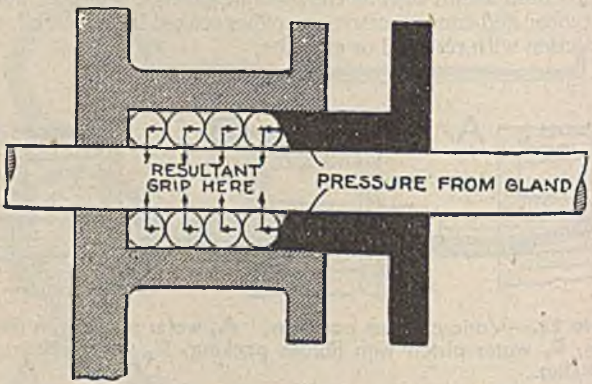


Fig. 24.—Principle of "Moncky wrench" packing. Packings of this type include the ordinary square flax, round core and gum core packings and what is commonly known as "red core spiral." The round and flat gum core packings are made by braiding successive sheaths about a flat or round gum core. These packings form about the cheapest steam packings that are on the market. They depend entirely for their success on a pressure from the gland and for that reason they are classed as "Moncky wrench" packing. As the pressure from the gland is generally much stronger than necessary, there is produced on the rod an unnecessary friction, and as this friction is on a moving rod there is done at every stroke a certain amount of work that is unnecessary, which work is required simply to pull the rod through the packing. This, of course, requires steam and steam requires coal and there is actually consumed a certain percentage of the work from the engine in overcoming the excess friction caused by the packing. Sometimes this packing is made by wrapping a piece of duck around a gum core, but the principle of application is the same and from an economical standpoint this class of packing, which was the class first brought-out, is really the poorest, the friction almost invariably being excessive and the consumption of the steam on account of the packing very large.

The packings, particularly the soft ones, are sometimes soaked with tallow, or graphite powder is rubbed into them to increase their durability and resistance to wear.

Ques. What kind of packing is used on steam pistons?

Ans. They are packed by means of carefully fitted cast iron spring packing rings.

They are self-adjusting and need no attention whatever; replacement on account of wear is necessary only after years of service.

Ques. When the water pistons are packed with fibrous packing what trouble is sometimes encountered?

Ans. Stiff operation of the pump due to the swelling of the packing.

It causes the pump to make uneven strokes, especially so when pumping hot liquids, and sometimes it is necessary to take out the packing and thin it down.

Methods of Applying Packings.—The way in which a packing is applied varies greatly according to material, position, finish of faces of joint, nature and duty of part in need of packing. It is expedient to apply the hand made packings in a heavier layer than the nearer uniform commercial packings.

On rough surfaces more packing material is needed than on smooth or finished surfaces. Stationary parts, with a higher clamping pressure, need much less packing than movable parts, where the packing is compressed comparatively lightly.

The pressure upon the packing is generally exerted by bolts, nuts, or tap screws.

For Stationary Parts.—The packing is clamped between sti- flanges which with the numerous bolts, exert a very heavy pre- sure upon the packing. The bolts should be spaced closer to- gether with light flanges than with heavy flanges, as otherwise a slight bulging of the flange between the bolts may take place, leading to leakage and final tearing and blowing-out of the pack- ing material.

For Moving Parts.—The packing is forced into a special recess of the stuffing box or piston by a gland or follower ring

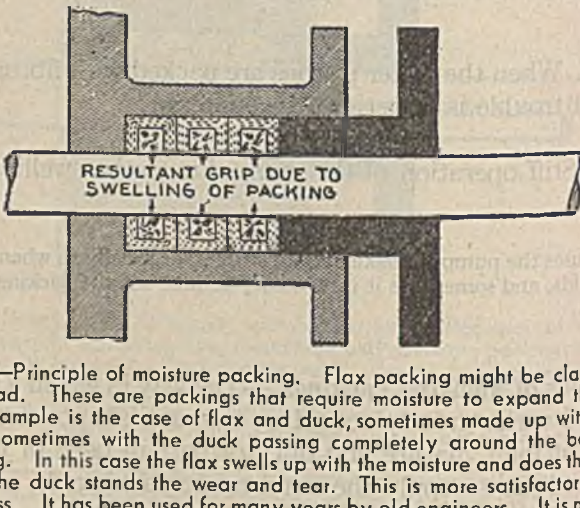
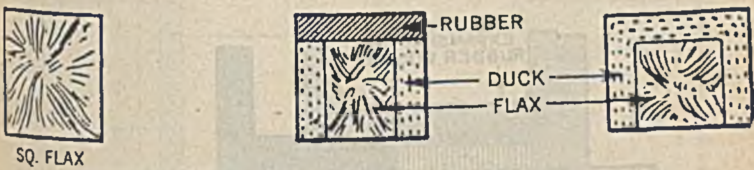


Fig. 25.—Principle of moisture packing. Flax packing might be classed un- this head. These are packings that require moisture to expand them. Th- best example is the case of flax and duck, sometimes made up with a rub- back, sometimes with the duck passing completely around the back of packing. In this case the flax swells up with the moisture and does the pack- while the duck stands the wear and tear. This is more satisfactory than first class. It has been used for many years by old engineers. It is not a pac- ing, however, that would be available where the steam was entirely dry - it reached the box, or where it was superheated. Of course steam that is - when it reaches the cylinder is not dry when it reaches the box, or whe- enters the box, for the reason that the box is more exposed and conseque- is much cooler and the rod is one-half the time out in the cool air, and s- as high a temperature as the live steam cannot well be maintained, the ste- in the box will be moist. If the steam be originally superheated, however, may get in the box as dry steam, in which case it is very hard on a pack- that requires moisture to cause it to swell and thus produce the requisite - on the rod.

As the danger of blowing out is practically removed in these places, the area under pressure being generally much smaller than with stationary parts, and, as the desired elasticity of the packing makes a low clamping pressure advisable, fewer bolts are required.

Graphite.—This is applied, in conjunction with metallic, asbestos, rubber and fibrous packings as powder, or made up with a little oil or tallow to a paste. It is excellent for all joints that are frequently broken; prevents burning of the packing to



Figs. 26 to 28.—Cross section of moisture packing.

the faces, and effects a saving of material by keeping the gasket in good condition for renewed employment.

Canvas, Paper, and Hemp Packings.—These are employed generally only temporarily, where lack of the higher grade forces the engineer to use a substitute.

They are used soaked in oil and coated with graphite, grease or red lead in order to give more body and tightness.

Hemp packing was formerly made by the engineers from the raw material into braids of varying thickness, shapes and length. The modern commercial packings, however, made up and ready for use, save so much time and labor, and prove in the end so much cheaper, that the practice has practically died out, being employed only in emergencies and special cases.

Asbestos Packing.—This is made in the shape of sheets, round or square strands, wicks, etc., from the fibrous raw mineral material. It has the valuable quality of being incombustible and imperishable, and is thus eminently adapted to all high pressure pipe and machinery joints around boilers and engines.

It ranks with metallic packings in this particular. One great advantage is its softness, fibrous nature and comparative elasticity, while it is, on the other hand, not nearly as durable as metallic packing, and therefore, more expensive. Some eingineers employ asbestos packing, together with metallic packing, in stuffing boxes, thus combining the softness and elasticity of the asbestos with durability of the metallic packing.

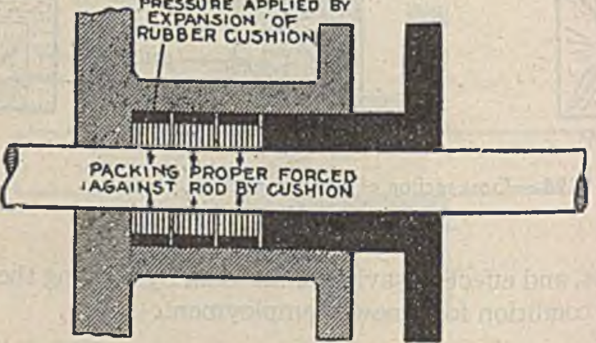
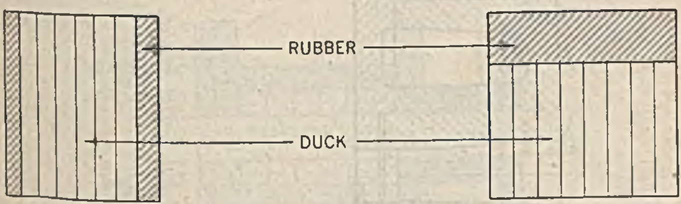


Fig. 29.—Principle of expansion packing. This packing expands but not from the pressure from the gland. This is generally caused by the expanding of some material used as a cushion, under the influence of heat. The material almost universally used is rubber. Rubber when heated will expand, and if it be used in connection with the packing, it will when heated force the packing against the rod and give an easy cushion effect on the rod, which is entirely independent of the gland. Since this is true, there is no reason for the engineer to use a Moncky wrench, and consequently there is no opportunity to force the packing too tightly against the rod and produce undue friction. It is therefore a packing that is more economical so far as consumption of steam is concerned, since there is a lighter load on the engine due to the decrease in the friction of the packing on the rod. An expansion packing is a better packing than that which depends on pressure from the gland, such as a Moncky wrench packing, being not only more economical of steam, but also easier on the rod. This same result is produced in certain metallic packings by the use of springs.

Rubber Packing.—This is the most frequently employed material. Its great elasticity, comparative strength and toughness, tightness and moderate price make it an excellent packing for all water service, joints around pumps, pipes, manifolds and valves. Numerous engineers prefer some one of the particular makes of this packing for low pressure and exhaust steam joints.

Rubber packing is made in sheets of all sizes, gaskets, round and square strands, washers, etc., generally with a body of canvas in one or more layers, that increase the strength and resistance against blowing out.



Figs. 30 and 31.—Typical cross sections of expansion packing.

Graphite, rubbed on the faces of rubber packings, effectively prevents burning on, thus allowing the gaskets to be used over.

Metallic Packing.—Usually metallic packing is made from the soft metals, which, under pressure, yield and shape themselves to the bearing surface. It is almost universally employed, and proves excellent for high steam pressure, where fibrous packing would burn and last but a very short time.

The elasticity of metallic packing is, however, considerably less than that of the fibrous packing, and this is a certain disadvantage, particularly in stuffing boxes. To overcome this lack of elasticity, certain constructions of metallic stuffing boxes employ springs, which tend to prevent gripping of the packing on the rods. The greater complication of this arrangement is objected to by many engineers, and the simple, more rigid arrangement preferred.

Copper, lead, babbitt metal and similar compositions of soft nature a employed with advantage in form of rings, sheets, stuffing box coils, et Copper is employed in rings of round, or triangular, sections, or in corr gated discs for packing on flanged boiler valves, manholes, pipe flanges engine flanges. It is very durable and lasting, but with iron flanges faces, it sets up galvanic action in some liquids, thus quickly destroy the smoothness of the faces.

The lead or babbitt metallic packings are used in sheets for flanges, in split rings for stuffing boxes. Their melting point should be conside

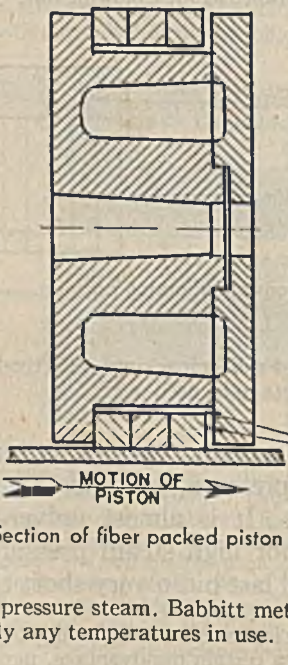


Fig. 32.—Section of fiber packed piston illustrating method of packing.

for high pressure steam. Babbitt metal may be mixed so as to resist ve effectively any temperatures in use.

Ques. What should be done previous to installing ne fibrous packing?

Ans. Soak in warm water overnight before fitting.

Ques. Mention a necessary requirement for fiber packing.

Ans. The packing must have clearance in the piston packing space both in depth and length, as in fig. 33.

It must not be jammed between the piston and liner or clamped between the head and follower. The packing rings should be carefully cut to length, each length being a little short of the exact measure to allow for the extension which is bound to occur after the packing becomes wet.

In fig. 33, the clearances are shown exaggerated so as to make the illustration clearer.

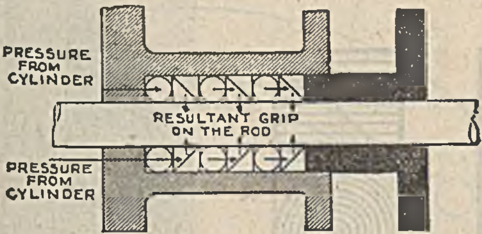


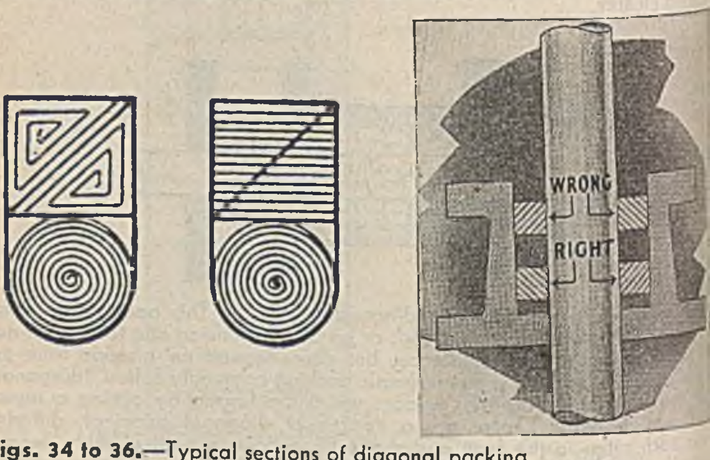
Fig. 33.—Principle of automatic diagonal packing. This packing does not depend on pressure from the gland, or on any expansion due to heat or due to the swelling of any substance, but does depend on pressure from the cylinder. This packing is an automatic packing commonly called "diagonal" packing, because it contains wedges which are formed by cutting a square section diagonally. There are a number of diagonal packings, differing from each other in the form and shape of wedges, but all of them depend upon the same fundamental principle that the pressure from within the cylinder will force the wedge next to the rod in toward the rod, and, of course, that means that the grip which it has on the rod will vary with the pressure forcing it, in other words, with the cylinder pressure.

When the piston is moving, as shown, the discharge pressure comes in between the piston and packing and presses the back of the packing, thus holding it against the liner to prevent leakage.

When the piston reverses, the packing shifts to the opposite end of the space and the pressure comes in from the other side of the piston. With this arrangement the pressure between the packing and the liner is proportional to the discharge pressure, while if the packing be compressed in the packing space, the pressure on the liner may be much greater than is nec sary and thus cause excess wear on the liner and packing.

Ques. What quality should a packing have to permit prop fitting and why?

Ans. It must have sufficient initial stiffness or tension to s it out snugly against the liner so when the piston move pressure will build up on the back of the packing and not b pass between the packing and the liner.



Figs. 34 to 36.—Typical sections of diagonal packing.

Ques. What precaution should be taken in adjusting stuffi box glands?

Ans. Do not screw up the glands too tight.

Ques. What is the useful life of packing?

Ans. It may be used until it begins to harden.

Do not allow it to remain in the stuffing box after it hardens becaus will score the piston rod or valve stem.

4. Repairs

A thorough overhauling of a pump is occasionally required in order that general repairs can be made. The principal reason for taking a pump apart is to ascertain the exact condition of the cylinder walls and working parts.

Ques. In disassembling what is the correct procedure?

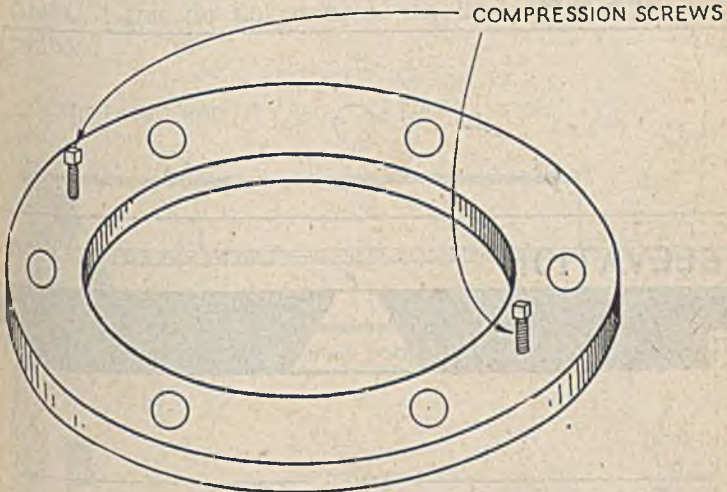


Fig. 37.—Cylinder head illustrating compression screws to facilitate removal of the piston if stuck to the cylinder flange.

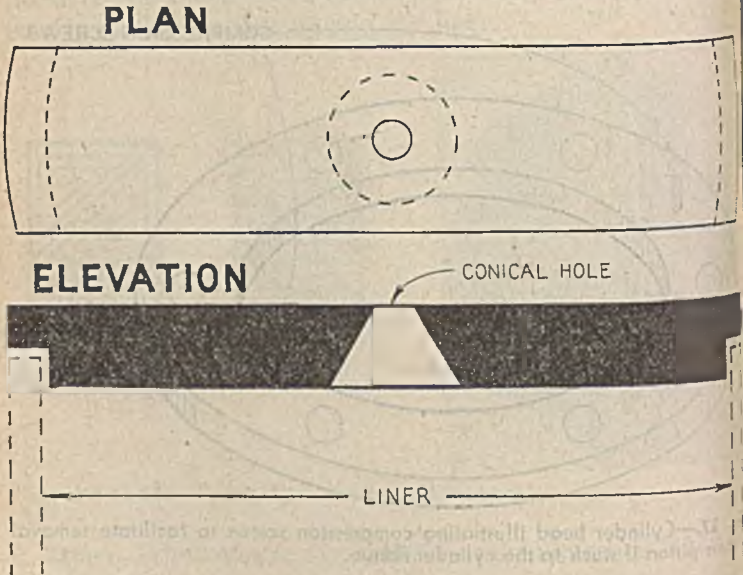
Ans. Each part as it is removed should be cleaned.

As soon as one part is unjointed or uncoupled, insert its pins or screws in their proper place before laying aside. This will prevent any small parts being misplaced.

Removing Cylinder Heads.—Pumps of size and good construction will have heads provided with compression screws to force the piston away from a sticking gasket. Usually two are provided as in fig. 37. By applying a small wrench to these screws and turning them clockwise, the head is easily removed.

Ques. What precaution should be taken?

Ans. Turn the screws equally, giving a revolution to one and then a revolution to the other one.



Figs. 38 and 39.—Plan and elevation of strong back for removal of liner.

This insures that the receding head remains parallel with the flange instead of cockeyed, where it may bind.

Removing Cylinder Liner in One Piece.—In the case of a liner difficult to remove and to avoid splitting the liner, first make a strong back as shown in figs. 38 and 39. The conical hole permits passing the strong back through the liner by tilting it at an angle, as in fig. 40, to get the strong back in position against the end of the liner.

Ques. How is the liner removed with the strong back?

Ans. First tighten the "persuader" nut on the rod, as at **A** in fig. 41, to give a good pull.

Ques. If this do not remove the liner what is the next operation?

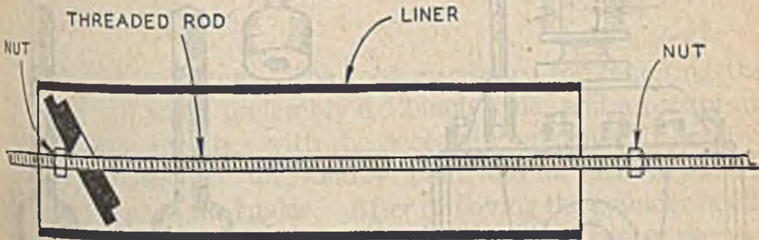


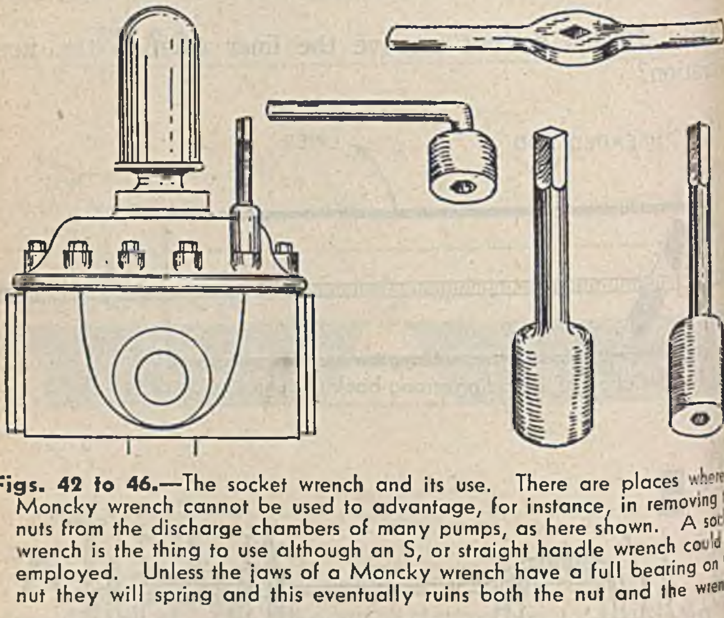
Fig. 40.—Method of threading strong back through liner.

Fig. 41.—Section through pump showing method of removing liner without splitting by aid of strong back.

Ans. Hit the free end of the pull rod with a hammer, as at **B**, in fig. 41.

Ques. What is the advantage of this method?

Ans. If the liner be not standard it may be rebored in a lathe and replaced.



Figs. 42 to 46.—The socket wrench and its use. There are places where a Moncky wrench cannot be used to advantage, for instance, in removing the nuts from the discharge chambers of many pumps, as here shown. A socket wrench is the thing to use although an S, or straight handle wrench could be employed. Unless the jaws of a Moncky wrench have a full bearing on the nut they will spring and this eventually ruins both the nut and the wrench.

Removing Crank Shaft on a Power Pump.—The following instructions are for pumps of the type shown in figs. 11 to 13.

To remove the crank shaft proceed as follows:

1. Remove the crank case cover.
2. Remove the connecting rod caps.

3. Remove the outside liquid and cylinder head.

4. Remove the bolts that hold the crank case stuffing box in place.

5. Travel the connecting rods as far as possible toward the liquid cylinder.

6. Roll the crank shaft until one of the cranks is horizontal and pointed toward the cylinder.

7. Remove the crank shaft bearings.

8. Work the crank through the opening in the bearing housing, swinging the crank slowly until the crank can be tipped up and lifted out of the main frame. Be careful not to jam the crank shaft fits where they run in the bearings.

9. In re-assembling reverse the process, being careful not to jam any parts.

Centering a Piston.—The tools necessary for centering the piston are, a scale, preferably a 12-inch scale, and a pair of inside calipers, together with the necessary wrenches for removing the cylinder head and follower plate and for turning the adjusting screws in the spider. After removing the cylinder head, have the crank turned to an approximate dead center nearest the cylinder. This will bring the piston to the end of the counterbore in the cylinder as in fig. 47, where it may be easily reached.

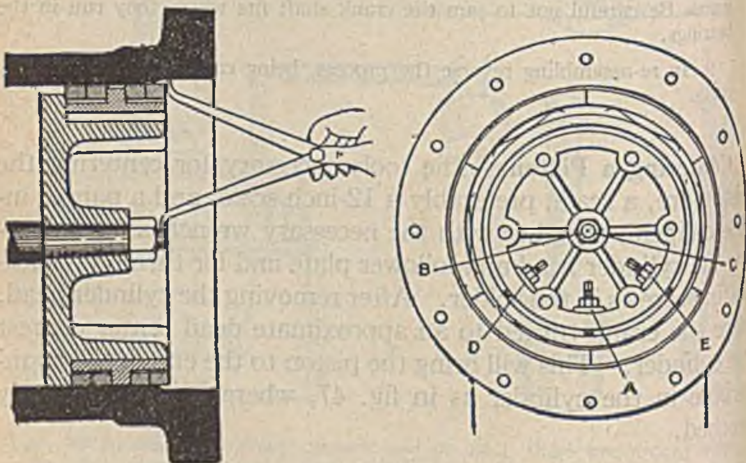
After removing the follower plate, the exterior of the shaft will be brought into plain view. In many instances the end of the rod will be found to project a trifle beyond the face of the nut. In this case open the calipers, and, placing one leg against the projecting end of the rod and the other against the counterbore, take the distance between the rod and the counterbore, being careful to have the extremities of the legs parallel to the face of the spider as shown.

Ques. Why is the counterbore preferred to the cylinder wall?

Ans. It is not subject to wear and accordingly retains its cylindrical form.

Ques. How is the cylinder calipered?

Ans. Place the calipers, first at the bottom, then at the top, ascertaining the distance by means of the scale in both instances. Then set out or compress the legs of the calipers an amount equal to one-half the difference between these two measurements, that is, if the bottom measurement be found to be, say, 6⅞ inches and the top measurement is 7 inches, then the calipers



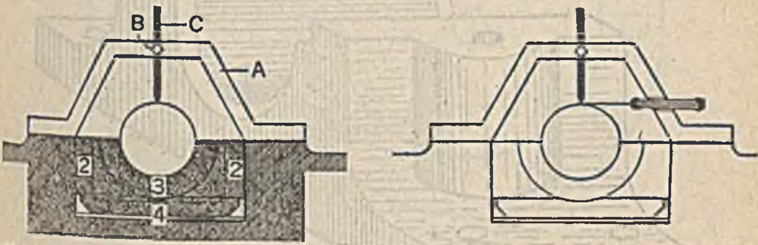
Figs. 47 and 48.—Centering a piston.

would be set to measure either 1/16 less than 7 inches or 1/16 more than 6⅞ inches, because the difference is seen to be ⅛, one-half of which is 1/16 of an inch.

Ques. What is done after setting the calipers?

Ans. Loosen the jamb nut on the center adjusting screw A (fig. 48) at the bottom of the spider, if three screws be used and on both of the screws if only two be found. Turn the center screw to the right a very little and then place the calipers on the rod, as in fig. 47 to see if the spider has been raised enough.

If not, give the screw another turn and again try the calipers, continuing in this way until the spider has been raised to the proper position. Now place the calipers at the sides of the rod (B and C), to see whether the piston is centered sidewise. If it be not exactly in the center, turn one of the side screws D or E, until it is centered. Turning the side screw will also have a tendency to raise the piston slightly, so the calipers must again be placed on top of the rod to see that it is not too high.



Figs. 49 and 50.—Tram and method of measuring height of shaft.

If it be too high, the center screw must be turned back a very little so as to lower the piston. The calipers are to be placed, first at the top and bottom then at the sides, and the screws in the spider are to be turned very carefully until the end of the rod occupies a perfectly central position.

Before putting on the follower plate, caliper the rod very carefully all around the cylinder (counterbore).

If the rod be exactly in the center of the cylinder the calipers should just "feel" the rod at all positions around the counterbore. When tightening the jamb nuts on the three adjusting screws care should be taken to see that the screws do not turn either way.

Using Feeler to Determine Shaft Level.—This method is shown in fig. 50. The bearing caps are all removed, and the tram is placed on bearing No. 1 as shown and the pin set so that it does not touch the shaft. It is then removed to the next bearing and that tested in the same way to determine what adjustments are necessary to bring the shaft to the level with the planed parts of the bed at each bearing. Try all bearings and set tram to the highest point, then go back to No. 1, and by the thickness gauge, or feeler, find the amount that the shaft must be raised.

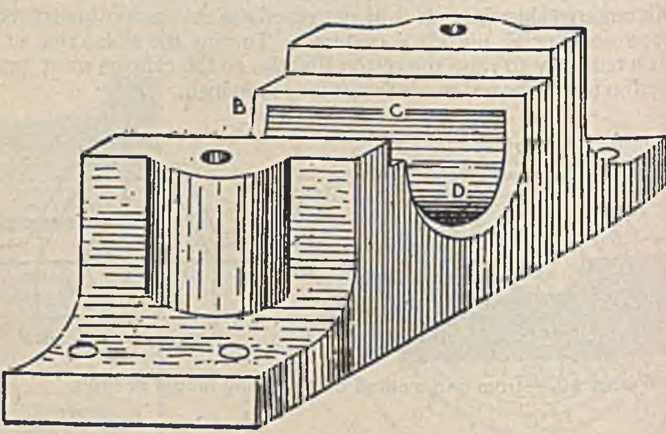
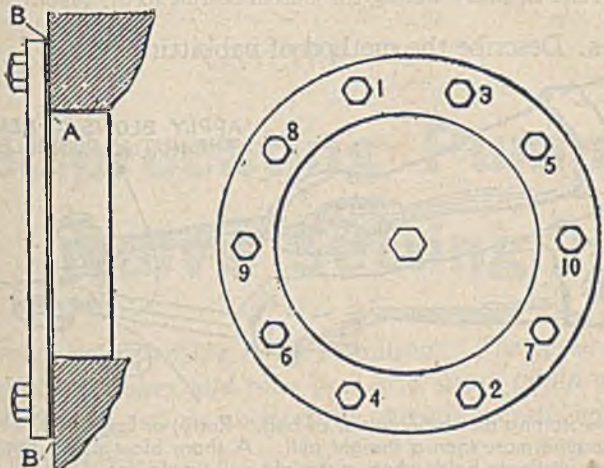


Fig. 51.—Method of babbitting a bearing.

Method of Babbitting a Bearing.—Babbitt or so called anti-friction metal is composed of tin, antimony and copper mixed in various proportions, and may be purchased, or if it be desired, it may be easily made. A good mixture, suitable for general use when the duty imposed is light, is composed of fifty parts tin, five parts antimony and one part copper.

A harder composition, sometimes termed white metal, is composed of 96 parts tin, 4 parts copper and 8 parts antimony. This mixture is especially suited for journal boxes or bearings, and is mixed as follows: First melt 12 parts of copper and then add 36 parts of tin; 24 parts of antimony are put in and then 36 parts of tin, the temperature being lowered as soon as the copper is melted, in order not to oxidize the tin and antimony; the surface of the bath being protected from contact with the air. The alloy thus made is subsequently remelted in the proportion of 50 parts of alloy to 100 parts of tin.



Figs. 52 and 53.—Tightening a cylinder head. All joints should be pulled up square and even all around from start to finish, especially where a metal joint is used. Dirt being left on joint surfaces often causes leaks, because the two cannot be brought evenly together, and just as often the leak is caused by the uneven strain on the bolts. Take for example, the cylinder head shown in the figures, which has a shoulder all around the inside of the flange. It will be seen that by pulling on one nut first, the head could be tipped out of true, and only one edge of the shoulder joint would touch. When first starting to set up on the nuts, a good method to follow is to set up on No. 1 nut lightly until the surfaces of the joint meet, then take up the same on No. 2 nut opposite to No. 1, then Nos. 3 and 4 in succession, after which the nuts can be taken up the same amount in the order given. Then go over them all again in the same order until the joint is tight. The space B, will be equal all around if the pulling up has been properly done. This rule applies equally well on all joints, taking any nut for No. 1 and making No. 2 come opposite.

For brass bearings or boxes, a mixture of 64 parts of copper, 8 parts tin and 1 part zinc will be found to answer very well; but for bearings not requiring so hard a metal, the quantity of zinc is increased and that of the tin diminished. Bearings that are to be babbitted are usually cast with a receptacle for the babbitt metal, as shown in fig. 51, there being a rib at A, B, and C, forming the cavity D, into which the melted metal is poured. These ribs, in new boxes, are sometimes bored out, or for rougher work may be chipped and filed out to fit the shaft and hold it in line. To prevent the ribs A, B, and C, bearing and cutting the shaft, a piece of pasteboard is laid on ribs A and B, thus confining the journal bearing to the babbitt.

Ques. Describe the method of babbitting.

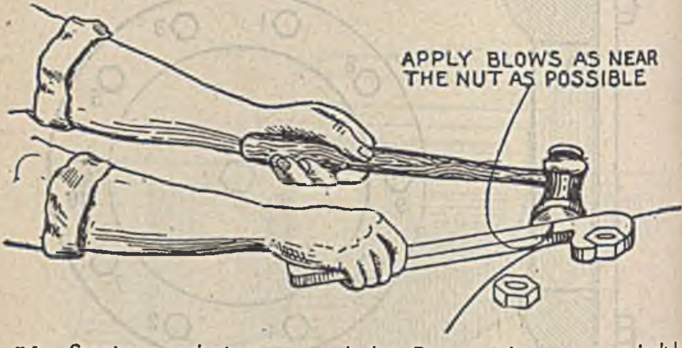


Fig. 54.—Starting an obstinate nut or bolt. Rusty, or large nuts or bolt heads often require more than a straight pull. A sharp blow with a hammer often starts an obstinate hold, where a straight pull would not. It is not advisable only in extreme cases to use the hammer on the wrench, but a hardwood block will do as well. In extreme cases a steady pull aided with blows will do the work. The blow should be delivered **as near the nut as possible,** as shown in the figure, instead of at the other end of the wrench as is usually and **ignorantly** done by greenhorns, thus avoiding the spring and inertia of the wrench and delivering the full energy direct to the nut. The author is indebted to Capt. Henry E. Raabe, M.E., for this suggestion.

Ans. The best method is to pour the bearing and then rivet the babbitt well into the cavity D, which is made wide at the bottom to prevent the babbitt coming loose, and then bore out the bearing in the usual manner. As the babbitt metal in a bearing is apt to close across the bore when cooling after being poured, a mandrel of slightly larger diameter than that of the journal should be used to run the bearing on in place of the working journal or shaft. Some mechanics effect the same purpose by wrapping paper.

CHAPTER 16

Reciprocating Pumps: Valve Setting

The term valve setting is by definition: *The act or process of adjusting the valves and valve gear of a steam pump or steam engine so that the various events of its operating* cycle *will occur at the proper times.*

The instructions here given are for the various types of pump, classed as:

1. Simplex
   - *a.* With *separate* auxiliary valve.
   - *b.* With *combined* auxiliary valve and auxiliary piston.
2. Duplex
   - *a.* D valves
   - *b.* D and B valves
   - *c.* Internal lost motion
   - *d.* External lost motion

1. Simplex Pumps

The various simplex pumps may be divided into two types with respect to the valve gear, as those having:

Class 1. A separate auxiliary valve.

Class 2. Main valve and auxiliary piston combined.

Some makes of Class 1 pumps have stems or tappets which project into the cylinder at the ends and are moved by contact with the main piston as it nears the end of the stroke. An example of this type pump is shown on page 320. Pumps of this type require no valve setting. Various other pumps of Class 1 do require valve setting.

Valve Setting Class 1 Pumps.—For this type which has a separate auxiliary valve, two examples are given for setting the valves. One for auxiliary valve on top and the other on the side.

Example 1.—In this model the auxiliary valve is on top and the lost motion adjustment outside. Fig. 1 shows detail of the valve gear.

To avoid confusion cut does not show starting or cushioning ports in either steam chest or cylinder. Cut shows steam chest having small slide valve above chest piston. This construction is used on large sizes.

Small sizes have small slide valve at side of chest. The same principle of operation and setting of valve movement applies to either type.

The setting of valve movement is accomplished by adjustment of small slide valve, no attention being paid to chest piston or main slide valve. Loosen valve rod collars X and Y before adjusting.

First.—Remove chest cover exposing small slide valve to view, or if small slide valve be at side of chest remove steam chest.

Second.—Move valve rod out until small slide valve A, opens steam port completely, as shown. Scratch valve rod outside stuffing box at point C.

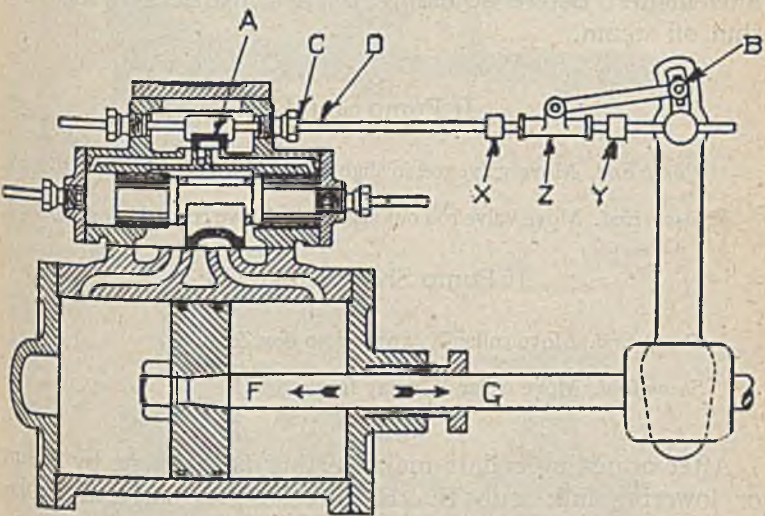


Fig. 1.—Class 1 simplex pump with separate auxiliary valve on top.

Next move valve rod out until small slide valve has opened steam port completely at opposite end. Scratch valve rod outside stuffing box, making second scratch, D, as shown in fig. 1.

Third.—Replace steam chest cover or steam chest.

Fourth.—Place link stub B, in center of slot at top of lever.

Fifth—Place mark C, even with gland, as shown. Move lever in direction of F, to end of stroke as far as piston will go. Put collar Y against dog Z and tighten set screw.

Sixth.—Move lever in opposite direction G, as far as piston will go. Place mark D, on valve rod even with gland. Place collar X, against dog Z. and tighten set screw.

Seventh.—Turn steam on slowly. Pump will not start immediately, due to cold parts and water in cylinder. Use starting device to shift chest piston and main slide valve if pump stop at either end of stroke.

The foregoing instructions give theoretical setting of valve movement. Before adjusting, as per instructions following, shut off steam.

If Pump Stop On:

Pump End. Move valve rod in slightly. Move collar X closer to dog Z.

Steam End. Move valve rod out slightly. Move collar Y closer to dog Z.

If Pump Short Stroke On:

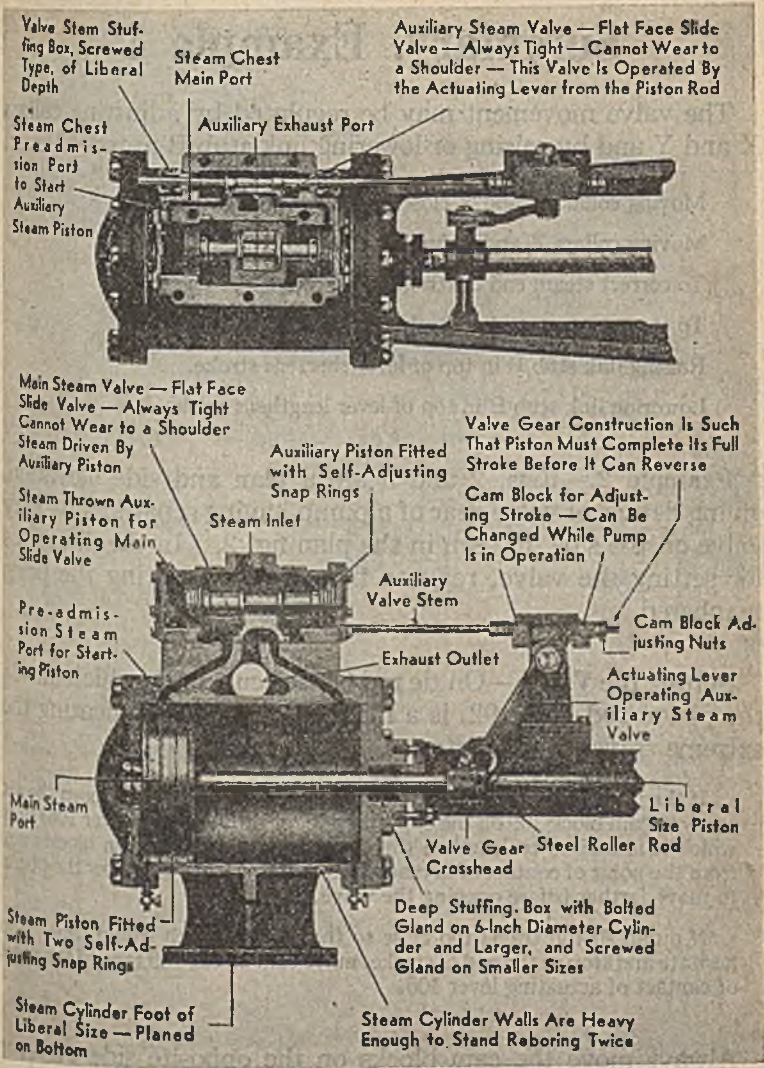
Pump End. Move collar X, away from dog Z.

Steam End. Move collar Y, away from dog Z.

After adjusting collars make further adjustment by raising or lowering link stub B. Raising stub B shortens stroke. Lowering stub B, lengthens stroke.

The raising or lowering of stub B, may be done without shutting off steam. This adjustment of valve movement should be made to get full stroke and proper operation. Pump should make full stroke, reversing slowly at each end.

Tie bar guiding cross head is marked showing contact and normal working stroke. Operator can see at a glance how pump is stroking, and if not making normal stroke, tell which end needs adjustment.



Figs. 2 and 3.—Class 1 simplex pump with separate auxiliary valve on side.

Summary. Example 1

The valve movement may be regulated by adjusting collars X and Y and by raising or lowering link stub B in top of lever.

Moving collars X and Y closer together shortens stroke.

Moving collars X and Y further apart lengthens stroke.

To correct steam end of stroke move collar Y.

To correct pump end of stroke move collar X.

Raising link stub B in top of lever shortens stroke.

Lowering link stub B in top of lever lengthens stroke.

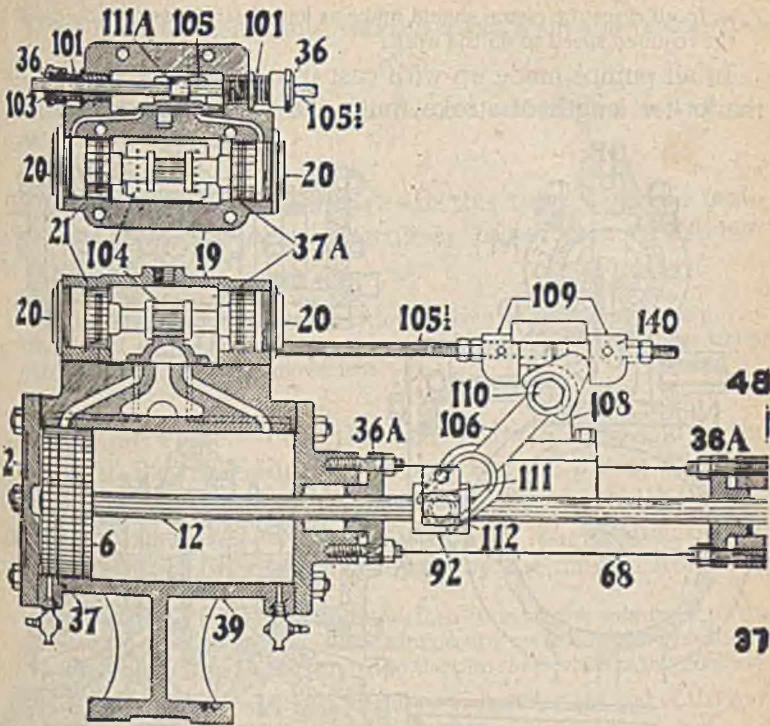
Example 2.—Figs. 2 and 3 are a plan and side elevation, giving details of valve gear of a pump having separate auxiliary valve on the side, as seen in the plan fig. 2. In the directions for setting the valve, refer to figs. 4 and 5 having the parts numbered.

Setting the Valve.—On tie rod 68 (see cut) upon which moves the piston rod guide 92, is a mark at each end, indicating the extreme travel of the piston.

If the pump do not run as close to the mark as practical, loosen the set screw in cam block 109 on the opposite side (of the actuating lever 106) from which it is desired to lengthen the stroke, and move the cam block away from the point of contact of actuating lever 106. This will allow the piston to move farther before opening the valve.

If pump should travel too close to the marks, which would cause it to hesitate and stop at the end of stroke, move cam blocks 109 toward the point of contact of actuating lever 106.

Always move the cam blocks on the opposite side of lever from which it is desired to change the stroke.



Figs. 4 and 5.—Class 1 simplex pump with separate auxiliary valve on the side. Detail of steam end with numbered parts to accompany directions for valve setting.

List of parts

The parts referred to in valve setting are:

68 Tie rod

92 Piston rod guide

109 Cam block

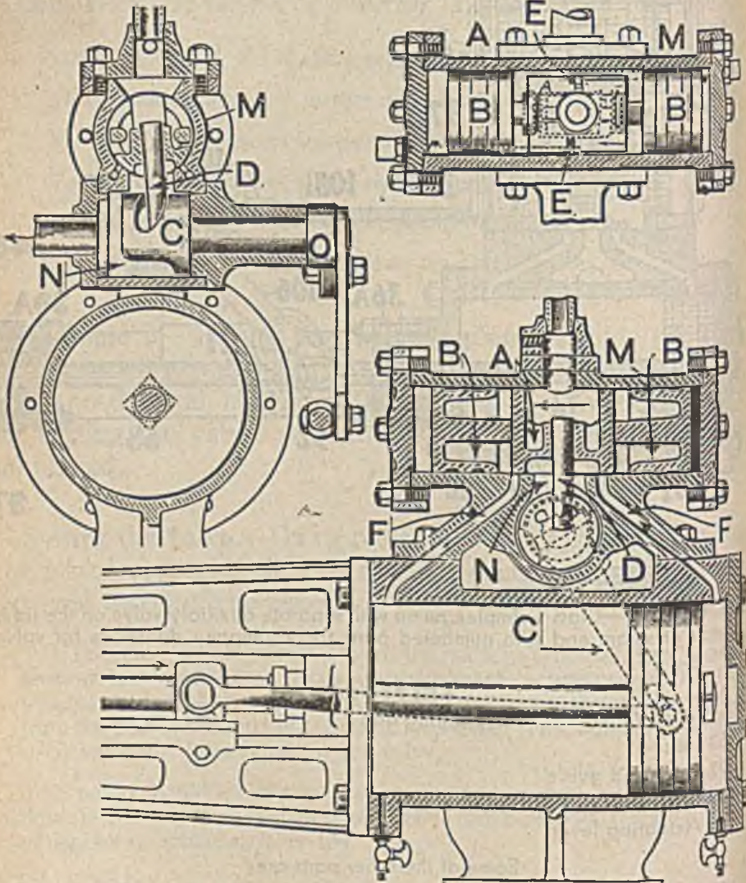
106 Actuating lever

Some of the other parts are:

111A, main steam valve; 101, valve stem; 36, valve stem stuffing box; 21, auxiliary valve; 20, cover; 106, rocker; 92, cross head; 68,] cradle; 6 piston; 12, piston rod.

In all cases the piston should make as long a stroke as possible and give the required speed to do the work.

In all pumps made up with cast iron yokes not showing the marks for length of stroke, push the piston to one end until



Figs. 6 to 8.—Class 2 simplex pump in which the main and auxiliary valves are combined.

it strikes cylinder head, and scratch a mark on the piston rod at the stuffing box.

Repeat this same operation on the other end, and use these marks to adjust the valve as described.

Valve Setting Class 2 Pumps.—In this type *there is a* **main** *slide valve combined with an* **auxiliary** *piston valve*, as shown in figs. 6 to 8.

In this type an initial rotary motion is given to the auxiliary piston by the external gear, causing it to uncover ports which give the proper steam distribution for the linear movement.

Setting the Valve.—Push the main pistons to the end of the stroke until the inner edge of the port and the piston coincide, then loosen the side lever, turn the cam C, until the valve piston uncovers the auxiliary steam port E, leading to the same end of the steam chest occupied by the main piston.

After setting, secure the cam and then connect the side lever to the connecting rod. The side lever and cam occupy correct relative positions, therefore, the lever should be secured to the cam shaft while in this position.

The stroke may be regulated by raising or lowering the end of the connecting rod in the slotted end of the slide lever.

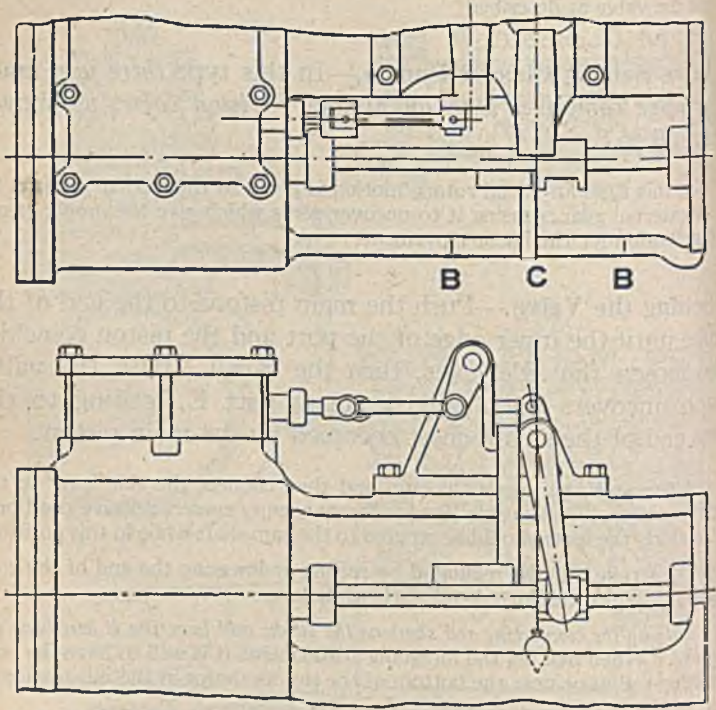
Raising the connecting rod shortens the stroke and lowering it lengthens the stroke. When making the foregoing adjustments it is well to have the connecting rod at or near the bottom of the slot as shown in the illustrations.

2. Duplex Pumps

If two steam pumps be placed side by side, it is found that the valve of each may be operated directly from the piston rod of the other without the aid of any auxiliary pistons or valves as is necessary with simplex pumps.

The duplex pump *has one main valve for each side, there being no auxiliary valves* as just stated.

The general arrangement is shown in figs. 9 and 10.



Figs. 9 and 10.—Plan and elevation of one side of a duplex pump showing steam cylinder and valve gear.

There are two types of main valves known as:

1. **D**
2. **B**

as shown in figs. 53 and 54, page 344. They are used on pumps having:

1. Five ports
2. Three ports.

That is, five ports for the D valve and three ports for the B valve.*

Locating Cross Head Centers.—Usually the cross heads of duplex pumps are held in position by a pin, which is driven through cross head and rod, after the former has been adjusted. It is impossible for the cross head to shift its position accidentally, unless the pin should drop out, and even then, there is a set screw, holding the cross head against slippage by ordinary use, and if such a thing should happen, the best way to readjust is to find its former position by the pinhole in the piston rod.

Ques. What must sometimes be done in case of old pumps?

Ans. Old scored or pitted piston rods must be replaced.

Sometimes, however, it is necessary to replace the old piston rods by new ones, which may be quite frequently, if the water be bad, and steel rods be used. In most cases the engineer will find that the rods can be put into their proper places without any trouble, as the builders have always exact fitting duplicates in stock, but it is better to be sure of this.

Ques. How can it be ascertained if the rods fit?

Ans. By making the following test: Mark the extreme position of the cross head on both sides of the pump on the frame or on a wood lathe, wedged in between the cylinder heads as shown in fig. 11.

*NOTE.—As elsewhere explained the D, valve is used on gears in which the valve follows the piston movement; the B, valve is used on gears in which the valve moves in reverse direction to the piston movement.

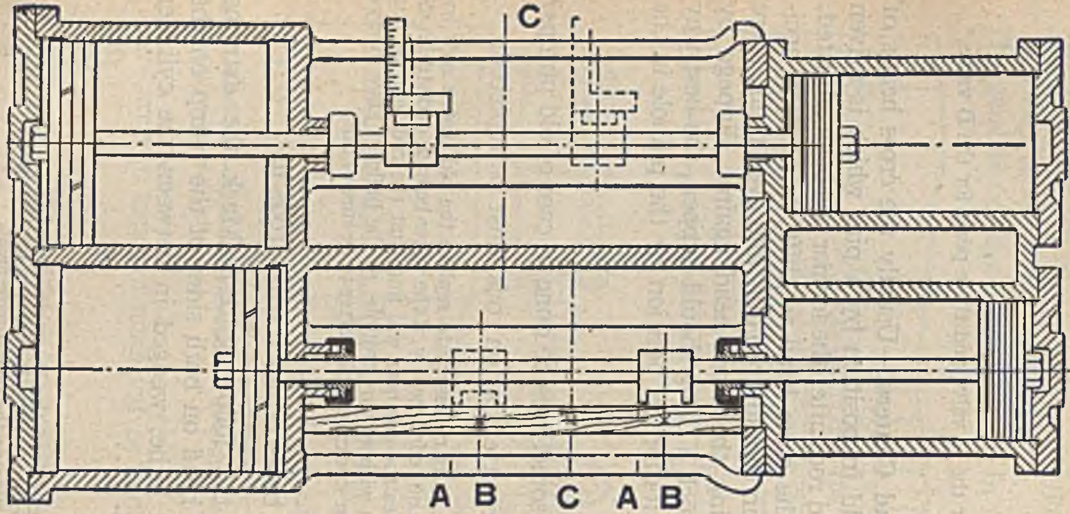


Fig. 11.—Location of cross head centers on duplex pumps.

Ques. How can the cross head positions be transferred without using a lathe?

Ans. Put a mark on either the frame or the lathe, to correspond with the central or mid-stroke position of the cross head.

Ques. How is this done?

Ans. There are two methods as shown in figs. 9 and 10.

Ques. What precaution should be taken with the plumb bob method?

Ans. It should be used only if the pump be leveled with precision.

When dropping the plumb bob in line with the center of the rock shaft as in fig. 10, the cross head may be moved close to the line, and its position be transferred to the frame, as in fig. 9.

Ques. What should be noted in using the other method?

Ans. When the square is used against the hub of the rocker arm, it will be seen by examining fig. 10 that the heel of the square does not indicate the center of the rock shaft, but is out an amount equal to one-half the diameter of the hub of the rocker arm, and the cross head should therefore not be set close to the square, but a distance equal to the radius of the hub, away from it.

Ques. How is this offset distance measured?

Ans. With an inside caliper or a rule.

Ques. How is the position of the cross head transferred?

Ans. It is transferred to the frame as in fig. 9, or marked on the lath as shown in fig. 11.

Ques. What is the advantage of the methods just described?

Ans. In both methods, no marks have been made on the piston rod, which is always best to avoid, the cross head having served for a mark in both cases.

If the pump be small, there is no difficulty to move the pistons for this purpose, but on a large pump, the cross head may be unfastened, so as to be free to slide on the piston rod.

Ques. What should be noted about the marks **AA**, fig. 11?

Ans. The marks AA, representing the extreme positions of the cross head have, however, been taken from one end of the cross head, and thus can not come equidistant from the mark C, representing the correct central position, even if the cross head be set correctly. Thus it will be necessary to transfer them toward the opposite end of the cross head, an amount equal to one-half the length of the cross head, BB, being the corrected marks.

If the position of the marks B B, be not equidistant from the center mark C, when the cross head is at the extreme ends of the stroke, it should be shifted on the piston rod, until in the proper position, the amount it is to be shifted will be indicated by the marks B B, fig. 11.

It will not be necessary to shift the cross head on the rod, if it be out only a small amount, as the duplex pump is not such a sensitive machine to require very delicate adjustment, and often it is found that if the entire mechanism be set correctly, the pump will not work as well under steam, as if slightly out of adjustment.

Ques. If the cross head be out of adjustment what should be done?

Ans. Test the pump under steam before making alterations.

Ques. What should be done in making this test?

Ans. The valves should be adjusted to suit the original position of the cross head, and if possible, it will be found very useful to attach a pointer to the cross head, pointing toward that part of the frame on which the center and extremes of the stroke have been marked.

By running the pump slow, it will be possible to ascertain the ends of the working stroke.

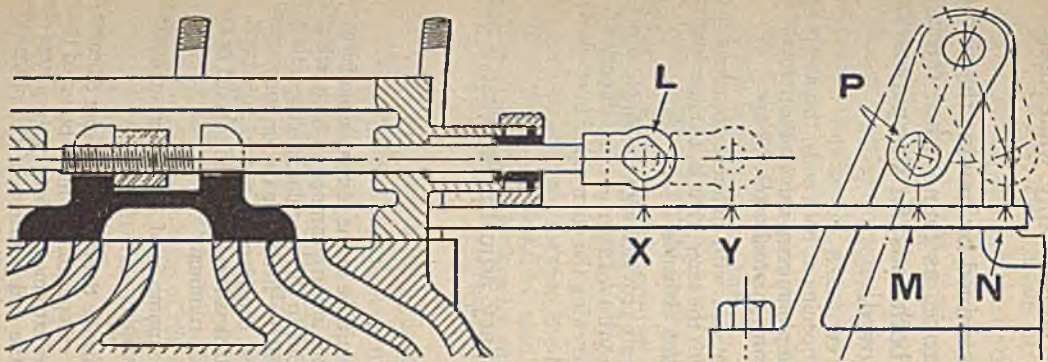


Fig. 12.—Valve and valve gear of duplex pump, showing rocker in its two extreme positions M and N, and the corresponding position of the valve stem end X and Y.

Ques. How are the ends of the working stroke determined?

Ans. By running the pump slow.

If the extreme positions of the pointer be marked, which can be done best by holding a lead pencil against the pointer, just touching the frame, the points, to which the lead pencil is pushed by the pointer on each end of the stroke, are the extremes of the stroke, when the pump is running, and by comparing these points with the marks previously obtained, indicating the true ends of the stroke, the clearance on each end can be obtained.

Ques. If the clearances differ considerably what should be done?

Ans. The valves should be examined by moving the pistons by hand to the extremes of stroke, as found when running, and noting the port opening at both ends; for this purpose the valve chest cover has to be removed.

If there be any difference in port opening at both ends, this may be the cause of the unequal clearance, and a preliminary valve adjustment should be made, by equalizing the port opening approximately by eye.

Various types of pumps are provided with different means for such adjustments, but the principle remains the same, that is, to either lengthen or shorten the valve stems, as occasion demands.

Most all types of the smaller sizes are provided with the simple adjusting device, as indicated in fig. 12, which consists of a square nut, through which the valve stem is screwed, and by screwing the stem either in or out, it is respectively shortened or lengthened.

Ques. After making preliminary adjustment what is next done?

Ans. Recheck under steam.

Should the marks denoting the clearance of the pistons again fall on the same points as before, and a difference in the clearance on both ends of the stroke be found, the trouble will be due to the irregular spacing of the ports in the cylinder bore, and there will be little chance for improvement, and, unless the cross head be found considerably out of adjustment, it should not be disturbed, and the final valve adjustment should be made to suit the extremes of the stroke while running.

It, however, rarely occurs that a pump is of such poor workmanship as to make proper adjustment impossible.

The location of the ends of the stroke does not make any difference in the manner of adjusting the valve, except, that it must be noted that in one case, by the end of the stroke, the extreme positions of the pistons when pried over, and in the other case the end positions of the pistons when allowed to run, are meant.

How to Set the Valves of a Duplex Pump.—Place a small stick or batten against the end of the valve chest, and mark the center of the pin P on the same, as indicated in fig. 12. Then move the piston, of the same side, to the other end of the stroke, and again mark the position of the pin P, on the same stick, as indicated by the dotted lines. The two marks M, and N, thus obtained, denote the extreme travel of the pin P.

It will now be necessary to obtain the marks X and Y on the same stick, which indicate the positions of the pin L when the valve has moved from one full port opening to the other.*

Now take a strip of stiff paper, and mark upon it the exact distance between the center of the holes in the valve connecting link, as D and E, fig. 15.

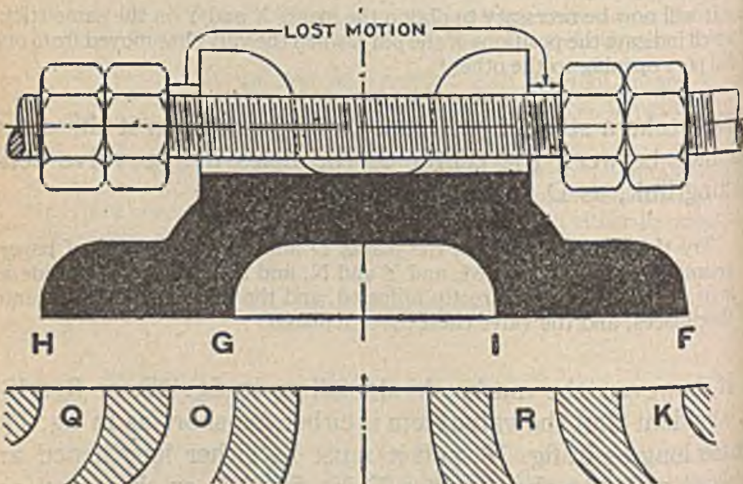
Try the distance between the marks D and E, on the strip of paper, against the marks X and M, and Y and N, and if they should coincide as in fig. 16, the valve is correctly adjusted, and the links should be put into their places, and the valve chest cover replaced.

If, however, the marks should fall as in fig. 17, or fig. 18, it is evident that the valve stem is either too short as in fig. 17, or too long as in fig. 18, and it must be either lengthened an amount equal to the distance E Y, fig. 17, or shortened an amount equal to the distance E Y, in fig. 18.

Figs. 19 and 20, show other positions in which the marks on the stick and the strip of paper may fall. In both cases, the travel of the valve between the two inside edges of the steam ports evidently does not coincide with the travel of the pin P, fig. 12, indicating that there is either too much lost motion between the valve stem and the valve, as in fig. 19, or not sufficient, as in fig. 20.

*NOTE.—When sliding the valve from one "full port" to the other, care should be taken to do this by moving the valve stem to obtain the full effect of the lost motion between the nut and the lugs on the back of the valve.

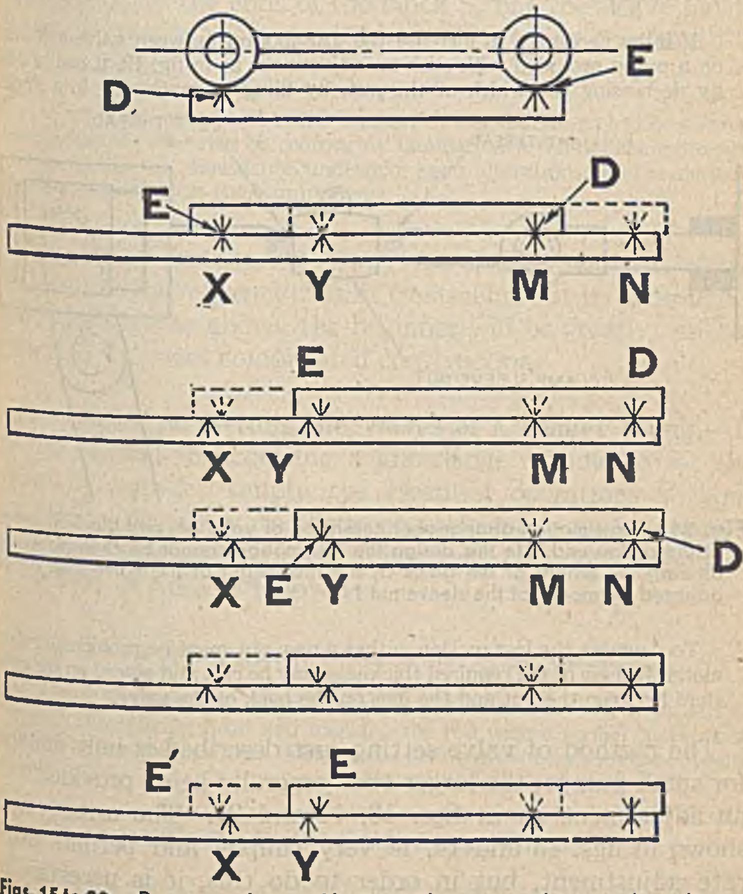
Before attempting to alter this, it is advisable to remove the valve entirely, and to see whether the distance between the steam and exhaust edges of the valve, as F and G, and H and I, fig. 13, correspond with the distances between the working edges of the ports K and O, and Q and R, respectively (fig. 14).



Figs. 13 and 14.—Main valve and valve seat of duplex pump; each "side" or pump is fitted with a valve and seat as here shown. H and F, are the steam edges of the valve and G and I, the exhaust edges. Q and K, are the steam ports and O and R, the exhaust ports; the exhaust cavity or outlet is seen at the center of the seat. Fig. 13, shows the lost motion between the stem and valve. The amount of lost motion given is such that the inlet ports are not closed and the exhaust ports opened too early in order to allow the piston to make a full stroke.

If these distances agree with each other, and the marks representing the valve and pin travel fall as in fig. 19, it indicates that the valve has not sufficient motion to fully open the ports, hence less lost motion has to be given. Fig. 20 shows the reverse of this condition.

Should the distance, between the edges F and G, or H and I, be found shorter than the distance between their respective port edges, an amount equal to one-half the difference between E′E and X Y, fig. 20, the steam edge of the valve will over



Figs. 15 to 20.—Paper template and batten with center marks as used in adjusting the valves of a duplex pump as fully explained in the accompanying text.

travel the inner edge of the steam port, when the valve is con-nected up, but the exhaust port would have just full opening indicating that there is some exhaust lap, and if the pump be found to run smooth, it is advisable not to tamper with the adjustment of the lost motion.

If it be necessary to increase the lost motion between valve and stem on a pump provided with such an adjustment as in fig. 12, it can be done by decreasing the width of the nut, by filing or machining in a shaper

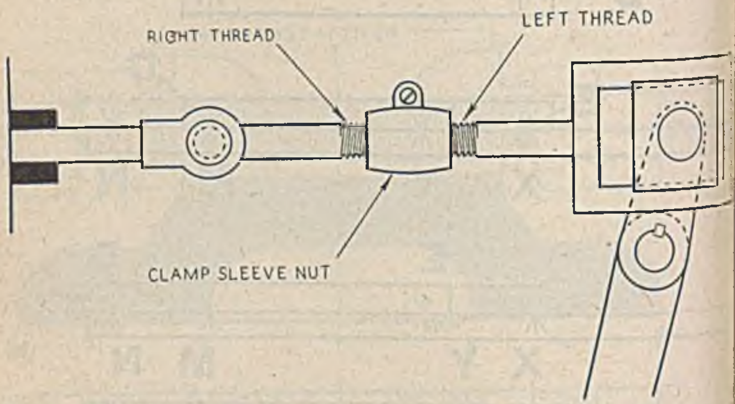


Fig. 21.—Lost motion arrangement consisting of yoke M, and block S, pivoted at the rocker end. In this design the lost motion cannot be changed without altering the length of the block S, but the length of the valve stem can be adjusted by means of the sleeve nut N.

To decrease the lost motion, either a new nut must be provided, or sheet metal washers of the required thickness may be cut, and placed on the valve stem between the nut and the lugs on the back of the valve.

The method of valve setting just described is only suitable for small pumps; the larger ones generally being provided with an adjustment as in figs. 13, 14 and 21. The arrangement shown in figs. 13 and 14, is very simple, and permits accurate adjustment, but in order to do this, it is necessary to remove the valve chest cover.

The type shown in fig. 21, is mostly used on large and more expensive pumps, and permits alterations in the adjustment being made while the pump is running.

In fig. 21, the lost motion can not be altered, without taking off or adding to the ends of the block S, but the sleeve nut in the connecting link is a good device for altering the length of the valve stem.

It is seldom the case that the amount of lost motion has to be altered, and unless the operator be thoroughly familiar with the details and design of the pump, he should not undertake such alterations, as the designer knows best what are the requirements.

These directions can not always be closely followed, as the different designs require different treatment, but by thoroughly understanding the above, the beginner will be greatly assisted even with the most complicated construction.

Short Rules for Setting the Valves of a Duplex Pump.—It may be helpful in acquiring a knowledge of how to set the valves to consider simply the essential operations without the various details or methods of performing them as given in the foregoing instructions. They may be briefly expressed in the form of rules as follows:

1. *Locate the steam piston in the center of the cylinder;*

This is accomplished by pushing the piston to one end of its stroke against the cylinder head and marking the rod with a scriber at the face of the stuffing box, and then bringing the piston in contact with the opposite head.

2. *Divide exactly the length of this contact stroke;*

Shove the piston back to this half mark; which brings the piston directly in the center of the steam cylinder.

3. *Perform the same operation with the other side;*

4. *Place the slide valves in their central position;*

5. *Pass each valve stem through the stuffing box and gland;*

The operation of placing the pistons in the center of their cylinders brings the levers and rock shafts in a vertical position.

6. *Screw the valve stem through the nuts;*

The stem is screwed until the hole in the eye of the valve stem head comes in a line with the hole in the links, connecting the rocker shaft.

7. *Put the pins in their places;*

8. *Adjust the nuts on both sides of the lugs.*

Leave about one-eighth to one-fourth inch lost motion on each side.

How to Set Duplex B and D Valve Gears.—*Since the levers of this type gear are duplicates, it is necessary that one of the steam valves should be a* D *valve and one a* B *valve, to give proper motion to the pump.*

Using a D valve and a B valve permits of a very unique arrangement of steam ports. There is but one cast port at each end of steam cylinder, this serving the purpose of steam and exhaust port.

The old duplex type of valve movement has two ports at each end of cylinder; one steam and cushioning port and one exhaust port, each port serving the single purpose of admitting steam to the cylinder or exhausting steam. The double ports cause excessive waste in steam, and also pump will short stroke.

Fig. 22 shows the gear. One side of the pump has been revolved 180° into the plane of the paper so as to show both sides in a single drawing—this is confusing to some but understand it first before reading further.

1 and 5 are **D** valve steam ports
2 " 6 " **B** " " "
3 " 4 " exhaust ports
7 " 8 " valve rod nuts

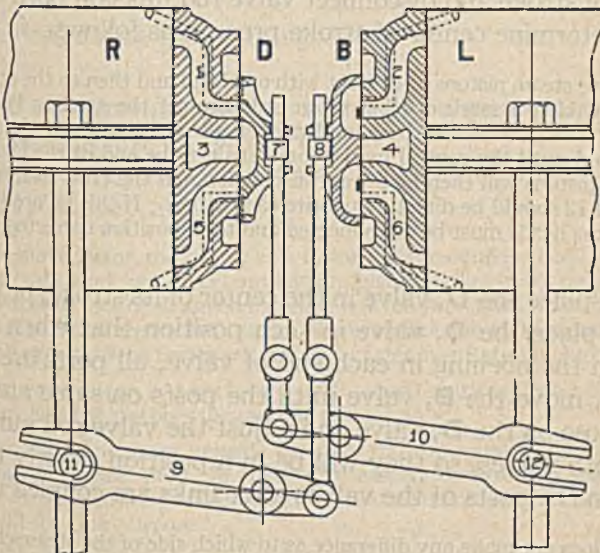


Fig. 22.—Valve gear detail of duplex pump having **B** and **D** valves.

Small drilled starting ports are shown by dotted lines in the illustration.

These starting ports may be disregarded when setting valves.

When erecting pumps, the valve rod is marked with a punch mark.

This punch mark is 1 inch from valve rod nuts 7 and 8. If valve movement be out of adjustment, simply place these nuts so that they are 1 inch from this mark and this will bring the parts back to their proper relation.

If, for any reason, this punch mark cannot be found, proceed in the following manner to set valves.

Setting the valves.—Place both steam pistons in the exact center of stroke and disconnect valve rod links on both sides.

To determine center of stroke proceed as follows:

Move steam pistons to contact with one head and then to the other, each time making a mark on the piston rods against the stuffing box glands. Then place a mark on each rod midway between the contact marks and move each rod until the central marks come against the end of the glands. The steam pistons will then be in central position and the cross head pins Nos. 11 and 12 should be directly opposite each other. If this be not true, the cross heads must be disconnected and their position corrected.

Next place the **D**, valve in the center of its stroke, or in other words, place the **D**, valve in such position that when looking through the opening in each end of valve, all ports are closed.

Now, move the **B**, valve until the posts on same are in line with those on the **D**, valve and adjust the valve rod nuts 7 and 8 on both valves, so they will be in a position exactly midway between the posts of the valves after links are coupled up.

It does not make any difference as to which side of the steam chest these valves are placed.

If these directions have been followed the steam valves and valve motion should be set correctly.

Finally, before placing chest cover in position, move either one of the slide valves against the nut on the valve rod. This is done so that in starting steam will be admitted through one of the ports.

Special Directions for Duplex Valve Setting.—The directions here given relate to valve setting on pumps having various lost motion arrangements. These may be classed as:

1. Internal

    a. Lost motion nut between valve lugs.

    b. Lost motion nuts at each end of valve.

2. External

These designs are shown in figs. 23 to 25.

Case 1.—*Instructions for gears moved by a single valve rod nut working between lugs on valve* (Fig. 22).

First open drip cocks so that water in steam cylinders will be completely drained away.

Now move piston rod of one side toward steam cylinder head by prying against cross head (not lever) until steam piston strikes head; make a mark on rod close to face of steam end stuffing box follower, then move piston rod to opposite end of stroke until steam piston strikes and make a mark on rod just half way between first mark and face of steam end stuffing box follower.

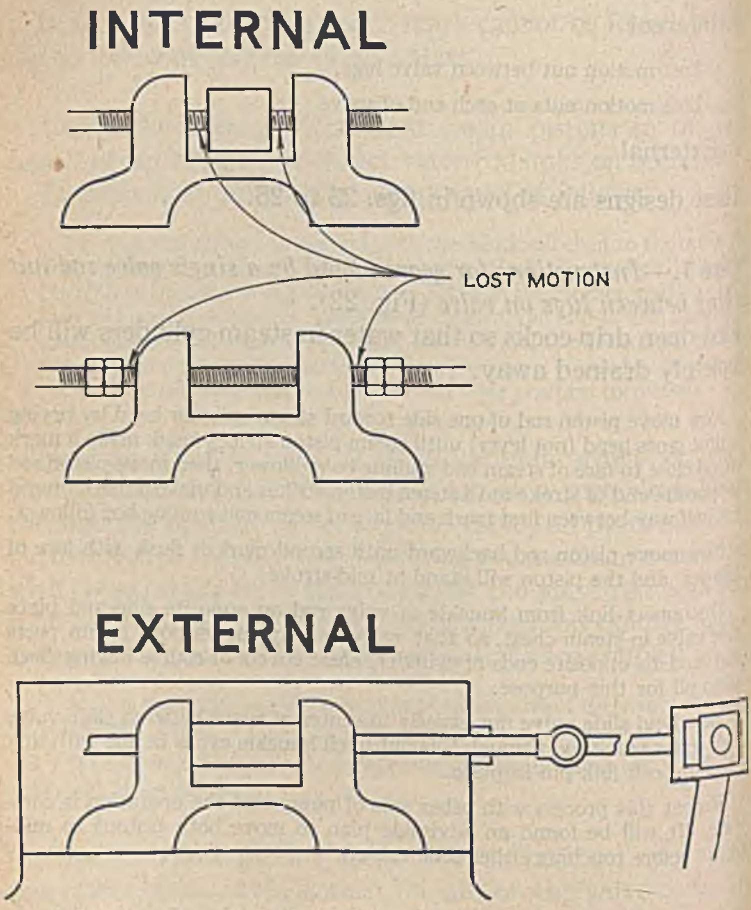
Now move piston rod backward until second mark is flush with face of follower, and the piston will stand at mid-stroke.

Disconnect link from knuckle of valve rod on opposite side and place slide valve in steam chest, so that valve exactly covers both steam ports that lead to opposite ends of cylinder, chest covers of course having been taken off for this purpose.

Now hold slide valve nut exactly in center of space between slide valve lugs; screw valve rod through this nut until knuckle eye is in line with link eye and push link pin in place.

Repeat this process with other side of pump and the operation is complete. (It will be found an advisable plan to move both pistons to mid-stroke before touching either slide valve.)

After everything is properly adjusted and before replacing chest cover, be sure to move one of the slide valves off center



Figs. 23 to 25.—Various lost motion arrangements, internal and external, to accompany special valve setting directions.

so as to leave one steam port open, otherwise pump cannot be started. *In operation* the valves can never move so that both will be on center at the same time under any condition of running. It is only when the valves are deliberately placed, as in the operation of setting, that this can happen.

Case 2.—*When valve rod has lock nuts at each end of the valve* (Fig. 24).

First place pistons and slide valves on centers, as previously mentioned, but do not disconnect valve rod from link; then set and lock nuts at equal distances from outer faces of valve lugs, allowing about half the width of steam port for lost motion on each side.

A good way to prove equality of lost motion is to move valve each way until it strikes nut and note if both port openings be equal. If it be found that this allowance gives pump too much or too little length of stroke, the lost motion will have to be altered by trial until pump makes desired stroke.

Too much lost motion lengthens stroke and may cause pistons to strike cylinder heads; too little lost motion shortens stroke, decreases pumping capacity and increases steam waste.

Case 3.—*For external lost motion adjustment.*

Set piston in the middle of its stroke, likewise the steam valve on the opposite side; move collars on the valve rod link so that they will be about half the width of the steam port away from the tappet.

Repeat this operation on the opposite side and the valves are set. Pump may now be started and if it be found that stroke is too short, the collars must be screwed farther apart, care being taken to turn back all collars the same amount, for otherwise the piston rod movement will be less than full stroke and nearer to one end of cylinder than other.

If stroke be too long so that pistons strike heads, collars must be set closer together.

With this type of valve gear, it is not actually necessary to stop pump and remove chest cover as the collars can be adjusted by trial until required length of stroke is reached, while pump is in regular operation. When adjustment is finally made, be sure to lock collars securely in place.

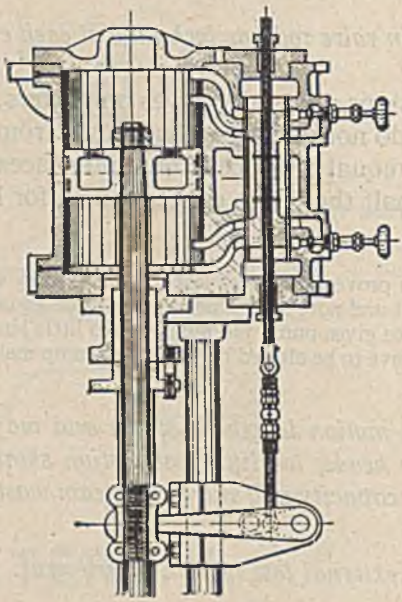


Fig. 26.—Vertical piston valve pump with outside lost motion adjustment.

Case 4.—*For piston type valve with external lost motion adjustment* (Fig. 26).

Set main steam piston at mid-stroke, likewise steam valve on the opposite side.

To set valve in mid-position, remove top valve chest cover and move valve until its upper face is just in line with the top edge of the uppermost steam port.

If valve cylinder be of so small a diameter that it is difficult to set valve on port line by inspection, measure distance from upper edge of top port to upper face of valve chest and move valve until its upper end is the same distance below face of chest. When thus set, lower end will be in line with lower edge of lower steam port as valve is made without lap.

Friction between valve and cylinder bore or stuffing box packing is usually sufficient to hold valve in any position it may be placed, but if this be not the case, valve must be prevented dropping, while adjustments are being made, by means of a temporary blocking of wood or whatever material may be available.

After main piston and steam valve are arranged in central position, collars are set in the same manner as for an ordinary duplex slide valve with outside adjustment, and, as is the case with these pumps having valves with outside adjustment, the valve travel and length of stroke can be adjusted by trial while pump is running without the necessity of removing chest cover.

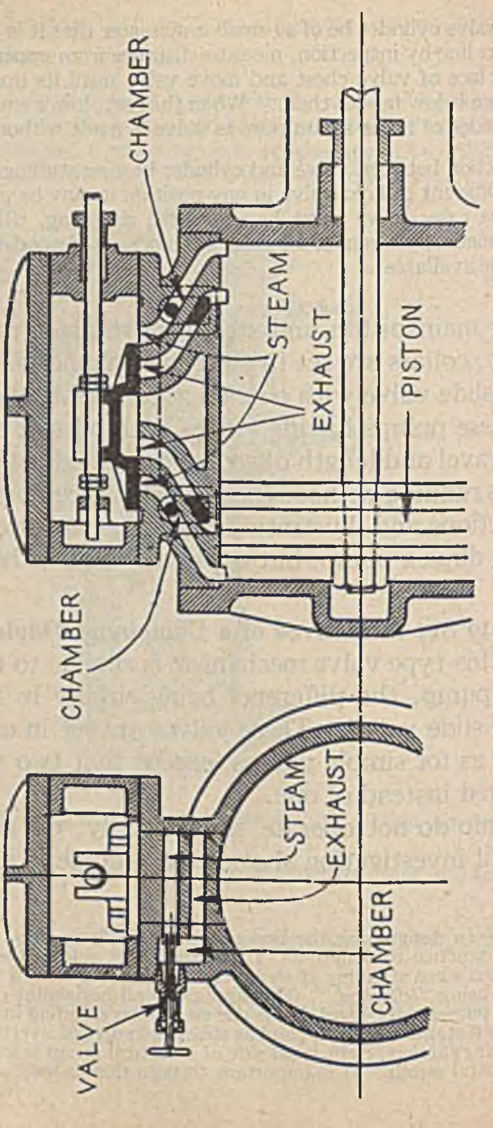
Directions and illustration refer to piston steam valves on a vertical duplex pump, but apply as well to a horizontal pump.

How to Set the Valves of a Compound Duplex Pump.—Evidently this type valve mechanism is similar to that of a simple duplex pump, the difference being simply in the addition of another slide valve. These valves are set in exactly the same manner as for simple pumps, except that two valves are to be considered instead of one.

If pump do not operate satisfactorily, do not touch steam end until investigation shows that trouble is not elsewhere.

NOTE.—In designating the two sides of a duplex pump, manufacturers have made it a practice to denote as "*right-hand*" that side of the pump at the right which is seen when standing at steam end and looking toward water end, the other of course, being "*left-hand.*" This applies to all horizontal pumps. In the case of vertical pumps right-hand side is one seen when standing at and facing front side of pump as it stands vertically, and as steam cylinders of a vertical pump are always above water cylinders, right hand side of a vertical pump is same as left-hand side of a horizontal pump. It is important to note this distinction in ordering repair parts.

CUSHIONING VALVES



Figs. 27 and 28.—Steam cylinder with cushioning valves. End and side sectional views.

Most pump troubles are due, not to steam end conditions, but to poor packing, fouled water cylinders, worn water valves or to faulty conditions.

The steam cylinders of certain types and sizes of direct-acting pumps are fitted with cushion valves, as shown in figs. 27 and 28.

The purpose of these, as previously explained, is to provide an adjustable steam cushion for the piston so as to insure a full-length working stroke, and at the same time, prevent piston striking cylinder heads when pump is working under widely varying conditions of load.

Referring to the figures a chamber, one at each end of the cylinder, forms a connection between steam port and exhaust port, and this connection can be shut off by means of a cushion valve.

As the piston approaches end of stroke it covers exhaust port, through which steam has been exhausting and confines it in space at cylinder end, thus forming a steam cushion that prevents piston striking head. Amount of this cushion is varied by opening or closing the cushion valve, the only means of escape for the confined steam being through chamber to exhaust port.

The more the cushion valve is open the longer the stroke will be.

If pump be running at low speed or working under heavy load, cushion valves should be open as much as possible without allowing piston to strike heads.

If pump be running at high speed or working under light load, valves must be closed more. Amount of steam cushion, and consequently length of stroke, can be properly regulated under different conditions of running by the simple adjustment of these valves. See also page 331.

The valve gear of pump should always be adjusted so that pistons will make as long stroke as possible without striking heads while cushion valves remain wide open.

Cushion valves are then partly or wholly closed, as varying conditions of load may demand.

Pumps of the smaller sizes are not fitted with cushion valves; instead of the valve, a plug is screwed into the wall separating ports ST and EX at C. A small hole is drilled through the plug of a size suited for the best general operation of the pump and through this hole the confined steam is slowly exhausted, the banked steam forming a proper cushion for the advancing piston.

CHAPTER 17

Reciprocating Pumps; Calculations

Numerous calculations, data and tables are here given relating to various types of pumps which will be found useful to the engineers as well as erectors and operators. The calculations are of a diversified nature and are intended to cover those ordinarily encountered.

Practical Lift.—Theoretical and practical lifts are quite different, there being a number of conditions causing the practical lift to be considerably less than the theoretical value. Accordingly for erectors, the correct determination of the practical lift is very important.

In fact, failure to correctly determine the lift has been the cause of numerous failures in installations.

Although Torricelli demonstrated by experiment that the pressure of the atmosphere at sea level, 14.7 lbs. per sq. ft., will support a column of cold water at its maximum density 33.83 ft. high, no pump will do this.

In making a calculation for theoretical lift referred to a 30 inch barometer which corresponds to atmospheric pressure of

14.74 lbs. per sq. in. A column of water approximately 2.31 feet high exerts a pressure of one lb. per sq. in. Accordingly, the corresponding theoretical lift is

$$\text{Ther. lift} = 2.31 \times 14.74 = 34.049 \text{ feet.}$$

(usually stated as 34 feet.)

The actual lift that can be obtained in practice is limited by the following opposing conditions:

1. Temperature of the water
2. Lowering of the air pressure with elevation
3. Frictional resistance through pipes, fittings and passages
4. Leakage

Practical or Permissible Lifts

For various temperatures and altitudes

ALTITUDE	Temperature of Water in Degrees F.														
	60	70	80	90	100	110	120	130	140	150	160	170	180	190	200	[illegible]
At sea level	−22	−20	−17	−13	−13	−11	−8	−6	−4	−2	0	+ 3	+ 5	+ 7	+10	[illegible]
At 2000 alt.	−19	−17	−15	−13	−11	− 8	−6	−4	−2	+1	+ 3	+ 5	+ 7	+10	+12	[illegible]
At 4000 alt.	−17	−15	−13	−10	− 8	− 6	−4	−1	+1	+3	+ 5	+ 7	+10	+12	+14	...
At 6000 alt.	−15	−13	−11	− 8	− 6	− 4	−2	+1	+3	+5	+ 7	+10	+12	+14	+16	...
At 8000 alt.	−13	−11	− 9	− 6	− 4	− 2	0	+3	+5	+7	+ 9	+12	+14	+16	...	...
At 10,000 alt.	−11	− 9	− 7	− 4	− 2	0	+2	+4	+7	+9	+11	+14	+16	+18	...	...

The above table according to Worthington gives the practical or permissible lifts corresponding to various temperatures and elevations. Where the values are preceded by a minus (−) sign, *lift* is indicated; where the values are preceded by a plus (+) sign *head* is indicated.

Ques. Does this table apply for pumping liquids other than water and why?

Ans. No. It depends upon the specific gravity of the liquid being pumped.

Thick liquids such as tar, molasses, should always flow to the pump by gravity that is, under inlet head frequently called *negative* lift.

The Inlet Pipe.—This pipe ignorantly called the "suction pipe" should be as direct as possible, that is, not tortured by a multiplicity of close elbows, which might offer enough frictional resistance to upset the values in the table just given.

Accordingly to reduce frictional resistance to a minimum, the elbows should be avoided and the inlet pipe proportioned for a flow of

250 ft. per minute*

Example.—A double acting duplex power pump runs at 50 *r.p.m.* cylinders 10 in. diameter by 12 in. stroke. Find diameter of inlet pipe.

Each pump makes two discharge strokes per revolution hence the piston speed of each pump is

$$\frac{12 \times 2}{12} \times 50 = 100 \text{ feet}$$

Since there are two pumps, the total distance traveled by the pistons or total piston speed is

$$2 \times 100 = 200 \text{ ft.}$$

Now area of the two pistons

$$= 2 \times 10^2 \times .7854 = 157.1$$

The area of the inlet pipe will be as much smaller than the area of the cylinders, as 200 is to 250, that is,

*NOTE.—This was the author's practice in designing pumps for City Water Works.

$$\text{Area inlet pipe} = 157.1 \times \frac{200}{250} = 125.7$$

$$\text{Diam.} = \sqrt{\frac{125.7}{.7854}} = 12\tfrac{5}{8} \text{ approx.}$$

Example.—What is the diameter of inlet pipe for a 1,000,000 gallon (per 24. hours) pump for 250 ft. flow through the pipe?

1,000,000 gallons per 24 hours

$$= \frac{1{,}000{,}000}{24 \times 60} = 694 \text{ gals. per min.}$$

One gallon = 231 cu. ins., and the volume of flow per minute is

$$694 \times 231 = 160314 \text{ cu. ins.}$$

Area inlet pipe for 250 ft. flow

$$= \frac{160314}{250 \times 12} = \frac{160314}{3000} = 53.4 \text{ sq. ins.}$$

$$\text{Diameter} = \sqrt{\frac{53.4}{.7854}} = 8\tfrac{1}{4} \text{ in. approx.}$$

As this is not a standard size take the next *larger* size which is 9 ins.

Ques. Why next *larger* size?

Ans. This is especially important for the inlet side, as increased flow with the smaller pipe would add to the frictional resistance of the pipe.

Manufacturers usually make the outlet one pipe size smaller than the inlet opening. The general practice is illustrated in the following table, which gives not only pipe sizes for water end, but also for steam end. It represents the latest practice of one of the leading manufacturers of pumps.

Ques. How is lift measured?

Ans. Vertically from the surface of the water to the center of the inlet opening on the pump.

Pump Pipe Sizes

For Simplex boiler feed and general service piston pumps

Size —Inches	Boiler feed capacity		CAPACITIES FOR CONTINUOUS SERVICE*										PIPE DIAMETERS —INCHES			
			Water and liquids up to 250 S.U.S. visc.		Liquids 250 to 500 S.U.S. visc.		Liquids 500 to 1000 S.U.S. visc.		Liquids 1000 to 2500 S.U.S. visc.		Liquids 2500 to 5000 S.U.S. visc.					
	Gal. per min.	Boiler horse-power	Gal. per min.	Piston speed —ft. per min.	Gal. per min.	Piston speed —ft. per min.	Gal. per min.	Piston speed —ft. per min.	Gal. per min.	Piston speed —ft. per min.	Gal. per min.	Piston speed —ft. per min.	Steam	Ex-haust	Suc-tion	Dis-charge
4½x2¾x6	8	110	13	45	11	39	10	35	8	27	6	22	¾	1	1¼	1
5¼x3¼x7	13	180	21	50	18	43	16	39	13	30	10	25	¾	1	1½	1¼
6 x4⅛x8	23	325	38	55	33	48	29	43	23	33	19	27	¾	1	2½	2
7½x4½x10	31	450	52	63	45	55	40	49	31	38	25	31	1¼	1½	3	2½
8 x5 x12	42	600	71	70	61	60	55	54	42	42	35	35	1¼	1½	3	3
10 x6 x12	62	875	103	70	88	60	79	54	62	42	51	35	1¼	1½	4	3
12 x7 x12	84	1200	140	70	120	60	108	54	84	42	70	35	1¼	1½	5	4
14 x8 x12	109	1575	182	70	156	60	140	54	109	42	91	35	2½	3	5	4
16 x10 x18	219	3150	365	90	316	80	285	70	220	54	183	45	2½	3	8	6

*These normal operating capacities are for continuous duty. For emergency service increase about 15 per cent.

Ques. What precaution should be taken in installation if the level of the water be liable to fluctuate?

Ans. The permissible lift should be measured from the lowest level.

Size of Discharge Pipe.—The size of the discharge pipe is generally made somewhat smaller than that of the inlet pipe.

The exact size will depend upon the nature of the discharge line, its length, number of elbows and other conditions tending

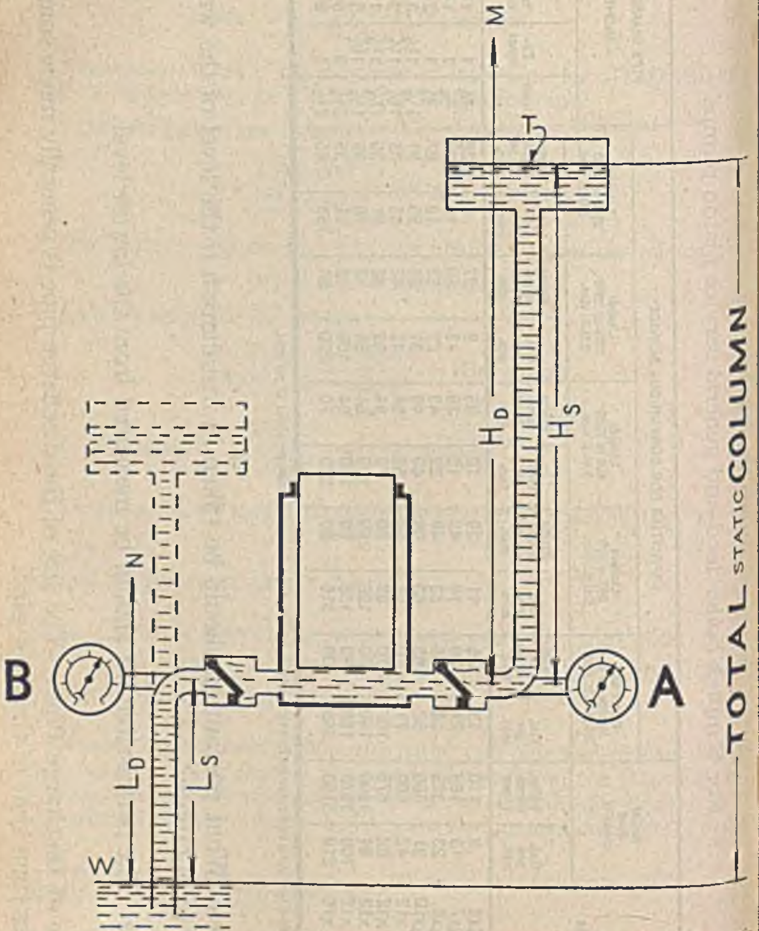


Fig. 1.—Gauge method of obtaining static and dynamic heads, also static and dynamic lifts. Note **total column** in place of the alleged and very objectionable total "head" as it is ordinarily called.

to set up resistance to the flow of water. For tank service or installations in buildings, the discharge pipe is usually calculated on a basis of 400 ft. per minute flow.

Head.—The static or dynamic head acting upon a pump may be found by a test gauge placed on the discharge pipe of a pump, as in fig. 1, gauge A. If the pump be at rest, the reading will give the *static* head H_S, if in operation, the *dynamic* head H_D.

Example.—With pump at rest what is the static head H_S when gauge A (fig. 1) reads 35 lbs.?

A column of water 2.31 ft. high exerts a pressure of 1 lb. per sq. in. Hence for a gauge reading of 35 lbs. the corresponding head or

$$H_S = 2.31 \times 35 = 80.9 \text{ ft.}$$

If the pump were in motion, the gauge **A**, would read higher owing to frictional resistance in pipe to flow and this would correspond to the *dynamic* head H_D, represented by the dimensions H_D in fig. 1, extending to some imaginary point as M.

Ques. Is the head the total load on the pump?

Ans. No, the lift L, must also be considered.

In fig. 1, evidently the pump must raise the water from level W, in the well to level T, in the tank, hence *total static column* C_S or

$$C_S = L_S + H_S$$

Ques. What name is very objectionably generally given to the "total column" $L_S + H_S$ (or $L_D + H_D$).

Ans. Total static head or total (dynamic) head.

In spite of the cast iron definitions for *lift* and *head* which have become firmly fixed by long usage, it would require a stretch of the imagination to agree with any argument advanced which would justify including the lift as part of the alleged total head. The author in order to avoid such messy misuse of words calls it the "total column" and that is just what it is, specifically *static* or *dynamic*.

Ques. How may static and dynamic lifts (L_S and L_D, fig. 1) be measured?

Ans. By taking readings of the vacuum gauge **B**, and converting the inches of vacuum into ft. elevation.

Example.—What is the dynamic lift (L_D) if vacuum gauge B (fig. 1) read 18 ins.?

$$\text{Lift in ft.} = \text{vacuum reading} \times .49116 \times 2.31 \ldots\ldots\ldots\ldots (1)$$

In the equation .49116 is lbs. per sq. in. corresponding to 1 in. of mercury. 2.31 ft. is the height of a column of water which exerts a pressure of 1 lb. per sq. in. Substituting vacuum reading in equation (1)

$$\text{Lift in ft.} = 18 \times .49116 \times 2.31 = 20.4 \text{ ft. approx.}$$

Ques. If the water flow to the pump under a head as indicated by the dotted outline in fig. 1, how is the total column obtained?

Ans. The inlet head should be subtracted from the discharge head.

That is, instead of $H + L$ for total column it becomes $H - L_N$ in which L_N is "negative" lift as sometimes questionably called.

In fig. 1 dynamic lift L_D is represented by the dimension L_D extending to some imaginary point as N

Ques. What allowance is made for velocity head in direct acting pumps and why?

Ans. It is generally negligible as the velocities are low.

Displacement.—By definition, the displacement of a reciprocating pump is: *The volume swept through or displaced by the piston or plunger in a single stroke.*

Displacement is stated in various ways as:

1. Cu. ins. per stroke

2. Cu. ins. per minute
3. Gallons per minute

However, stated in terms of cu. ins. per stroke adheres more closely to the meaning of the definition.

To determine the displacement of a pump cylinder in cu. ins. per stroke, apply the following rule:

Rule.—Multiply the effective area of the piston (or plunger) by the length of the stroke.

Example.—A double acting simplex piston pump has a water cylinder 5×12, 1 in. piston rod. What is its displacement per stroke in cu. ins.?

Area piston $= 5^2 \times .7854 =$ ....19.635 sq. ins.

½ Area piston rod $= \dfrac{1^2 \times .7854}{2} =$ .... .393*

Effective area piston (= Item A − Item B) = ...19.242 sq. ins.

Displacement = effective piston area × stroke

$= 19.242 \times 12 = 230.9$ cu. ins.

Example.—What is the displacement of the pump per minute in the preceding example, when running 92 strokes per minute?

Rule.—*Cylinder displacement per stroke multiplied by the number of discharging strokes per minute.*

Since the pump is double acting each stroke is a discharging stroke and accordingly—

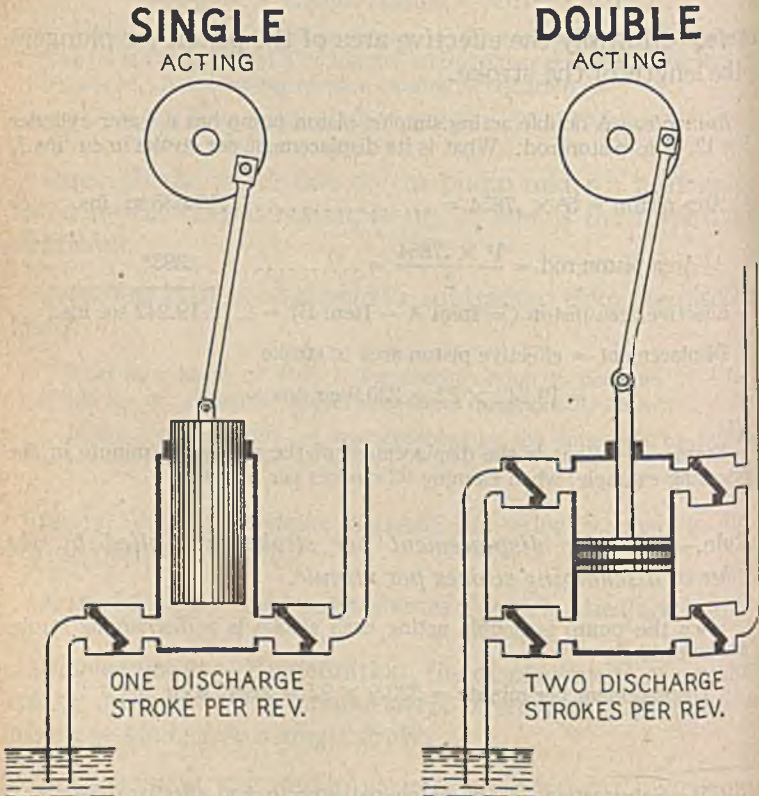
Displacement per minute $= 230.9 \times 92 = 21242.8$ cu. ins.

*NOTE.—Substracting *half* of piston rod area to find effective piston area relates, as must be evident, to a double acting pump. Clearly the whole rod acts to reduce the displacement on one side of the piston, but not on the other, hence subtracting the half area would give the average.

Ques. What is the displacement per minute in gallons?

Ans. The volume of one gallon is 231 cu. ins., hence displacement = 21242.8 ÷ 231 = 91.96 say, 92 gallons per minute.

Piston Speed.—By definition: *The total distance in feet traveled by a moving piston (or plunger) in one minute.*



Figs. 2 and 3.—Simplex single acting and double acting power pumps illustrating figuring displacement with piston speed.

Example.—A pump having a 16 in. stroke makes 60 strokes per minute.

$$\text{Piston speed} = \frac{16 \times 60}{12} = 80 \text{ ft. per min.}$$

Figuring Displacement with Piston Speed.—*Piston speed* can be used as a factor in figuring displacement, but a *coefficient* must be used whose value depends upon the type of pump stated in terms of number of *discharging strokes "per revolution."*

To make this plain, consider the so-called power pumps. For instance consider a single acting power pump as in fig. 2, and a double acting pump, fig. 3. Evidently the single acting pump, fig. 2, has *one* discharging stroke per revolution, and the double acting pump, fig. 3, has *two* discharging strokes per revolution.

Now *piston speed* is based upon both the up or *charging* stroke and the down or *discharging* stroke. Since in one revolution for the single acting pump (fig. 2) there is only one discharging stroke, the coefficient here is ½, that is the piston speed must be multiplied by ½. Evidently in the double acting pump the coefficient is 1, since there are two discharging strokes per revolution.

Example.—A single acting power pump with a displacement of 300 cu. ins. running at 100 *r.p.m.* What is the displacement per revolution?

Since (in fig. 2) there is only one discharging stroke per revolution, the displacement is

$$300 \times \tfrac{1}{2} = 150 \text{ cu. ins.}$$

Example.—A double acting pump (as in fig. 3) has a displacement of 300 cu. ins. running at 100 *r.p.m.* What is the displacement per revolution?

Here the pump discharges both strokes and evidently the coefficient is 1, that is

$$300 \times 1 = 300 \text{ cu. ins. per } r.p.m.$$

Evidently the various combinations can be tabulated with respect to piston speed.

Single Acting

	Coefficient ×	Piston Speed
Simplex	½	1
Duplex	1	1
Triplex	1½	1

Double Acting

Simplex	1	1
Duplex	2	1
Triplex	3	1

Example.—A triplex single acting pump has a displacement of 300 cu. in. per cylinder. What is the displacement per revolution?

Displacement per revolution = 300 × 1½ = 450 cu. ins.

Example.—A duplex double acting pump has a displacement of 300 cu. ins. per cylinder. What is the displacement per revolution?

Displacement per revolution = 300 × 2 = 600 cu. ins.

Piston Speeds for Pumps.—The capacity of a pump having voluntary opening valves is limited by the number of times the valves will open and close smoothly and quietly, and not by the piston speed.

The speeds listed in the following table are determined from years of observation and tests by manufacturers and are such as will result in quiet and satisfactory operation with minimum reversals of the pistons, both of which add to the durability of the pump under continuous operating conditions.

Piston Speeds for Pumps

Stroke of Pump	General Service Pumps		Light Service Pumps		Boiler Feed and Hot Water Pumps Viscous Liquid Pumps		Hot Oil Pumps		Wet Vacuum Pumps Jet Condensers		Pressure Pumps	
	Strokes Per Min.	Ft. Per Min.	Strokes Per Min.	Ft. Per Min.	Strokes Per Min.	Ft. Per Min.	Strokes Per Min.	Ft. Per Min.	Strokes Per Min.	Ft Per Min.	Strokes Per Min.	Ft. Per Min.
3	100	27	108	27	72	18	50	12.5	100	25	80	20
4	100	33.3	105	35	66	22	50	16.6	100	33.3	72	24
5	92	38.3	94	39	57	23.8	50	21	92	38.3	63	26.2
6	80	40	84	42	52	26	50	25	80	40	60	30
7	77	45	77	45	48	28	50	29	77	45	55	32
8	75	50	75	50	45	30	50	33.3	75	50	54	36
10	66	55	72	60	45	37.5	50	41.6	66	55	54	45
12	60	60	65	65	39	39	50	50	60	60	50	50
16	56	74.6	60	80	34	45.3	39	52	56	74.6	45	60
18	54	81	60	90	34	51	35	52.5	54	81	43	64.5
20	51	85	60	100	33	55	33	55	51	85	42	70
24	50	100	50	100	30	60	30	60	50	100	38	76

Slip.—The displacements in the preceding examples are only theoretical displacements—no pump in actual operation discharges a volume of water equal to its theoretical displacement because of:

1. Slip through the valves and
2. Leakage

By definition, slip of pumps is: *That amount by which the volume of water delivered per stroke falls short of the pump's displacement, generally expressed as a percentage of the displacement.*

Example.—The displacement of a certain pump is 300 cu. ins. but in operation the volume of water discharged is only 285 cu. ins. per stroke.

What is the slip in per cent of the displacement?

$$\text{Slip} = 300 - 285 = 15 \text{ cu. ins.}$$

$$= \frac{15}{300} \times 100\% = 5\%$$

Ques. What allowance is usually made for slip?

Ans. Slip varies in pumps from 2% to 10% depending on the type of pump, whether piston or plunger, also the condition of the pump and the pressure the pump is working against. For a plunger pump it is good practice to figure 2% in estimating the slip of a pump, for light service piston pumps 5%, and for pressure pattern piston pumps 10%.

Ques. What other factor except slip may reduce the output of a fast running pump?

Ans. If the speed be too great the water cannot flow through the inlet valves fast enough to completely fill the cylinder.

Ques. What is *negative slip?*

Ans. This relates to a discharge greater than the displacement.

This can happen with bucket valve pumps operating on low lift where the moving column of water has sufficient dynamic inertia, that is, momentum to continue in motion during part or all of the return stroke.

Capacity.—By definition, the capacity of a pump is: *The actual volume of water delivered.* It is usually stated in terms of gallons per stroke or gallons per minute when discharging at a given speed.

Example.—A certain single acting pump has a displacement of 300 cu. ins. What is its capacity in gallons per minute when making 100 strokes per minute with 5% slip?

Since it is single acting there are only $100 \div 2 = 50$ discharging strokes per minute, hence

Displacement per min. $= 300 \times 50 = 15000$ cu. ins.

5% slip $= 15000 \times .05 = 750$ cu. ins.

Capacity per min. $= 15000 - 750 = 14250$ cu. ins.

$= 14250 \div 231 = 61.59$ gallons.

How to Figure Capacity.—Rule: *Multiply the area of the piston in sq. ins. by the length of the stroke in ins., and by the number of delivery strokes per minute; divide the product by* **1,728** *to obtain the theoretical capacity in* **cu. ft.,** *or by* **231** *to obtain theoretical capacity in* **U. S. gallons.** *The result thus obtained is to be multiplied by an assumed* **factor** *representing the efficiency of the pump to obtain the* **approximate** *net capacity.*

The rule expressed as a formula is

$$\text{Approximate net capacity} = \frac{.7854 \times D^2 \times L \times N}{1{,}728} \times (1 - f) \textit{ cu. ft.,}$$

or

$$= \frac{.7854 \times D^2 \times L \times N}{231} \times (1 - f) \textit{ gallons}$$

in which

D^2 = square of piston or plunger diameter in sq. ins.;

L = length of stroke in ins.;

N = number of delivery strokes per minute;

f = factor representing assumed slip in per cent. of displacement;

1,728 = cu. ins. in one cu. ft.;

231 = cu. ins. in one U. S. gallon.

Example.—What is the approximate net capacity of a 3 × 5 double acting power pump running at 75 revolutions per minute with an assumed slip of 5 per cent., applying this formula?

$$\text{Approximate net capacity} = \frac{.7854 \times 3^2 \times 5 \times 150}{1{,}728} \times (1 - .05) = 2.92 \text{ cu. ft.}$$

$$= \frac{.7854 \times 3^2 \times 5 \times 150}{231} \times (1 - .05) = 22.8 \text{ gals.}$$

Water Valves.—It has been proven by practice after many years of experiments that a number of small valves are more satisfactory than a few large valves. In a well designed valve the valves seat opening area will equal the valve *port opening* area without, the necessity of too much valve lift to give equal area.

The author's practice in designing valves for City Water Works pumps limited the diameter to 4 or 4½ ins.—never larger, no matter how large the pump. The seat opening is made such that the seat opening and port opening will be equalized at ¼ in. valve lift.

Example.—Calculate valve port opening area for a simplex double acting pump having a capacity of 231 gallons per minute neglecting slip when running 100 strokes per minute.

Displacement per minute = 231 × 231* = 53361 cu. ins.

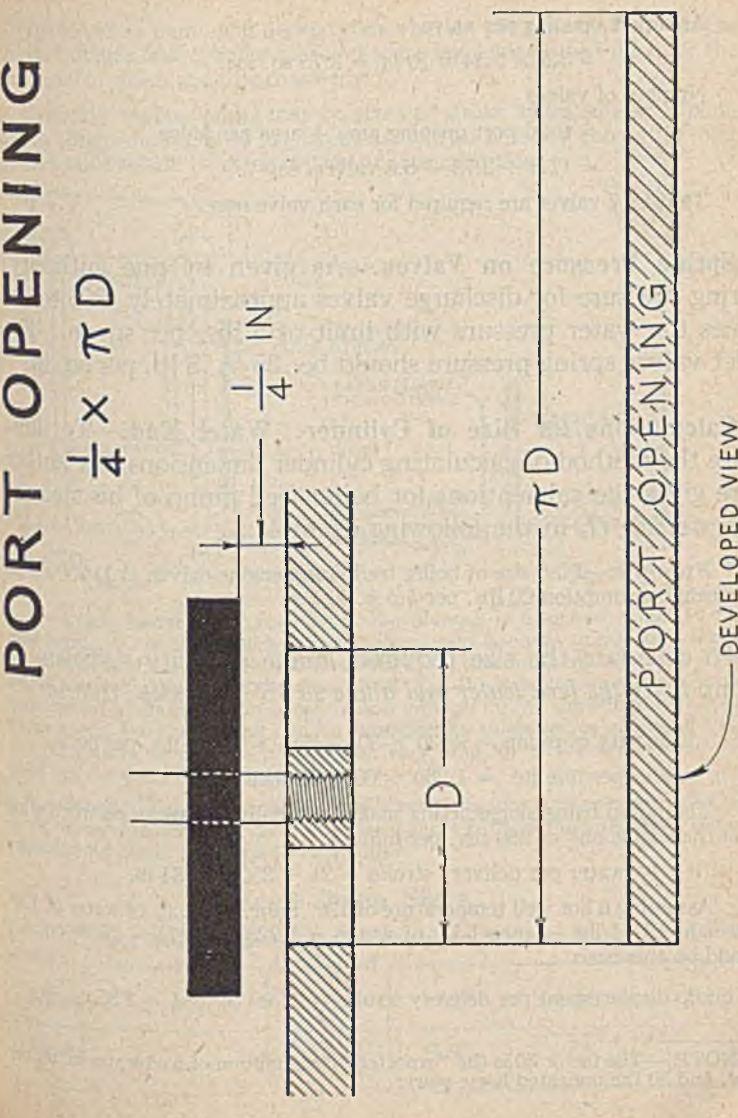
Using 4 in. valves diameter of port opening is 3½ ins. (see fig. 4) and for ¼ valve lift,

$$f\,\text{Area valve port opening} = \frac{53361}{{}^*250 \times 12} = 17.8 \text{ sq. ins.}$$

f NOTE.—The area of valve port opening should equal the area through valve ports.

*NOTE.—The second 231 = cu. ins. in 1 gallon. 250 = 250 ft. flow per minute.

NOTE.—Figs. 4 and 5 show graphically the valve port opening. In fig. 4, the valve port opening is cylindrical in shape, ¼ in. high and having a diameter equal to that of the seat opening.



Figs. 4 and 5.—Diagrams illustrating valve port opening at ¼ in. lift.

Area port opening per valve

$$= 3.5 \times 3.1416 \times \tfrac{1}{4} = 2.75 \text{ sq. ins.}$$

Number of valves

= total port opening area ÷ area per valve

= 17.8 ÷ 2.75 = 6.5 valves, say 7.

That is, 7 valves are required for each valve nest.

Spring Pressure on Valves.—As given by one authority, spring pressure for discharge valves approximately .005 to .01 times the water pressure with limit of 5 lbs. per sq. in. For inlet valves spring pressure should be .25 to .5 lb. per sq. in.

Calculations for Size of Cylinder: Water End.—To illustrate the method of calculating cylinder dimensions, the author here gives the calculations for boiler feed pump of his steamer *Stornoway II,* in the following example:

Example.—Find size of boiler feed pump engine driven (1:1) 250 *r.p.m.* Steam consumption 20 lbs. per *h.p.h.* 30 *h.p.*

To calculate the size required, make capacity sufficient to pump *twice the feed water and allow say* 5% *for slip*, that is

*Capacity of pump = 2 (20 × 30) + 5% = 1,260 lbs. per hour

water per minute = 1,260 ÷ 60 = 21 lbs.

The pump being single acting makes one delivery stroke per revolution of the engine and at 250 rev. per minute,

water per delivery stroke = 21 ÷ 250 = .084 lb.

Assuming a hot well temperature of 120° Fahr., 1 cu. ft. of water @ 120° weighs 61.74 lbs., hence 1 lb. of water = 1,728 ÷ 61.74 = 27.99 cu. in. and on this basis

pump displacement per delivery stroke = 27.99 × .084 = 2.35 cu. in.

*NOTE.—The factor 20 is the "expected" consumption of feed water in lbs. per hour, and 30 the indicated horse power.

The stroke of pump will depend somewhat on the distance between the engine columns and cylinder axis and space available on bed plate for the pump, all of which must be considered.

Evidently various values may be given to stroke and diameter of plunger to obtain the required displacement, and to properly choose the best ratio, a table should be made (by aid of slide rule) thus:

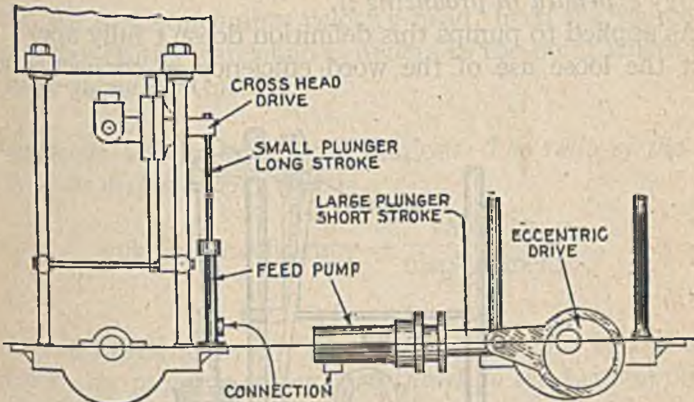


Fig. 6.—**Cross head drive** for pump. The plunger is attached direct to the cross head. This is a very simple arrangement and because of the long stroke the plunger is of very small cross sectional area, thus bringing very little stress on the arm projecting from the cross head. This forms a simple and desirable arrangement when the piston speed is slow enough for the satisfactory working of the pump. The speed limit can be extended by using valves of liberal size, thus the volume of water is passed with very little lift of the valves, and the noise and jar due to seating reduced.

Fig. 7.—**Eccentric drive** for pump. This is a satisfactory and common form of drive, though the eccentric drive introduces more friction than in other forms, necessitating closer attention to lubrication.

Feed Pump Sizes

(for 2.35 cu. in. displacement per delivery stroke)

Diameter.............	¾	⅞	13/16	1
Stroke................	5.35	3.92	4.54	3

A sectional view of the pump as designed for steamer *Stornoway II* is shown on page 520.

Efficiency of Pumps.—A general definition of efficiency is: *The ratio of the useful work performed by a prime mover and the energy expended in producing it.*

As applied to pumps this definition doesn't fully apply. In fact the loose use of the word efficiency in connection with

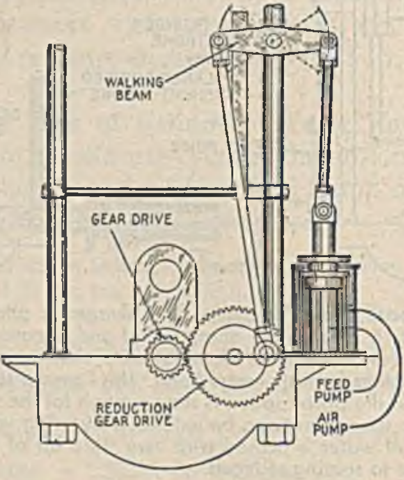


Fig. 8.—Geared drive for pumps. For small high speed engines this arrangement permits a speed reduction to any desired number of strokes per minute for the pumps and is a highly satisfactory method of driving pumps on high speed engines.

pumping machinery is ambiguous, and has led to unnecessary disputes in the interpretation of guarantees and contracts. There are several kinds of pump efficiency:

1. Hydraulic
2. Volumetric
3. Thermal
4. Mechanical
5. Overall

Hydraulic Efficiency.—By definition: *The ratio of the total column* pumped against to the total column pumped against plus the hydraulic losses.*

Ques. What are the hydraulic losses?

Ans. All losses including velocity head, from the source of supply through the water end cylinders to the point where the discharge gauge is attached.

Volumetric Efficiency.—By definition: *The ratio of the capacity to the displacement*, that is

$$\text{volumetric efficiency} = \frac{\text{capacity}}{\text{displacement}}$$

Thermal Efficiency.—By definition: *The ratio of the heat utilized by the pump in doing useful work to the heat supplied.* The formula for thermal efficiency is:

$$E_t = \frac{42.44 \times P \times 60}{S(H-h)}$$

in which

42.44 = heat equivalent of one horse power in *B.t.u.* per minute

P = horse power

S = steam consumed in lbs. per hour

H = total heat in one lb. of steam at initial pressure

h = total heat in one lb. of feed water

*NOTE.—Total **Column** = dynamic head + dynamic lift; very objectionably called total "head." It would require some imagination to consider *lift* as part of the *head* unless one wants to deliberately butcher definitions.

Mechanical Efficiency.—By definition: *The ratio of the indicated horse power of the water end to the indicated horse power of the steam end*, that is,

$$\text{mechanical efficiency} = \frac{\text{I.H.P. water end}}{\text{I.H.P. steam end}}$$

According to one manufacturer the mechanical efficiency of direct acting pumps varies with the size and type from 50% to 90%. This factor can be determined only by actual test.

The following table gives an approximate idea of the mechanical efficiency of direct acting pumps of the piston and outside packed types.

It is recommended that the mechanical efficiency be taken at 80% of the values given in this table:—

Mechanical Efficiency of Pumps

Stroke of Pump Inches	Piston Type Percent	Outside Packed Plunger Type Percent
3	55	50
5	60	56
6	65	61
7	68	64
8	72	68
10	76	72
12	78	75
16	80	77
20	83	80
24	85	82

Horse Power at the Water End.—The horse power required for a given pump capacity is obtained by the following formulae:

$$\text{T.H.P.} = \frac{\text{Cu. ft.} \times W(L_S + H_S)}{33{,}000}$$

$$\text{I.H.P.} = \frac{\text{Cu. ft:} \times W(L_D + H_D)}{33{,}000}$$

in which

T.H.P. = theoretical horse power

I.H.P. = indicated (actual) horse power

W = weight of one cu. ft. of water in lbs.

L_S = static lift in feet

L_D = dynamic lift in feet

H_S = static head in feet

H_D = dynamic head in feet

Example.—What is the theoretical horse power required to raise 100 cu. ft. of water 200 ft., with a 10 ft. lift when the water is at a temperature of 75° Fahr., and when at 35° Fahr?

For a temperature of 75°, one cu. ft. of water weighs 62.28 lbs. Substituting this and the other data in the formula,

$$\text{T.H.P. (at 75° Fahr.)} = \frac{100 \times 62.28 \times (10 + 200)}{33{,}000} = 39.63$$

Now if the water have a temperature of only 35°, as might be in very cold weather, the weight of one cu. ft. will increase to 62.42, and the horse power would accordingly increase in proportion to the ratio of the two weights, or

$$\text{T.H.P. (at 35° Fahr.)} = 39.63 \times \frac{62.42}{62.28} = 39.7$$

By observing the very slight difference in the two results it will be seen that for ordinary calculations, the temperature need not be considered, taking the usual value 62.4 lbs.

How to Calculate Size of Steam Cylinder.—In a direct connected pump the diameter of the steam cylinder is always greater than that of the water cylinder principally on account of the excessively low mechanical efficiency of the pump, that is, loss due to friction.

The size of the steam piston depends upon:

1. Mean effective pressure on piston

2. Hydraulic load

3. Mechanical efficiency

Example.—A simple direct connected pump has a 10 inch water piston. Dynamic hydraulic load is 100 lbs. per sq. in. Mean effective pressure on steam piston 80 lbs. per sq. in. Mechanical efficiency 55% including 10% allowance for possible drop in steam pressure. Find diameter of steam cylinder.

Area water piston $= \frac{1}{4}\pi D^2 = .7854 \times 10 = 78.54$ sq. ins.

Total hydraulic load $= 78.54 \times 100 = 7854$ lbs.

Total load to be overcome by steam piston (mechanical efficiency = 55%)

$$= 7854 \times 1.55 = 12174 \text{ lbs.}$$

$$\text{area steam piston} = \frac{12174}{80} = 152.2 \text{ sq. ins.}$$

$$\text{diameter piston} = \sqrt{\frac{152.2}{.7854}} = 13\tfrac{7}{8} \text{ (approx. from table)}$$

In this case make diameter 14 ins. (area 153.9)

Circumferences and Areas of Circles

Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area	Diam.	Circum.	Area
1/64	0.0491	0.0002	3	9.4248	7.0686	8	25.1327	50.265	16	50.2655	201.06
1/32	0.0982	0.0008	1/16	9.6211	7.3662	1/8	25.5254	51.849	1/8	50.6582	204.22
1/16	0.1964	0.0031	1/8	9.8175	7.6699	1/4	25.9181	53.456	1/4	51.0509	207.39
3/32	0.2945	0.0059	3/16	10.0138	7.9798	3/8	26.3108	55.088	3/8	51.4436	210.60
1/8	0.3927	0.0123	1/4	10.2102	8.2958	1/2	26.7035	56.745	1/2	51.8363	213.82
5/32	0.4909	0.0192	5/16	10.4065	8.6179	5/8	27.0962	58.426	5/8	52.2290	217.08
3/16	0.5890	0.0276	3/8	10.6029	8.9462	3/4	27.4889	60.132	3/4	52.6217	220.35
7/32	0.6872	0.0376	7/16	10.7992	9.2806	7/8	27.8816	61.862	7/8	53.0144	223.65
1/4	0.7854	0.0491	1/2	10.9956	9.6211	9	28.2743	63.617	17	53.4071	226.98
9/32	0.8836	0.0621	9/16	11.1919	9.9678	1/8	28.6670	65.397	1/8	53.7998	230.33
5/16	0.9817	0.0767	5/8	11.3883	10.321	1/4	29.0597	67.201	1/4	54.1925	233.71
11/32	1.0799	0.0928	11/16	11.5846	10.680	3/8	29.4524	69.029	3/8	54.5852	237.10
3/8	1.1781	0.1105	3/4	11.7810	11.045	1/2	29.8451	70.882	1/2	54.9779	240.53
13/32	1.2763	0.1296	13/16	11.9773	11.416	5/8	30.2378	72.760	5/8	55.3706	243.98
7/16	1.3745	0.1503	7/8	12.1737	11.793	3/4	30.6305	74.662	3/4	55.7633	247.45
15/32	1.4726	0.1726	15/16	12.3700	12.177	7/8	31.0232	76.589	7/8	56.1560	250.95
1/2	1.5708	0.1964	4	12.5664	12.566	10	31.4159	78.540	18	56.5487	254.47
17/32	1.6690	0.2217	1/16	12.7627	12.962	1/8	31.8086	80.516	1/8	56.9414	258.02
9/16	1.7672	0.2485	1/8	12.9591	13.364	1/4	32.2013	82.516	1/4	57.3341	261.59
19/32	1.8653	0.2769	3/16	13.1554	13.772	3/8	32.5940	84.541	3/8	57.7268	265.18
5/8	1.9635	0.3068	1/4	13.3518	14.185	1/2	32.9867	86.590	1/2	58.1195	268.80
21/32	2.0617	0.3382	5/16	13.5481	14.607	5/8	33.3794	88.664	5/8	58.5122	272.45
11/16	2.1598	0.3712	3/8	13.7445	15.033	3/4	33.7721	90.763	3/4	58.9049	276.12
23/32	2.2580	0.4057	7/16	13.9408	15.466	7/8	34.1648	92.886	7/8	59.2976	279.81
3/4	2.3562	0.4418	1/2	14.1372	15.904	11	34.5575	95.033	19	59.6903	283.53
25/32	2.4544	0.4794	9/16	14.3335	16.349	1/8	34.9502	97.205	1/8	60.0830	287.27
13/16	2.5525	0.5185	5/8	14.5299	16.800	1/4	35.3429	99.402	1/4	60.4757	291.04
27/32	2.6507	0.5591	11/16	14.7262	17.257	3/8	35.7356	101.62	3/8	60.8684	294.83
7/8	2.7489	0.6013	3/4	14.9226	17.721	1/2	36.1283	103.87	1/2	61.2611	298.65
29/32	2.8471	0.6450	13/16	15.1189	18.190	5/8	36.5210	106.14	5/8	61.6538	302.49
15/16	2.9452	0.6903	7/8	15.3153	18.665	3/4	36.9137	108.43	3/4	62.0465	306.35
31/32	3.0434	0.7371	15/16	15.5116	19.147	7/8	37.3064	110.75	7/8	62.4392	310.24
1	3.1416	0.7854	5	15.7080	19.635	12	37.6991	113.10	20	62.8319	314.16
1/16	3.3379	0.8866	1/16	15.9043	20.129	1/8	38.0918	115.47	1/8	63.2246	318.10
1/8	3.5343	0.9940	1/8	16.1007	20.629	1/4	38.4845	117.86	1/4	63.6173	322.06
3/16	3.7306	1.1075	3/16	16.2970	21.135	3/8	38.8772	120.28	3/8	64.0100	326.05
1/4	3.9270	1.2272	1/4	16.4934	21.648	1/2	39.2699	122.72	1/2	64.4026	330.06
5/16	4.1233	1.3530	5/16	16.6897	22.166	5/8	39.6626	125.19	5/8	64.7953	334.10
3/8	4.3197	1.4849	3/8	16.8861	22.691	3/4	40.0553	127.68	3/4	65.1880	338.16
7/16	4.5160	1.6230	7/16	17.0824	23.221	7/8	40.4480	130.19	7/8	65.5807	342.25
1/2	4.7124	1.7671	1/2	17.2788	23.758	13	40.8407	132.73	21	65.9734	346.36
9/16	4.9087	1.9175	9/16	17.4751	24.301	1/8	41.2334	135.30	1/8	66.3661	350.50
5/8	5.1051	2.0739	5/8	17.6715	24.850	1/4	41.6261	137.89	1/4	66.7588	354.66
11/16	5.3014	2.2365	11/16	17.8678	25.406	3/8	42.0188	140.50	3/8	67.1515	358.84
3/4	5.4978	2.4053	3/4	18.0642	25.967	1/2	42.4115	143.14	1/2	67.5442	363.05
13/16	5.6941	2.5802	13/16	18.2605	26.535	5/8	42.8042	145.80	5/8	67.9369	367.28
7/8	5.8905	2.7612	7/8	18.4569	27.100	3/4	43.1969	148.49	3/4	68.3296	371.54
15/16	6.0868	2.9483	15/16	18.6532	27.688	7/8	43.5896	151.20	7/8	68.7223	375.83
2	6.2832	3.1416	6	18.8496	28.274	14	43.9823	153.94	22	69.1150	380.13
1/16	6.4795	3.3410	1/8	19.2423	29.465	1/8	44.3750	156.70	1/8	69.5077	384.46
1/8	6.6759	3.5466	1/4	19.6350	30.680	1/4	44.7677	159.48	1/4	69.9004	388.82
3/16	6.8722	3.7583	3/8	20.0277	31.919	3/8	45.1604	162.30	3/8	70.2931	393.20
1/4	7.0686	3.9761	1/2	20.4204	33.183	1/2	45.5531	165.13	1/2	70.6858	397.61
5/16	7.2649	4.2000	5/8	20.8131	34.472	5/8	45.9458	167.99	5/8	71.0785	402.04
3/8	7.4613	4.4301	3/4	21.2058	35.785	3/4	46.3385	170.87	3/4	71.4712	406.49
7/16	7.6576	4.6664	7/8	21.5984	37.122	7/8	46.7312	173.78	7/8	71.8639	410.97
1/2	7.8540	4.9087	7	21.9911	38.485	15	47.1239	176.71	23	72.2566	415.48
9/16	8.0503	5.1572	1/8	22.3838	39.871	1/8	47.5166	179.67	1/8	72.6493	420.00
5/8	8.2467	5.4119	1/4	22.7765	41.282	1/4	47.9093	182.65	1/4	73.0420	424.56
11/16	8.4430	5.6727	3/8	23.1692	42.718	3/8	48.3020	185.66	3/8	73.4347	429.13
3/4	8.6394	5.9396	1/2	23.5619	44.179	1/2	48.6947	188.69	1/2	73.8274	433.74
13/16	8.8357	6.2126	5/8	23.9546	45.664	5/8	49.0874	191.75	5/8	74.2201	438.36
7/8	9.0321	6.4918	3/4	24.3473	47.173	3/4	49.4801	194.83	3/4	74.6128	443.01
15/16	9.2284	6.7771	7/8	24.7400	48.707	7/8	49.8728	197.93	7/8	75.0055	447.69

Table of Theoretical Horse Power Required to Raise Water to Different Heights

Gallons per Minute	5 feet	10 feet	15 feet	20 feet	25 feet	30 feet	35 feet	40 feet	45 feet	50 feet	60 feet	75 feet	90 feet	100 feet	125 feet	150 feet	175 feet	200 feet	250 feet	300 feet	350 feet	400 feet	Gallons per Minute
5	.006	.012	.019	.025	.031	.037	.044	.05	.06	.06	.07	.09	.11	.12	.16	.19	.22	.25	.31	.37	.44	.50	5
10	.012	.025	.037	.050	.062	.075	.087	.10	.11	.12	.15	.19	.22	.25	.31	.37	.44	.50	.62	.75	.87	1.00	10
15	.019	.037	.056	.075	.094	.112	.131	.15	.17	.19	.22	.28	.34	.37	.47	.56	.66	.75	.94	1.12	1.31	1.50	15
20	.025	.050	.075	.100	.125	.150	.175	.20	.22	.25	.30	.37	.45	.50	.62	.75	.87	1.00	1.25	1.50	1.75	2.00	20
25	.031	.062	.093	.125	.156	.187	.219	.25	.28	.31	.37	.47	.56	.62	.78	.94	1.09	1.25	1.56	1.87	2.19	2.50	25
30	.037	.075	.112	.150	.187	.225	.262	.30	.34	.37	.45	.56	.67	.75	.94	1.12	1.31	1.50	1.87	2.25	2.62	3.00	30
35	.043	.087	.131	.175	.219	.262	.306	.35	.39	.44	.52	.66	.79	.87	1.08	1.31	1.53	1.75	2.19	2.62	3.06	3.50	35
40	.050	.100	.150	.200	.250	.300	.350	.40	.45	.50	.60	.75	.90	1.00	1.25	1.50	1.75	2.00	2.50	3.00	3.50	4.00	40
45	.056	.112	.168	.225	.281	.337	.394	.45	.51	.56	.67	.84	1.01	1.12	1.41	1.69	1.97	2.25	2.81	3.37	3.94	4.50	45
50	.062	.125	.187	.250	.312	.375	.437	.50	.56	.62	.75	.94	1.12	1.25	1.56	1.87	2.19	2.50	3.12	3.75	4.37	5.00	50
60	.075	.150	.225	.300	.375	.450	.525	.60	.67	.75	.90	1.12	1.35	1.50	1.87	2.25	2.62	3.00	3.75	4.50	5.25	6.00	60
75	.093	.187	.281	.375	.469	.562	.656	.75	.84	.94	1.12	1.40	1.69	1.87	2.34	2.81	3.28	3.75	4.69	5.62	6.56	7.50	75
90	.112	.225	.337	.450	.562	.675	.787	.90	1.01	1.12	1.35	1.68	2.02	2.25	2.81	3.37	3.94	4.50	5.62	6.75	7.87	9.00	90
100	.125	.250	.375	.500	.625	.750	.875	1.00	1.12	1.25	1.50	1.87	2.25	2.50	3.12	3.75	4.37	5.00	6.25	7.50	8.75	10.00	100
125	.156	.312	.469	.625	.781	.937	1.094	1.25	1.41	1.56	1.87	2.34	2.81	3.12	3.91	4.69	5.47	6.25	7.81	9.37	10.94	12.50	125
150	.187	.375	.562	.750	.937	1.125	1.312	1.50	1.69	1.87	2.25	2.81	3.37	3.75	4.69	5.62	6.56	7.50	9.37	11.25	13.12	15.00	150
175	.219	.437	.656	.875	1.093	1.312	1.531	1.75	1.97	2.19	2.62	3.28	3.94	4.37	5.47	6.56	7.66	8.75	10.94	13.12	15.31	17.50	175
200	.250	.500	.750	1.000	1.250	1.500	1.750	2.00	2.25	2.50	3.00	3.75	4.50	5.00	6.25	7.50	8.75	10.00	12.50	15.00	17.50	20.00	200
250	.312	.625	.937	1.250	1.562	1.875	2.187	2.50	2.81	3.12	3.75	4.69	5.62	6.25	7.81	9.37	10.94	12.50	15.72	18.75	21.87	25.00	250
300	.375	.750	1.125	1.500	1.875	2.250	2.625	3.00	3.37	3.75	4.50	5.62	6.75	7.50	9.37	11.25	13.12	15.00	18.75	22.50	26.25	30.00	300
350	.437	.875	1.312	1.750	2.187	2.625	3.062	3.50	3.94	4.37	5.25	6.56	7.87	8.75	10.94	13.12	15.31	17.50	21.87	26.25	30.62	35.00	350
400	.500	1.000	1.500	2.000	2.500	3.000	3.500	4.00	4.50	5.00	6.00	7.50	9.00	10.00	12.50	15.00	17.50	20.00	25.00	30.00	35.00	40.00	400
500	.625	1.250	1.875	2.500	3.125	3.750	4.375	5.00	5.62	6.25	7.50	9.37	11.25	12.50	15.62	18.75	21.87	25.00	31.25	37.50	43.75	50.00	500

This table gives the actual water horse power. When selecting motors, turbines allowance must be made for pipe friction and loss in the pump, gears, belts, etc. One foot head equals .43 pounds pressure to the square inch.

Duty of Pumps.—The word "duty" is used in engineering to express the efficiency of a steam pumping engine as measured by the work done by a certain quantity of fuel, or steam. Duty then, stands for foot pounds or work done, and means the number of pounds of water lifted one foot, or its equivalent, by 100 pounds of coal, or 1,000 pounds of saturated steam.

Formerly duty was expressed on the coal basis, but this has fallen into disuse owing to the variations in the quality of the latter.

Duty expressed per 1,000 lbs. of steam is equivalent to 100 lbs. of coal when the evaporation is 10 to 1.

This result is readily obtainable with good grades of coal when the boilers are correctly proportioned and in proper working condition.

Duty per 100 lbs. of coal would show the combined efficiency of the pump and boiler; when expressed on the steam basis, the efficiency of the pump alone is obtained.

The latter, therefore, is generally used, as the result sought is to determine how economical the pump is in the use of steam.

Another unit of duty is the *foot pounds of work at the water end per million heat units furnished by the boiler.*

This is the equivalent of 100 pounds of coal where each pound imparts 10,000 heat units to the water in the boiler, or where the evaporation is 10,000 ÷ 965.7 = 10.355 pounds of water from and at 212°, per pound of coal.

The last mentioned unit which was reported in 1891 by a committee of the *A. S. M. E.* (*Trans.* XII, 530), reaffirmed it in 1915 as the standard unit and defined it as follows: *The duty per million heat units is found by dividing the number of foot pounds of work done during the trial by the total number of heat units consumed, and multiplying the quotient by* 1,000,000.

The amount of work is found in the case of reciprocating pumps by multiplying the net area of the plunger in sq. ins., the so called total head in lbs. per sq. in. by the length of the stroke in feet, and the total number of single strokes during the trial; finally allowing for the percentage of leakage of the pump. In cases when the water delivered is determined by weir or other measurement, the work done is found by multiplying the weight of water discharged during the trial by the total head in feet.

CHAPTER 18

Air and Vacuum Chambers

Those who have had experience in designing pumps found out early in their careers that *water is an* **unyielding** *substance* making it necessary that pumps, especially fast running reciprocating types, be of very substantial construction to withstand hydraulic shocks and that cushioning chambers should be provided to soften these shocks. These shocks are generally called "*water hammer*." Usually an air chamber is provided and sometimes a vacuum chamber in addition.

1. Air Chambers

Ques. Where is an air chamber placed on a pump?

Ans. On the discharge side of the discharge valve.

Ques. What is the usual shape of an air chamber?

Ans. It is shaped like a cone.

Ques. How is it connected?

Ans. Vertically with the neck or small end (which has the opening) down.

Ques. Why is it connected in this way?

Ans. To reduce the surface area of the water in contact with the air.

Ques. Why should the contact surface of the water be reduced?

Ans. To obtain the minimum amount of absorption of the air by the water.

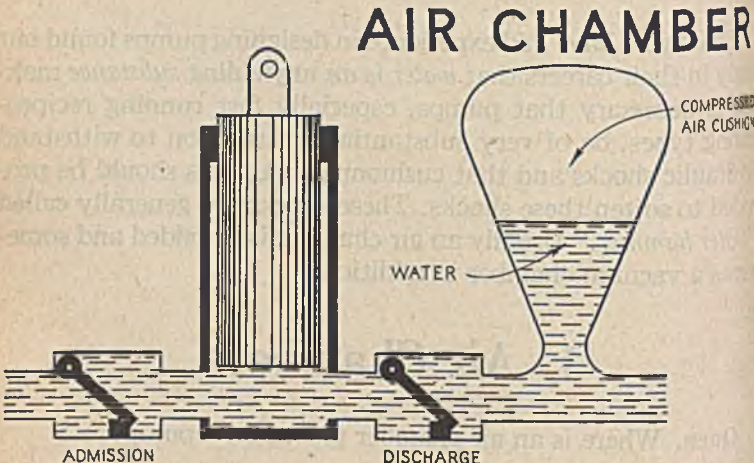


Fig. 1.—Elementary pump showing placement of the air chamber.

Ques. What is the result of this air absorption?

Ans. The air chamber gradually fills up with water and is rendered ineffective in cushioning shocks.

Ques. How does an air chamber work?

Ans. Fig. 1 shows an elementary pump with air chamber.

The water being under pressure in the discharge chamber compresses the air in the air chamber during each discharge stroke of the water piston.

Ques. What happens at the end of the stroke?

Ans. When the piston momentarily stops at the end of the stroke and during the return stroke, the air expands to a certain extent and tends to produce a gradual stopping of the flow of water, thus permitting the valves to seat easily and without shock.

Ques. What trouble is experienced with air chambers?

Ans. They are a useless adjunct unless provided with some device to keep them full of air.

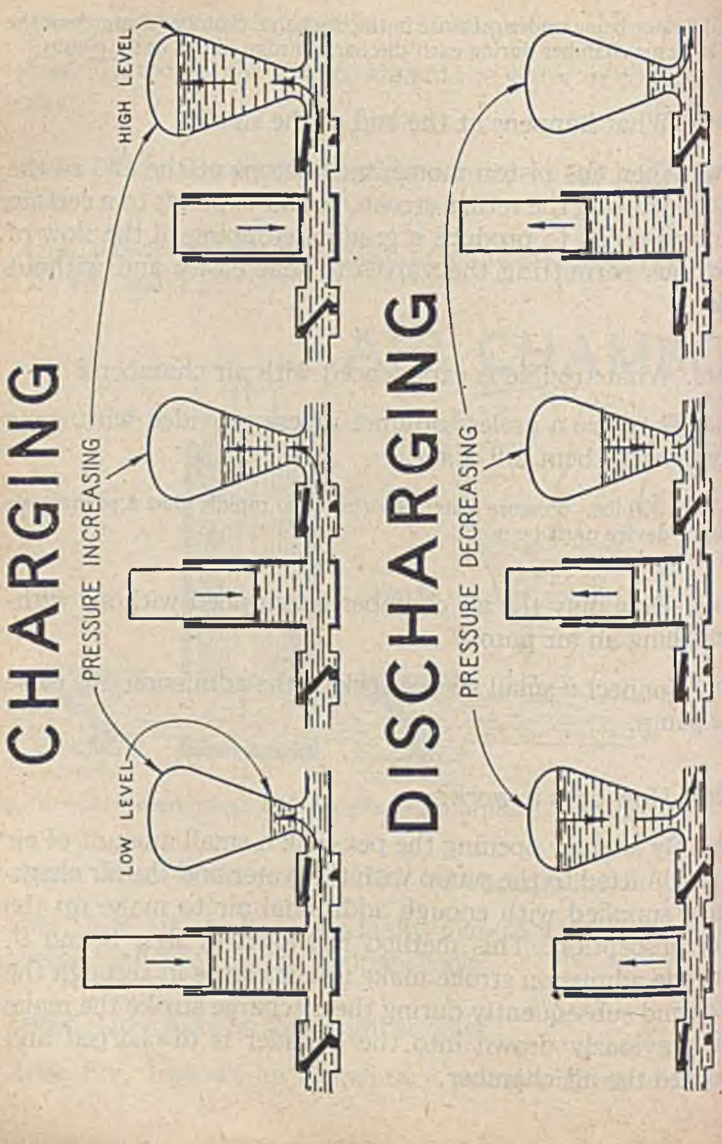
Above 300 lbs. pressure water absorbs air so rapidly that a positive air charging device must be used.

Ques. How may the air chamber be supplied with air without installing an air pump?

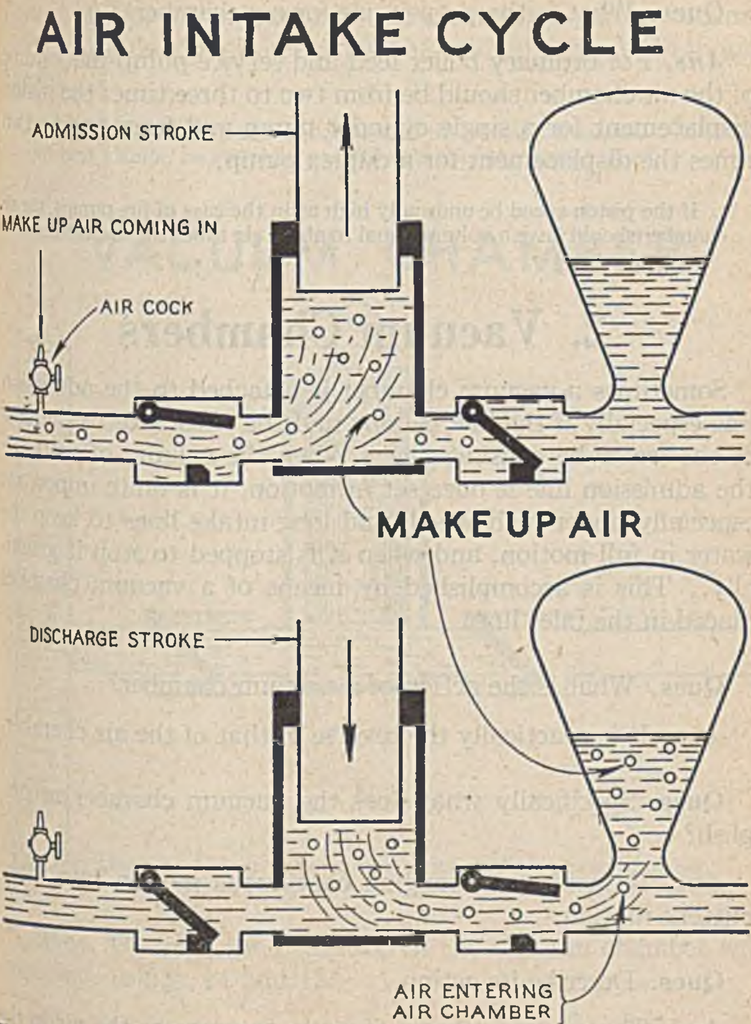
Ans. Connect a small air pet cock to the admission line close to the pump.

Ques. How does it work?

Ans. By slightly opening the pet cock a small amount of air may be admitted to the pump with the water and the air chamber kept supplied with enough additional air to make up the loss by absorption. This method is shown in figs. 8 and 9. During the admission stroke make up air comes in through the air cock and subsequently during the discharge stroke the make up air previously drawn into the cylinder is discharged and passes into the air chamber.



Figs. 2 to 7.—Action of air chamber progressively shown. Figs. 2 to 4, charging during the discharge stroke; figs. 5 to 7, discharging during the intake stroke.



Figs. 8 and 9.—How "make up air" is supplied to the air chamber by means of small air cock connected to the intake pipe.

Ques. What is the proper size for air chambers?

Ans. For ordinary boiler feed and service pump the volume of the air chamber should be from two to three times the piston displacement for a single cylinder pump and from one to two times the displacement for a duplex pump.

If the piston speed be unusually high as in the case of fire pumps, the air chamber should have a volume equal to about six times the displacement.

2. Vacuum Chambers

Sometimes a vacuum chamber is attached to the admission line especially if the line be long and the resistance to the flow of the water be considerable. When the column of water in the admission line is once set in motion, it is quite important especially under high speeds and long intake lines to keep the water in full motion, and when it is stopped to stop it gradually. This is accomplished by means of a vacuum chamber placed in the inlet line.

Ques. What is the action of a vacuum chamber?

Ans. It is practically the reverse of that of the air chamber.

Ques. Specifically what does the vacuum chamber accomplish?

Ans. It facilitates changing continuous motion into intermittent motion.

Ques. Describe its action.

Ans. The moving column of water compresses the air in the vacuum chamber at the end of the pump stroke and when the piston starts, the air expands (thus creating a partial vacuum) which aids the piston in setting the column of water in motion again.

Fig 10 shows the best arrangement of the vacuum chamber. It should, for best results, be virtually a continuation of the intake pipe.

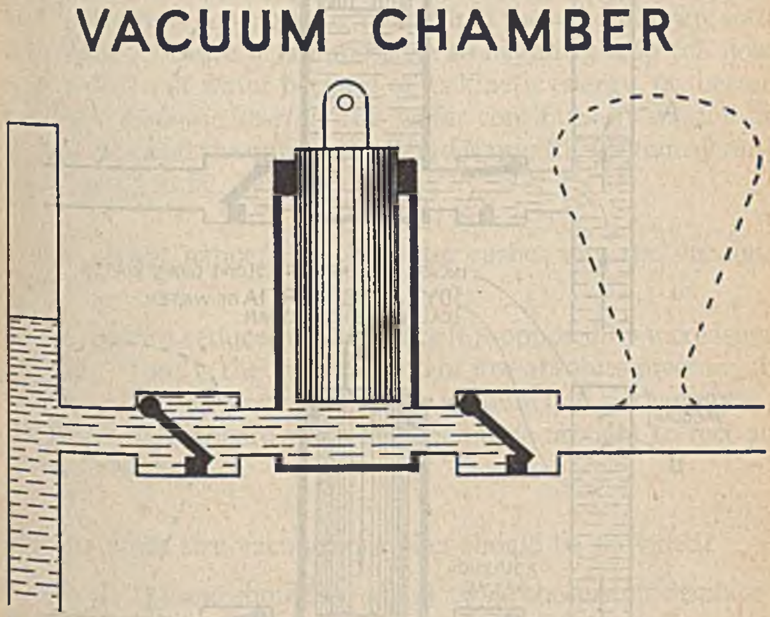
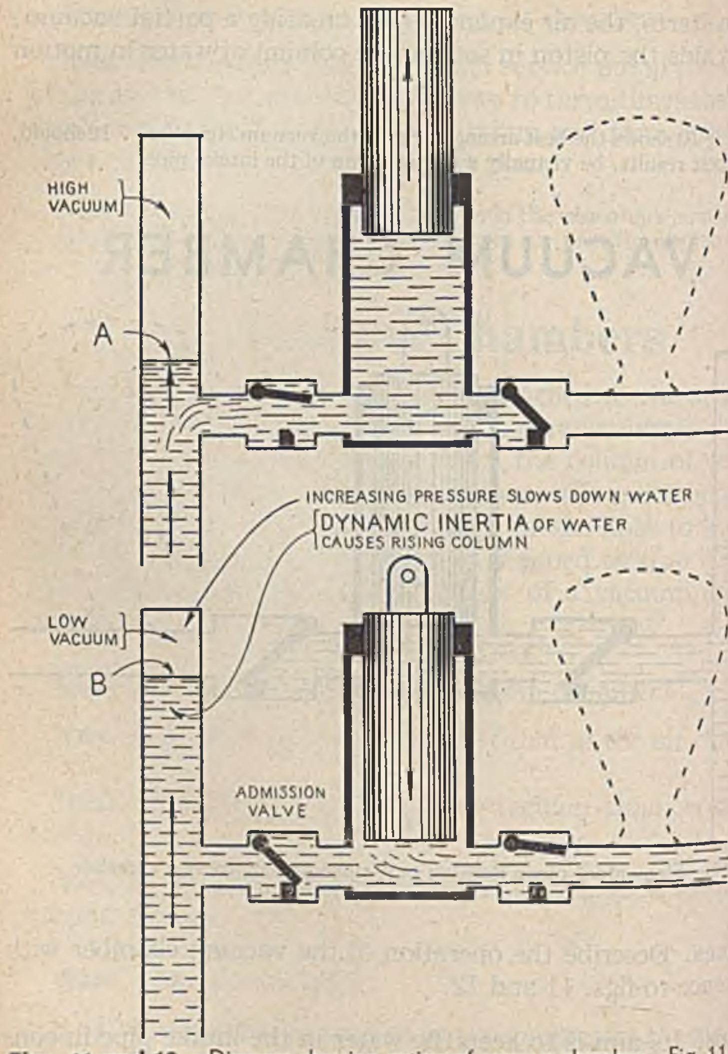


Fig. 10.—Elementary pump showing best placement of vacuum chamber.

Ques. Describe the operation of the vacuum chamber with reference to figs. 11 and 12.

Ans. Its aim is to keep the water in the intake pipe in constant motion, that is, without stopping or reversing.



Figs. 11 and 12.—Diagram showing action of vacuum chamber. Fig. 11, condition at end of admission stroke; fig. 12, condition at end of discharge stroke.

In fig. 11 the piston moving on the up stroke is drawing water into the cylinder. During this operation the vacuum in the vacuum chamber is at its highest and has brought the receding column of water to rest at some point as A.

Ques. What happens on the down or discharge stroke?

Ans. As soon as the plunger starts moving downward the inlet valve slams shut, closing the path of the water flowing into the cylinder. Since it is impossible to instantly stop the flow of the column of water because of its kinetic energy, or, better expressed, *dynamic inertia*, the water continues its motion in the inlet pipe and the only path offered is into the vacuum chamber as shown in fig. 12.

Ques. What happens as the water rushes into the vacuum chamber?

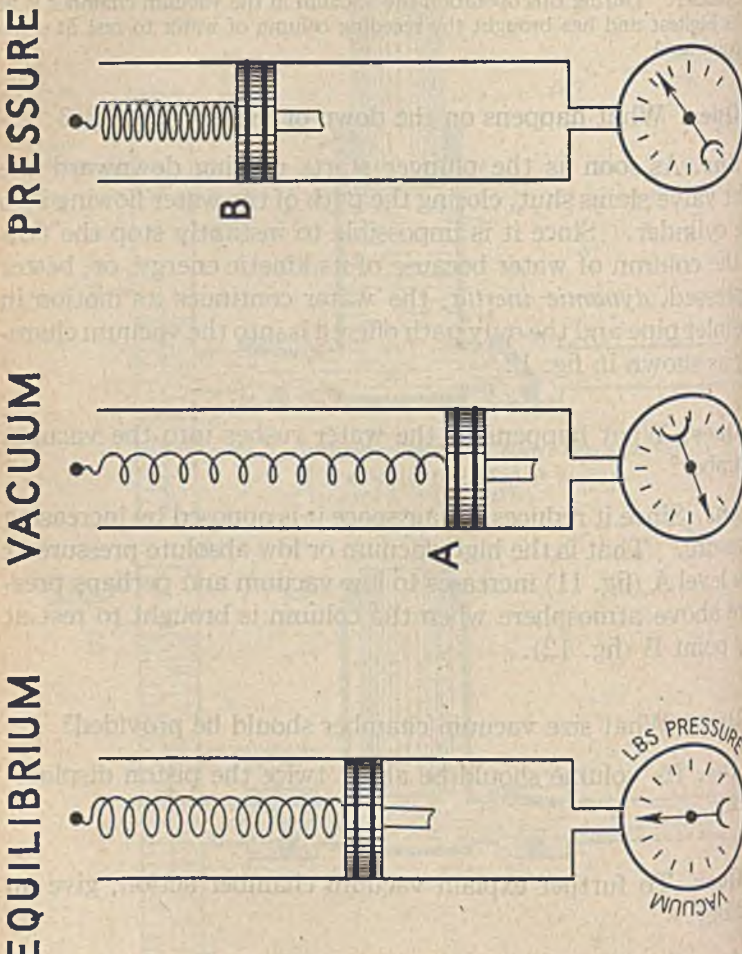
Ans. Since it reduces the air space it is opposed by increasing pressure. That is the high vacuum or low absolute pressure at low level A (fig. 11) increases to low vacuum and perhaps pressure above atmosphere when the column is brought to rest at say point B (fig. 12).

Ques. What size vacuum chamber should be provided?

Ans. Its volume should be about twice the piston displacement.

Ques. To further explain vacuum chamber action, give an analogy.

Ans. Consider a tube to represent the walls of a vacuum chamber. Place a piston inside the tube attached to an anchored spring as in fig. 13.



Figs. 13 to 15.—Spring analogy of vacuum chamber action.

Ques. What condition does fig. 13 represent?

Ans. Since the spring is in its natural position, no up or down force is acting on the piston. This in analogy represents the condition within a vacuum chamber at some intermediate point of the stroke when there is neither vacuum nor pressure within the chamber* as indicated by the gauge.

Ques. In analogy what happens during the up stroke?

Ans. In fig. 14 suppose the piston be pulled down to some position as A. The spring will resist this force with an equal upward force. This corresponds to the conditions in fig. 11 where the inrush of water into the cylinder has caused the water level to drop in the vacuum chamber to same point A, causing a high vacuum which causes atmospheric pressure to act upward bringing the water to rest at A. In fig. 14 this action is represented by the extended spring.

Ques. In analogy what happens during the down or discharge stroke?

Ans. In fig. 15 suppose the piston to be pushed up to some position B. The spring will resist this force with an equal downward force. This corresponds to the condition in fig. 12, where the sudden closing of the inlet valve causes the water to flow into the vacuum chamber by virtue of its dynamic inertia, meeting a constantly increasing opposing pressure as its flow reduces the air space. Position B, of the piston in fig. 15, corresponds to water level B in fig. 12.

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*NOTE.—The term pressure is here used as gauge pressure or pressure above that of the atmosphere as distinguished from vacuum or pressure below that of the atmosphere.

Air Pumps and Jet Condensers

Size Pump	Strokes per Minute	Displacem't Gallons per Minute	Steam Condensed per Hour 26" Vacuum, 30" Barometer				Gallons per Minute Cooling Water Required			
			Temp. Cooling Water				Temp. Cooling Water			
			50°	60°	70°	80°	50°	60°	70°	80°
4½x 5x 8	100	68	840	740	630	500	31	32	34	36
4½x 6x 8	100	98	1200	1070	910	730	43	46	49	53
5 x 6x10	100	122	1500	1330	1130	910	54	58	61	66
5 x 7x10	100	166	2050	1810	1540	1230	73	78	83	89
6½x 8x10	100	217	2700	2380	2000	1610	98	103	108	116
6½x 9x10	100	275	3400	3000	2550	2040	122	130	138	148
6½x10x10	100	340	4200	3700	3160	2530	152	160	170	182
8 x10x12	100	408	5050	4450	3800	3000	180	192	205	217
8 x12x12	100	587	7250	6400	5450	4350	260	275	290	315
8 x12x16	75	587	7250	6400	5450	4350	260	275	290	315
10x14x16	75	800	10000	8750	7400	5950	360	375	400	430
10x16x16	75	1044	13000	11400	9750	7800	470	495	525	560
12x16x20	60	1044	13000	11400	9750	7800	470	495	525	560
12x18x20	60	1322	16400	14400	12300	9800	590	625	670	710
14x20x24	50	1632	20000	17700	15000	12000	720	765	810	870
14x22x24	50	1974	24400	21500	18300	14600	880	930	970	1060
14x24x24	50	2350	29200	25600	21700	17400	1050	1100	1170	1250
16x26x24	50	2758	34000	30000	25500	20300	1230	1300	1380	1470
16x28x24	50	3199	40000	35400	30000	23800	1440	1500	1620	1720
16x30x24	50	3672	45500	40000	34000	27300	1640	1720	1840	1970

CHAPTER 19

Power Pumps

The term power pump, by universal usage, has come to mean: *A reciprocating pump driven by any external prime mover, as distinguished from a direct connected reciprocating pump.*

Briefly *any reciprocating pump having a shaft to which the power is applied externally either by belt, by toothed gears or a combination of the two elements of power transmission.* This does not include, by common usage, the centrifugal pump which usually receives its power from an external source, the same as a power pump.

A power pump differs from a direct connected steam pump in many of its operating characteristics.

These differences should be clearly understood as often a power pump replacing a steam pump does not give complete satisfaction until the user becomes educated to the requirements for satisfactory power pump operation.

Ques. What is the characteristic (in most cases) of a power pump?

Ans. As usually hooked up to the power source, it is a constant *speed machine*, regardless of the fluctuations in load.*

*NOTE.—An exception is the direct connected marine pump whose speed is governed by the marine engine. In such marine installations, there are usually two pumps: 1, boiler feed, and 2, air pump, both of which supply the needed output demanded by the speed of the engine.

Ques. What precaution should be taken in determining the speed at which a power pump should run?

Ans. A power pump, especially when driven from a line shaft or other constant speed source, does not have the capacity flexibility as a steam pump and accordingly more care must be taken to insure that the power pump speed is correct for the conditions to be met.

If the capacity conditions be *variable*, a power pump operated from a constant speed source should not be used.

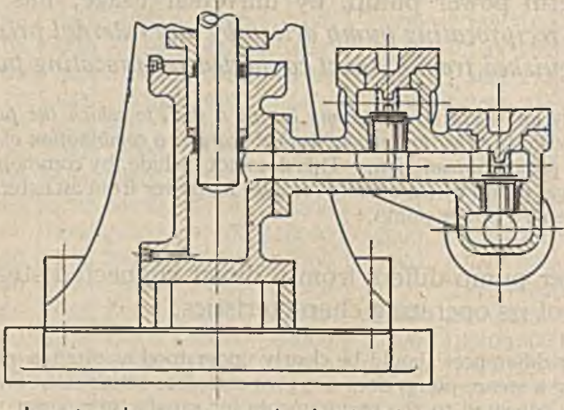


Fig. 1.—Single acting plunger water end with side valves cast separate. The valves are for 500 lb. pressure, drop forged steel wing type.

Ques. Give a comparison with a steam pump for variable operating conditions.

Ans. If the intake pipe be too long or too small, or if a full supply of water be not always available, a steam pump can readily be slowed down until its capacity is equal to the maximum amount of water which can be handled through the intake line. A power pump under similar conditions must run along at full speed, but only partially filling.

Ques. What is the result?

Ans. A pump only partially filling will operate with severe pounding and possible eventual breakage.

Ques. What happens in the case of excess pressure, comparing steam and power pumps?

Ans. Under such conditions, as may happen with a closed or partially closed valve on the discharge line, the steam pump automatically slows down and often stalls if the pressure becomes high enough. Under the same conditions the power pump keeps running at full speed until something breaks.

Ques. What protection should be provided in the case of a power pump?

Ans. A relief valve should be placed on the discharge line.

This functions the same as a safety valve on a boiler.

Power **Pump Classification.**—These pumps may be classified:

1. With respect to the working cycle, as:

   a. Single acting

   b. Double acting.

2. With respect to the number of cylinders, as:

   a. Simplex

   b. Duplex

   c. Triplex

   d. Quadruplex, etc.

3. With respect to the position of the cylinders, as:
   a. Horizontal
   b. Vertical

4. With respect to the pumping element, as:
   a. Piston
   b. Plunger

5. With respect to the stuffing box, as:
   a. Inside packed
   b. Outside packed $\begin{cases}\text{center}\\ \text{end}\end{cases}$

6. With respect to the valve, as:
   a. Single
   b. Multi
   c. Bucket
   d. Pot
   e. Ball

7. With respect to the valve arrangement, as:
   a. Single deck
   b. Double deck
   c. Side deck

8. With respect to pressure as:
   a. Low
   b. Medium
   c. High
   d. Hydraulic (super high)

9. With respect to the velocity reduction of the drive, as:

   a. Single reduction
   b. Double reduction
   c. Multi-reduction

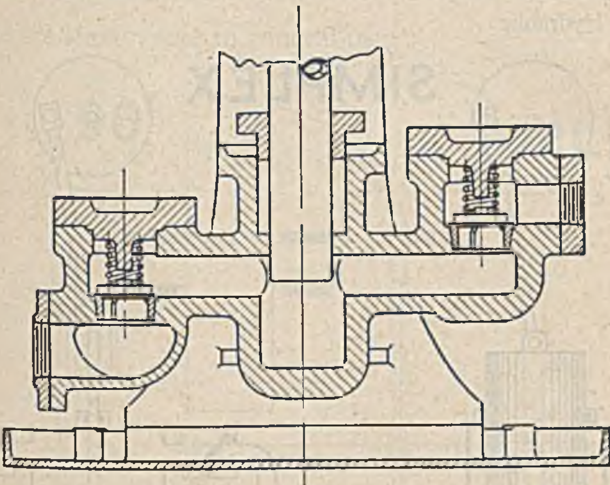


Fig. 2.—Single acting plunger power pump water end with pot valves for 1,000 lbs. working pressure. Each inlet and each discharge valve is contained in a pot, readily accessible by removing a cover plate. Inlet and discharge valve chests are cast integral with the liquid cylinders and the lower half of the frame. The upper half of the frame contains the power end parts, as illustrated. The upper and lower sections are doweled and bolted together. This construction is as strong as a one-piece frame, and on these large size pumps, it permits easy separation for major repairs or transportation. Drop forged steel wing valves.

10. With respect to the type of transmission, as:

    a. Belt
    b. Spur gear
    c. Helical gear
    d. Worm gear
    e. Combined silent chain and tooth gear
    f. Combined belt and tooth gear

11. With respect to the type of prime mover, as:

a. Steam
b. Internal combustion engine
c. Electric motor
d. Hydraulic

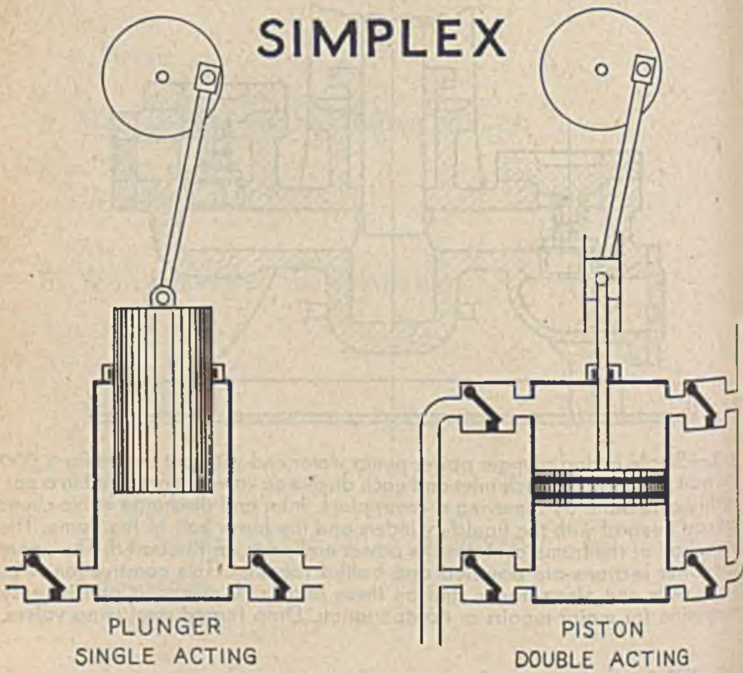


Fig. 3.—Simplex single acting plunger pump.
Fig. 4.—Simplex double acting piston pump.

12. With respect to housing, as:

a. Open
b. Enclosed (self-oiling)

Power Pump Types and Water Flow.—The power pump is inherently a multi-cylinder machine to secure a flow of water approaching uniform delivery instead of spasmodic discharge. The stresses coming on the transmission are not so heavy.

There are three types in general use:

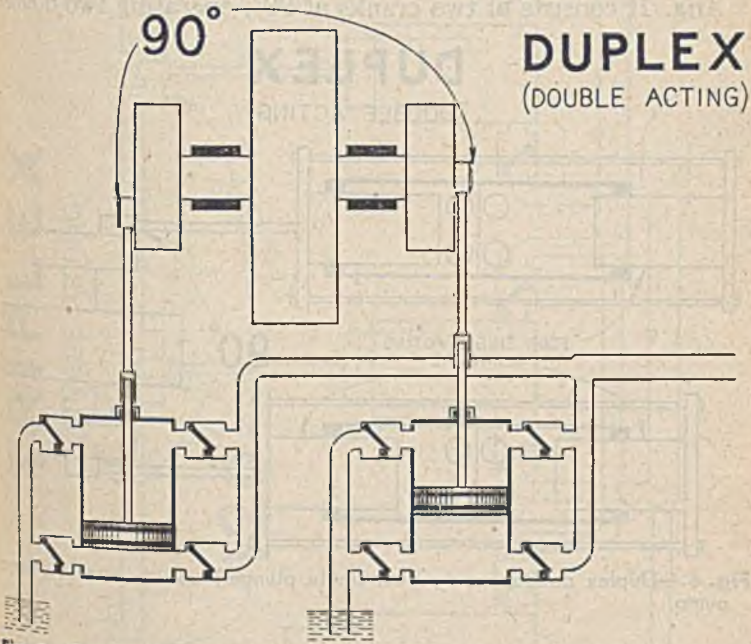


Fig. 5.—Duplex double acting piston pump.

1. Single cylinder or Simplex {single acting, double acting}
2. Duplex {single acting, double acting}
3. Triplex

Ques. Describe the single cylinder power pump.

Ans. It consists of one crank which operates one single acting plunger or a double acting piston, as indicated in figs. 3 and 4.

Ques. Describe a duplex power pump.

Ans. It consists of two cranks at 90°, operating two double

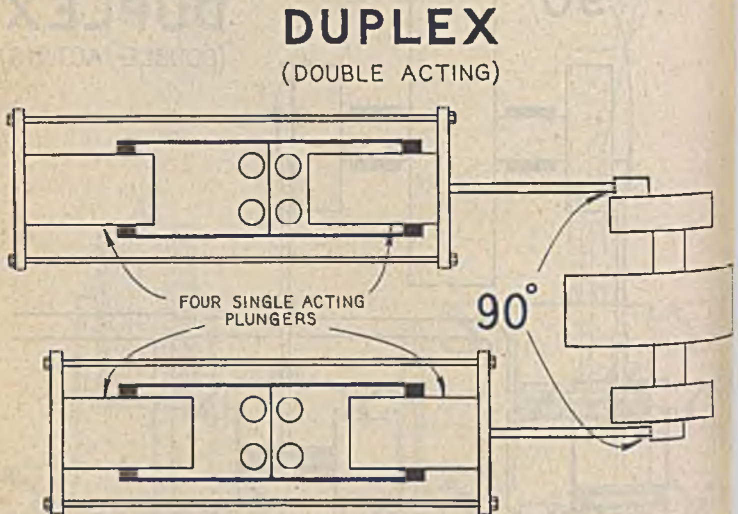


Fig. 6.—Duplex double acting (four single plunger) outside packed plunger pump.

acting pistons or the hydraulic equivalent—4 single acting plungers, as shown in figs. 5 and 6.

Ques. Describe the triplex power pump.

Ans. It consists of three cranks at 120° operating three single acting plungers or sometimes three double acting pistons, as shown in figs. 7 and 8.

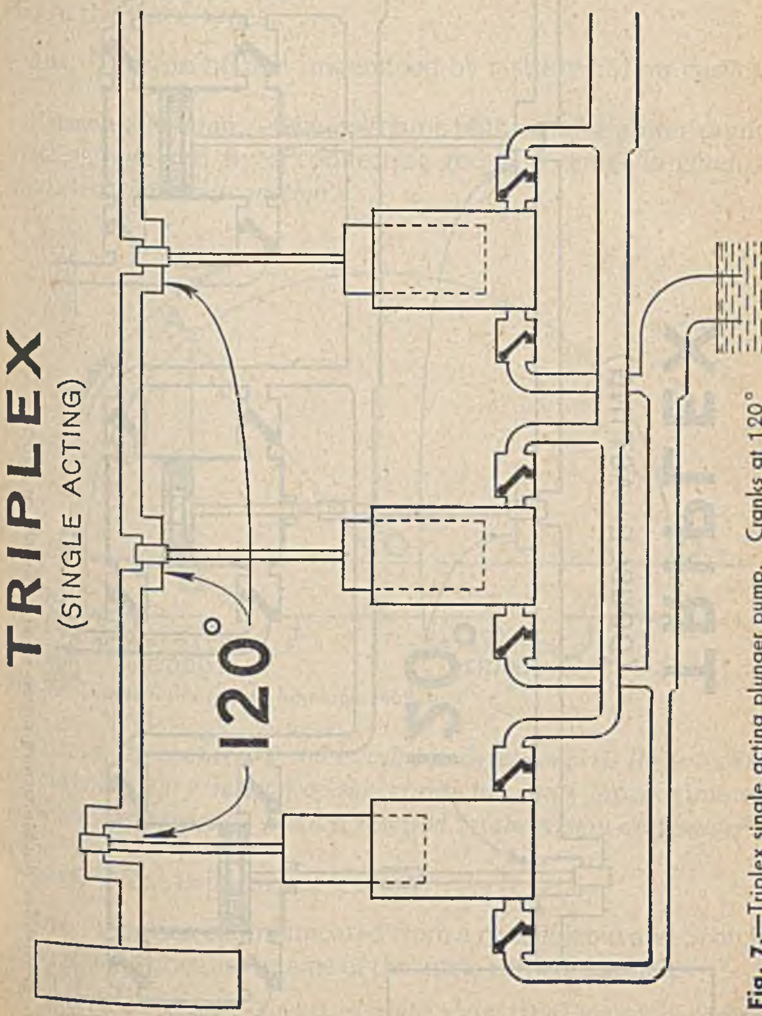


Fig. 7.—Triplex single acting plunger pump. Cranks at 120°

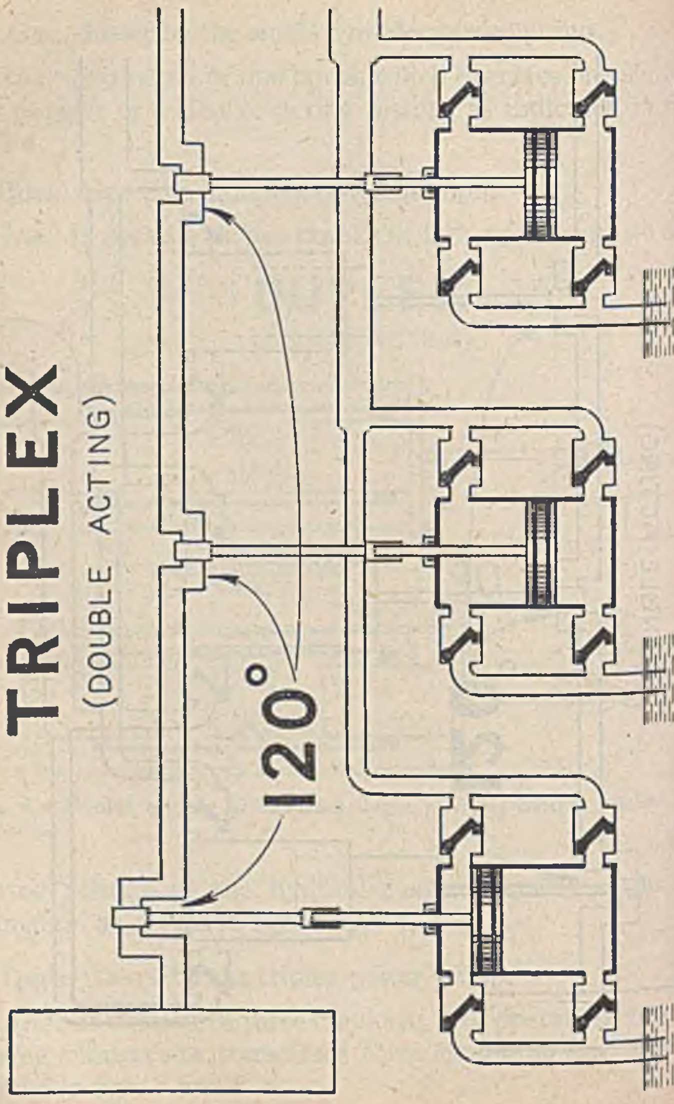


Fig. 8.—Triplex double acting piston pump. Cranks at 120°.

Ques. What are the characteristics with respect to water flow of the three types.

Ans. This can be best understood by a study of flow curves.

Harmonic Motion.—The piston or plunger of a power pump which is operated by a connecting rod and crank is given a (distorted) *harmonic motion.*

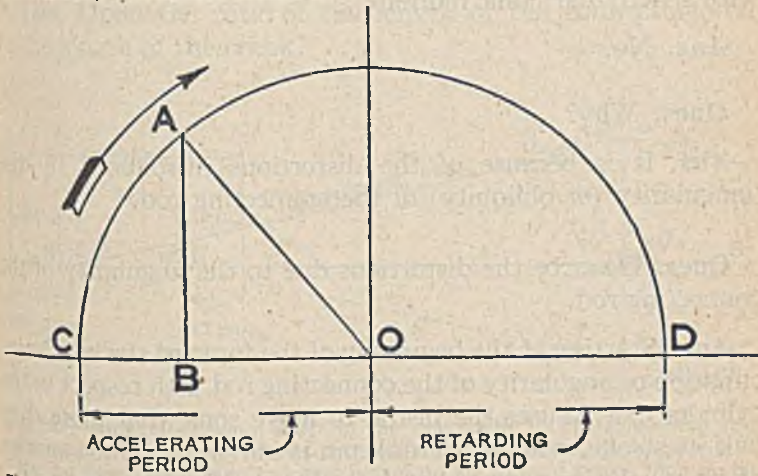


Fig. 9.—Diagram illustrating harmonic motion.

That is, *the connecting rod crank pin drive converts the uniform or constant rotary motion of the crank pin into* (approximate) *harmonic reciprocating motion received by the piston or plunger.*

Ques. What is harmonic motion?

Ans. A motion communicated from a rotating pin and Scotch yoke, which varies as the sine of the angle turned through.

If in fig. 9 a point **A,** as indicated by the arrow, travel around the circumference of a circle with uniform velocity and another point **B,** travel across the diameter at the same time at such variable velocity that it is always at a point where a perpendicular let fall from A, would meet the diameter, the point B, would be said to have *harmonic motion*.

The velocity of B, will increase from zero at the starting point C, till it reaches the center O, and from that point its velocity will decrease to zero when it reaches D, the end of its travel.

Ques. Does the piston or plunger of a power pump travel with strictly harmonic motion?

Ans. No.

Ques. Why?

Ans. It is because of the distortions introduced by the "angularity (or obliquity) of the connecting rod."

Ques. Describe the distortions due to the angularity of the connecting rod.

Ans. Starting at the beginning of the forward stroke, the inclination or angularity of the connecting rod with respect to the cylinder axis, causes the piston to move somewhat more than half its stroke, while the crank pin is moving the first quarter of its revolution, somewhat less than half stroke during the second and third quarters, and again somewhat more than half stroke during the fourth or last quarter of the revolution.

This is illustrated in fig. 10. In the diagram the piston is not shown since its position with respect to the stroke corresponds exactly to that of the wrist pin C. The connecting rod is shown in two positions CR and C'R' such that the crank pin has traveled equal distances AR, and BR' from the dead centers (ends of stroke positions).

The piston positions are indicated by C and C', the piston having traveled on the forward stroke the distance M, and on the return stroke the distance S. For equal crank pin travel from each end of the stroke, it is thus seen that the piston, or plunger, travels further on the forward stroke than on the return stroke.

Were it not for the angularity of the connecting rod, the piston would travel the equal distances M′ and S′. The connecting rod then increases the piston travel by a distance NC, on the forward stroke, and diminishes it by a distance N′C′, on the return stroke.

Ques. On what does the amount of the distortion due to the angularity of the connecting rod depend?

Ans. Upon the ratio of the length of the connecting rod to the length of the crank.

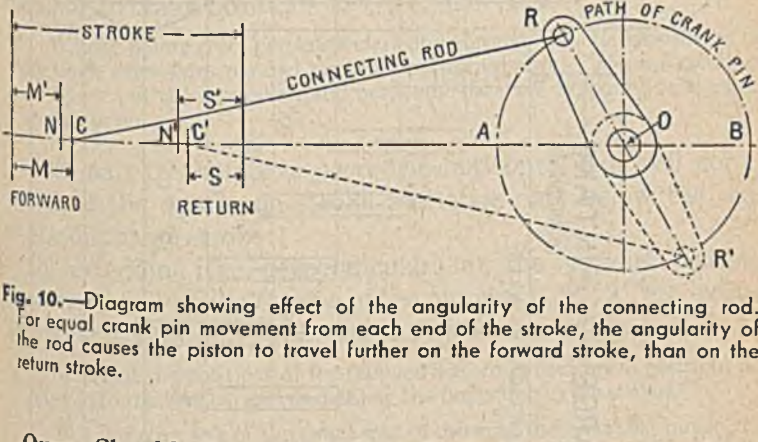


Fig. 10.—Diagram showing effect of the angularity of the connecting rod. For equal crank pin movement from each end of the stroke, the angularity of the rod causes the piston to travel further on the forward stroke, than on the return stroke.

Ques. Should the angularity of the connecting rod be considered in explaining flow characteristics of power pumps, and why?

Ans. No. For simplicity it is best disregarded.

The Harmonic Flow Curves.—These curves represent the variable flow *of power pumps, neglecting the angularity of the connecting rods.*

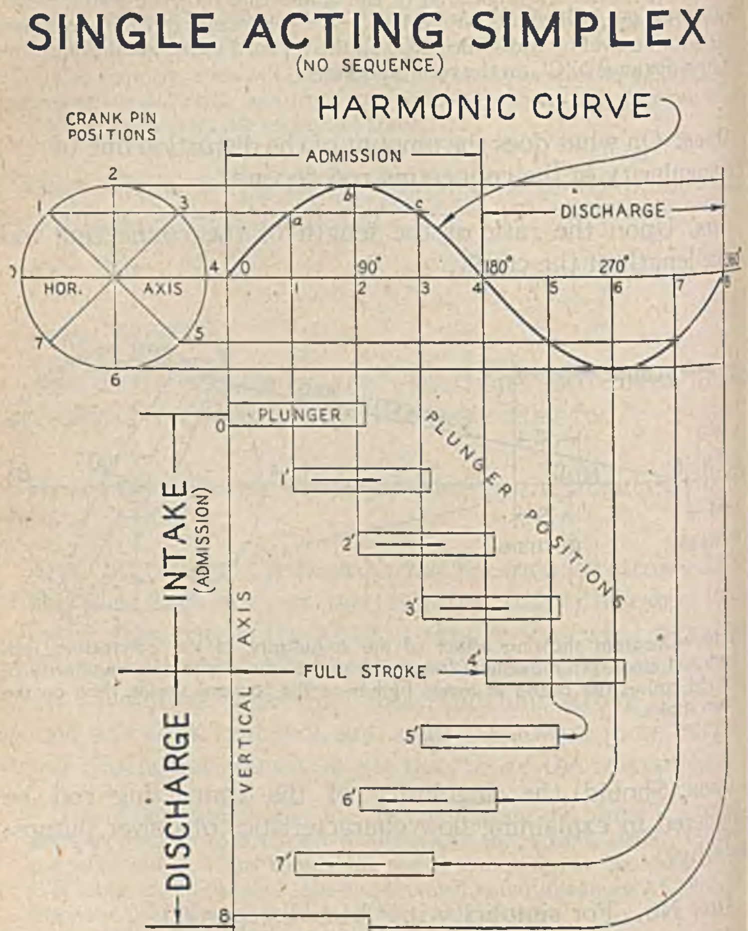


Fig. 11.—Flow curve for simplex single acting pump showing progressive plunge position. Here admission is plotted above the horizontal axis and discharg in order to more clearly show the plunger positions for both strokes, but th is the reverse of the method usually employed.

Curve for Simplex Single Acting Pumps.—To illustrate how they are generated, the construction of a single curve for a single acting one cylinder pump is shown in fig. 11.

First describe a circle whose diameter equals the length of the stroke and lay off any number of crank positions as 1, 2, 3, etc.

Here eight positions are taken for one revolution, so that crank position 1, corresponds to 45°, crank position 2; 90°, etc.

Through the center of circle draw a straight line and rectify the crank positions on this line by points 1, 2, 3, etc., corresponding to crank positions 45°, 90°, 120°, etc.

At these points erect perpendiculars and through the crank positions on the circle draw lines parallel with the horizontal axis. The intersections *a.b.c.*, etc., of these parallels and perpendiculars will be points on the harmonic curve.

That part of the curve *above* the horizontal axis will correspond to the admission stroke and that part below the axis to the discharge stroke.

By extending the perpendiculars at the various stations downward will give positions of the plunger 1′, 2′, 3′, etc. corresponding to positions (1, 2, 3, etc.) of the crank pin.

Note that the positions of the plunger (shown in horizontal position) are referred to the vertical axis as defining the beginning of the stroke.

This is the reverse of the usual way of drawing these curves, but in this case, they are drawn so that plunger positions for different points of the curves are more directly and plainly shown.* The diagram fig. 12 shows the order in which the curves are drawn.

Curves for Simplex Double Acting Pumps.—In this pump two operations are going on at the same time, that is, *admission* is taking place in unison with discharge. However, there is a

NOTE.—It should be understood that the curves may be drawn with admission either above or below the horizontal axis, depending upon which is the more direct with respect to piston or plunger movements.

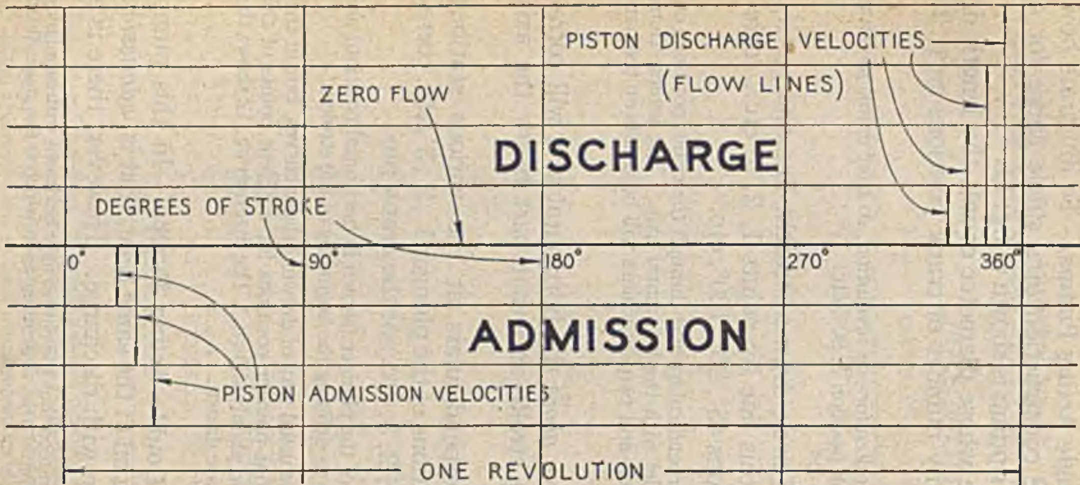


Fig. 12.—Diagram showing conventional placement of discharge and admission periods with respect to the horizontal axis in plotting flow curves.

"phase difference" of 180° between discharge by one side of the piston and by the other side.

In other words, when discharge through one set of discharge valves ends, discharge begins through the other set of discharge valves, that is, the sequence between like operations is 180°.

Accordingly in fig. 13 sine curve OADM will represent the operations taking place on the **A** side of the piston.

Now since the sequence is 180° curve ODAM taken at a phase difference of 180° will represent the operations taking place on the **B**, side of the piston.

The operations taking place at the same time on both sides of the piston are shown graphically for the 90° and 270° positions. Here the piston is in

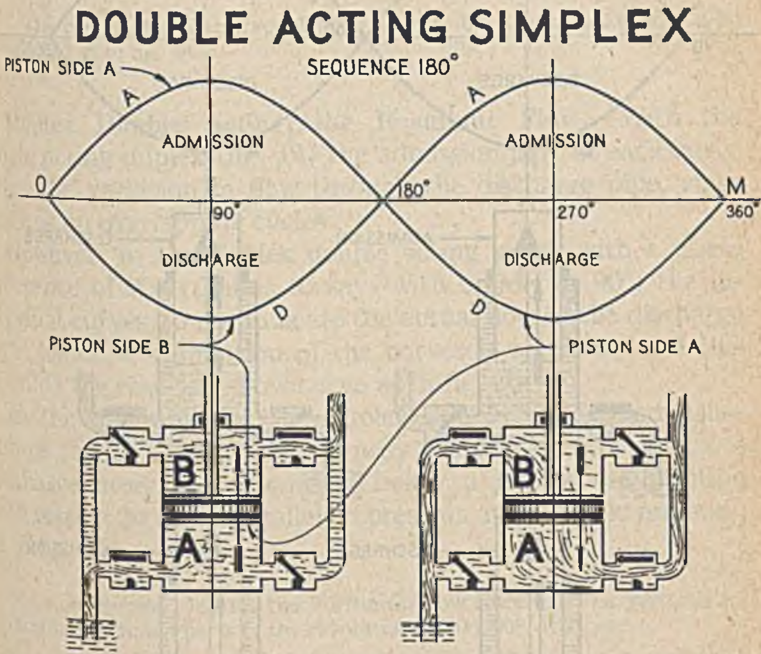


Fig. 13.—Flow curves for simplex double acting piston pump. This type is equivalent to a duplex single acting plunger pump with cranks at 180°.

the same material position, the only difference is that it is moving in opposite directions as indicated by the arrows.

Ques. What other type pump would the flow curves of fig. 13 apply?

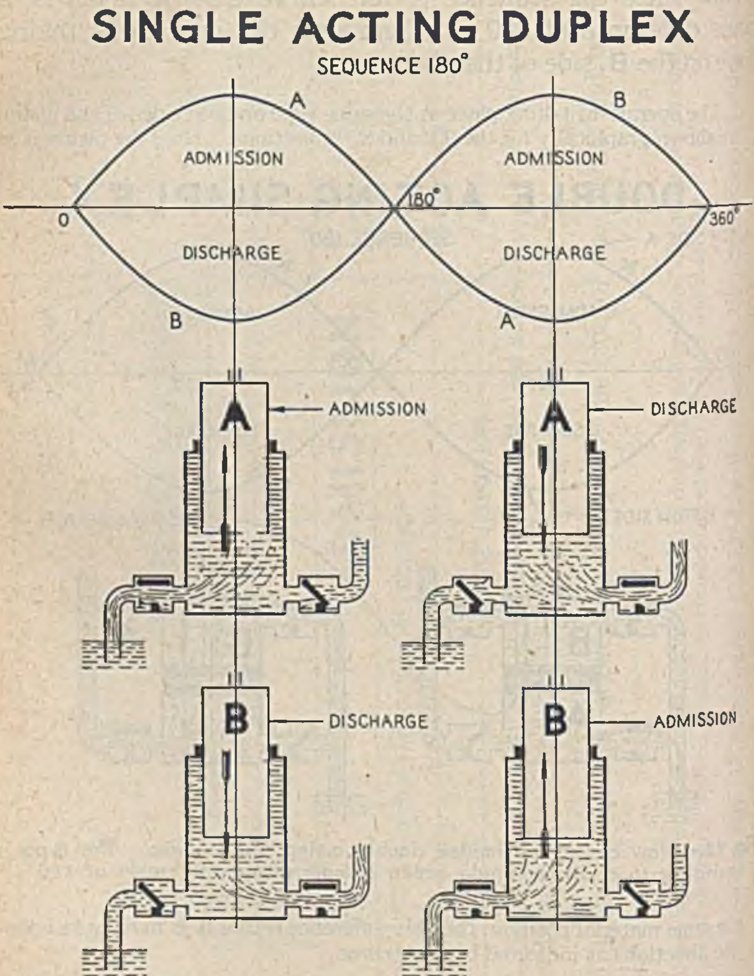


Fig. 14.—Flow curves of single acting plunger with cranks at 180°. This duplex pump is the equivalent of the simplex double acting pump shown in fig. 13.

Ans. To a duplex single acting pump with cranks at 180°.

Ques. Why?

Ans. Because hydraulically it is virtually the equivalent of a double acting simplex pump.

The curves, fig. 13, serve for both pumps, but the notation should be modified as in fig. 14.

Duplex Double Acting; the Resultant Flow.—With the single acting duplex (fig. 14) the admission part of each curve gives the variation in flow through the discharge pipe, since there is no overlapping cycles.

However, in the duplex double acting pump with a phase difference of 90° (that is to say, with cranks at 90°) the individual curves do not indicate the actual flow in the discharge pipe, hence, a summation of the curves is necessary and this is called the resultant flow curve as in fig. 15.

In the diagrams for the duplex double acting and other pumps the curves will be shown by the usual method, that is discharge *above* and *admission* below a line whose position with respect to other parallels represents atmospheric pressure, as in fig. 15.

In construction, lay off the horizontal flow lines and the verticals indicating fractional parts of the revolutions as 0°, 90°, 180°, etc.

At the left describe a semi-circle and draw two 45° radii as shown, which is sufficient to lay off the height of the sign for each 45° over the whole revolution or 360°.

In laying off the verticals it is convenient to let 8 in. represent 360° then each inch along any horizontal = 45°.

In the duplex pump **double** *acting* there will be four curves, one pair for each cylinder; call them curves A and B (full) for

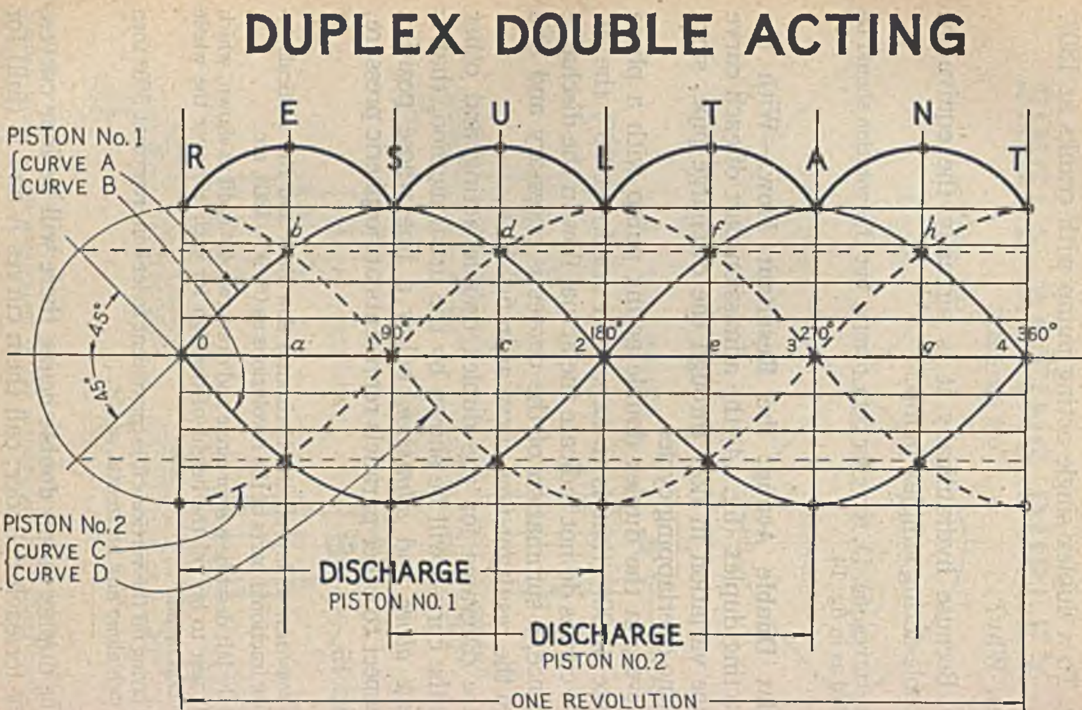


Fig. 15.—Flow curves for duplex double acting piston pump including curve of **resultant** flow, as explained in the text. Sequence of cranks 90°.

cylinder No. 1, and curves C and D (dotted) for cylinder No. 2.

Construct curve A, in the usual way, the first 180° showing discharge and from 180° to 360° admission.

Now for the other side of the double acting No. 1 piston construct curve B, with phase difference of 180°, that is discharge for curve B, will begin 180° later than for curve A.

Now for No. 2 piston, curve C, will begin discharge 90° after curve A begins the discharge period.

For the other side of the double acting No. 2 piston, construct curve D with phase difference 180° later than curve C.

Briefly the beginning of discharge for the four curves is as follows:

Curve	*Beginning of discharge*
A	0°
B	180°
C	90°
D	270°

The resultant flow is obtained by adding the ordinates of any two curves discharging at the same time. Plotting these additions for 90°, 180°, etc. will give points on the resultant *flow* curve. Thus,

Position	*Pistons Acting*	*Resultant*
At 0°	One	OR
45°	Both	2 ab = aE
90°	One	1 S
135°	Both	2 cd = cU
180°	One	2 L
225°	Both	2 ef = eT
270°	One	3 A
315°	Both	2 gh = gN
360°	One	4 T

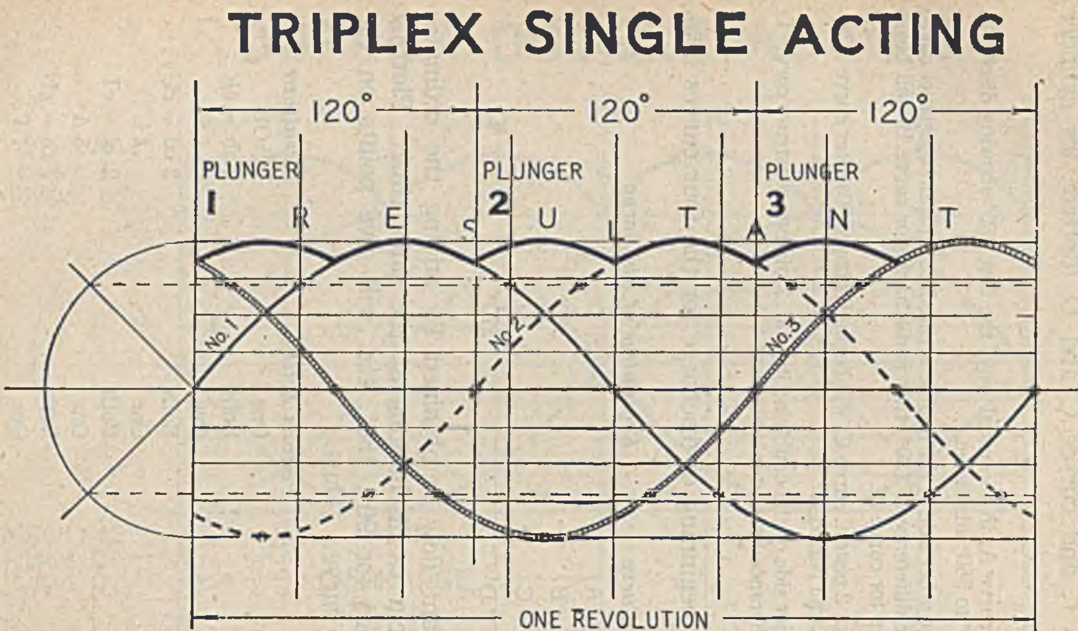


Fig. 16.—Flow curves for triplex single acting plunger pump including curve of **resultant** flow. Sequence of cranks 120°.

Evidently a curve passing through the points, R, E, S, U, L, T, A, N, T will represent the resultant flow or discharge.

Ques. What is the characteristic of this type pump?

Ans. There are four points of minimum flow and four of maximum flow, the velocity increasing from minimum to maximum and back to minimum every 90°.

Triplex Single Acting Pump.—This is a three cylinder plunger pump in which the plungers are operated by cranks at 120° apart. That is, each cylinder begins the discharge 120° or ⅓ of a revolution after the preceding cylinder.

Since the points of maximum flow are not as great and the points of minimum flow not as low as with duplex double acting pumps, the flow is nearer uniform. This is clearly seen by comparing the flow diagrams for the two pumps, figs. 15 and 16. The flow diagram for the triplex single acting as shown in fig. 16 is constructed in a manner similar to the preceding diagrams.

In fig. 16 note that the plunger sequence is 120° or 60° before the preceding plunger finishes its discharge stroke.

Water Ends.—In construction of the water end on the "pump itself" the practice is similar to the direct connected reciprocating type; however, in most instances the pump is vertical. The pumping element may be either:

1. Piston, or
2. Plunger
3. Single acting
4. Double acting.

However, in the multi-cylinder pump, single acting plungers are usually provided.

Various types of valves are used, such as disc, bronze, ring, guided, etc., depending upon the service.

Figs. 17 and 18 show two designs of single valve, single acting plunger pump.

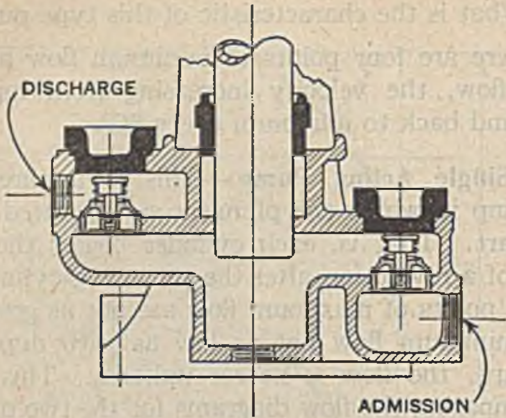


Fig. 17.—Single acting plunger pump with single side valves and chambers cast integral.

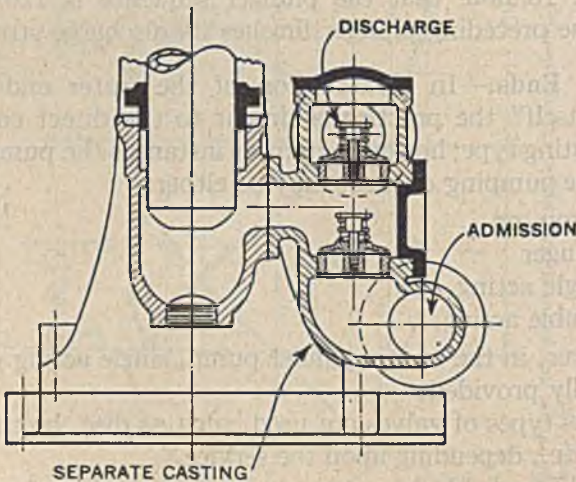


Fig. 18.—Single acting plunger pump unit with double deck valve chamber cast separately.

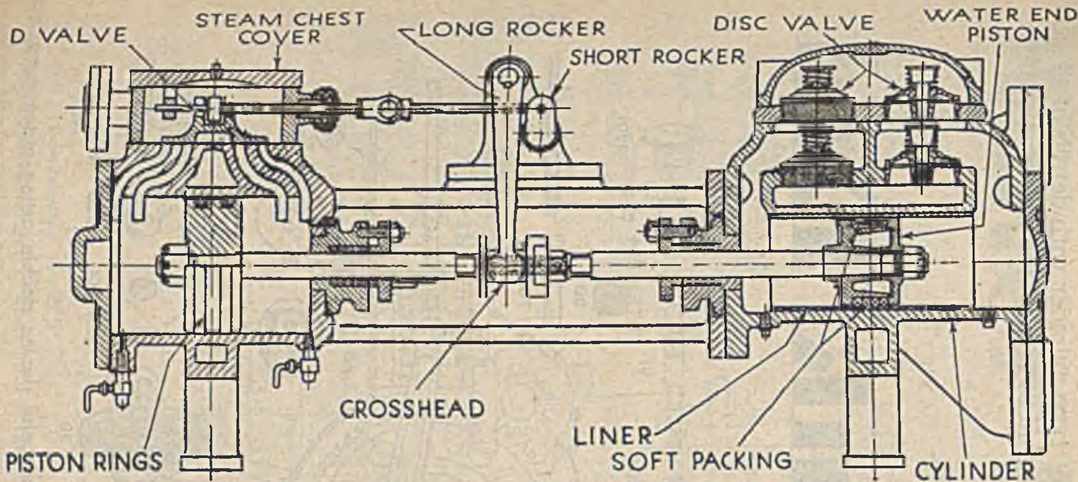


Fig. 19.—Water end showing one cylinder of duplex double acting pump with double deck valve and plate construction.

For very heavy duty, pot valves are used. Valve arrangement may be double deck, single deck or special, depending on the design and purpose of the pump.

Fig. 19 shows a water end with double deck valves above the cylinder, with plate construction.

This pattern is known as the submerged type and, as can be seen, the construction is the same as or the direct connected steam pump. The cylinder shown is one of a pair of cylinders of horizontal duplex pump.

Fig. 22 shows side pot type cylinder with details of all metal wing valve assembly and fibre faced wing valve assembly.

The parts are as follows:

Crank End

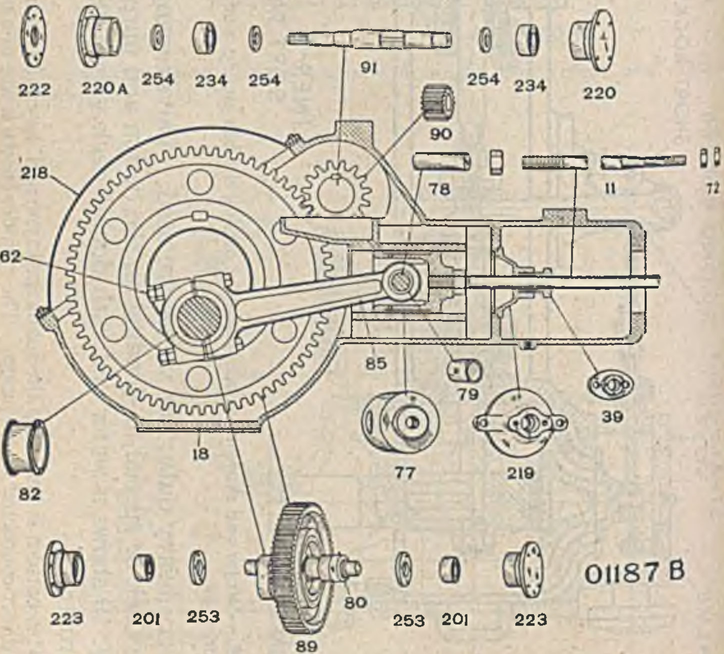
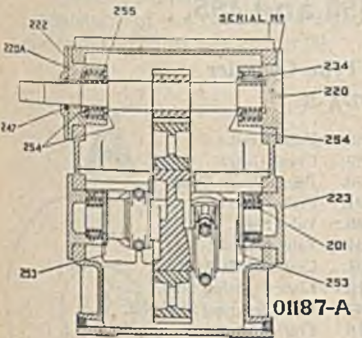
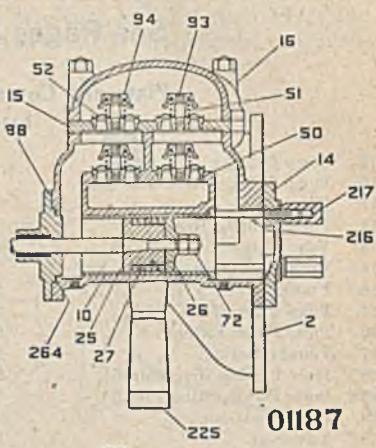
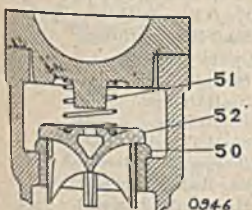


Fig. 20.—Crank end of horizontal duplex double acting piston pump. See page 496 for list of parts.

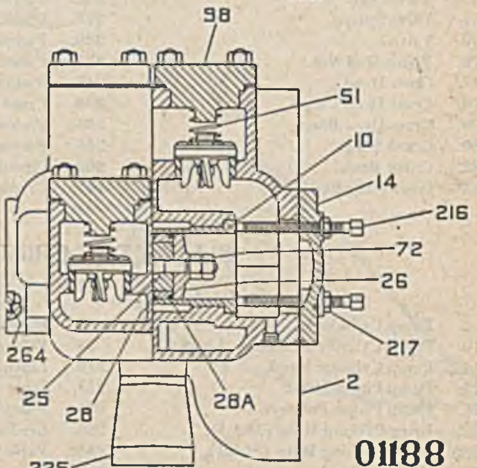
Water End

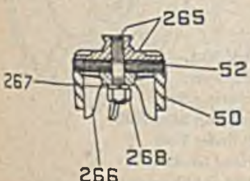






All Metal Wing Valve Assembly





Fibre Faced Wing Valve Assembly

Figs. 21 and 22.—Water end of horizontal duplex double acting piston pump. See page 496 for list of parts.

LIST OF PARTS

Horizontal Duplex Double Acting Piston Pumps

See Pages 494 and 495.

Plate and Cover Type Cylinder

01187-A-B

2. Pump Cylinder.
10. Pump Cylinder Removable Liner.
11. Piston Rod.
14. Pump Cylinder Head.
15. Valve Plate.
16. Valve Chest Cover.
18. Frame.
25. Pump Piston Head.
26. Pump Piston Follower.
27. Fibrous Packing.
28. Inner Packing Ring (Metal).
28A. Outer Packing Ring (Metal).
29. Piston Rod Gland.
50. Valve Seat.
51. Valve Spring.
52. Valve.
72. Piston Rod Nut.
77. Cross Head.
78. Cross Head Pin.
79. Cross Head Brass.
80. Crank Shaft.
82. Crank Brass.
85. Connecting Rod.
88. False Head.
89. Crank Gear.
90. Pump Pinion Gear.
91. Pinion Shaft.
93. Valve Stem.
94. Valve Guard.
201. Crank Shaft Bearing.
216. Liner Securing Stud.
217. Liner Securing Stud Nut.
218. Frame Cover.
219. Piston Rod Oil Flange.
220. Pinion Shaft Bearing Housing.
220A. Pinion Shaft Bearing Housing (Pulley S
222. Pinion Shaft Bearing Housing Cover.
223. Crank Shaft Bearing Housing.
225. Pump Cylinder Foot.
234. Pinion Shaft Bearing.
247. Pinion Shaft Oil Seal.
253. Crank Thrust Ring.
254. Pinion Shaft Thrust Ring.
255. Pinion Bearing Outer Race Lock Ring.
262. Connecting Rod Cap Bolts.
264. Cradle Head Gasket (Copper).

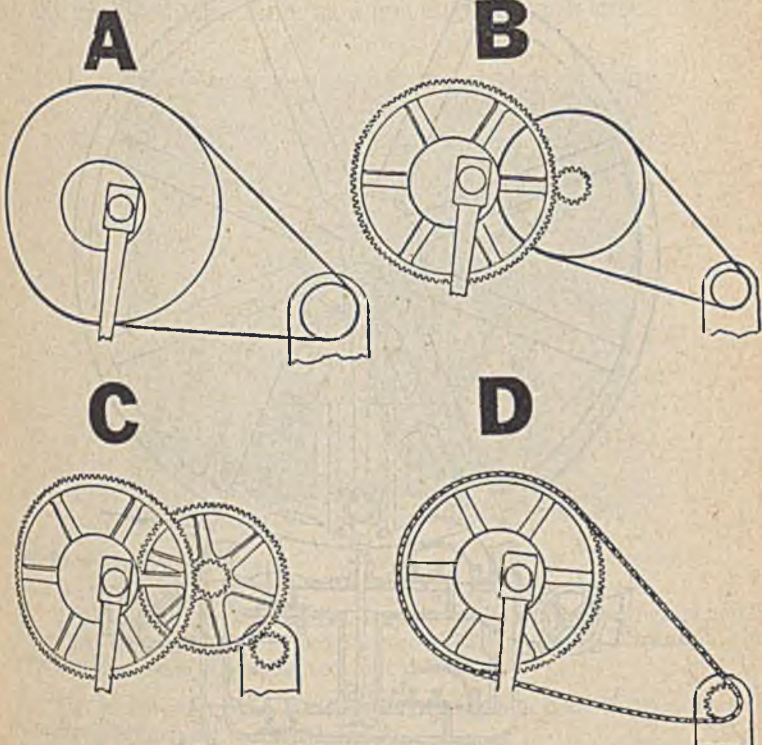
Side Pot Type Cylinder

01188

2. Pump Cylinder.
10. Pump Cylinder Removable Liner.
14. Pump Cylinder Head.
25. Pump Piston Head.
26. Pump Piston Follower.
28. Inner Packing Ring (Metal).
28A. Outer Packing Ring (Metal).
50. Valve Seat.
51. Valve Spring.
52. Valve.
72. Piston Rod Nut.
98. Valve Pot Cover.
216. Liner Securing Stud.
217. Liner Securing Stud Nut.
225. Pump Cylinder Foot.
264. Cradle Head Gasket.
265. Valve Stem and Washer.
266. Valve Wing Guide.
267. Valve Stem Lock Washer.
268. Valve Stem Lock Nut.

The Transmission.—By definition, a transmission is *the assembly of any form of gearing or mechanism by which power is transmitted from the prime mover to the water end of the pump.*

There are numerous types of gear used such as belts, tooth gears and a combination of the two. The essentials of these various gears or transmissions are shown in figs. 23 to 26.



Figs. 23 to 26.—Various forms of drive for power pumps. **A,** belt single reduction; **B,** combined belt and toothed gear double reduction; **C,** toothed gear, double reduction; **D,** chain single reduction.

An example of direct belt drive (indicated in fig. 23) is shown in fig. 27, being one of a line of power pumps designed by the author some years ago for City Water Works.

This design employs a flat belt but the tendency of later practice on smaller units, is to use multi V belts as in fig. 28.

The multi V belt transmission is especially adapted for short center drives

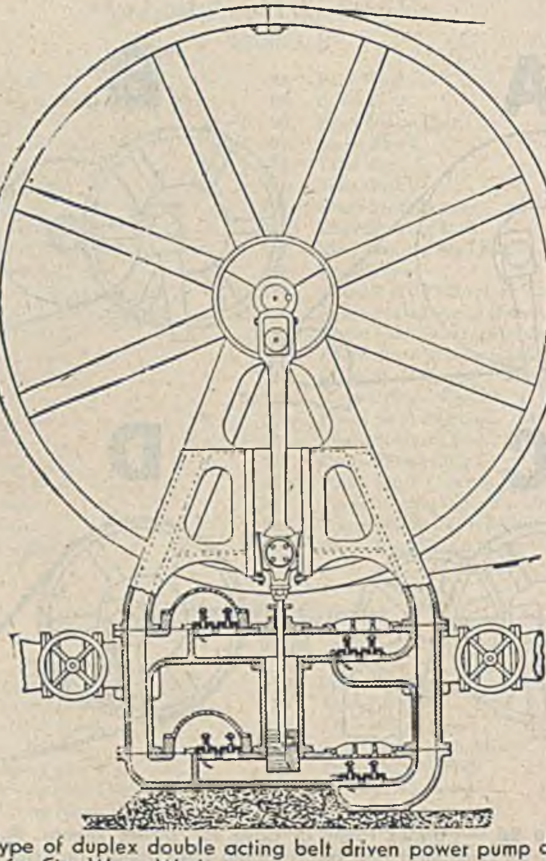


Fig. 27.—Type of duplex double acting belt driven power pump designed by the author for City Water Works.

In place of belts, toothed gears or a combination of belt and gears are frequently used.

Fig. 29 shows a vertical single acting triplex pump with *open* toothed gears operated by pulley and belt, the shaft projecting at the left to receive the pulley. Note that this design is the *open* type.

Another type known as the *closed* type has the power end totally enclosed, the same as a gas engine crank case.

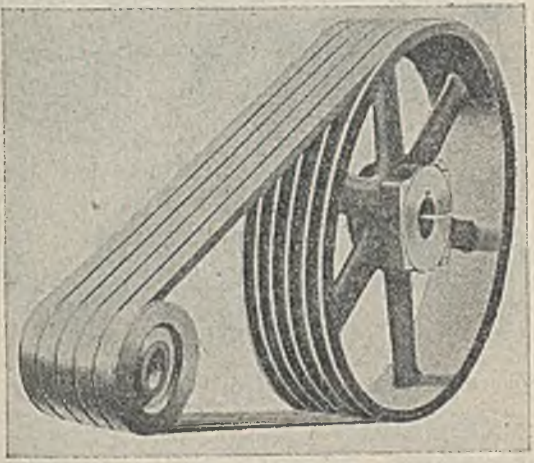


Fig. 28.—Multi V belt drive.

The object of this construction is to make the transmission gears and all bearings dust proof and to provide automatic "oil bath" lubrication, thus adding life to the pump and reducing attention.

Fig. 30 shows the construction of a closed water end. This is the same pump shown in part in figs. 31 and 32.

In the closed type pump since all the working parts are completely enclosed, they are arranged to operate in a flood of oil which insures complete lubrication of all parts.

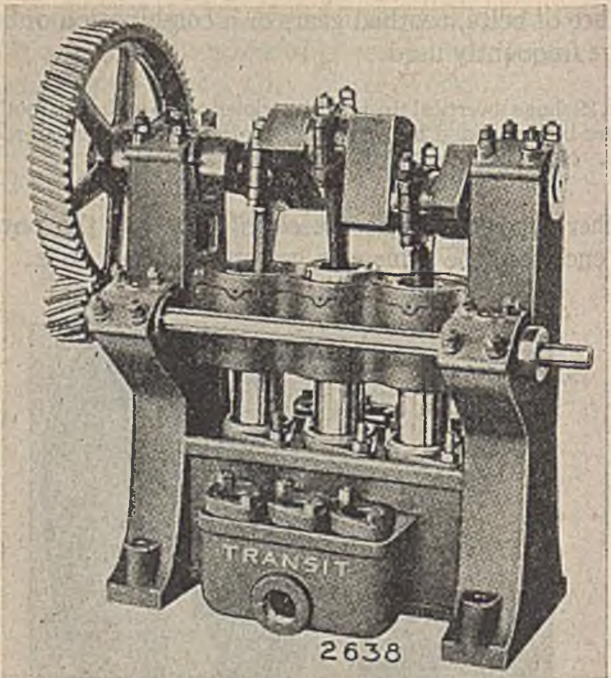


Fig. 29.—Vertical triplex single acting tooth geared pump.

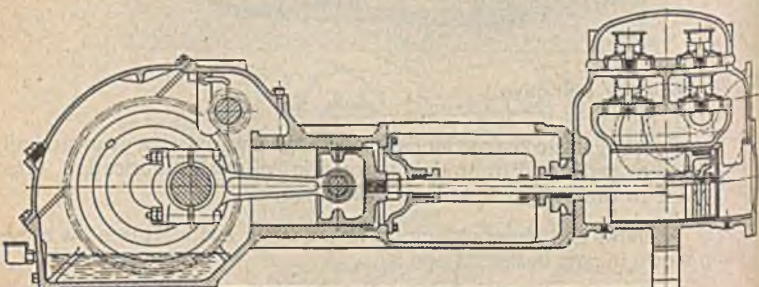


Fig. 30.—Horizontal duplex double acting enclosed type pump. Sectional view showing lubrication system.

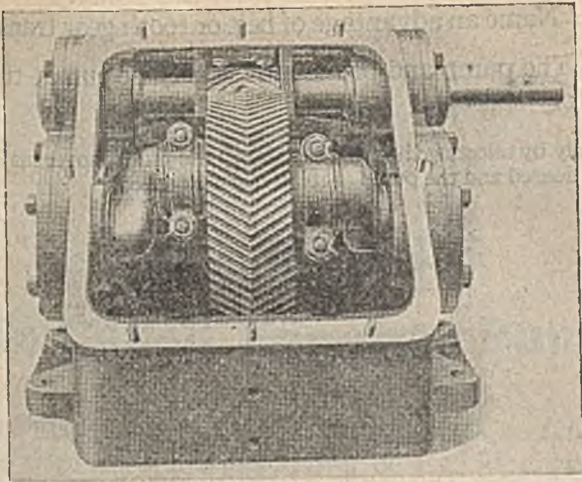


Fig. 31.—End view showing interior of enclosed crank case of horizontal duplex double acting pump.

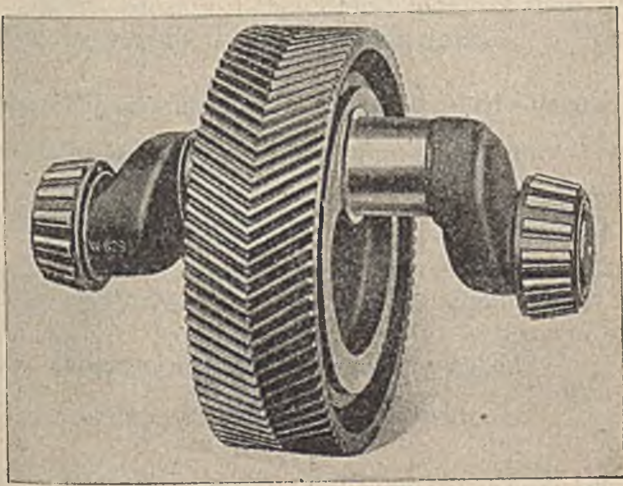


Fig. 32.—Main gear and bearings removed from the crank case of fig. 31.

Ques. Name an advantage of belt or tooth gear transmission

Ans. The pump and prime mover may be run at their most satisfactory speeds.

Simply by selecting the right size pulley the prime mover may be run as fast as desired and the pump at relatively slow speed.

CHAPTER 20

Air Pumps

(So Called *Vacuum* Pumps)

In the first place, there is no such thing as a vacuum pump and the word vacuum should never be used for air pump.

Manufacturers who know perfectly well that the term vacuum pump is incorrect, frequently are forced to use it, even in print, because if they didn't many of their customers wouldn't know what they were talking about.

Ques. Why shouldn't an air pump be called a vacuum pump?

Ans. Because no pump can "pump a vacuum."

Ques. What does the pump do?

Ans. The pump in operation pumps out most of the air (or other gas) from an enclosed space resulting in a partial vacuum.

Ques. What is this vacuum ordinarily called?

Ans. A vacuum regardless of its pressure.

Ques. Why does the air pump not extract all the air and obtain a perfect vacuum?

Ans. That is a condition regarded as practically impossible in nature.

In the receiver of the ordinary air pump the vacuum can only be partial since with each stroke of the piston only a certain fraction of the air is removed (depending upon the relative size of the cylinder and the receiver) and hence, theoretically, an infinite number of strokes would be necessary.

Ques. Amplify the reason for not obtaining a perfect vacuum.

Ans. Practically the degree of exhaustion obtained falls short owing to the imperfections of the machine. Thus, in the common form, the exhaustion is limited to the point where the remaining air has not sufficient elasticity to raise the valves.

Ques. How is the nearest approach to a perfect vacuum obtained?

Ans. By chemical means, using a chemical to absorb the last traces of gas left being exhausted by a mercury air pump.

Ques. Give a general definition for a partial vacuum.

Ans. An enclosed space in which the pressure is less than that of the atmosphere and greater than absolute zero.

Ques. How does the pressure of the atmosphere vary with elevation?

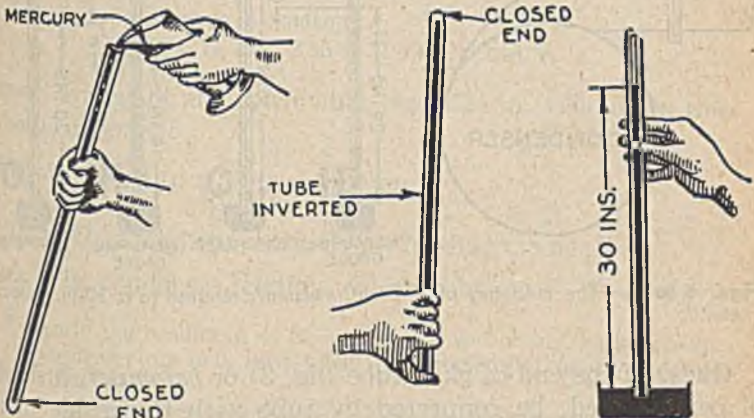
Ans. It decreases approximately ½ lb. for each 1,000 ft. of ascent.

NOTE.—*Torcellian vacuum:* The space above the mercury in a carefully manipulated barometer tube. It is practically perfect, but the space contains a small amount of the vapor of mercury.

At sea level the pressure of the atmosphere is ordinarily 14.7 lbs. per sq. in., measured above absolute zero, that is, the zero pressure of a perfect vacuum. The atmospheric pressure gradually decreases with increasing elevation. For instance, at ¼ mile above sea level it is 14.02 lbs.; at ½ mile, 13.33; at ¾ mile, 12.66; at 1 mile, 12.02; at 1¼ miles, 11.42, at 1½ miles, 10.88, and at 2 miles, 9.8 lbs. per sq. in.

Ques. Does atmospheric pressure vary at any given point and why?

Ans. It varies continually being influenced by weather conditions.



Figs. 1 to 3.—How to make a "mercury" column and to measure the pressure of the atmosphere with the column.

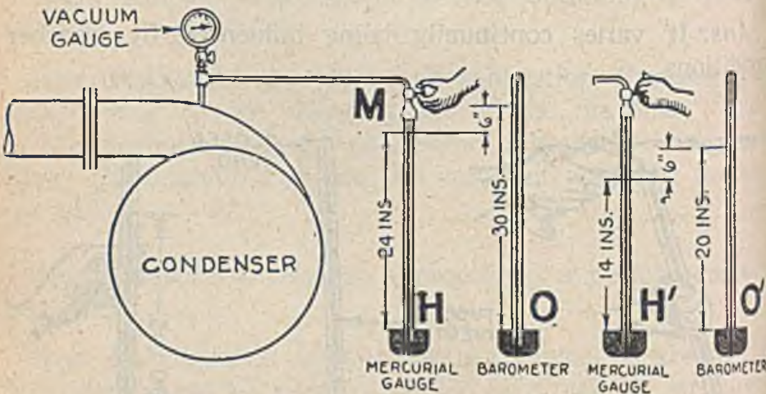
To measure the pressure of the atmosphere take a glass tube about three feet long, closed at one end, and fill it with mercury as in fig. 1. Close the open end with the thumb to prevent premature escape, and invert it as in fig. 2. Now place the open end in a cup of mercury as in fig. 3.

Ques. What happens when the thumb is removed from the open end?

Ans. The mercury inside the glass tube will recede from the closed end of the tube until the column stands approximately 30 ins. above the level of the mercury in the cup, the exact height depending upon the pressure of the atmosphere.

Since a cu. in. of mercury weighs .4916 lb. the pressure of the atmosphere when the mercury stands 30 ins. high is

$$.4916 \times 30 = 14.748 \text{ lbs. per sq. in.}$$



Figs. 4 to 7.—The meaning of a "24-in. vacuum" "referred to a 30-in. barometer."

Ques. If the end of glass tube (fig. 3) or *barometer*, instead of being closed, be connected by tube with the inside of a condenser, what happens?

Ans. The mercury will fall in the tube until its height indicates the difference between the atmospheric pressure and the pressure in the condenser.

Thus, in fig. 5, the barometer O, which registers the pressure of the atmosphere stands at 30 ins. If a similar barometer H, fig. 4, have its upper end connected to a condenser in which is a 24 in. vacuum, on opening the valve M, the mercury will rise in the tube to a height of 24 ins. or 6 ins. less than the height of the barometer O.

Ques. What does this 6 ins. difference between the readings of the barometers represent?

Ans. The absolute pressure in the condenser.

That is,

absolute pressure in condenser = $6 \times .49116 = 2.95$ lbs. per sq. in.

Ques. What does the 24 in. vacuum in the condenser represent?

Ans. The difference between atmospheric pressure and the absolute pressure (2.95 lbs.) in the condenser.

That is, 24 in. vacuum in condenser

$$= 14.75 - 2.95 = 11.8 \text{ lbs. per sq. in.}$$

Ques. In figs. 4 and 5, to what is the 24 in. vacuum in the condenser referred?

Ans. Referred to a 30 in. barometer.

Ques. What happens if the 30 in. reading change?

Ans. The condenser reading will also change.

Evidently the conditions of 24 in. vacuum with 6 ins. back pressure in the condenser can only exist when the barometer O, fig. 5, stands at 30 ins., that is

$$\begin{array}{ccccc} \text{vacuum} & + & \text{condenser pressure} & = & \text{atmospheric pressure} \\ 24 & + & 6 & = & 30 \end{array}$$

Accordingly, the 24 in. vacuum in the condenser is said to be *"referred to a 30 in. barometer"* because with the constant condenser pressure of 6 ins. the column of mercury H (fig. 4) will only remain at 24 ins. as long as the barometer O (fig. 5) reads 30 ins. varying the height with O, being always 6 ins. less than O.

Ques. If the barometer drop to say 20 ins. (fig. 7) as at very high elevation, how would this affect the condenser vacuum?

Ans. The vacuum in the condenser would also drop to 20-6 = 14 ins. as at **H′** fig. 6.

Ques. If the barometer were to fall from say 30 ins. to 20 ins. how would this affect condensing engine operation?

Ans. Considerably less work would be done below atmospheric pressure as indicated by the shaded area **S**, fig. 10, which is much smaller than the area **M**, fig. 9.

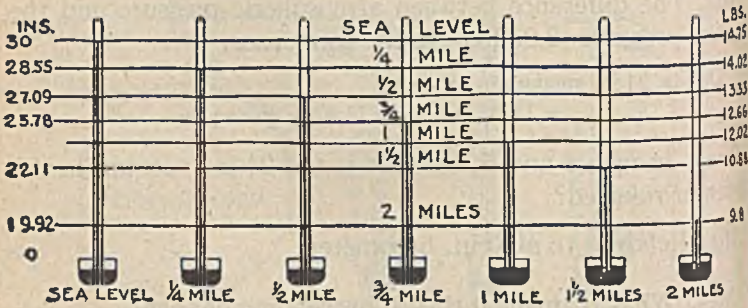


Fig. 8.—Readings of the barometer in ins. of mercury with equivalents in lbs. per sq. in. for different elevations.

Ques. How would similar conditions affect the operation of a direct connected non-condensing steam pump?

Ans. It would act to increase the power of the pump.

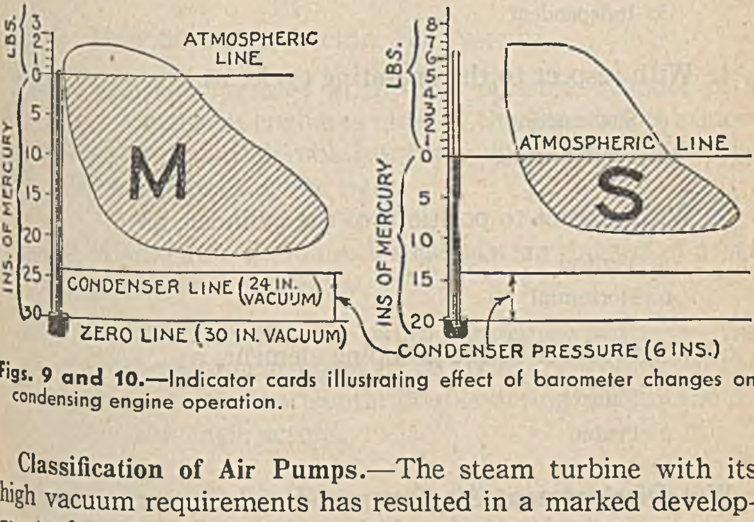
Ques. Why?

Ans. When the valve opens to exhaust on the return stroke there is less back pressure.

That is, the power developed is proportional to the mean effective pressure, which is equal to the mean forward pressure minus the mean back pressure.

Figs. 11 and 12 are two indicator cards representing a pump working non-condensing with barometer at 30 ins. and 20 ins. respectively. The *area* of these cards represent work done. Now for non-condensing operation, that is, pump exhausting into the atmosphere, the back pressure or lower line of the cards will be a little above the atmospheric line.

Accordingly in fig. 12 the atmospheric line being lowered, that is, the pressure of the atmosphere being reduced, the pump exhausts at a lower pressure, hence the area of the card will be increased by the shaded area **R**, which represents additional work that is increase in power.



Figs. 9 and 10.—Indicator cards illustrating effect of barometer changes on condensing engine operation.

Classification of Air Pumps.—The steam turbine with its high vacuum requirements has resulted in a marked development of condenser air pumps, giving rise to a multiplicity of types, some of which are very efficient.

In general air pumps may be classified:

1. With respect to the duty they perform, as

a. Wet { for jet condensers / for surface condensers }

b. Dry { single stage / double stage }

2. With respect to the *principle* of operation, as

    a. Displacement $\left\{\begin{matrix}\text{force}\\\text{impulse}\end{matrix}\right.$
    b. Impulse
    c. Entrainment
    d. Centrifugal
    e. Jet or ejector
    f. Rotary

3. With respect to the *mode* of operation or drive, as

    a. Direct connected
    b. Independent

4. With respect to the operating cycle, as

    a. Single acting
    b. Double acting

5. With respect to positions of the cylinder, as

    a. Vertical
    b. Horizontal

6. With respect to the pumping element, as

    a. Plunger
    b. Piston

Wet Displacement Air Pumps.—By definition, a displacement air pump is *one in which a plunger or piston displaces the atmosphere, thus creating a vacuum.*

In a surface condenser the pump has to handle only the condensate and air, the cooling or "*circulating*" water being pumped by a separate pump called a circulating pump.

As compared with a jet condenser, the pump handles a much smaller quantity of water and a relatively large volume of air. The latter item is important because the air volume to be displaced is much larger than the water volume.

Ques. How much air is mechanically mixed in water?

Ans. About 1/20 or 5% of its volume of air at atmospheric pressure.

Ques. How much air enters a condenser by leakage?

Ans. It may be 3 or more times as much as was liberated by the water.

Ques. How does this air enter the system?

Ans. Since the pressure in the low pressure cylinder of a multi-stage expansion engine is most of the time below atmospheric pressure, considerable air may leak in through the stuffing boxes unless they be tight.

Ques. Give a rule o' thumb for calculating the size of a wet air pump for surface condenser.

Ans. The practice of some pump companies is to give the air pump a displacement equal to 20 times the volume of the condensate, if it be a horizontal double acting pump, and 12 times if vertical single acting.

Ques. Why is the air pump for a surface condenser sometimes made one-half the capacity of one required for a jet condenser?

Ans. In order to enable the surface condenser to be used as a jet condenser in case of emergency.

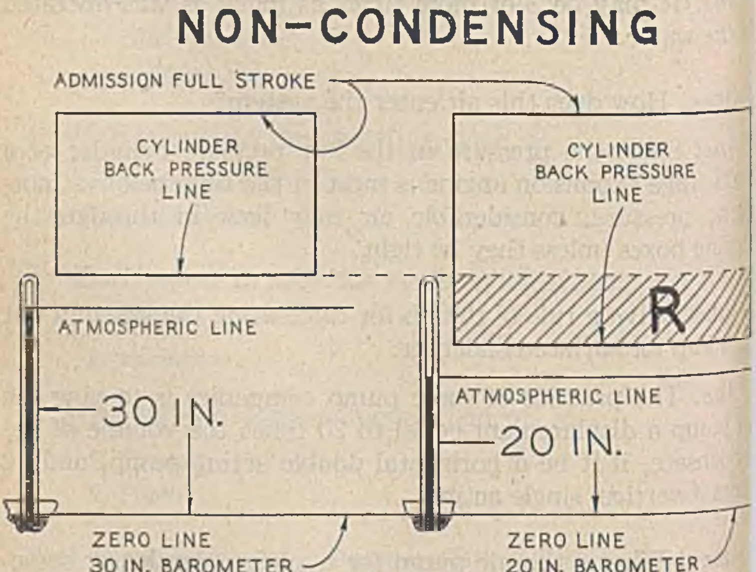
Ques. How should the air pump be located with respect to the condenser?

Ans. It should always be placed below the condenser.

Ques. Why?

Ans. In order that the condenser will drain into the pump, which condition is necessary for obtaining the best vacuum within limit of the pump.

Ques. Is it always possible to locate pump below condenser?



Figs. 11 and 12.—Indicator cards illustrating effect on noncondensing pump operation with barometer at 30 ins. and at 20 ins.

Ans. In the case of a keel condenser the pump cannot be lowered enough for the condenser to entirely drain except in case of some very special construction.

For instance fig. 13 shows result of not having the air pump below enough as installed in the author's steamer *Stornoway I.* The vacuum obtained due to this faulty arrangement was around 18 ins. with resultant loss of power and economy.

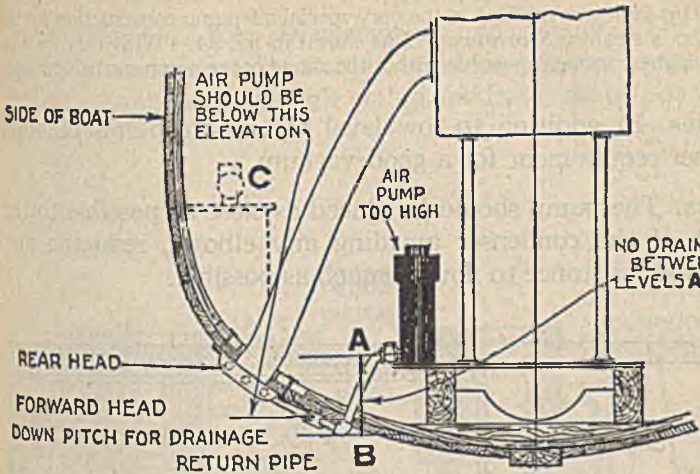


Fig. 13.—Section through author's steamer **Stornoway I** showing faulty installation of pump for keel condenser.

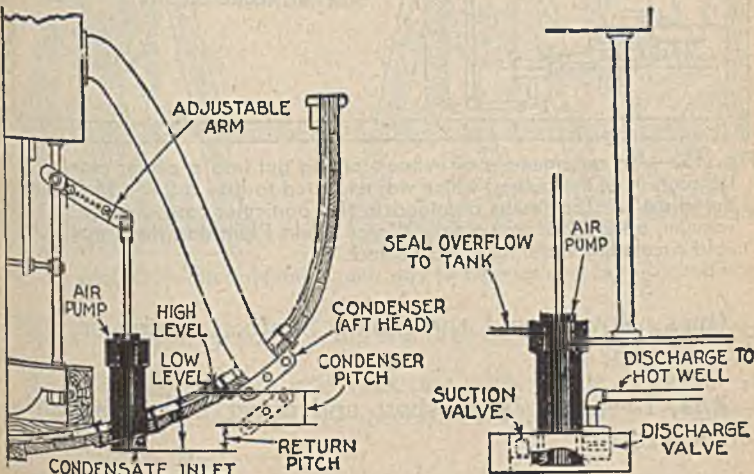


Fig. 14.—Author's method of getting the air pump at a level lower than the condenser so the condenser will drain into the pump. Installation as designed for the author's steamer **Stornoway II.**

This fault was avoided in the very special air pump construction for the author's steamer *Stornoway II* as shown in fig. 14. With this location the highest vacuum possible within the range of the pump can be obtained.

Ques. In addition to low level of the air pump mention another requirement for a good vacuum.

Ans. The pump should be placed as close as possible to the outlet of the condenser avoiding any elbows, reducing any frictional resistance to flow as much as possible.

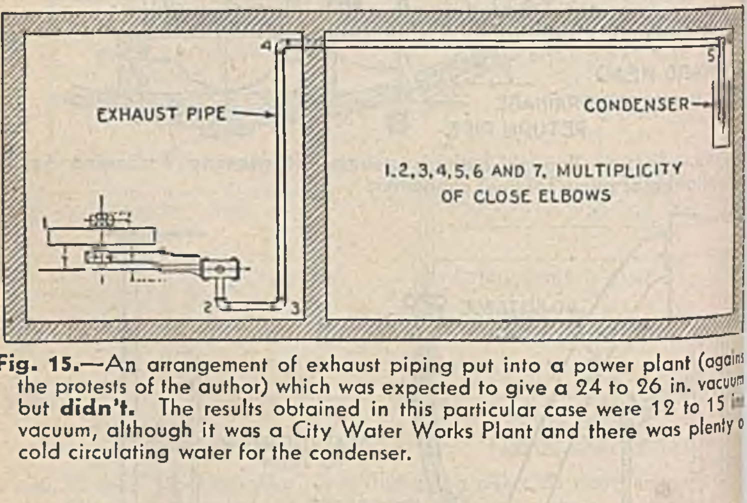


Fig. 15.—An arrangement of exhaust piping put into a power plant (against the protests of the author) which was expected to give a 24 to 26 in. vacuum but **didn't.** The results obtained in this particular case were 12 to 15 in. vacuum, although it was a City Water Works Plant and there was plenty of cold circulating water for the condenser.

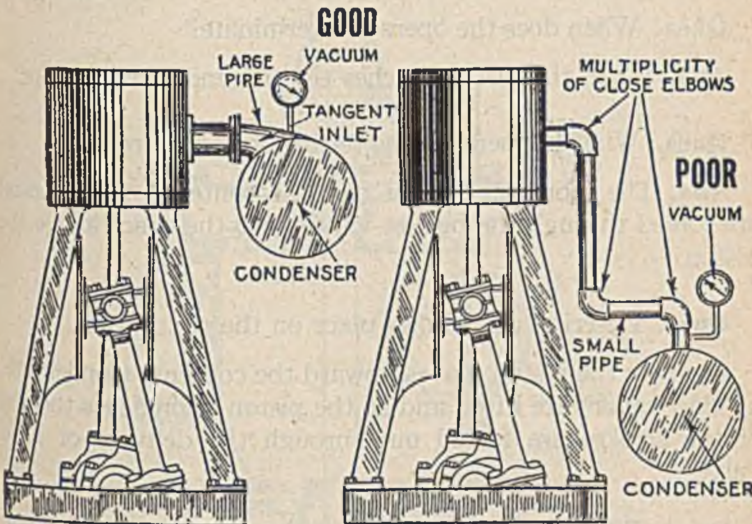
Ques. How should the piping between engine or turbine be arranged?

Ans. It should be as short and direct as possible and of ample size.

A striking example of disregarding these requirements which came to the notice of the author is shown in fig. 15. Good and questionable practice in piping an engine to a condenser is shown in figs. 16 and 17.

Operation of Wet Displacement Air Pump.—The type generally used direct connected on marine engines, is shown in fig. 18. As shown it is single acting and has three sets of valves known as:

1. Foot
2. Bucket
3. Head



Figs. 16 and 17.—The right and wrong way to pipe an engine to a condenser.

Ques. What happens during the first up stroke of the piston?

Ans. As the piston rises the air and vapor between its lower face and foot valves become rarefied with resultant decrease of pressure.

Ques. What additional event takes place as the piston continues its up stroke?

Ans. Soon a point is reached where the pressure in the condenser is decidedly greater than that in the space above the foot valves and in response to this difference of pressure the valves open and admit air, vapor and water from the condenser.

It must be evident that if the condensate "drop" into the pump as when the latter is placed below the condenser instead of having gravity act against the entrance of the condensate, a better vacuum will be obtained.

Ques. When does the operation terminate?

Ans. When the piston reaches the top end of the stroke.

Ques. What happens on the return or down stroke?

Ans. The foot valve closes and the contents of the barrel are forced through the bucket valves into the space above the piston.

Ques. Describe what takes place on the next stroke.

Ans. As the piston moves upward the contents (condensate, air and vapor) are lifted and as the piston approaches the end of the stroke, are forced out through the delivery or head valves.

Ques. What becomes of the contents after being forced out through the head valves?

Ans. The air and vapor escape and the condensate flows to the *hot well* whence it is pumped by the feed pump back into the boiler.

Ques. Upon what does the amount of vacuum that can be obtained depend?

Ans. The pump construction.

Ques. Explain.

Ans. The pressure in the condenser cannot be reduced below that necessary to leave a sufficient excess of pressure in condenser over that in pump barrel to raise the foot valves.

Ques. How may the vacuum range of the pump be increased?

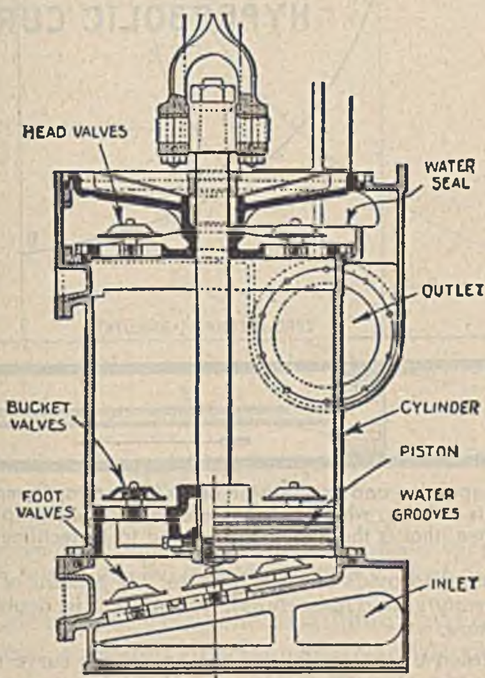


Fig. 18.—Vertical single acting direct connected bucket piston wet air pump.

Ans. By making the pump valves as light as possible and using valve springs no stiffer than necessary to give proper closing.

Ques. Name another detail in pump design upon which the vacuum depends.

Ans. The amount of clearance space between the foot valves and piston when at lower end of its stroke.

To begin with, assume this space full of air at atmospheric pressure. Now during the up stroke this air will expand and the pressure fall approximately

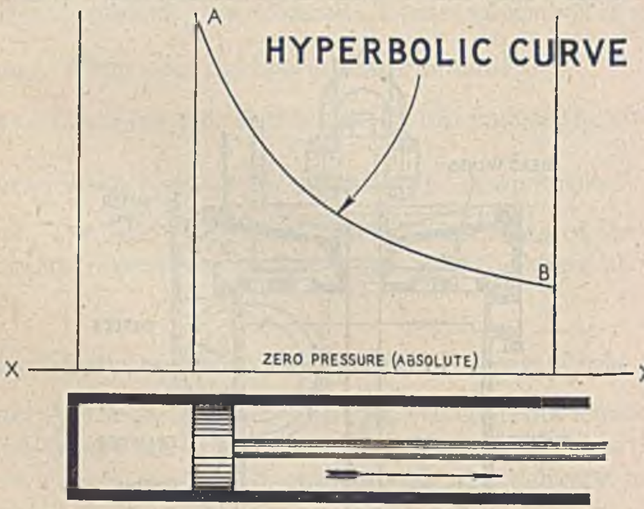


Fig. 19.—The expansion and compression of air (at constant temperature) varies inversely as its volume, which changes are represented graphically by the hyperbolic curve, that is the hyperbola referred to its rectilinear asymptotes.

in accordance with Boyle's law which states: *The pressure of a perfect gas at constant temperature varies inversely as its volume*, that is, double the volume, half the pressure.

The expansion of the air follows the hyperbolic curve referred to its rectilinear asymptotes as in fig. 19. In the figure, the curve constantly approaches the XX axis as the gas expands (as by movement of the piston to the right) but never reaches it. That is, a pump with any clearance cannot create a perfect vacuum.

Ques. Is the best vacuum within the range of the pump obtained on the first stroke?

Ans. No, it would require an infinite number of strokes.

In fig. 20 assume clearance OA and stroke AL. Now L′L = atmospheric pressure. The expansion of the air charge in the clearance space during the first stroke will be represented by the hyperbolic curve AB. Similarly for additional strokes expansion curves CD, EF are obtained, but the corresponding terminal pressures L′B, L′D, L′F, etc., gradually approach but never reach axis XX, or zero pressure.

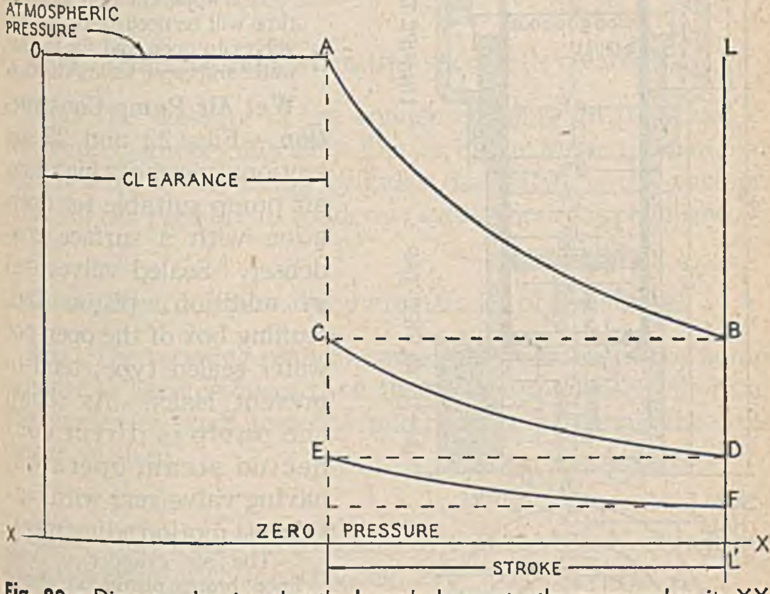
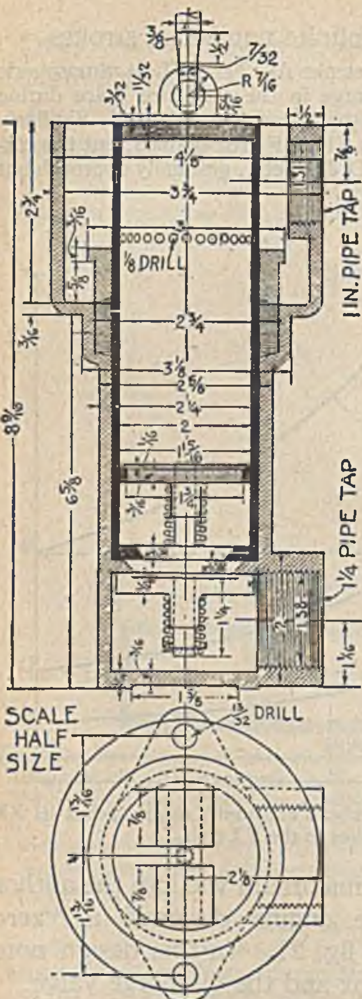


Fig. 20.—Diagram showing that the hyperbola constantly appproaches its XX axis but never reaches it. This also applies to the YY axis.

With the idea of obtaining the maximum vacuum the author for his special jacketed marine engine designed a "*zero clearance*" air pump as shown in fig. 21. In this design note the water seals for the stuffing box and the discharge valve.*

Ques. What is the effect of excessive speed?

*NOTE.—For further explanation see Audels Engineers and Mechanics Guide, Vol. 4. Page 1677 to 1732 *b*.



Ans. The valves may not have time to open between strokes, and accordingly the vacuum will become poorer and the condenser may become flooded with water.

It is apparent that a certain time will be needed for the foot valves to open and for the air, water and vapor to flow through.

Wet Air Pump Construction.—Figs 22 and 23 are sectional views of a high duty air pump suitable for operation with a surface condenser. Sealed valves and in addition, piston rod, stuffing box of the open pot water sealed type, tend to prevent leaks. As shown the pump is direct connected steam operated, having valve gear with outside lost motion adjustment.

The air cylinder is brass lined, bronze piston rod; also of bronze are the valve seats, guards, springs and stems. Valves are durable Monel metal, disc type. The accompanying list gives names of parts and their numbers which refer to the illustrations figs. 22 and 23.

Fig. 21.—Author's **zero clearance** wet air pump adapted to installations where the hot well inlet is below the level of the air pump discharge. The bucket valve is always under a head of water, hence is water sealed at all times. The arrangement of the valves permits of practically zero clearance.

Ques. What happens during the first stroke of the author's zero clearance air pump?

Ans. When the plunger starts on the intake stroke and before the foot valve opens, a momentary practically perfect vacuum is created.

Ques. What happens when this vacuum is created?

Ans. The pressure in the condenser being greater than in the pump cylinder causes the foot or intake valve to open and the pressure in the pump cylinder rises (that is the vacuum falls) till the pressure in condenser and cylinder is equalized.

Ques. What happens during remainder of the stroke?

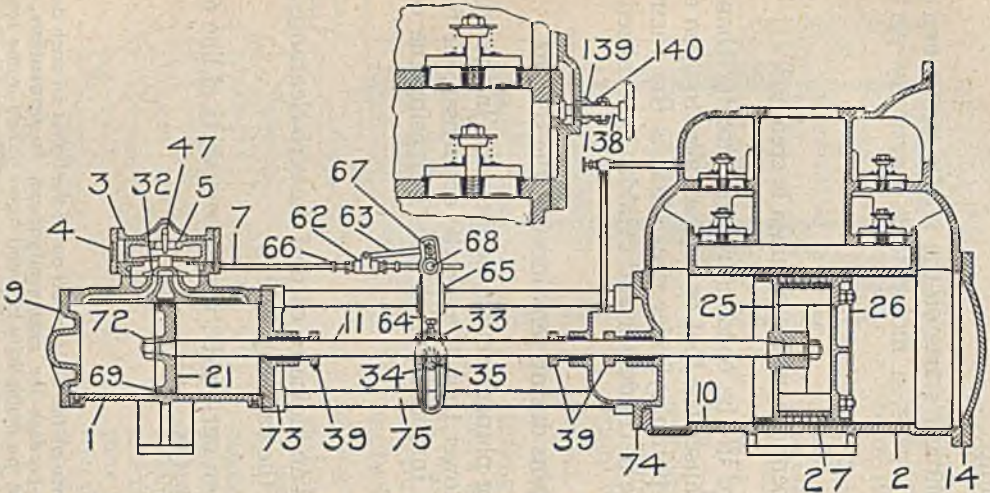
Ans. The receding plunger causes the pressure in the pump cylinder to become lower than that in the condenser, air from the condenser flows into the pump, thus increasing the condenser vacuum.

Ques. How would an indicator card of the zero clearance air pump compare with the card, fig. 20?

Ans. The expansion curve AB, would be a vertical line and coincide with the axis OX.

Fig. 21.—*text continued.*

by close adjustment, hence all the air is forced out of the barrel on each down stroke, thus tending to produce the maximum vacuum. **In operation,** the condensate oozes out of the multiplicity of small holes drilled in the top of the hollow plunger and flows to hot well by gravity. The stuffing box is water sealed. The discharge valve also is water sealed.



Figs. 22 and 23.—Double acting piston air pump with double deck valves.

Wet Air Pumps; Jet Condenser Type.—Pumps of this type are considerably larger than surface condenser wet air pumps, because in addition to the air and condensate the pump must remove the cooling water which is—to—times the condensate.

The jet type pump and its condenser are usually arranged in one unit. In operation a spray of cold water is "*injected*" into the condenser, hence the name *injection water*

LIST OF PARTS

DOUBLE ACTING PISTON AIR PUMP

FIGS. 22 AND 23

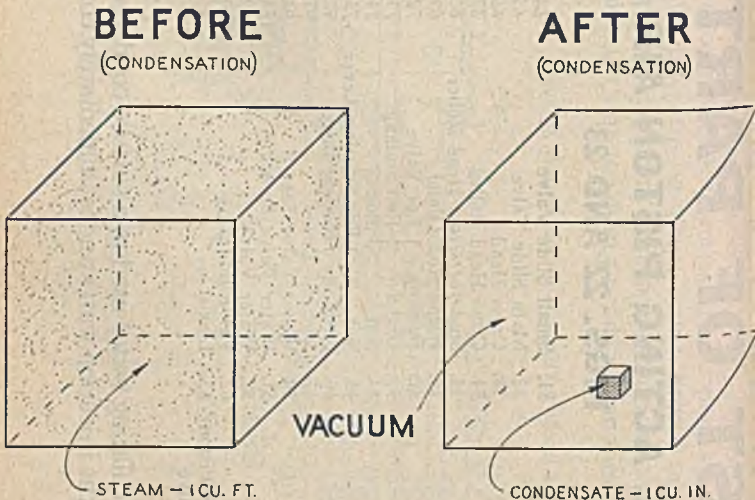
1. Steam Cylinder
2. Pump Cylinder
3. Steam Chest
4. Steam Chest Head
5. Chest Piston (Note)
7. Valve Rod
9. Steam Cylinder Head
10. Pump Cylinder Lining
11. Piston Rod
13. Hand Hole Plate
14. Pump Cylinder Head
15. Discharge Valve Plate
16. Water Chest Cover
21. Steam Piston
25. Pump Piston Head
26. Pump Piston Follower
27. Fibrous Packing for Pump Piston
30. Clamp for Small Slide Valve
31. Small Slide Valve
32. Main Slide Valve
33. Cross Head
34. Cross Head Roller
35. Stud for Cross Head Roller
39. Piston Rod Gland
42. Bushing } For Valve
43. Gland } Rod Stuffing
44. Nut } Box
45. Nut } For Starting Lever
46. Bushing } Stuffing Box
47. Starting Lever
50. Water Valve Seat
51. Water Valve Spring
52. Water Valve
62. Valve Rod Dog
63. Link
64. Lever
65. Stand
66. Valve Rod Dog Collar
67. Link Stud
68. Lever Stud
69. Steam Piston Ring
72. Piston Rod Nut
73. Steam Cradle Head
74. Pump Cradle Head
75. Cradle Bar
92. Plates for Valve
93. Valve Stem
94. Valve Guard
138. Relief Valve
139. Relief Valve Stuffing Box
140. Relief Valve Stuffing Box Nut

as distinguished from circulating water used in a surface condenser. The steam comes into actual contact with the injection water and is immediately condensed, which results in a vacuum.

Ques. Why does a vacuum form on the condensation of steam?

Ans. Because each cu. ft. of steam in condensing shrinks to about 1 cu. in. as in figs. 24 and 25.

Ques. Does an air pump create the vacuum? Mention a misleading idea.



Figs. 24 and 25.—One cu. ft. of steam and shrinkage after condensation—the reason a vacuum forms in a condenser.

Ans. It is popularly supposed that the air pump creates the vacuum, but this is not correct and probably accounts for the erroneous name of "*vacuum* pump" for *air* pump.

Ques. What does create the vacuum?

Ans. The condensation of the steam as in fig. 25.

Ques. What does the air pump (as such) really do?

Ans. It merely prevents the vacuum already created by the condensation of the steam being destroyed by the gradual accumulation of air.

Ques. What would happen (with surface condenser) if the air pump stop?

Ans. The condenser would continue in operation for a considerable time with no change but a gradual drop in vacuum.

Ques. What kind of pump is generally used with jet condensers?

Ans. The ordinary direct connected double acting horizontal piston pump.

An exception is the vertical pump used on some side wheel jet condensing steamers.

Ques. How does the pump differ from the wet air pump used with surface condensers?

Ans. Chiefly in size.

The general appearance of a jet condenser and air pump unit is shown in fig. 26.

Dry Air Pumps.—The object of this type pump is to create a vacuum higher than can be obtained with a wet air pump. The reason for the development of the dry air pump is the very high vacuum required in the operation of steam turbines.

Ques. What does a dry air pump remove from the condenser?

Ans. Air and any vapor.

Ques. How is the condensate removed?

Ans. By a separate pump called a condensate pump.

Ques. What is an objectionable name sometimes given to a condensate pump?

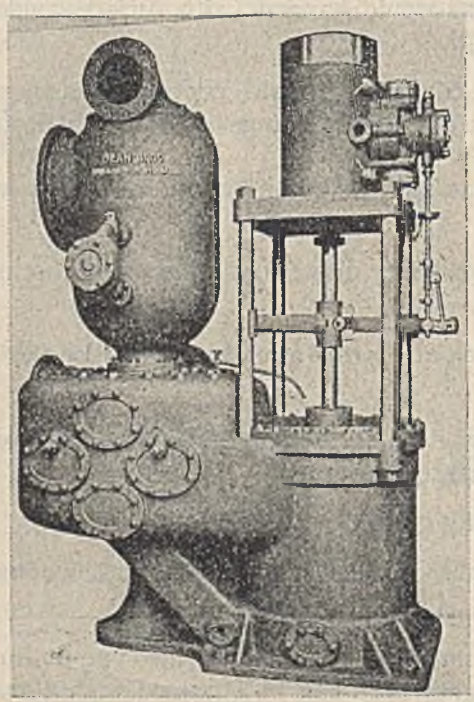


Fig. 26.—Vertical direct connected air pump and jet condenser unit.

Ans. Hot well pump.

In the first place on some fresh water installations there is no hot well, the condensate being discharged overboard. The important thing is what the pump removes from the condenser—not where it moves it.

Ques. What advantage is inherent in dry air pumps?

Ans. Handling air and gases only, the pump may be designed especially for such service (very light) and thus maintain a considerably higher vacuum than possible with the wet air pump.

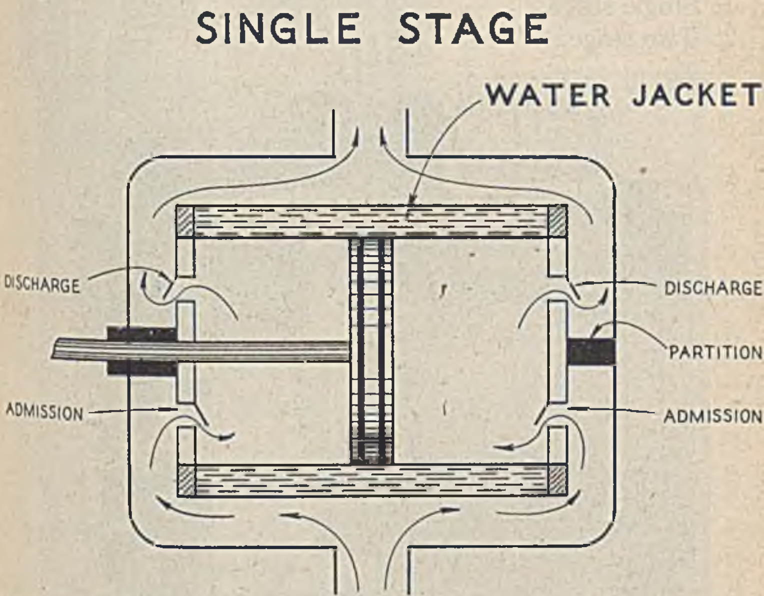


Fig. 27.—Elementary drawing of **single** stage dry air pump.

Reciprocating Dry Air Pump Types.—Construction details which differ from the wet air pump are, the valves, less clearance, water jacketed air cylinders.

In order to obtain the high vacuum, the valves must be exceptionally light and the spring must not be any stronger than required to insure proper seating of the valve. The clearance should be a minimum.

In order to reduce clearance to a minimum, the valves are placed in the cylinder heads and in some special constructions the heads themselves form the valves.

With respect to the working cycles, dry air pumps may be classed as:

1. Single stage
2. Two stage

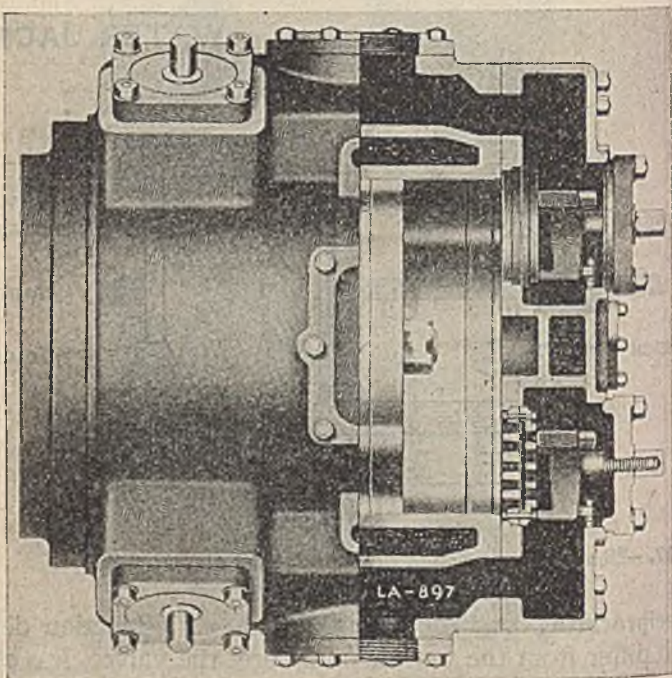


Fig. 28.—Single stage dry air pump showing typical construction.

Single Stage Dry Air Pumps.—Fig. 27 shows an elementary drawing of the single stage type showing essential features. Note placement of the valves in the head to reduce clearance

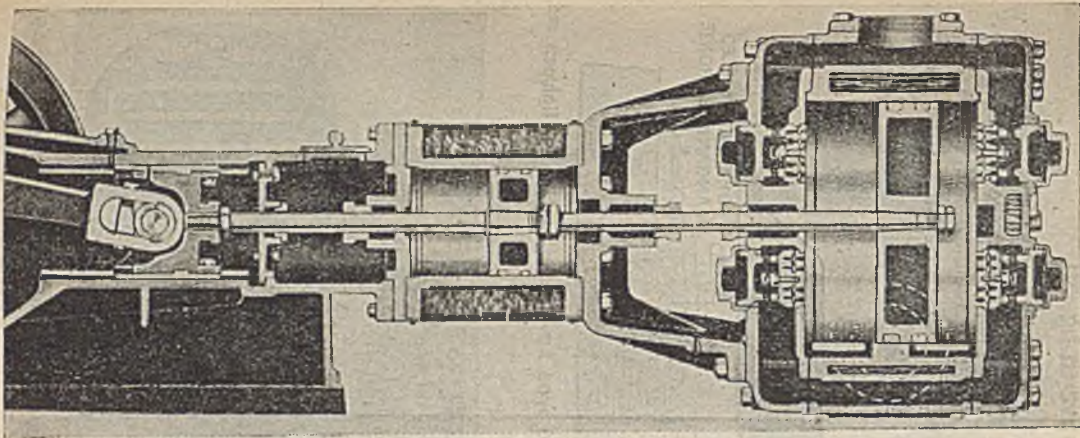


Fig. 29.—Sectional view showing construction of steam driven single stage dry air pump.

the admission valves being at the bottom and the discharge valves at the top. To carry out this arrangement the inlet and outlet passages are separated by a partition at each end.

A pump of this type as constructed is shown in figure 28.

Since as before stated the pressure in the condenser cannot be reduced (*i.e.*, vacuum increased) below that pressure which is necessary to raise the admission valves, the latter are made as light as possible. Two outstanding designs are here shown.

First is the multi-disc valve in fig. 30.

It is made of metal discs with a special spring. The parts making up the second type known as the "feather" valve are shown in figs. 31 to 34.

The complete valve element consists of but three parts: 1, valve strip; 2, valve seat; 3, valve guard.

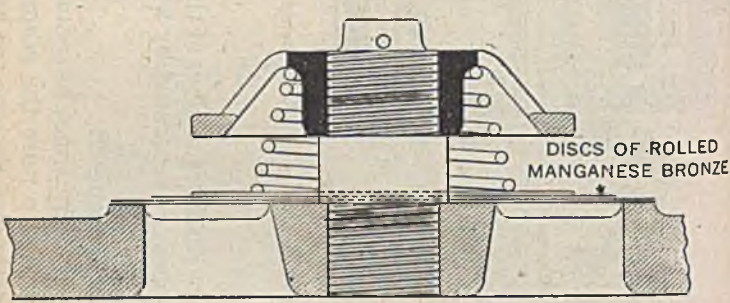
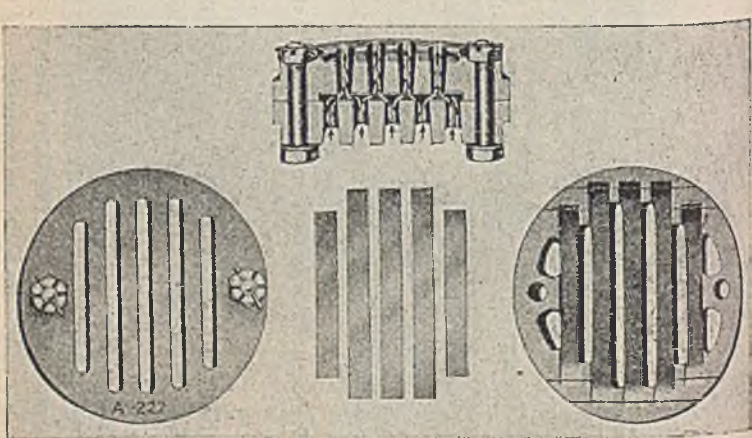
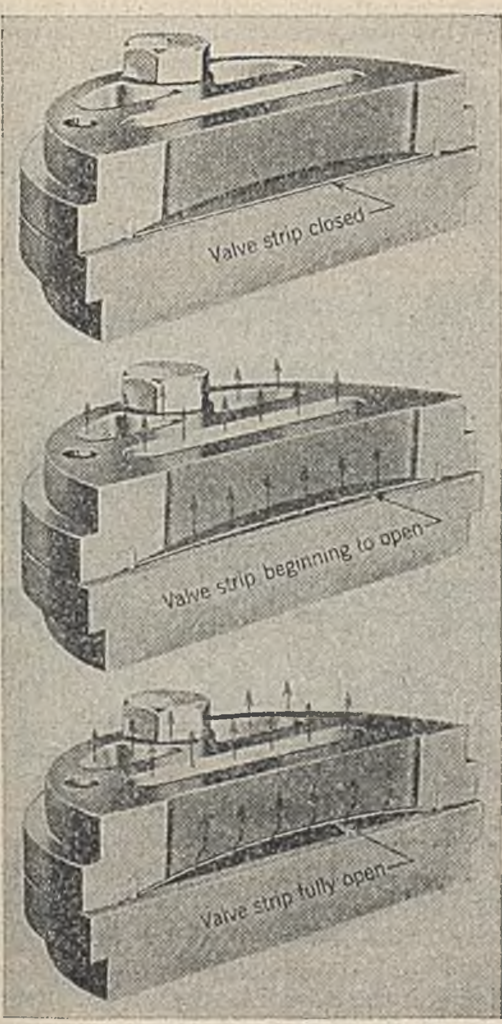


Fig. 30.—Multi-disc air valve; a type having the features of lightness and low lift.



Figs. 31 to 34.—Assembly and parts of "feather valve". A very light weight type used on dry air pumps.



Figs. 35 to 37.—Cut away views showing progressively the action of a feather air valve.

The valve strips are the only moving parts of the valve and they are no rigidly held at any point, being restrained from lateral movement only b recesses in the curved guard.

Complete freedom for the strip is assured by steel inserts placed acro the ends of the guard milling.

In opening, the valve strip is permitted to lift in the middle again a perfectly curved guard, to allow passage of air on either side of the stri The opening of the valve is shown progressively in figs. 35 to 37.

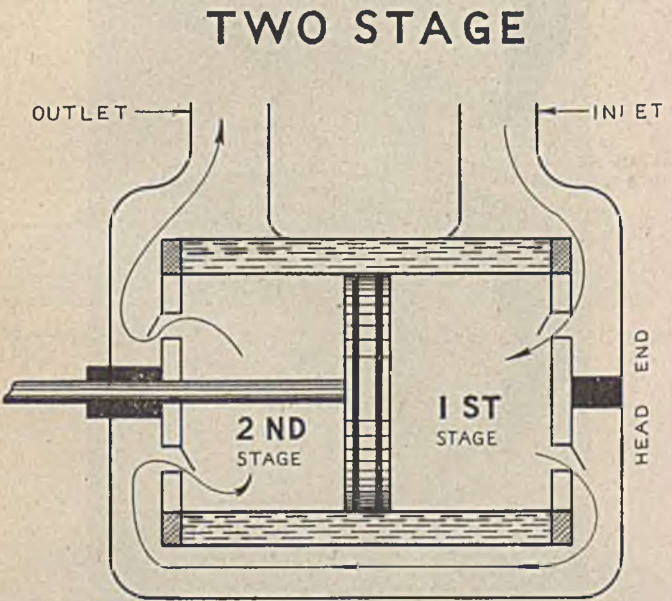


Fig. 38.—Elementary drawing of a **two** stage dry air pump.

Two Stage Dry Air Pumps.—An elementary drawing of t one cylinder two stage pump is shown in fig. 38, which presen in a very simple way the principles involved.

The head end of the cylinder, which is the first stage, dra the vacuum and therefore receives any moisture or vapo which might come over from the inlet.

Since the ratio of compressions of the first stage to the second stage is one to one, any entrained moisture or vapor which enters the first stage passes directly to the second stage without any noticeable evaporation.

Furthermore, since there is no compression in the first stage, there is no re-expansion of the air in the clearance space. As a result of both these factors, the entire cylinder volume of the two stage pump is available for removing air and vapors from the condenser or inlet instead of having the cylinder very considerably filled with re-evaporated vapor from cylinder condensation. Therefore, a very much higher volumetric efficiency is obtained than with any other type.

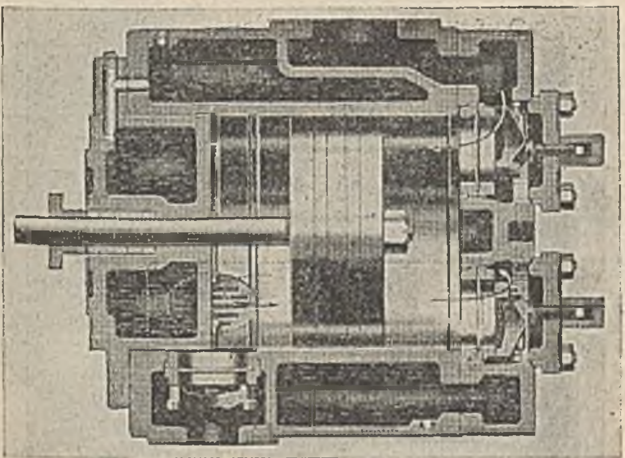


Fig. 39.—Sectional view of two stage feather valve air pump. Single horizontal cylinder.

The first stage or the high vacuum, being on the head end, does not have a stuffing box; consequently there is no stuffing box leakage. Should the stuffing box on the second stage leak slightly, it would not seriously affect the vacuum, since the degree of vacuum is almost entirely a function of the first stage.

Figs. 39 and 40 show construction of two designs of two stage cylinder.

Impulse Wet Air Pumps.—In this type pump the condensate and air is transferred from the lower side to the upper side of the piston by a slapping action or *impulse* given when the piston nears the lower end of the stroke—hence the name.

The pump is a reciprocating pump, but of the impulse type. Figs. 41 and 42 show a vertical impulse pump invented by Edwards.

Ques. What is the essential feature of the impulse pump?

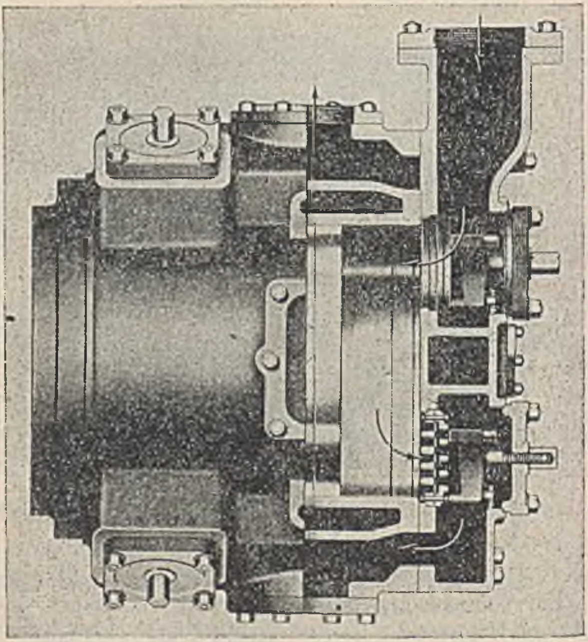


Fig. 40.—Another design of two stage feather valve air pump. Single horizontal cylinder.

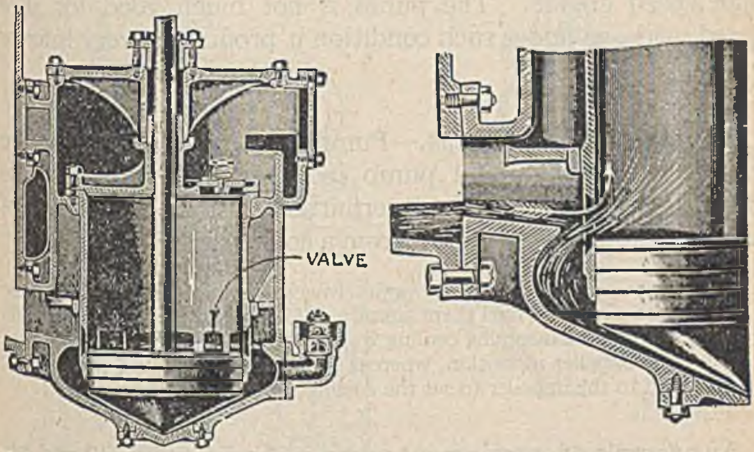
Ans. The piston as it nears the lower end of the stroke uncovers ports in the cylinder walls which form a mechanically operated inlet or admission valve.

Ques. What other valve is necessary for the operation of the pump?

Ans. A discharge or head valve.

Ques. Describe the operation of the pump.

Ans. The condensate flows continuously by passing from



Figs. 41 and 42.—Single acting impulse type vertical wet air pump.

the condenser into the base of the pump and is there dealt with mechanically by the conical bucket working in combination with a base of similar shape.

The conical shaped piston "*slaps*" the accumulation of condensate in the bottom of the cylinder giving it an impulse and the momentum thus acquired causes the water and air to flow at high velocity through the ports into the cylinder.

Ques. Describe what happens on the up stroke.

Ans. The rising piston closes the ports, trapping the water and air in the barrel and carries it up. Approaching the top end of the stroke it forces the condensate and air out through the discharge or head valve.

Ques. For what service is the impulse pump adapted?

Ans. For high speed operation as by direct connection on high speed engine. The pump is not much good for slow speed work, as under such condition it produces a very inferior vacuum.

Entrainment Air Pumps.—Pumps of this classification are a species of centrifugal pump in that centrifugal force is employed to hurl spirally intermittent pistons or "slugs" of water to expel air and vapors from a condenser.

In these pumps an impeller or jet wheel is employed to break up the water into slugs and hurl them spirally. They differ from the centrifugal pump in that the incoming cooling water (entering under pressure) is used to set the impeller in motion, whereas in the true centrifugal pump power is applied to the impeller to set the cooling water into motion.

An example of entrainment pump is shown in figs. 43 and 44.

Ques. Describe its operation.

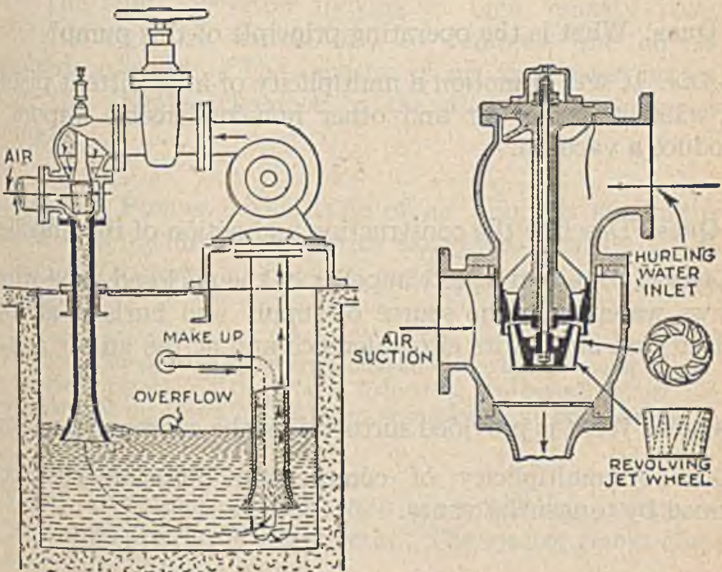
Ans. It removes the air and non-condensate vapors from the condenser by hurling jet of water at high velocity approximately rectangular in cross section from a revolving wheel.

The shape of the revolving jet and diffuser wheel is clearly seen in fig. 44.

Ques. How do the water jets rushing through the impeller discharge?

Ans. In the form of a spiral enclosing the vapors which enter around the transforming wheel between the jets or pistons of water.

Ques. How is the cooling water delivered to the impeller?



Figs. 43 and 44.—Entrainment air pump. Assembly and details.

Ans. Under pressure by a centrifugal pump.

Ques. What is the source?

Ans. The supply of water comes from a tank into which the air pump discharges.

Make up and overflow connections are provided as shown in fig. 43.

Turbine Air Pump.—The essential elements of this pump are an impeller, air gap and diffusion vanes.

This is really a type of centrifugal pump because power is applied to the impeller and centrifugal force utilizes to throw the water all around the circle.

Ques. What is the operating principle of this pump?

Ans. It sets in motion a multiplicity of intermittent pistons of water to expel air and other non-condensible vapors to produce a vacuum.

Ques. Describe the construction and action of the impeller?

Ans. It is a high speed impeller of the enclosed type which draws water from the source of supply and hurls it at high velocity all around its circumference and across an *air gap*.

Ques. What is provided surrounding the concentric air gap?

Ans. A multiplicity of compression diffusion chambers formed by tangential vanes.

Ques. How is the water disposed of?

Ans. It passes off through a surrounding passage formed by the shape of the casing.

Ques. Describe the operation of the three elements?

Ans. The impeller in its rotation hurls the water at high velocity across the air gap and it is divided into a great number of separate "pistons" or intermittent slugs of water on entering the diffusing chambers. These intermittent slugs of water entrap successive portions of air around the air gap which produces the vacuum.

Ques. How does the water get out?

Ans. The slugs of water moving at high velocity have considerable dynamic inertia and so compress the air to atmospheric pressure. The mixture of air and water passes through the annular chamber to the discharge opening of the pump.

Ejector Air Pumps.—This type of air pump is similar in operation to a steam injector, with exception that the jet of steam is used to force air out of a condenser instead of water into a boiler.

The increasing demand for high vacuum and large air and vapor capacities in power plants and the process industries, has brought about the development of the steam jet air pump or ejector to its present state of high efficiency.

Inherent advantages of this pump are simplicity, compactness, no moving parts, low cost, etc. The ejector pump consists essentially of these parts:

1. Steam nozzle
2. Combining tube
3. Delivery tube.

The elementary drawing fig. 45 shows the arrangement of these parts. In construction the combining tube and delivery are made in one part instead of separate as in the case of the injector.

Ques. What is the principle of operation?

Ans. The kinetic energy of the steam issuing from the nozzle at very high velocity is utilized to pick up the air in passing across the air gap and eject it from the condenser against atmospheric pressure.

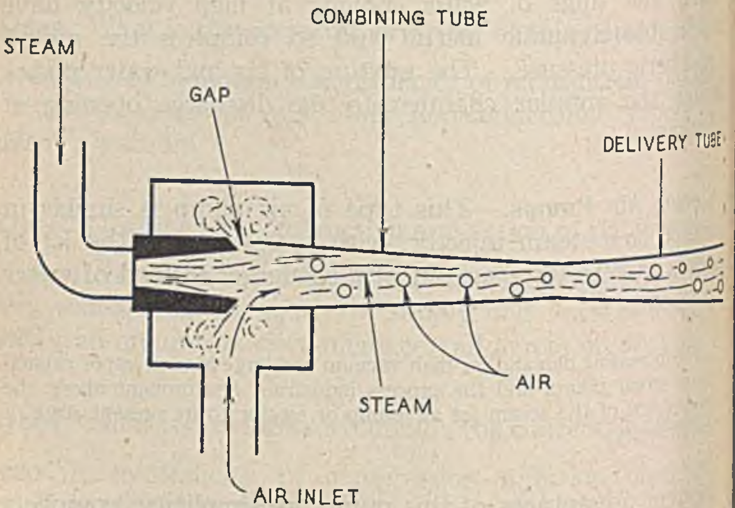


Fig. 45.—Elementary drawing showing essential elements of an ejector type air pump.

Rotary Air Pumps.—In principle, this type is the rotary equivalent of a single acting reciprocating pump. That is, in place of a piston it has an eccentric mounted disc, an oscillating cam for an admission valve and feather-weight discharge valves. A typical construction is shown in fig. 46.

Ques. Describe the construction?

Ans. The disc rotor is mounted eccentrically on a shaft supported in outboard bearings. An oscillating cam operated from the rotor shaft by means of external cranks and connecting rod separates the admission and discharge sides of the cylinder and follows the motion of the rotor with a small clearance.

Ques. How is the rotor adjusted?

Ans. It is so adjusted as to maintain during its entire rotation a close clearance with the inside of the air chamber.

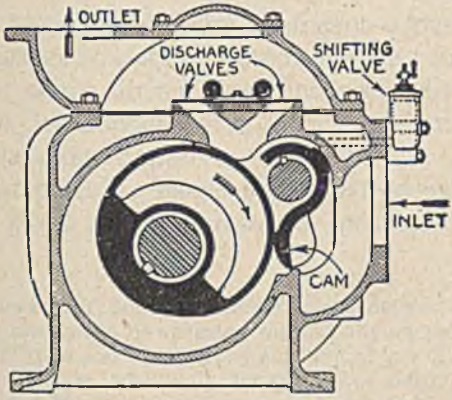


Fig. 46.—Rotary air pump. It operates at speeds rendering it suitable for direct connected engine and geared or direct connected motor drive. **In construction,** the cylindrical rotor is mounted eccentrically on a shaft supported in outboard bearings. An oscillating cam, operated from the rotor shaft by means of external cranks and connecting rod, separates the inlet and discharge sides of the pump cylinder, and follows the motion of the rotor with a small clearance. The rotor is so adjusted as to maintain during its entire rotation a close clearance with the inside of the air cylinder. Air is drawn through the intake, at the lower right, and discharged through the valves shown on the deck above the cam. The outlet is at the top and may be either right or left hand. Freedom from leakage is assured because all clearances are water sealed. The rotor is at no time in contact with the cylinder, nor does the cam touch the rotor, being controlled in its motion by an external driving gear which keeps it close to, but not quite touching, the rotor. One end of the rotor shaft has a crank operating another crank on the end of the oscillating cam shaft, by means of a rod.

Ques. What is the path of the air?

Ans. Air is drained through the inlet at the right and discharged through the valves shown on the deck above the cam and outlet at the top.

Ques. With the clearances how is leakage avoided?

Ans. All clearances are water sealed.

CHAPTER 21

Jet Condensers

Although a condenser is by no means a pump, condensers and air pumps are so closely related (one can't function without the other) it seems fitting that condensers should be treated in this book—Jet Condensers in this chapter and Surface Condensers in the next chapter.

In the first place, *it should be understood* **once and for all**, that the *air pump does* **not** *create the vacuum*, but only maintains it.

As a matter of fact, any first class surface condensing equipment in good operating condition can be operated without any air pump for a considerable length of time and without any substantial loss of vacuum. This in no case will continue indefinitely, for quickly or slowly, but in any event surely, the air will accumulate in the condenser and build up a pressure, and it is only a question of time until that pressure will be atmospheric and greater.

Therefore, the air pump is quite necessary even though the heat removing capability of the condenser itself remains as the essential factor of the process.

By definition a jet condenser is: *In steam engineering, an apparatus in which exhaust steam is reconverted into water by mingling with a spray of cooling water*. That is, in other words the function of a condenser is to cool exhaust steam so as to reduce its pressure and volume to a minimum by extracting its latent heat and thus changing its state, that is reconverting it into water, or as usually put, condensing the steam.

Ques. What name is given to the cooling water and why?

Ans. The *injection* water because it is "injected" into the condenser.

Classification of Condensers.—All condensers with respect to the method of transferring the heat from the steam to the cooling water may be divided into two classes:

1. Direct contact (**Jet** condensers)................Class 1

2. Surface contact (**Surface** condensers)..........Class 2

(See Chapter 22)

Class 1. Condensers

(Classification)

The numerous types of condensers of the direct contact ("jet") type may be divided into several classes, with respect to various distinguishing features.

1. With respect to the method of circulating the cooling water as,

- *a*. Rain
- *b*. Jet
- *c*. Barometric
- *d*. Steam ejector

2. With respect to elevation, as

   a. Low level (jet)

   b. High level (*barometric*)

3. With respect to the flow of the steam and cooling water, as

   a. Parallel flow

   b. Counter flow

   c. Combined, parallel and counter flow

   d. Disc flow

4. With respect to the degree of baffling, as

   a. One pass

   b. Two pass, etc.

5. With respect to the method of introducing the cooling water, as

   a. Low level

   b. High level (*Barometric*)

6. With respect to the method of removing the cooling water, as

   a. Barometric

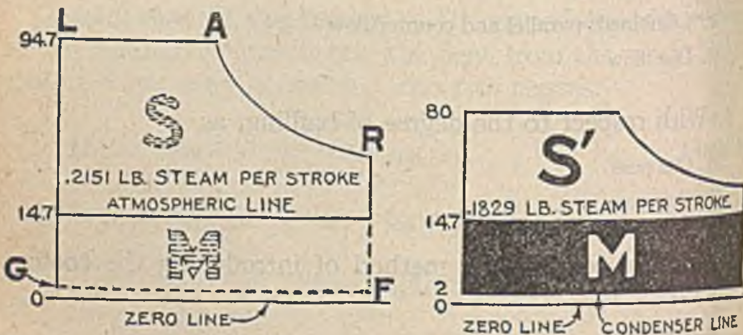
   b. Eductor

   c. Pump

The Vacuum.—When the absolute pressure in an enclosed space is less than the barometer reading outside, it is customary to call it a "vacuum." However, it should be noted that this alleged vacuum is not a vacuum only a partial vacuum. That is to say, any reading of the vacuum gauge less than barometric is not a vacuum—only a partial vacuum.

By definition (as before defined) a vacuum is *a space void of matter*, that is, a space in which the pressure is zero absolute. The meaning of a partial vacuum referred to as a 30 in. barometer has been explained on page 506.

Saving Due to Condensing.—When an engine is run without a condenser the steam must be exhausted against the pressure



Figs. 1 and 2.—Theoretical diagrams for equal power of throttling engines operating non-condensing (sometimes ill-advisedly called "high pressure") as in fig. 1 and condensing as in fig. 2. It should be distinctly understood that these are theoretical cards for engines without clearance being shown for simplicity and in practice the actual saving by condensing depends on many conditions. The solid black area **M**, is due to the condenser; hence, it must be evident that in governing by throttling when changing from non-condensing to condensing operation, the initial pressure is lowered until the card area S′ and **M** (fig. 2) is equal to S (fig. 1) thus maintaining constant load. Also if the initial pressure remained the same and condenser be added, the card area S, would be increased by the area M, giving card LARFG (fig. 1) increasing the power by area M. This is one way of increasing the power of an engine.

of the atmosphere, or 14.7 lbs. per sq. in. Now the nature of steam is such that most of this back pressure can be removed, that is, if at the end of the forward or steam admission stroke, the cylinder full of steam be chilled as by injecting cold water or exhausting into a cold chamber, the steam will condense leaving a vacuum into which the piston can return without having to force back the atmosphere.

It must be evident that since, by condensation, most of the back pressure is removed from the exhaust side of the piston, a considerable *gain* in power or *saving* is the result. The extent of this saving depends largely upon conditions of operation, the net economic effect being equal to the saving in fuel less the cost of condensing.

To obtain an idea of the nature and extent of the saving due to condensing, consider first the theoretical results obtained with

1. Throttling engine. 2. Automatic cut off engine.

In the first instance, assume a throttling engine running non-condensing with 80 lbs. initial gauge pressure, and ½ cut off, as indicated by the theoretical diagram, fig. 1. The corresponding mean effective pressure is

$$\frac{94.7 \times 1.69}{2} - 14.7 = 65.3 \text{ lbs.} \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Now, if the engine be operated condensing, the condenser will reduce the back pressure to say 2 lbs., thus removing 14.7 − 2 = 12.7 lbs. pressure from the exhaust side of the piston.

Now since the cut off remains the same, for equal power, the throttling governor will reduce the initial pressure to approximately 80 lbs. absolute, giving a *m.e.p.* of

$$\frac{80 \times 1.69}{2} - 2 = 65.6 \text{ lbs.} \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

being practically the same *m.e.p.* as obtained in (1). Fig. 2, is the condensing diagram, the solid black area M, being the portion of the diagram due to the condenser.

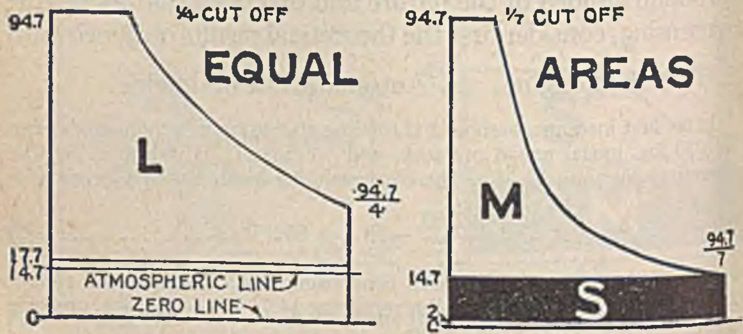
Again, if the volume of the cylinder up to the point of cut off be one cu. ft. (no clearance), it would require when running non-condensing one cu. ft. of steam at 94.7 lbs. which weighs .2151 lb., and when running condensing, one cu. ft. of steam at 80 lbs. pressure which weighs .1829 lb. Hence, when running condensing there would be an ***apparent saving*** of .2151 − .1829 = .0322 lb. per stroke, or

$$\frac{.2151 - .1829}{.2151} \times 100 = 14.97\%$$

Consider now the automatic cut off engine, running non-condensing, with 80 lbs. initial gauge pressure, and ¼ cut off, 17.7 lbs. abs. back pressure as indicated by the theoretical diagram in fig. 3, giving a mean effective pressure of*

$$\frac{94.7 \times 2.39}{4} - 17.7 = 38.9 \text{ lbs.} \qquad (1)$$

Now if the engine be run condensing, the condenser reducing the back pressure to say 2 lbs., this will remove 17.7 − 2 = 15.7 lbs. pressure from the exhaust side of the piston, hence the engine governor would automatically shorten the cut off to some point such that the portion of the



Figs. 3 and 4.—Theoretical diagrams for equal power of an automatic cut off engine operating non-condensing, as in fig. 3, and condensing as in fig. 4. Here the area of the card remains the same, but its contour changes. The solid black portion S, is the portion due to condensing, hence to keep the power constant the portion M, above the atmospheric line is reduced by shortening the cut off till M + S = L. The cut offs here shown: ¼ non-condensing, and 1/7 condensing are usually the most economical cut offs.

card produced above the atmospheric pressure line would give a mean effective pressure of 37.9 − 17.7 = 20.2 lbs. or the same *m.e.p.* of 38.9 lbs. for the entire card, thus giving constant power.

By trial and error the shortened cut off is found to be approximately 1/7th, thus:

$$\frac{94.7 \times 2.95}{7} - 2 = 37.9 \text{ lbs. } m.e.p. \qquad (2)$$

The diagram corresponding being shown in fig. 4.

*NOTE.—In equations (1) and (2) above 94.7 is *absolute* initial pressure; 2.95 and 2.39 is 1 + *hyp. log* of 4 and 7 (expansions) respectively.

Now, $1/7 = 1/7 \div 1/4 = 57\%$ of $1/4$, hence, the volume of steam to be admitted for $1/7$ cut off is only 57% of that required for $1/4$ cut off, and accordingly the *apparent saving* is

$$\frac{1 - .57}{1} \times 100 = 43\%$$

In these two cases the saving is the *apparent* saving for, as must be evident, if a feed water heater be used, the feed water could be returned to the boiler at a higher temperature when operating non-condensing than with a condenser, because some of the heat is carried off in the circulating water of the condenser, which otherwise would be absorbed by the feed water.

Thus the temperature of the exhaust steam at atmospheric pressure is 212° Fahr., and at 2 lbs. absolute, 126°, and assuming that with a feed water heater the water could be heated to these temperatures, its temperature would be $212 - 126 = 86°$ higher non-condensing than condensing. Now since there is a saving of approximately 1% for each 10° that the feed water is heated, the saving in this case would be 8.6%, which must be deducted from the apparent saving, and also the work of the condenser pump to obtain the net (theoretical) saving. The work of the condenser pump consists in pumping the water used in condensing the steam and removing air from the condenser.

Assuming 2° for this work, the net theoretical saving would be as follows:

		Throttling engine	Cut off engine
Apparent theoretical saving		14.97%	43%
Increased saving by feed water heating (non-condensing)............	8.6%		
Work of condenser pump..........	2 "		
	10.6%	10.6%	10.6%
Net theoretical saving condensing...		4.07%	32.4%

Thus considerably more saving is obtained with the cut off engine than with the throttling engine, which is to be expected, because of the increased expansion of the steam.

Now in practice the net economy of condensing is somewhat less as explained in the note below.*

Power Gain Due to Condensing.—There still remains a surprisingly large number of power producing plants (including high powered tugs) which could very profitably be changed over to condensing operation. This is due to gross ignorance of the owners who do not realize the extent to which their power output can be increased by utilizing the energy which while operating non-condensing is thrown away, which if operating condensing could be converted into useful work.

According to C. H. Wheeler, taking full account of feed water temperatures and the power consumed by the condenser auxiliaries, the actual increase in power gained by changing over a single expansion reciprocating engine from non-condensing to condensing operation amounts to from twenty to thirty per cent. Similarly, with a compound reciprocating engine operating condensing at about five inches absolute, if the cut-off be lengthened and the engine exhaust to a suitable low pressure turbine, an actual increase in power can be gained which also amounts to from twenty to thirty per cent. In this case the combined cost of the turbine and the condensing equipment would still be less than that of the additional engine, boiler and condenser which would otherwise be required for the production of the additional power. In the same connection, another interesting comparison is that of a small high pressure turbine, operating non-condensing, and requiring say forty pounds of steam per hour per *k.w.* with a similar turbine, but designed for and operating condensing, and therefore requiring say only twenty pounds.

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*NOTE.—*Economy of Condensing.* "It is held in the popular mind that the economy of condensing is, in round numbers, 25%. This percentage usually relates to simple engines and it refers to the economy as measured by the difference in the coal consumption produced by a condenser." The evidence of some of Barrus' tests show that "this belief is not well founded except in special cases." "If the feed water be heated by the exhaust steam of the non-condensing engine to a temperature of 100° Fahr., which is that of the ordinary hot well, to a temperature of 210° Fahr., the non-condensing engine can be credited with about 11% less coal consumption, which should be considered in determining condenser economy." The average of a number of Barrus' tests gives a saving produced by condensing of 22.3%. "If we allow for the steam or power used by an economical condenser, it would be seen that the net economy of condensing is at best not much over 20%, based on steam consumption. If furthermore, we allow for the difference produced by heating the feed water to the extent above mentioned, the *saving of fuel* would be reduced to about 10%."—*Barrus.*

Class 1. Jet Condensers

By definition, a jet condenser is: *A closed chamber within which exhaust steam comes in* **direct contact** *with a spray or jet of cold water and is condensed.*

Ques. What is the cold water called and why?

Ans. The injection water because it is "injected" into the condenser.

As distinguished from the circulating water of the surface condenser.

Ques. What happens when the steam comes into contact with the finely divided injection water?

Ans. It is *almost* instantly condensed.

Ques. What happens when condensation takes place?

Ans. Since each cu. ft. of exhaust steam shrinks to about 1 cu. in. of water when condensed, an empty space or vacuum is thus created in the condenser.

Ques. What causes the injection water to enter the condenser?

Ans. When the condenser is not too high, atmospheric pressure forces it in; when higher than the barometric column, a pump is required.

Ques. What must be removed from the condenser in addition to the condensate and injection water?

Ans. A small amount of air.

Ques. How are the water and air removed?

Ans. On the low level type, by a single pump.

In the larger sizes sometimes a dry air pump is used to remove the air only. Fig. 5 shows the essentials of a low level condenser.

Ques. What name is generally given to the low level condenser?

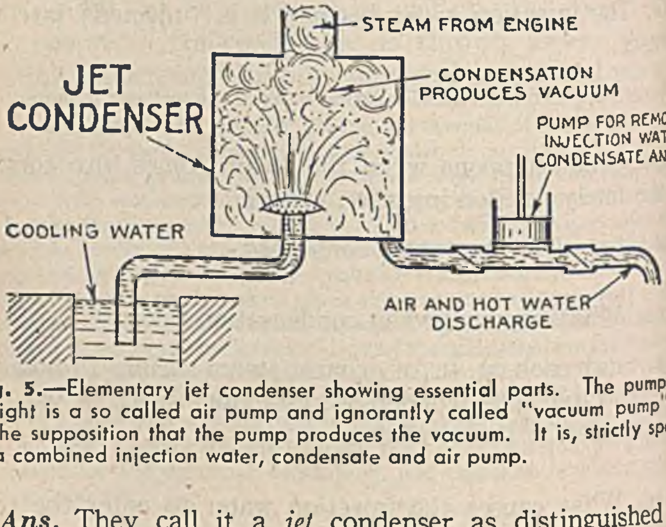


Fig. 5.—Elementary jet condenser showing essential parts. The pump at the right is a so called air pump and ignorantly called "vacuum pump" under the supposition that the pump produces the vacuum. It is, strictly speaking, a combined injection water, condensate and air pump.

Ans. They call it a *jet* condenser as distinguished from the barometric type of jet condenser.

Low Level or "Jet" Condensers.—The term *jet*, although it is applied too broadly to all condensers in which the steam and cooling water come into direct contact, is generally used to designate a low level condenser in which a pump is required to remove the water as distinguished from a high level or barometric condenser, which, as later explained, requires no pump to remove the injection water.

There are three general types of jet condensers:

1. Parallel flow.
2. Counter-flow.
3. Combined counter and parallel flow.

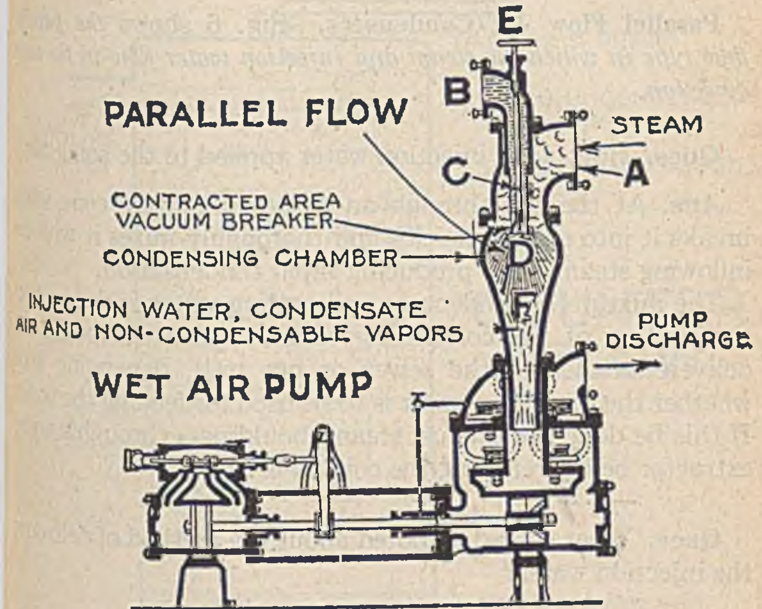


Fig. 6.—Parallel flow condenser. **It consists of** a conical or bottle shaped casting projecting down into the water end of the air pump, and having openings in its upper parts for steam and cooling water. **In operation,** the exhaust from the engine enters the condensing chamber at A, and the injection water at B. C, is the spray pipe which has at its lower extremity a number of vertical slits through which the water passes and becomes spread into thin sheets. The spray cone D, breaks the water passing over it into a fine spray, and thus causes a rapid and thorough mixture of the steam and water. The spray cone is adjusted to give the proper amount of water by means of a stem passing through the top of the condenser to wheel E. The injection water and condensed steam fall together through the opening F, into the pump and are discharged into a convenient waste pipe, or into a hot well when the discharge water is to be used for feeding the boilers.

Ques. What should be provided on every low level or jet condenser and why?

Ans. An automatic vacuum breaker to protect the engine or turbine in case the pump stop.

Parallel Flow Jet Condensers.—Fig. 6 shows *the parallel flow type in which the steam and injection water flow in the same direction.*

Ques. How is the injection water applied to the condenser?

Ans. At the top through an adjustable spray cone which breaks it into small particles and thoroughly mixes it with the inflowing steam, thus producing rapid condensation.

The mixture of condensate and cooling water is drawn from the bottom of the condensing chamber into the pump and delivered either to the sewer or hot well, depending upon whether the discharge water is to be used for feeding the boiler. If this be done the exhaust steam should pass through a grease extractor before reaching the condenser.

Ques. What should be noted about the method of delivering the injection water?

Ans. It depends upon the lift.

If the lift be not over 20 ft. no pump is needed, if greater, a pump should be used as a booster to atmospheric pressure in lifting the water.

Counter Flow Jet Condensers.—Fig. 7 shows the counter flow type of jet condenser *in which the steam and water flow in opposite directions.*

Ques. What should be noted about this type condenser?

Ans. In this arrangement the coldest water comes in contact with the coldest steam, that is, with steam in its last stage of condensation, thus tending more to complete condensation in the condensing chamber and requiring a smaller amount of cooling water.

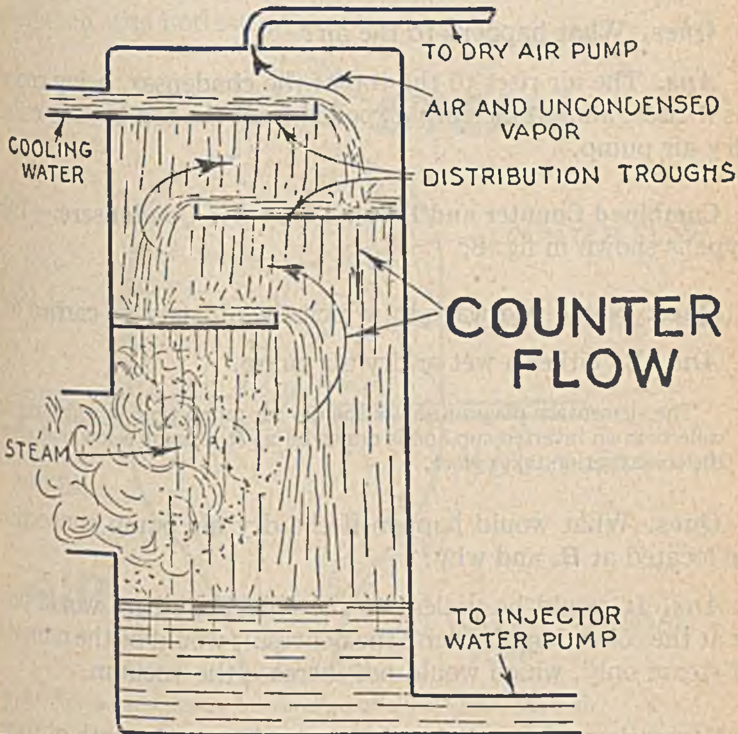


Fig. 7.—Counter-flow "jet" condenser. **In this arrangement** the steam and water flow in opposite directions, that is, the entering steam encounters the warmest water and condenses as it rises, passing through successive curtains of water, obtained by suitably arranged overflow trays. Thus the temperature of the vapors is gradually reduced as they approach the top of the condenser, due to the proximity of the incoming injection water. Ultimately the mixture entering the pipe to the dry air pump consists of air of relatively high density compared with that of the residual water vapors.

Ques. What is the path of the injection water and condensate?

Ans. They both fall to the bottom of the condenser and are carried off by a pump.

Ques. What happens to the air?

Ans. The air rises to the top of the condenser, being cooled as it rises, and is efficiently expelled from the top by a so called dry air pump.

Combined Counter and Parallel Flow Jet Condensers.—This type is shown in fig. 8.

Ques. Name two ways by which the air may be carried off.

Ans. By either a wet or dry air pump.

The elementary diagram shows the dry air pump type in which the air collects in an inverted cup and is drawn off at **A**, a point below that where the condensation takes place.

Ques. What would happen if the dry air pump connection be located at **B**, and why?

Ans. It would be useless as the entering steam would prevent the collection of air and the net result would be the removal of steam only, which would not increase the vacuum.

Comparison of Parallel and Counter Flow.—In both of these types the flow of the injection water is downward as is plainly seen in the diagrams.

Ques. What are the characteristics of the parallel flow condensers?

Ans. The steam enters at the top with result that it meets the injection water when it is coldest.

Ques. What obtains with counter flow?

Ans. The steam enters at the bottom and meets the injection water when it is hottest.

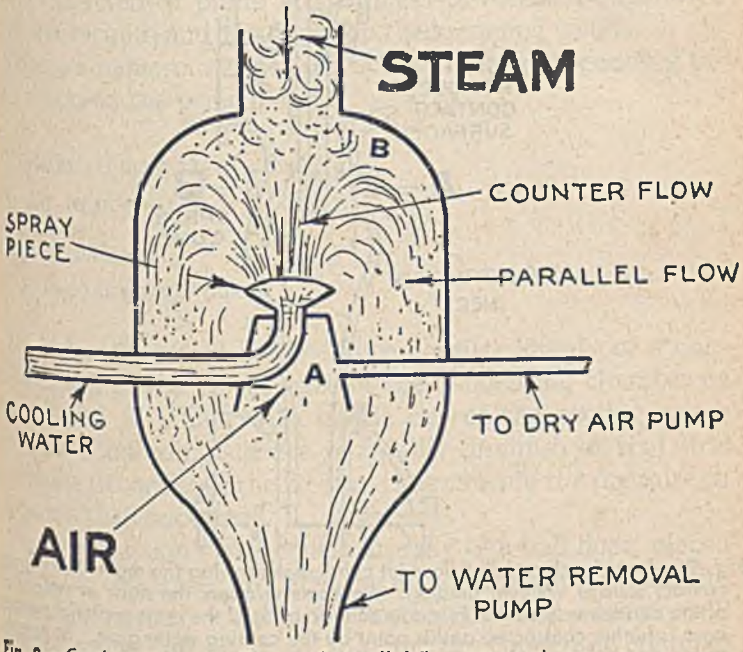


Fig. 8.—Combined counter flow and parallel flow jet condenser.

Ques. Name a feature of the parallel flow condenser.

Ans. It is the possibility of utilizing the *ejector* action of the steam to assist in air removal.

Ques. What should be provided in either type?

Ans. Means for cooling the vapors before they pass to the air pump and at the same time, for heating the condensing water as nearly as possible to the temperature of the vacuum.

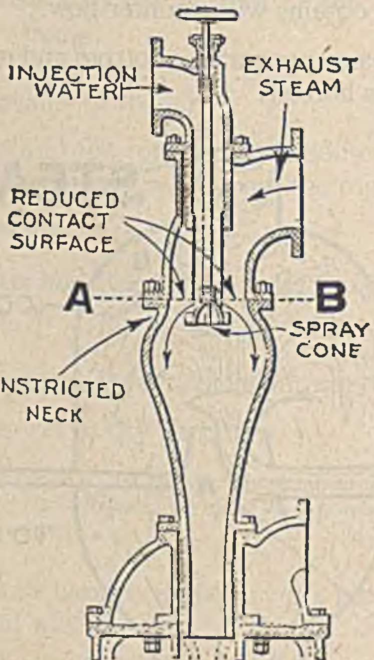


Fig. 9.—Low level parallel flow jet condenser showing the method of reduced contact surface vacuum breaker. **In construction** the neck or upper part of the condenser chamber is made quite small and the cross sectional passage area is further constricted at this point by the cooling water pipe. **In operation,** rapid condensation is due only to the large surface exposed by the cooling water as it passes through the large section of the condensing chamber. Due to the constricted neck, any accumulation of water rapidly diminishes the condensing surface until the spray cone itself is submerged, leaving only the small annular ring of water at AB, to act on the large volume of entering steam. The surface of this ring being far too small to condense the steam, the pressure immediately rises causing the relief valve between the engine and condenser to open and allow engine to run non-condensing, or in the absence of a relief valve the exhaust steam will blow out through the cooling water pipe and pump valves, thus forcing all the water out of the condenser.

Automatic Vacuum Breakers.—To protect the main engine or turbine from flooding, every jet condenser which depends upon a pump for the removal of the water is, or should be, provided with an automatic vacuum breaker, in case the water removal pump should fail. At the usual rate of flow a jet condenser would be entirely filled with water in a few seconds should the removal pump stop, unless provision be made to break the vacuum and thereby stop the incoming water.

There are numerous types of vacuum breaker depending for their action on the principle of

1. Reduced contact surface.
2. Air admission.
   a. To condenser.
   b. To cooling water pipe.

The reduced contact surface type consists simply of a constricted neck at the upper part of the condensing chamber as shown in fig. 9, which with undue rise of the cooling water causes the condensing surface to rapidly diminish so that it is inadequate to condense the steam, thus causing the pressure to rise within the condenser.

The air admission types consist usually of a ball float, placed either in the condenser proper, or in an adjoining and communicating chamber, and which upon flooding of the condenser, will operate a valve and allow air to enter the condenser chamber as in fig. 10.*

A better arrangement is to allow the air to enter the injection line as in fig. 12.

*NOTE.—A vacuum breaker should not be confused with an atmospheric relief valve which is placed in the exhaust line between the engine or turbine and the condenser to provide a means of escape for the steam in case the condenser become disabled.

Low Level Jet Condenser Construction.—An example of modern low level jet condenser construction is shown in fig. 11.

The course of the condensing water through the condenser is clearly shown.

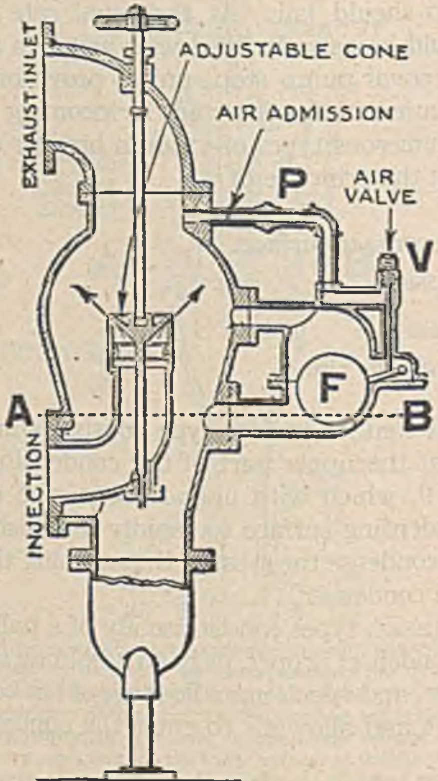


Fig. 10.—Low level combined counter and parallel flow condenser showing typical example of air admission vacuum breaker. **It consists of** a separate and communicating chamber with float operating an air valve which admits air into the condenser. **In operation,** when the water rises in the condenser to the level AB, it lifts the float F, which in turn lifts the air valve V, from its seat, admitting air into the condenser through pipe P, thus breaking the vacuum.

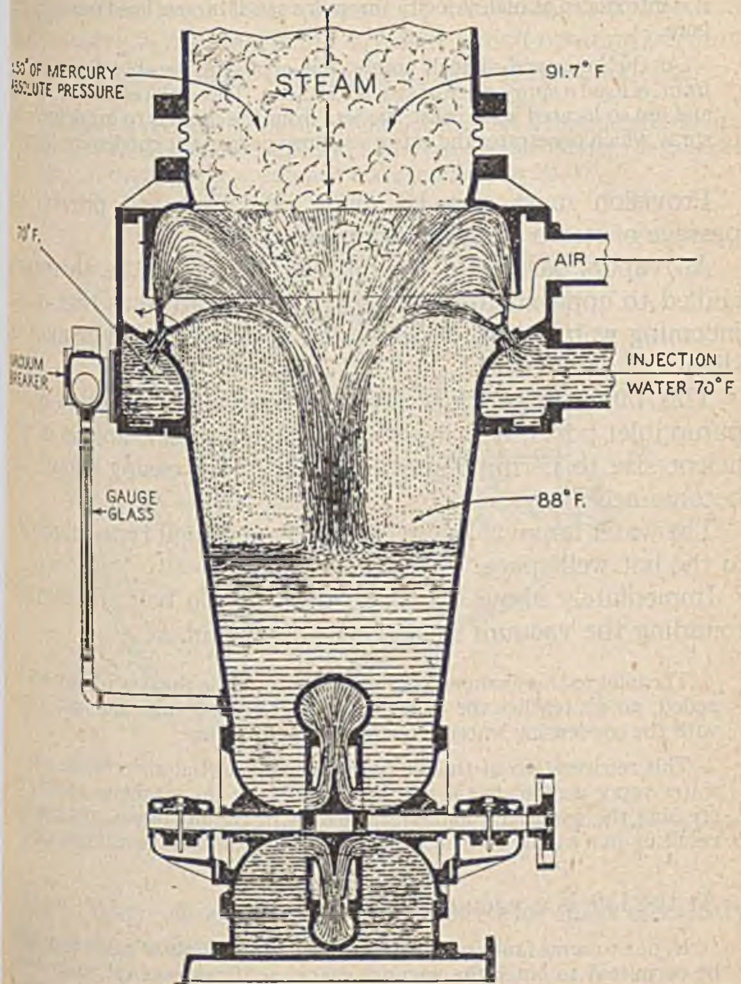


Fig. 11.—Cross section of a modern low level jet condenser with pressure and temperatures indicated to show condition of operation.

It enters a distribution belt surrounding the vacuum space, to which it is introduced at high velocity through a set of bronze lined nozzles of large bore.

At the inner end of each nozzle, but at a considerable distance therefrom, is fixed a spray plate of special design. These plates are of such shape and are so located as to cause the jets from the nozzles to break into a fine spray which penetrates the entire vacuum space of the condensing chamber.

Provision must also be made effectively to prevent the passage of steam to the air chamber.

All vapors passing to the air chamber must be thoroughly chilled to approximately the same temperature as that of the incoming water; and the resistance to the water flow must be slight.

This manhole provides for access to and inspection of the pump inlet ports, wearing rings, impeller, etc., and is of sufficient size to permit removal of the pump casing should this become necessary.

The water removal pump is of the centrifugal type submerged in the hot well space.

Immediately above the water distribution belt and also surrounding the vacuum space is the air chamber.

The inlet to this chamber is so placed that, while the flow of air is not impeded, no air reaches the chamber without coming into thorough contact with the condensing water at its lowest temperature.

This refrigeration of the air produces high partial air pressure and low water vapor content in the mixture handled by the air pump, thereby decreasing the required volumetric capacity of the air pump, and therefore resulting in a saving of space as well as of capital and operating costs.

At the left is a vacuum breaker.

If, due to some fault in operating, too great a supply of condensing water be permitted to enter the vacuum space, or if the removal pump should fail, or if the water discharge line should become clogged, it is a matter of necessity that the vacuum be broken immediately. Otherwise the main turbine may be flooded and much damage done within a very few moments.

To break the vacuum the common device is such that, with a higher water level than normal in the condenser vacuum space, a valve is automatically opened, which opening permits air from the atmosphere to enter the body of the condenser.

This valve is necessarily small and on this account a considerable length of time is required to admit sufficient air into the condenser for the purpose in view. A detail of the vacuum breaker is shown in fig. 12.

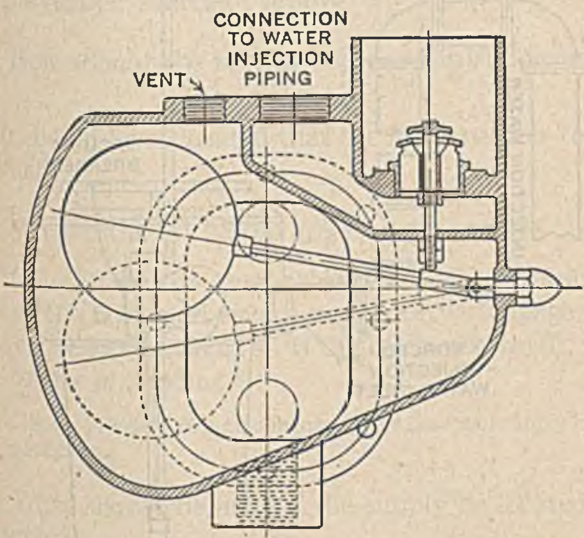


Fig. 12.—Detail of vacuum breaker as installed on the low level jet condenser of fig. 11.

Ques. Where should the air be admitted for most efficiently breaking the vacuum?

Ans. In the water injection line rather than in the condenser. This method of connection is shown in fig. 13.

Two Sources of Injection Water Necessary.—On low level jet condensers two sources of injection water are necessary—one for starting and one for operating.

Ques. What is "forced injection"?

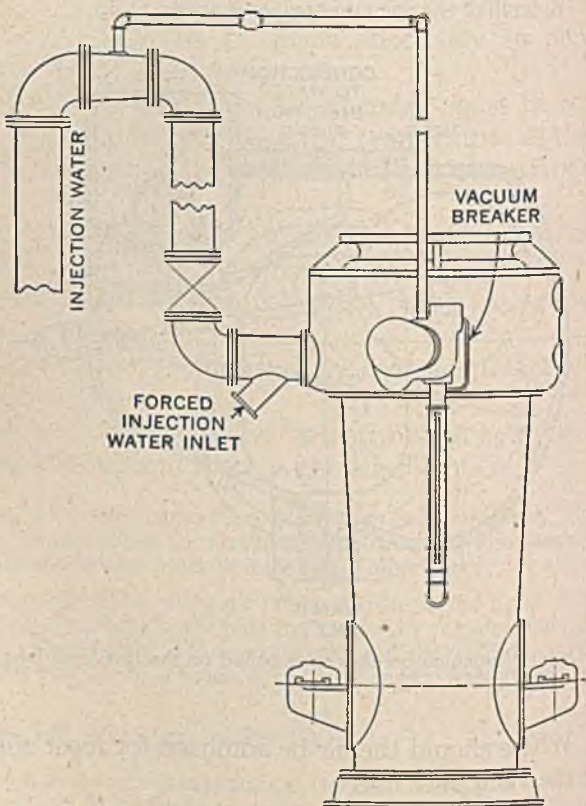


Fig. 13.—Method of connecting vacuum breaker to the injection water inlet line of a low level jet condensing equipment. The connection to the water line should be at its highest point.

Ans. The method employed for starting a condenser by injecting water into the condenser by pressure.

Ques. Why is this necessary?

Ans. Because there is no vacuum in the condenser in starting. As soon as the vacuum is established, water from the starting source is discontinued, and simultaneously connection is made to the regular source of supply.

Ques. How should the regular source of injection water be located?

Ans. It should be so located that the lift is neither too great nor too small.

Ques. What if the lift be too great or too small?

Ans. If the lift be too great, then the slightest obstruction will prevent the flow, and even a momentary stoppage will result in an entire loss of vacuum. If the lift be too small, there is constant danger of flooding.

Such a danger becomes an actual menace if the source of supply be located above the condenser.

Ques. What should be done if the supply be located above the condenser?

Ans. Proper precaution should be taken by installing a cold well, with an ample overflow, located at a suitable distance below the condenser level.

Barometric Condensers.—By definition a barometric condenser is *a high level jet condenser*. The essential details are quite similar to if not exactly the same as those of a low level jet condenser, with the exception that no removal pump is used.

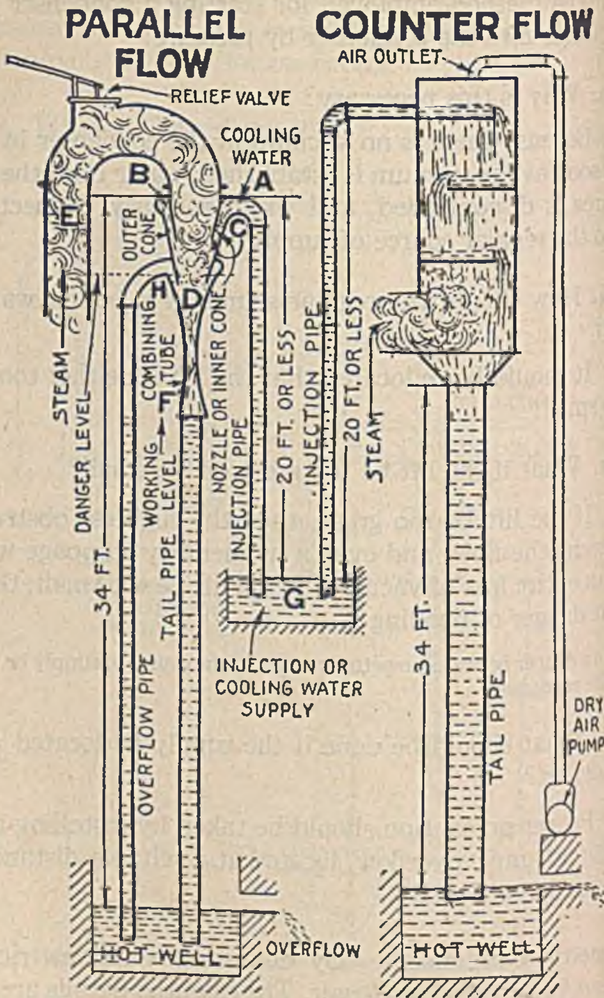


Fig. 14.—Parallel flow barometric condenser or so-called injector condenser.

Fig. 15.—Counter flow barometric condenser or dry air pump type.

Ques. What takes the place of the removal pump?

Ans. A *tail pipe* of comparatively large diameter and over 34 ft. long, attached to and submerged at its lower end in a hot well.

Due to the length of the tail pipe it removes the water from the condenser without the use of a pump. That is, a steam discharge nozzle, combining tube and tail pipe terminating in a hot well. Two annular passages provide openings for the cooling water and exist to relief pipe.

Types of Barometric Condensers.—There are two general classes of barometric condenser:

1. Parallel flow or so called injector type, fig. 14.
2. Counter flow or dry air pump type, as shown in fig. 15.

Figs. 14 and 15 illustrate the principles of operation as explained in the accompanying text.

Parallel Flow Barometric Condensers.—The essentials of this type are shown in the elementary diagram, fig. 14.

Ques. What is its construction?

Ans. It consists of a steam discharge nozzle, combining tube and tail pipe terminating in a hot well. Two annular passages provide openings for the cooling water and exit to relief pipe.

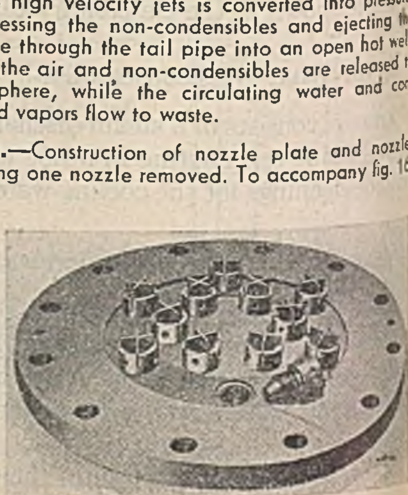
Ques. How does the injection water circulate?

Ans. The injection water enters the condenser at A, and circulates around the annual passage B, falling through the annular space C, between the outer and inner cones forming a moving cone of water D, with a sharp vortex.

Ques. Describe the action of the steam.

Fig. 16—Modern parallel flow or ejector type barometric condenser. **In operation,** water at atmospheric pressure or slightly higher enters near the top of the condenser and is injected into the steam space through a number of efficient hydraulic nozzles, forming high velocity streams or jets of water which converge in a throat piece at the bottom. As the steam is condensed, air and non-condensible gases are entrained to the converging streams and carried to the throat piece, where the kinetic energy of the high velocity jets is converted into pressure, compressing the non-condensibles and ejecting the mixture through the tail pipe into an open hot well. Here the air and non-condensibles are released to atmosphere, while the circulating water and condensed vapors flow to waste.

Fig. 17.—Construction of nozzle plate and nozzles showing one nozzle removed. To accompany fig. 16.



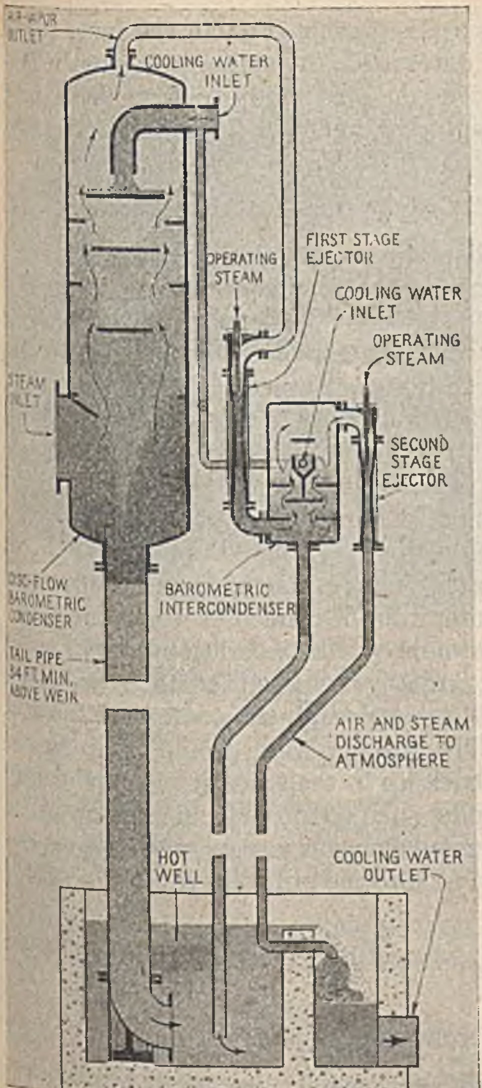


Fig. 18.—Modern counter flow (**disc** flow type) barometric condenser. **In operation,** steam and air or other non-condensibles enter the condenser near the bottom where they come in direct contact with the cooling water flowing down through the condenser by directed, gravity flow. A considerable portion of the steam is condensed immediately while the remaining steam and air are drawn up through successive water curtains by the air ejector. By the time the mixture has penetrated several water curtains, all the remaining steam has been condensed, but the air is hot and saturated with vapor which can be removed only by cooling. The upper curtains, through which the saturated air must pass, are designed to give the best possible cooling and devaporization. After thorough cooling, the air is withdrawn from the condenser by the steam-jet ejector or vacuum pump.

Ans. Steam enters at E, passing through the inner cone or nozzle and meets the water at C, imparting to it what velocity it has on being condensed. This tends to force more water through the annular space between the inner and outer cones.

Ques. What happens to the condensate air and vapors?

Ans. They are brought down through the hollow cone of falling water D, and the air and non-condensate vapors discharged through the contracted throat F, of the combining tube. Since there is parallel flow of the steam and water, air and non-condensable vapors, the latter are forced by their own inertia to enter the vortex which effectively removes them.

After passing the throat F, the combining tube expands to the size of the tail pipe, thus reducing the frictional resistance of the pipe to a minimum. For proper siphoning of the injection water the lift of the supply G, should not be over 20 feet. Now if the supply G, were very near the exhaust A, the pressure tending to force the water into the condenser would be considerably increased. Accordingly if this condition obtain, or if a pump be used and not properly regulated the water would enter A, faster than it could pass through the contracted throat F, and without the overflow pipe the water would fill up the condenser and probably back up in the exhaust pipe and damage the engine. Hence, under such conditions, an overflow pipe as shown should be provided for safety. A relief valve is provided, permitting exhaust into the atmosphere when desired.

Ques. What name is sometimes given to the parallel flow type?

Ans. The ejector barometric condenser.

Modern construction of this type is shown in figs. 16 and 17.

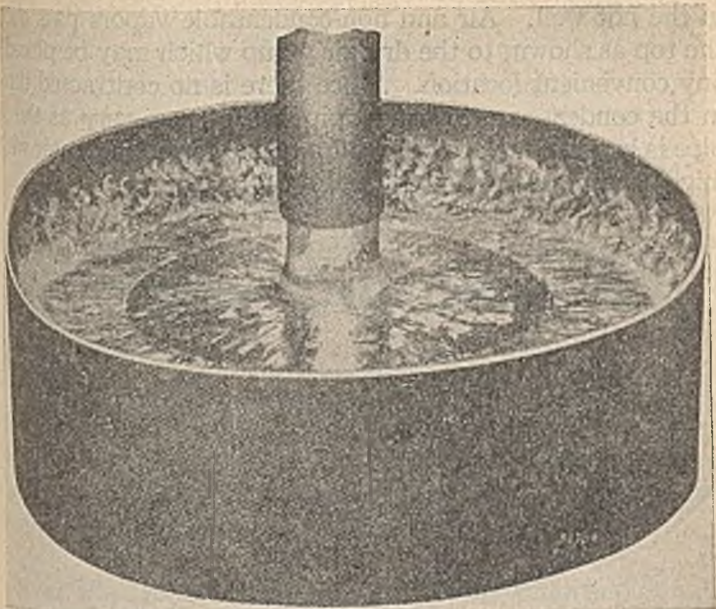


Fig. 19.—Photograph of an operating model illustrating **disc** flow. In accordance with a well-known principle of hydraulics, a low velocity jet of water impinging on a flat surface produces a thin sheet of water without rebound or splashing. As the water continues to spread across the annular space between the impingement plate and the cylindrical wall of the condenser, it forms a thin "disc" or curtain of water, which can be maintained with water quantities ranging from full rating to 1/4 or lower. The thin uniform water curtain reduces the pressure differential required to pull the non-condensibles through the condenser. At the same time a greater surface of cooling water is exposed to the air and non-condensibles to obtain thorough devaporization, thereby decreasing the load on the steam jet ejector or vacuum pump.

Counter Flow Barometric Condensers.—The essential elements of this type are shown in the elementary diagram, fig. 15.

Ques. Describe its operation.

Ans. The injection water passes out to the hot well through the tail pipe, the steam inlet being 34 feet above the water level in the hot well. Air and non-condensable vapors pass out a the top as shown, to the dry air pump which may be placed in any convenient location. Since there is no contracted throa in the condenser an overflow pipe is not necessary as the tai pipe is large enough to carry off any excess cooling water which might enter the condenser.

CHAPTER 22

Surface Condensers

By definition a surface condenser is: *In steam engineering, an apparatus in which exhaust steam is reconverted into water by contact with cooled surfaces.* In other words: *A device for condensing steam in which the steam and cooling water do not come into contact with each other, but are separated by metal surfaces.*

Ques. What name is given to the cooling water and why?

Ans. The *circulating* water because it is "circulated" on the water side of the cooling surface.

Class 2. Condensers

(Classification)

As with jet condensers, there are numerous classes of the surface type and they may be classed,

NOTE.—The surface condenser was first employed by James Watt, but was dismissed by him on account of the cumbrous nature of the apparatus, and its use was not revived until 1835.

1. With respect to conditions relating to the cooling medium, as

   a. Keel
   b. Inboard (commonly called "surface")
   c. Atmospheric

2. With respect to the flow of the steam, as,

   a. Cross
   b. Radial

3. With respect to the flow of the steam and cooling water as,

   a. Parallel
   b. Counter
   c. Return

4. With respect to the degree of baffling,

   a. For water $\left\{\begin{array}{l}\text{one pass}\\ \text{two pass}\end{array}\right.$
   b. For steam $\left\{\begin{array}{l}\text{one pass}\\ \text{two pass}\end{array}\right.$

5. With respect to the arrangements of the tubes, as

   a. Semi dual bank
   b. Dual bank
   c. Radial

6. With respect to the method of circulating the cooling water, as

   a. Keel
   b. Pump
   c. Inductor (scoop)

7. With respect to the cooling water circuit, as

   a. Single
   b. Divided

8. With respect to the method of removing the air, as

   a. Wet
   b. Dry

9. With respect to the pressure in the condenser, as

   a. Bastard (atmospheric)
   b. Vacuum

10. With respect to which side of the cooling surface the cooling water is in contact, as,

   a. Standard
   b. Water works

11. With respect to the shape of the cooling surface, as,

   a. Tubular
   b. Flat (Graham type)

12. With respect to the function of the condenser, as

   a. Main
   b. Auxiliary (booster)

13. With respect to regional application, as

   a. Marine
   b. Stationary (land)

14. With respect to outlets and connections as,

a. Wet

b. Dry

15. With respect to the location of the condenser in marine practice.

a. Outboard (keel)

b. Inboard ("surface")

So Called Keel Condensers.—It is difficult to understand why this type of condenser was ever called a keel condenser unless because of some early faulty installations ignorantly placed low down outside attached to and parallel with the keel. As a matter of fact, when properly installed, the inlet to a keel condenser is nearer the level of the water line than that of the keel.

By definition, a keel condenser is: *A type of marine* **outboard single pass** *surface condenser attached to the side of the hull below the water line.*

Evidently such condenser needs no circulating pump, and owing to the unlimited quantity of circulating water and the rapidity of circulation, due to the motion of the boat, the cooling surface is more efficient than in the inboard type. This is because the rise in temperature of the cooling water is very small in comparison with that in the inboard type.

The application of keel condensers is to small passenger vessels running in shallow water, launches, tugs and lighters.

Keel condensers may be classed,

1. With respect to the internal pressure, as

a. Bastard (atmospheric)

b. Vacuum

2. With respect to the arrangement of the cooling surface, as

a. Single tube
b. Multi-tube

These condensers consist of *a tube* (*or pipe*) *or several tubes* in parallel *attached to the outside of the hull at an elevation between the water line and keel*. Steam is exhausted into this assembly at one end and the condensate and air removed at the other end, or at the end of a "return" pipe usually by a wet air pump.

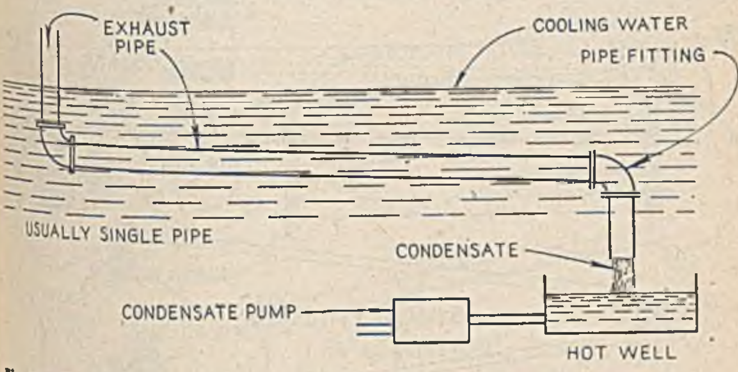


Fig. 1.—Bastard keel condenser. A makeshift nondescript contraption.

Ques. What is the most difficult problem connected with keel condensers?

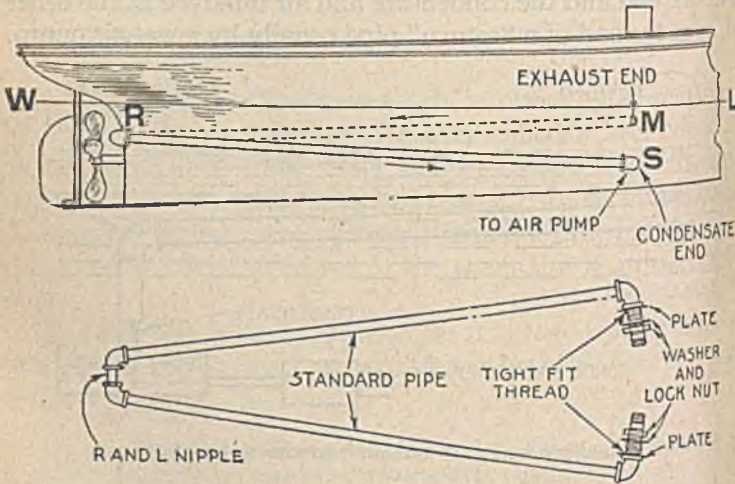
Ans. Drainage.

It must be evident that the cooling surface of any condenser must not be covered up or submerged under *condensate* but exposed so that the steam will come in contact with it, otherwise the steam will not condense. Moreover the steam must condense and be removed with the liberated air in order to get a vacuum.

Bastard or Atmospheric Keel Condensers.—By definition a bastard condenser is: *A type of keel condenser with outlet open to* *the atmosphere (hence the name "atmospheric") and high enough to drain into the hot well,* as shown in fig. 1.

Ques. What is the application of such condensers?

Ans. They are sometimes fitted to canal boats or other nondescript vessels.



Figs. 2 and 3.—Ordinary keel condenser made of standard pipe and fittings. Fig. 2, assembly on boat; fig. 3, construction details. The exhaust should be piped through the hull at **M**, very near the water line so that there will be as much pitch as possible between **M** and **S**, as it should be remembered that thorough and quick drainage is very important in keel condensers.

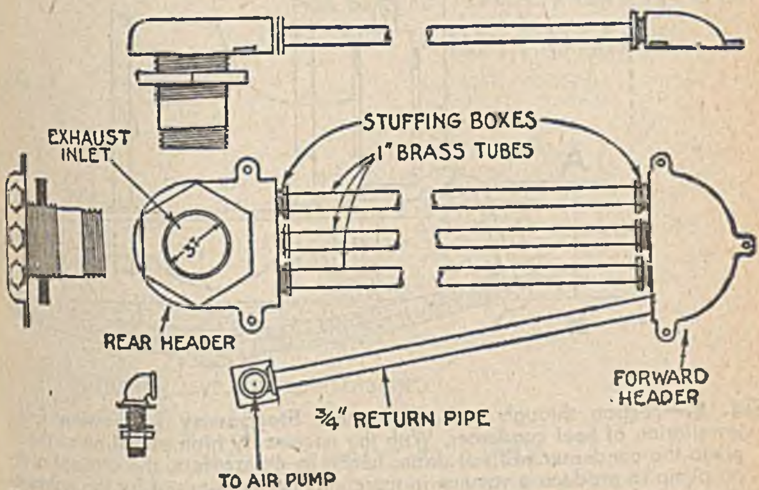
Ques. Why?

Ans. Owing to the ignorance of the owner—in fact such makeshift apparatus operating without vacuum is inexcusable.

Ques. What other reason?

Ans. The frightful fear of some people of spending a couple of dollars for an air pump.

Vacuum Keel Condensers.—The word vacuum is here used to distinguish this type from bastard condensers; however, practically all keel condensers work with vacuum.



Figs. 4 to 7.—Construction details of keel condenser of the author's steamer **Stornoway I.** The heads are provided with stuffing boxes for the tubes, and in addition the forward header has an additional outlet tapped with ¾ pipe to tap for the return to air pump, the return being standard weight brass pipe. A special elbow with flange for fastening to hull, passes through the planking and is secured on the inside by a washer and nut. The return pipe is fastened to the elbow by long screw and lock nut joint.

By definition, a vacuum keel condenser is: *A type of keel condenser, having tubular condensing surface, a return pipe for the condensate, the end of which connects with a wet air pump.*

There are two types: 1, the single pipe as in figs. 2 and 3, and 2, the multi-tube type as in figs. 4 to 7.

The single pipe assembly is a cheap and semi-makeshift outfit but the multi-tube design as shown in figs. 4 to 7, represents the finest construction.

Ques. What material should these condensers be made of?

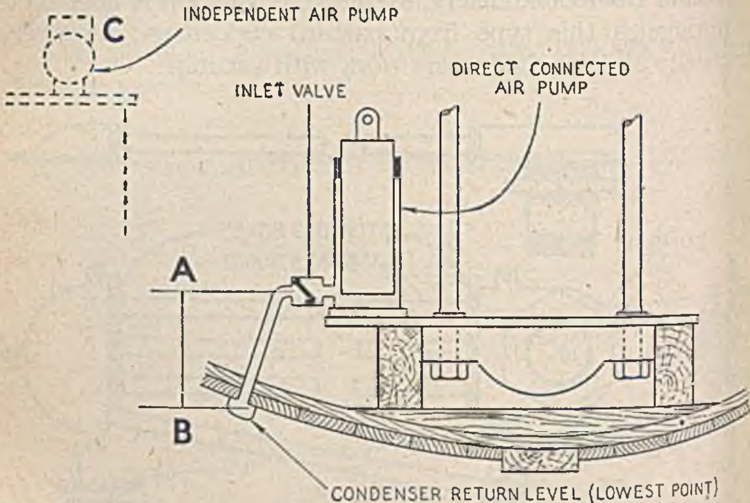


Fig. 8.—Section through author's steamer **Stornoway I,** showing faulty installation of keel condenser. With the necessarily high elevation of the air pump the condenser will not drain, hence **in operation,** the attempt of the air pump to produce a vacuum in the condenser is opposed by the resistance due to a column of water whose height equals the dimension **BA. For example,** if the temperature of condensate be 100° Fahr., the maximum vacuum possible is 28 ins., and further if the column of water in inlet pipe between levels B and A, be 1 ft., it will offer a resistance of .43 lb. per sq. in. which is equal to .87 in., thus reducing the vacuum from 23 to 28 − .87 = 27.13 ins. in the condenser. This vacuum is further reduced by 1, the inefficiency of the pump 2, friction of the condensate en route to pump, and 3, the non-draining feature which causes spasmodic flooding in the return pipe. Hence, **in practice,** if a 28 in. vacuum were aimed at under the above conditions, probably not more than 24 or 25 ins. would be obtained, and accordingly the importance of arranging the apparatus, as shown in fig. 9, so as to reduce these losses to a minimum. The importance of this is further emphasized by an experience of the author with an independent air pump **C** (shown in dotted lines) located at a high level, and connected to the condenser by an inlet line having a multiplicity of elbows. With this faulty rig only about 15 or 16 ins. of vacuum could be obtained. Note how this trouble was avoided in the installation of **Stornoway II,** fig. 9.

Ans. Copper or brass pipe may be used on vessels running in fresh water, but if running in salt water, copper pipes are permissible only if the propeller wheel and the tail shaft with its bearing are made of bronze.

Ques. What will happen otherwise?

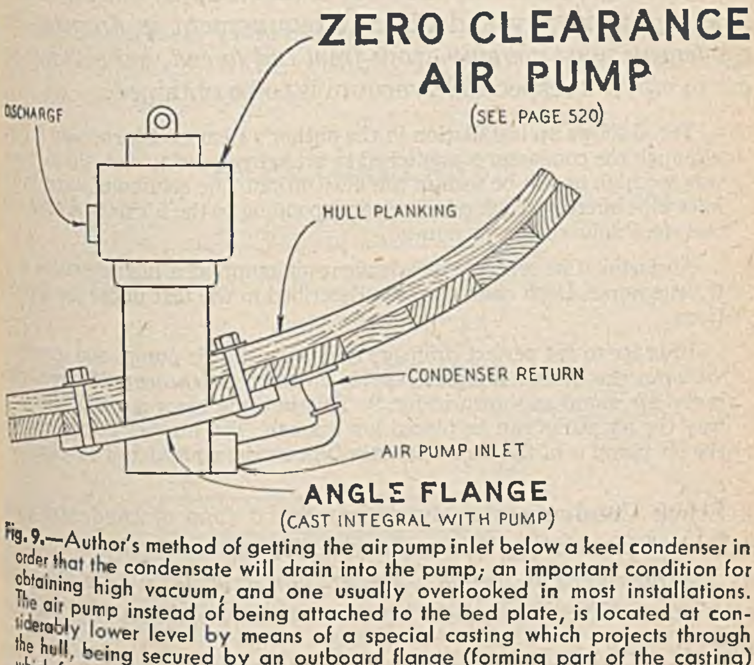


Fig. 9.—Author's method of getting the air pump inlet below a keel condenser in order that the condensate will drain into the pump; an important condition for obtaining high vacuum, and one usually overlooked in most installations. The air pump instead of being attached to the bed plate, is located at considerably lower level by means of a special casting which projects through the hull, being secured by an outboard flange (forming part of the casting) which forms a tight joint as shown. With this arrangement the inlet valve is at the lowest level of the return pipe thus securing the ideal working conditions. Proposed arrangement for author's steamer **Stornoway II.**

Ans. Destructive galvanic action will take place.

Importance of Drainage.—The keel condenser has several advantages for small installations: It is cheap, simple, easily installed and needs no circulating pump as the movement of the boat through the water takes the place of the pump. Moreover, considerably less cooling surface is required than with the inboard type because the rise in temperature of the cooling water is relatively very small.

Again no room inside the boat is taken up by the apparatus. The one necessary and difficult requirement is *drainage—the condensate must not only drain from end to end, but also into the air pump*, if a respectable vacuum is to be obtained.

Fig. 8 shows an installation in the author's steamer *Stornoway I*. Here although the condenser was pitched to drain from end to end, the air pump was too high as can be seen in the illustration. The column of water in the inlet pipe offered a back pressure corresponding to the lift from A to B, and therefore reduced the vacuum.

An installation with an independent air pump at a high elevation as at C, was worse. Both cases are fully described in the text under the illustrations.

In order to get perfect drainage right into the air pump, and maximum vacuum, the author designed for his steamer *Stornoway II* a semi-outboard air pump as shown in fig. 9. This in some cases is about the only way the air pump can be placed low enough. The interior construction of the air pump is of the zero clearance type as shown in detail on page 520.

Scoop Condensers.—By definition, *a type of condenser with a flow of circulating water induced through an enclosed chamber by the movement of the boat, rather than externally as with a keel condenser*. The distinction between the two types is shown in figs. 10 and 11.

The adaptation of the scoop condenser is for fast vessels particularly on fast yachts and destroyers. The speed of the vessel creates enough pressure in entering the scoop to force the water through the condenser at a speed comparable with that imparted by a circulation pump.

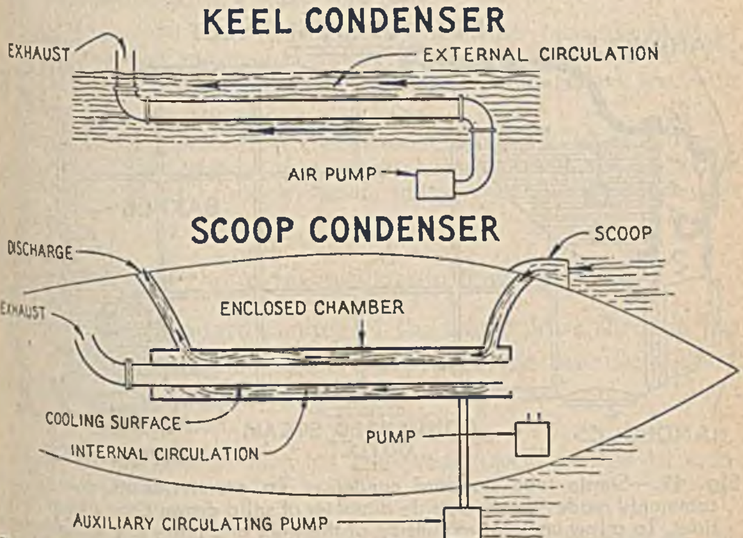
Ques. What must be provided on these condesners?

Ans. A small circulating pump as shown in fig. 11.

Ques. What for?

Ans. To circulate sufficient water through the condenser when the vessel moves slowly as in port.

Inboard Surface Condensers.—The name *inboard* is here applied to distinguish the ordinary form of surface condenser from the keel or scoop types.



Figs. 10 and 11.—Difference between keel and scoop condensers.

In general the surface condenser is *that form in which the steam to be condensed is on one side of the cooling surface and the cooling water on the other side*, as distinguished from the jet, or direct contact condenser.

There are many different types of condenser and they may all be classed as either:

1. Wet, or
2. Dry.

Both types have in common,

1. An enclosed chamber, in which is:
2. The cooling surface.
3. Inlet opening for the steam.
4. Inlet and discharge openings for the circulating water.

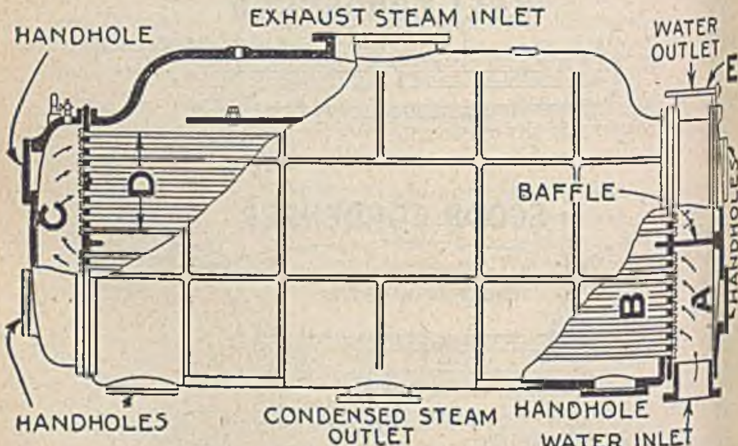


Fig. 12.—Single tube standard condenser. **In construction,** the tubes are commonly made ¾ inch outside diameter of solid drawn brass tinned on both sides. To allow unequal expansion of the shell and tubes screwed glands and stuffing boxes are provided; these are packed with cotton cord or corset lacing. The tube sheets or plates to which the ends of the tubes are attached are of brass and usually from 1.1 to 1½ times the diameter of the tube in thickness. The type of joint determines the thickness. With screwed glands a thinner plate may be used than when the packing extends through it. Usually the tubes are spaced in a zigzag manner, pitched from 1.5 to 1.7 of their diameter on centers. The tubes, plate, ferrules, nuts and washers should be of brass to prevent corrosion. The shell is generally made of cast iron; no wrought iron should be used when the parts are exposed to the distilled water.

In addition the wet condenser has:

1. A common opening for the discharge of the condensate and air. The dry type has:
1. An opening for the discharge of the condensate, and
2. An opening for the discharge of the air.

Typical Wet Condenser Construction.—The typical construction will be clear from fig. 12.

In general the steam chamber consists of a cast iron cylinder, to each end of which is attached a brass "tube plate".

To the two tube plates are secured the condenser tubes, generally by means of threaded ferrules and packing. Attached to each tube plate is a water chamber to or from which the condensing water enters or leaves the tubes.

To the steam chamber there are two essential openings (three in the dry type), one for the admission of the steam to be condensed, and one for the discharge or removal of the condensate and air and other non-condensible vapors.

Ques. Describe the water and steam flow.

Ans. In the standard condenser the water flows through the tubes while the exhaust steam flows over the outside surface of the tubes and is there condensed.

Ques. Describe the flow in the water works type of condenser.

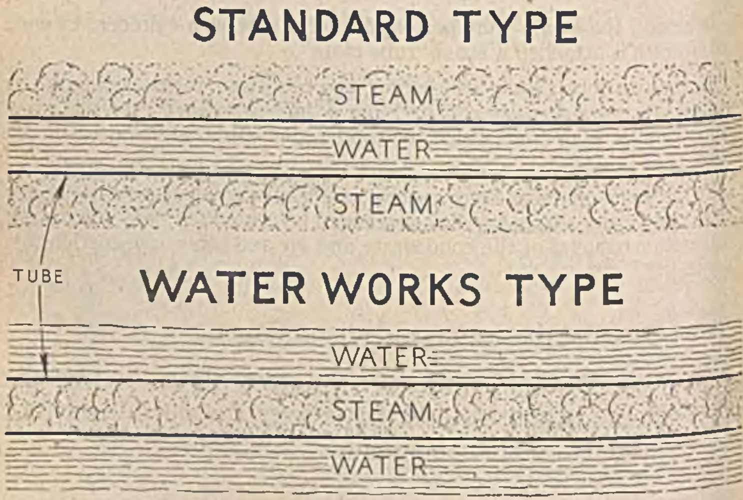
Ans. In this type, the flow is reversed with respect to the tubes, that is, the water flows over the outside of the tubes while the steam flows through the tubes, and is condensed therein.

The latter type is a special type and is used for the special conditions usually found only in water works pumping stations. These differences are shown in figs. 13 and 14.

Ques. For ordinary practice what is the advantage of water flow through the tubes and steam flow outside the tubes?

Ans. In this arrangement, as shown in fig. 15, the heat is drawn from every direction, indicated by the arrows, and absorbed by the rapidly moving cooling water.

Ques. Explain the advantages of the surface condenser as compared with jet condensers.



Figs. 13 and 14.—Standard and water works condensers. Note reversal of steam-water flow arrangement.

Ans. The surface condenser permits the use of impure or salt cooling water in marine practice without bringing same into contact with the condensate, hence the condensate is available for use as boiler feed.

For this reason the only type of condenser that can be used for marine service on salt water where the condensate is to be used as feed water, is the surface condenser.

Pumps for Wet Condensers.—The usual assembly of pumps for a wet condenser consists of:

1. Wet air pump
2. Circulation pump
3. Steam unit

These are arranged in tandem with the steam cylinder between the air and circulating pump all connected to a common

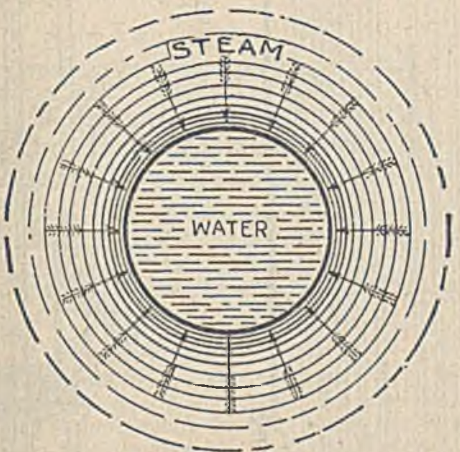


Fig. 15.—Section through condenser tube illustrating the action of condenser with cooling water passing through the tubes and steam outside. The almost universal practice is to circulate the water through the tubes as here shown.

piston rod. On the top of this assembly is mounted the condenser as shown in fig. 16.

It must be quite evident that this arrangement is not only compact, but has the very necessary feature of the condenser so that the condensate simply drops by gravity into the air pump.

As already explained under Keel Condensers, the latter condition is necessary for obtaining the best vacuum within the range of the system.

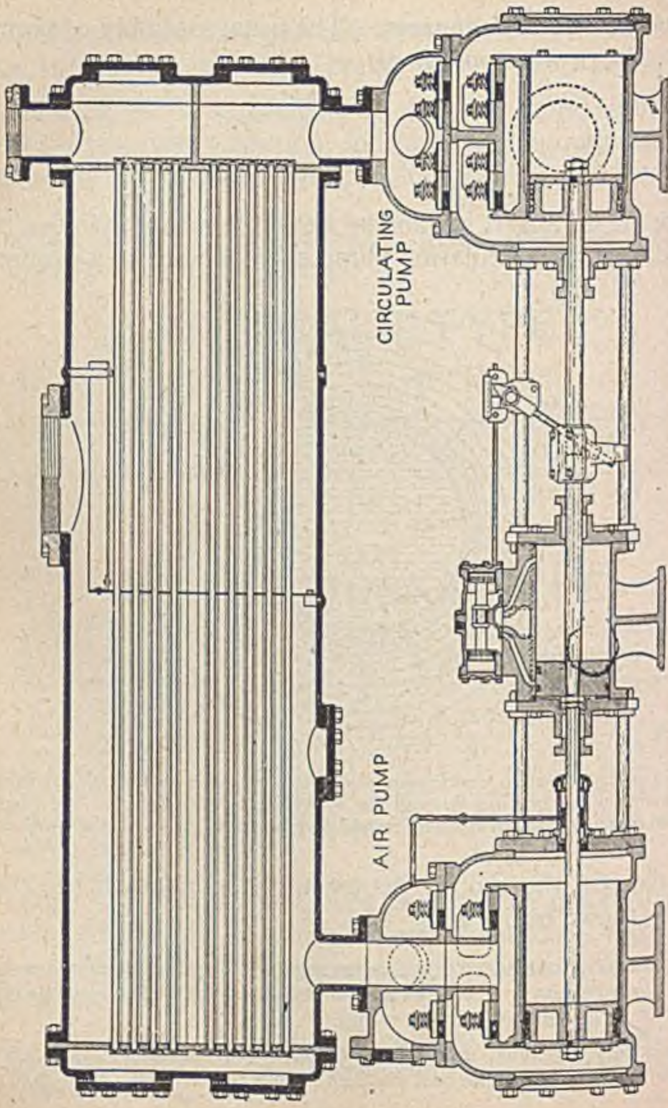


Fig. 16.—Assembly of condenser and wet air pump and circulating pump.

Dry Condensers.—The dry system is generally used for large units. These condensers with respect to essential features are the same as the wet type, except for the extra outlet or outlets for the dry air pump, and the fact that the condensate outlet or outlets need not be as large as in the wet type. The term "dry" condensers is a misnomer as the condenser inside is anything but dry regardless of the type of pump.

Fig. 17 shows a two pass dry type condenser, a type adapted to high vacuum.

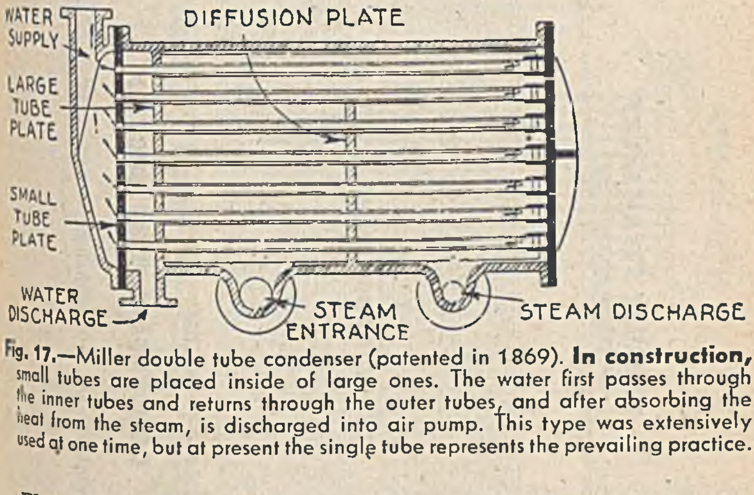


Fig. 17.—Miller double tube condenser (patented in 1869). **In construction,** small tubes are placed inside of large ones. The water first passes through the inner tubes and returns through the outer tubes, and after absorbing the heat from the steam, is discharged into air pump. This type was extensively used at one time, but at present the single tube represents the prevailing practice.

Water Works Condensers.—These condensers may be designed for either *wet* or *dry* operation, depending upon the requirements. The essentials of a small water works condenser are shown in fig. 19.

Ques. How is a water works condenser connected up?

Ans. They may be connected in either the inlet or discharge lines of the main pump, as indicated in figs. 20 and 21.

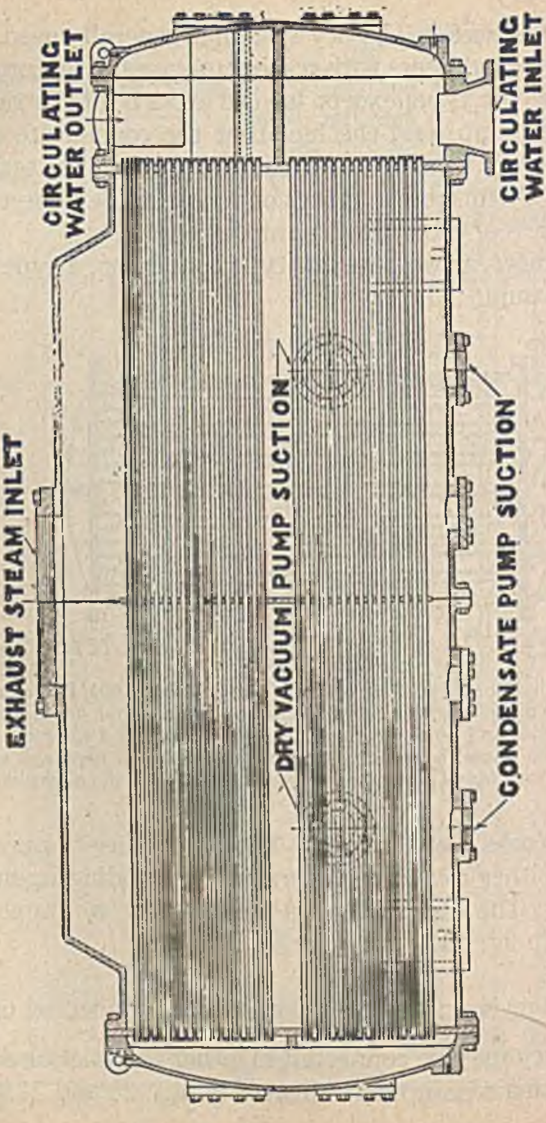


Fig. 18.—Two (water) pass high vacuum dry condenser.

Ques. What is the advantage of connecting the condenser in the inlet line?

Ans. It avoids the high pressure in the discharge line and accordingly the condenser itself may be of lighter construction.

Ques. When is it necessary to connect the condenser in the discharge line?

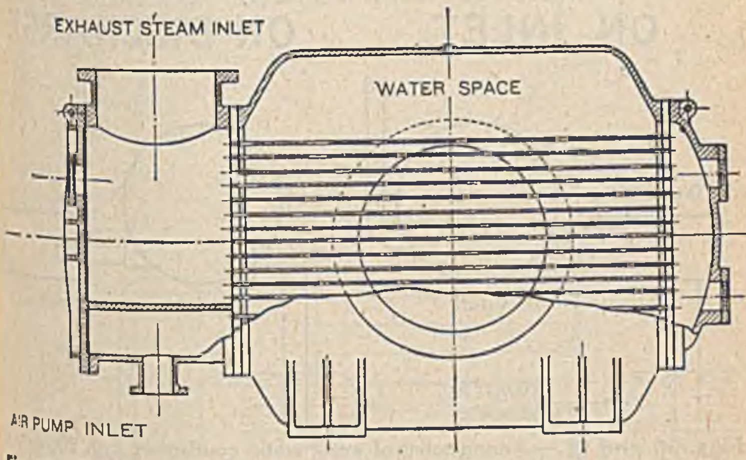


Fig. 19.—Small water works wet condenser.

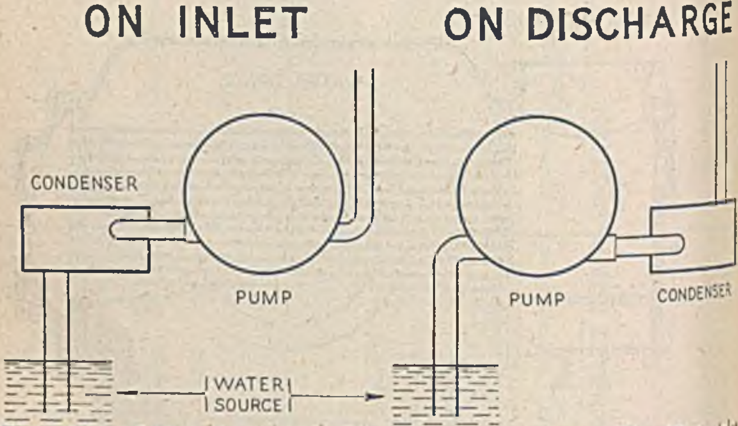
Ans. When there is no space available for the condenser between the pump and its water supply.

Evaporative Condensers.—In this type condenser *the cooling surface is kept cool by the evaporation of the cooling water which is sprayed over the outer surfaces of the tubes, the evaporation usually being increased by a fan blast.*

Less circulating water is used in this type than in the case of surface condensers.

The condensation takes place and the condensed water and air are extracted by auxiliary pumps in the same way as in the surface condensers.

Condenser Tubes.—The material for condenser tubes most commonly employed is muntz metal for fresh water and admiralty metal for salt water, pure copper being used only for exceptional conditions.



Figs. 20 and 21.—Arrangement of water works condensers. Fig. 20, on inlet line; fig. 21, on discharge line.

Ques. What should be considered in selecting the proper metal for the tubes?

Ans. Cost, life and thermal conductivity.

The thermal conductivity being within a range of 5% is ordinarily negligible.*

* NOTE.—The two metals having the greatest thermal conductivity are silver and copper. The metals chiefly used for condenser tubes are copper, admiralty metal (70% copper, 29% zinc and 1% tin) and muntz metal (60% copper and 40% zinc). Compared with silver the thermal conductivity of these three metals is about 90%, 88% and 86% respectively.

Ques. What is the latest practice with respect to tube diameter, length and thickness?

Ans. The smallest outside diameter of tube usually employed is ⅝ inch and the largest 1 inch. Lengths run approximately from 4 to 22 feet. The thickness ordinarily varies from No. 16 to No. 20 *B.w.g.*

Ques. Upon what does the thickness of tube depend?

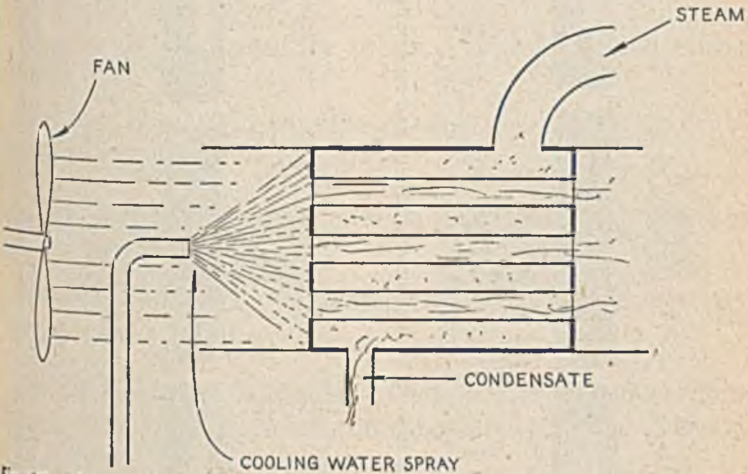


Fig. 22.—Evaporative condenser.

Ans. Upon the length; the larger the tube the greater its thickness to resist bending between points of support because of the weight of the tube and water within it.

Ques. Why are tubes made No. 18 *B.w.g.* thickness?

Ans. Because the useful life of a tube usually does not depend on its resistance to actual wear and therefore the metal

Fig. 23.—Method of securing tube to tube plates by ferrules and especially prepared corset lace packing at both ends of the tube.

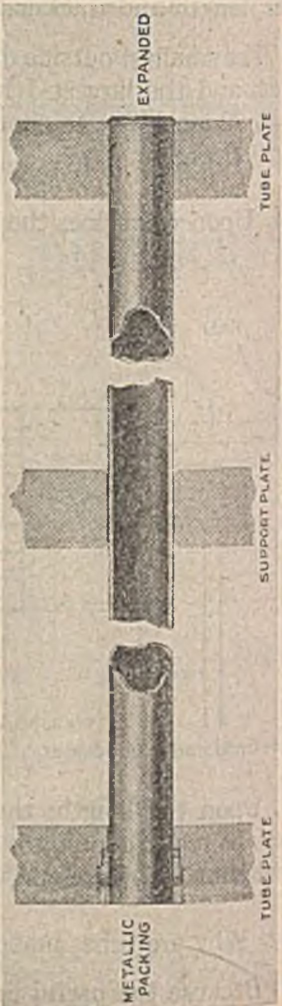


Fig. 24.—Method of securing tube to tube plates by expanding at one end of the tube and packing at the other with metallic packing and expansion rings, but with no ferrules.

thickness is not influenced by this consideration but by the cost of the thicker tube metal as compared with the cost of support plates.

Tube Plates or Sheets.—These are ordinarily made of muntz metal rolled 60% copper and 40% zinc, or of the admiralty mixture. Sometimes brass is used. The thickness of the sheet ranges from 1.1 to 1½ tube diameters. Tube plates must be well stayed to prevent collapse. They are held to the shell by alternate collar bolts.

Ques. How long should be the free or unsupported length of tube?

Ans. Not over 100 times the outside diameter of the tube depending upon the tube thickness.

Ques. What precaution should be taken in drilling the tube holes in the plates?

Ans. They should be well chamfered so as to prevent cutting the tubes.

Tube Connections to Plates.—There are several methods of making tight joints between the tubes and the plates, as by:

1. Stuffing box
2. Expanding.

Ques. What is the important point to be considered in any method?

Ans. Provision for expansion and contraction.

Ques. What other requirement is essential?

Ans. The joints must be both water and air tight.

Ques. Describe the stuffing box method.

Ans. It consists in the use of ferrules and packing at both ends of each tube, as shown in fig. 23.

The tube plates are bored with a slight clearance for the tubes. They are then counterbored to provide a shoulder for the packing, and the counterbored holes are next threaded for the ferrules.

Ques. How are the tubes installed?

Ans. Each tube is first put in place in both tube plates then the packing inserted and the ferrule screwed on.

The operation is about the same as packing an ordinary stuffing box.

Ques. Of what does the packing consist?

Ans. The standard packing is corset lacing impregnated with pure paraffin.

Ques. What other kinds of packing are used?

Ans. Fibre and metallic.

Ques. What trouble is sometimes encountered with packed joints?

Ans. Creeping of the tubes due to expansion and contraction.

Ques. What prevents the tube from creeping too far?

Ans. On the inner face of each ferrule is a shoulder as shown in fig. 23, which effectively prevents displacement of the tube, but the total length between the shoulders of each pair of ferrules is such as to provide ample clearance for the expansion of the tube when it is heated by the exhaust steam within the condenser.

Ques. Describe the combined expanded and packed method.

Ans. The tubes may be packed and ferruled at one end only and be expanded into plain holes in the tube plate at the other end, as shown in fig. 24.

Ques. What are the features of this method?

Ans. It cuts in half the number of joints where probable leakage may take place.

Ques. Describe a third method of making tube joints.

Ans. The tubes are sometimes expanded into the tube plates at both ends and provision is made for expansion and contraction by placing a suitable expansion joint between one of the plates and condenser shell.

Ques. What is a plate with this expansion joint called?

Ans. A "floating" plate.

Ques. When is this method used and why?

Ans. When the condensing water is of extremely bad quality because it reduces chances of joint leakage to a minimum.

This method is quite expensive.

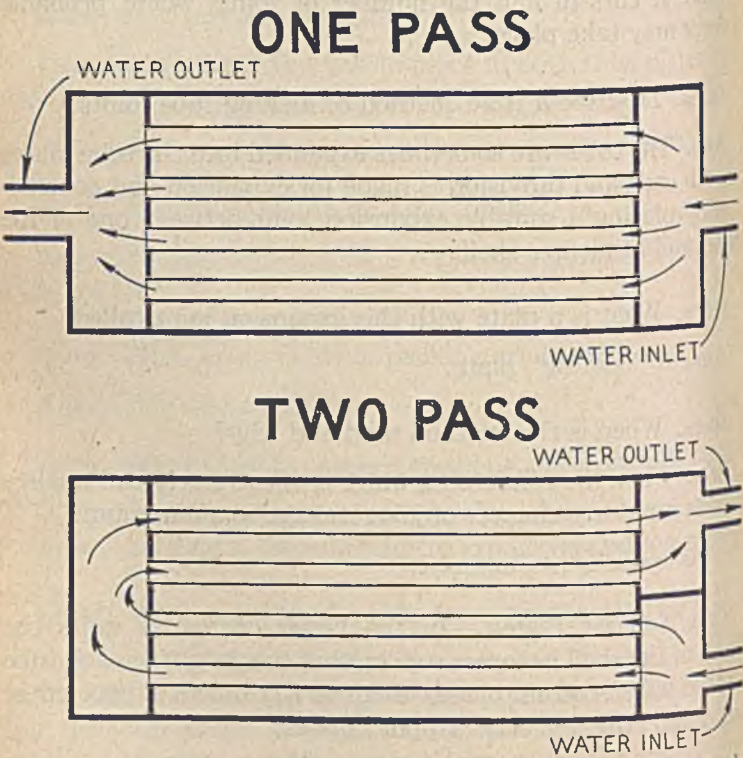
Tube Support Plates.—In the larger condenser units the length of the shell becomes so long that the weight of each tube and the water therein causes them to sag unless supported at proper intervals by tube support plates.

These supporting plates are usually of cast iron and should be spaced about 60 to 70 tube diameters. In no case should the spacing exceed 100 diameters.

The holes should be ⅛ in. larger than the diameter of the tubes and should be well chamfered to prevent cutting the tubes.

They are usually made of cast iron for merchant vessels or brass or galvanized steel for naval vessels.

Water Passes.—Various conditions of operation determine the choice of the water circuit in its passage from inlet to out-



Figs. 25 and 26.—One and two pass condensers. It should be noted that the word "pass" relates to the water circuit and not the steam circuit. Steam circuit not shown in these diagrams.

let of the condenser. According to the arrangement of the water circuit, condensers may be classed as:

1. Single pass
2. Two pass,
   etc.

The flow in these two types and method of obtaining it is shown in figs. 25 and 26.

Ques. How does the water flow in the single pass condenser?

Ans. It enters all the condenser tubes from the water entry end of the condenser, passes through the tubes to the water box at the other end; and is thence discharged after having been heated during its one way passage through the tubes.

Ques. How does the two pass arrangement work?

Ans. The entry water box is divided into two sections. The circulating water is admitted to one of these sections, passes to the second water box, there enters the remaining tubes, is thus returned to the other section of the entry water box and is thence discharged.

Ques. How are more passes obtained in construction?

Ans. By dividing the water box into more sections the water can be made to traverse the condenser any number of times.

Ques. What chiefly determines the number of passes?

Ans. The quantity of water available; economy of construction of the condenser; terminal temperature of the water as related to the desired vacuum.*

Steam Flow.—Similarly with the guidance of the water, the flow of steam may be:

1. Single flow
2. Divided (dual) flow
3. Multi-flow
4. Cross flow

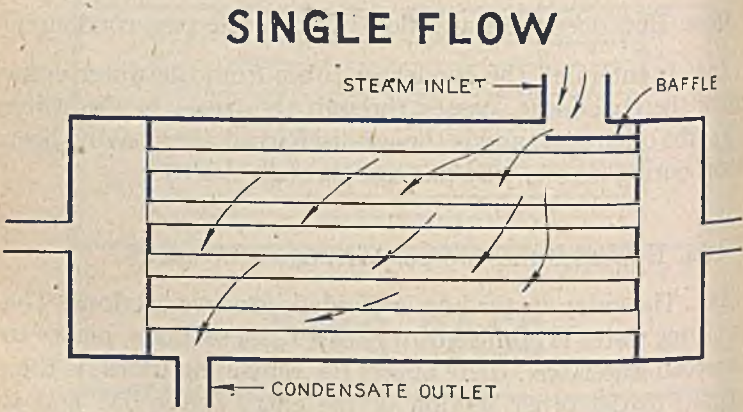


Fig. 27.—Single flow condenser. The steam finds its air path from steam inlet to condensate outlet with result that the condensing process and vacuum obtained is not as efficient as might be otherwise. Arrangement sometimes used on small condensers.

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* NOTE.—It is evident that, other conditions remaining the same, the greater the quantity of water the greater the water velocity. Likewise, and also with other conditions remaining the same, the greater the number of passes into which the tubes are divided the greater the water velocity. Thus the two extremes consist, first, of a large water quantity combined with multiple passes, and second, of a small water quantity and a single pass. In actual practice the tendency is toward an average between these two extremes.

NOTE.—The water works condenser is a good example of the single pass type, see fig. 19. Two pass construction is shown in fig. 18.

This leading of the steam is obtained by suitable baffles. Some of the various arrangements are shown in figs. 27 to 29.

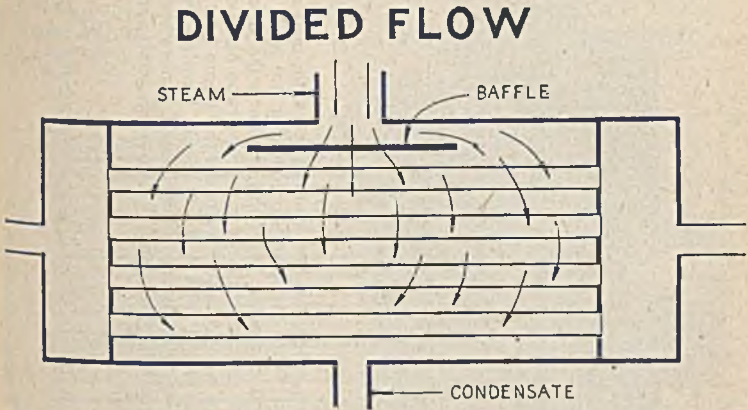


Fig. 28.—Divided flow condenser. Note placement of baffle plate causing steam to flow toward the two ends, then downward and converging toward the condensate outlet.

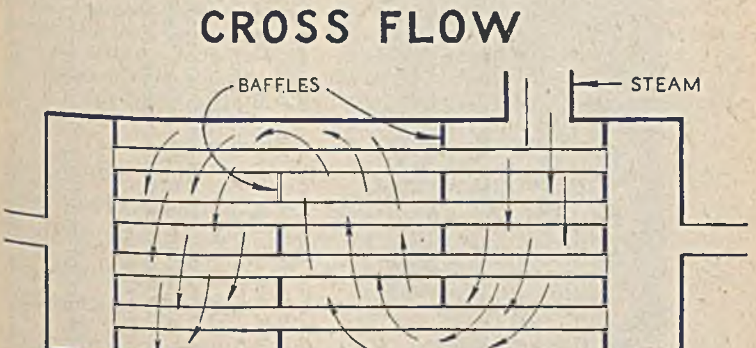


Fig. 29.—Cross flow condenser. Evidently any number of lanes may be obtained by baffling. The object of this arrangement is to completely control the steam flow, thus preventing short circuiting. Moreover the accumulation of condensate is discharged to the bottom of the condenser as formed in each alternate lane.

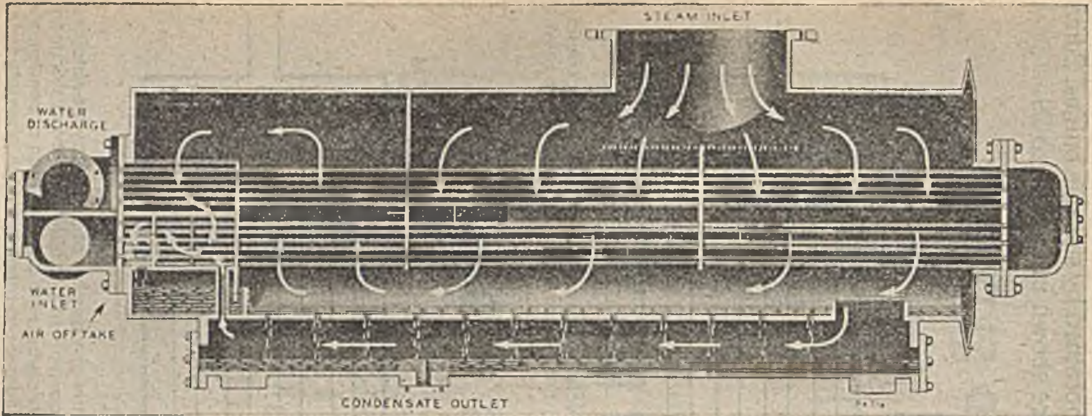


Fig. 30.—Typical cross flow condenser as constructed showing graduated flow path to the multi-pass air cooler located in the lower half of the tube bank at the inlet end. Note the long reactangular hot well, with provisions to assure hot air-free condensate. The distinguishing characteristic of this design is the arrangement of tube support plates which direct the flow of steam back and forth across a shallow tube bank along the entire length of the condenser. An additional support plate is placed directly under the steam inlet, and the lower rows of tubes are spaced on closer centers to prevent the high velocity steam blasting through at this point and so distribute it to the ends of this section. Also, a perforated plate is used, if necessary to protect the top row of tubes from direct steam impingement which is apt to be quite severe in small units due to the small turbine exhaust.

Ques. Aside from guiding the flow of steam, what is the object of placing a baffle at the steam inlet?

Ans. It prevents the steam eroding the outer row of tubes.

Ques. How are baffle plates usually attached to the shell?

Ans. By welding.

Ques. What is the object of the various baffling arrangements?

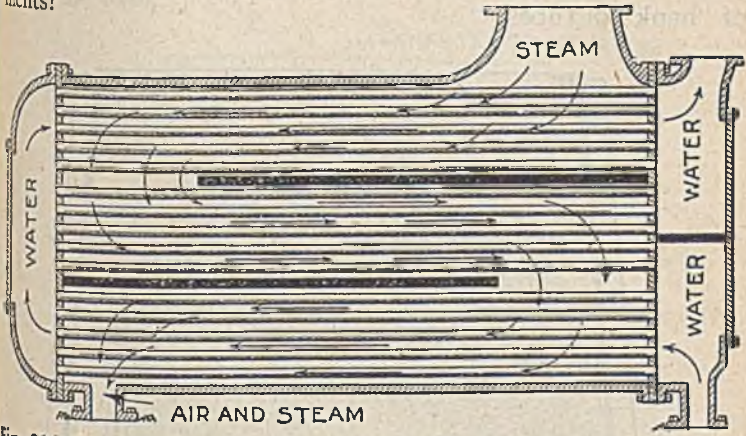


Fig. 31.—**Counter-flow** condenser. **In this arrangement** the steam flows parallel with the cooling water as indicated by the arrows. The baffle plates cause the entering steam to flow in a direction parallel with the upper condensing tubes; when striking the end of the condenser body, the direction of flow is reversed; this operation being repeated as often as there are baffle plates.

Ans. Baffle plates are located and arranged to provide against short circuiting of the steam flow to the air pump outlet and to ensure the refrigeration of all vapors passing to the air pump; also to minimize the dripping of condensate from the upper tubes to the lower.

Ques. What other provision is made to assist obtaining these results?

Ans. Troughs are introduced on the baffle plates leading to suitable channels for draining the collected condensate direct to the hot well.

Tube Banks.—In any assembly of closely spaced tubes, the entering steam readily comes into contact with the outer tubes, but not so effectively with the less accessible tubes near the center. This becomes more pronounced the larger the assembly of "bank" of tubes.

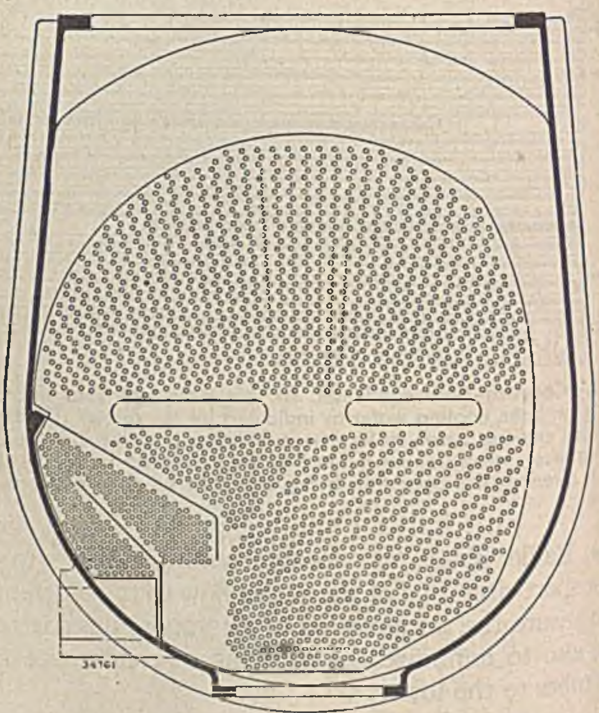


Fig. 32.—Semi-dual bank surface condenser tube plate arrangement.

Owing to the very large sizes of condensers now being built to avoid this steam flow defect the tubes are arranged in two or more banks with *lanes* between, rendering the cooling surface more accessible to the steam. Of the various arrangements, attention is called to

1. Semi-dual bank
2. Dual bank

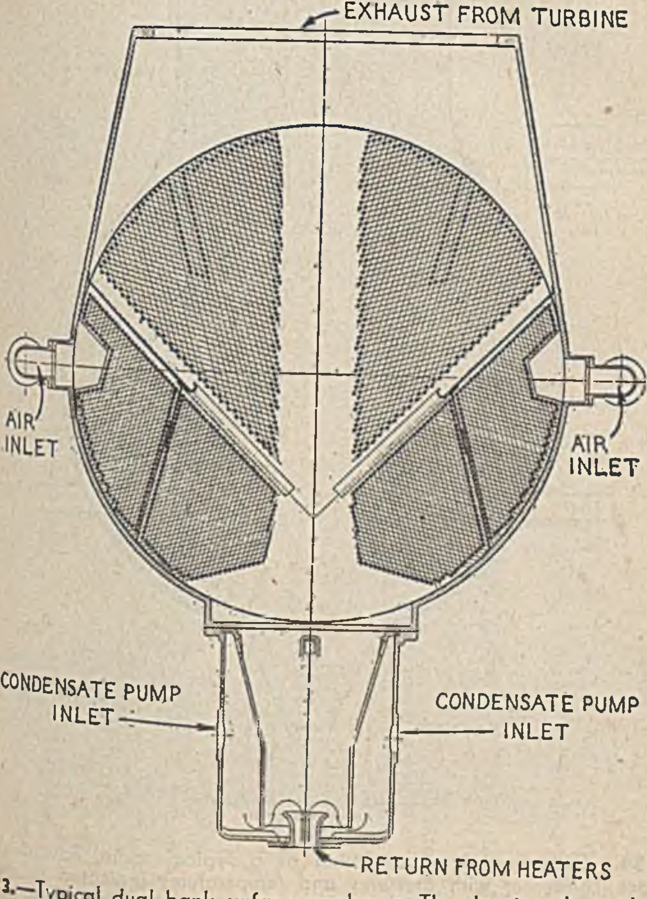


Fig. 33.—Typical dual bank surface condenser. The drawing shows the tube arrangement and a de-aerating hot well chamber.

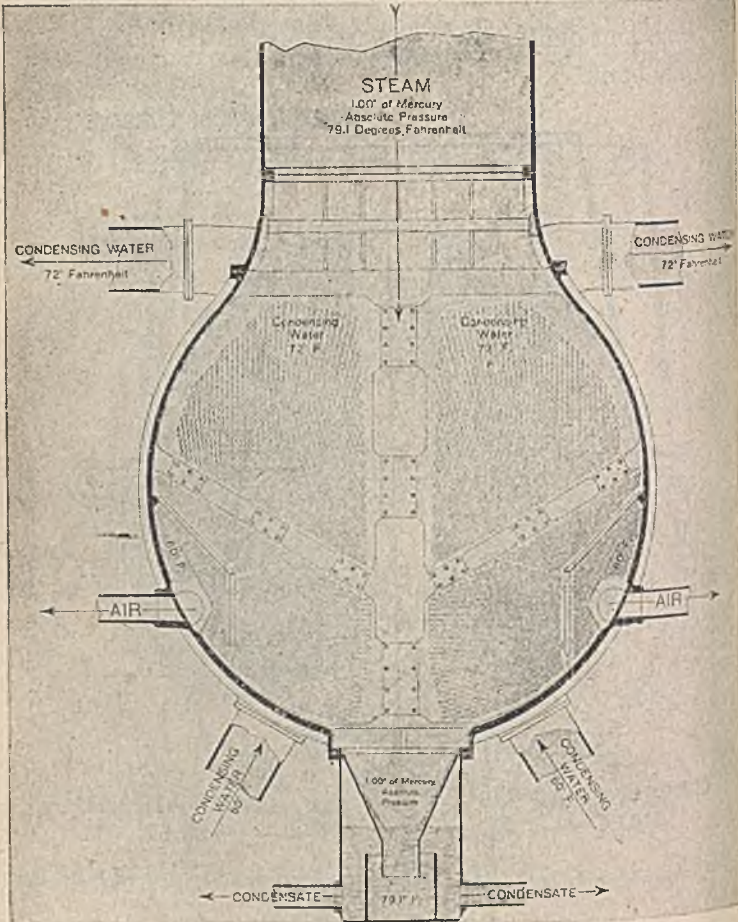
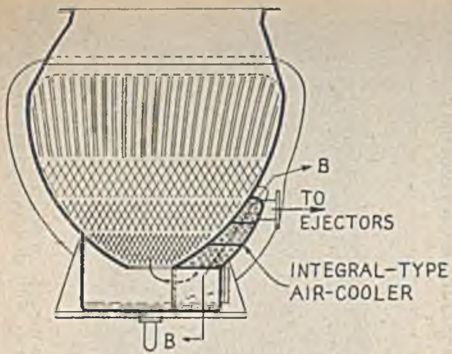
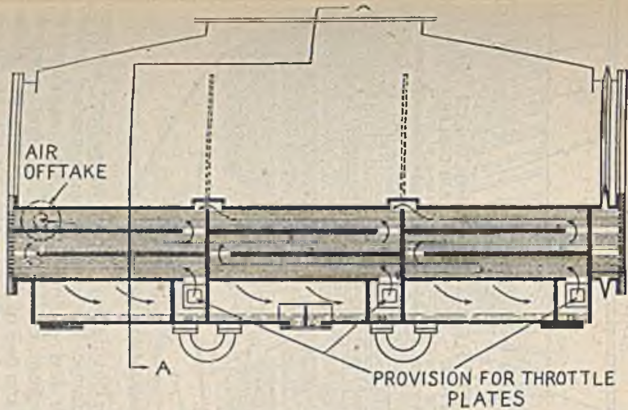


Fig. 34.—Diagrammatic cross-section of a typical radial flow dual bank surface condenser with pressures and temperatures indicated to show an excellent condition of operation. Note the condensate and vacuum temperatures and the partial air pressure at the outlet to air pump.





Figs. 35 and 36.—Multi-pass counter current air cooler built integral with main condenser shell. It is similar to the so called external cooler and the provisions for throttling vapor from the individual sections where necessary.

as shown in figs. 32 and 33 respectively.

Tube Spacing.—With respect to the spaces between the tubes in a bank and the spaces between the banks, they are known as:

1. Minor steam lanes
2. Major steam lanes

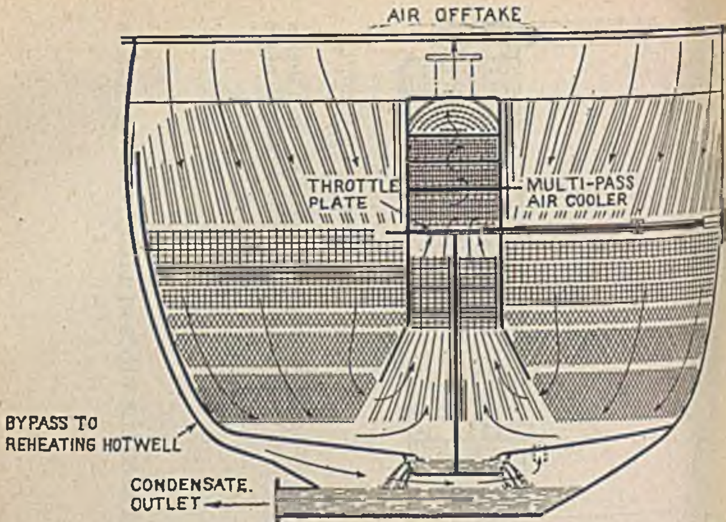


Fig. 37.—Condenser designed for double flow turbine where head room prevents the use of a suitable steam dome. The steam from a modern double-flow turbine exhausts in two more or less distinct paths. **The condenser consists of:** two heart-shaped sections built side by side into a single shell. Each section receives its steam from one of the two paths of the turbine exhaust. A single air cooler serves both sections. The cooler is of the multi-pass type.

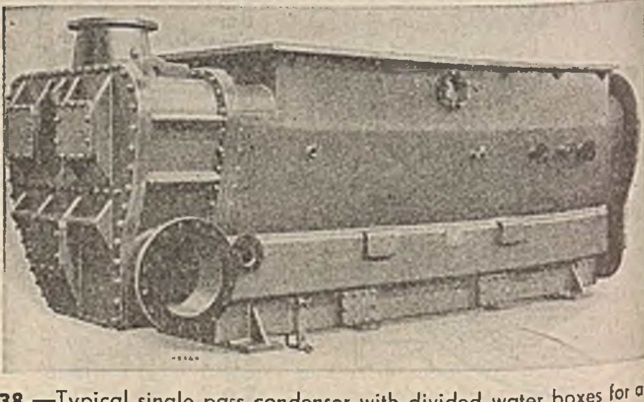


Fig. 38.—Typical single pass condenser with divided water boxes for a 3,500 kw. unit.

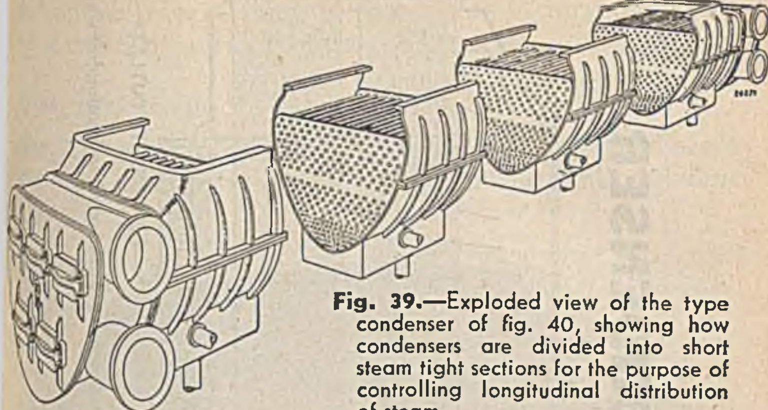


Fig. 39.—Exploded view of the type condenser of fig. 40, showing how condensers are divided into short steam tight sections for the purpose of controlling longitudinal distribution of steam.

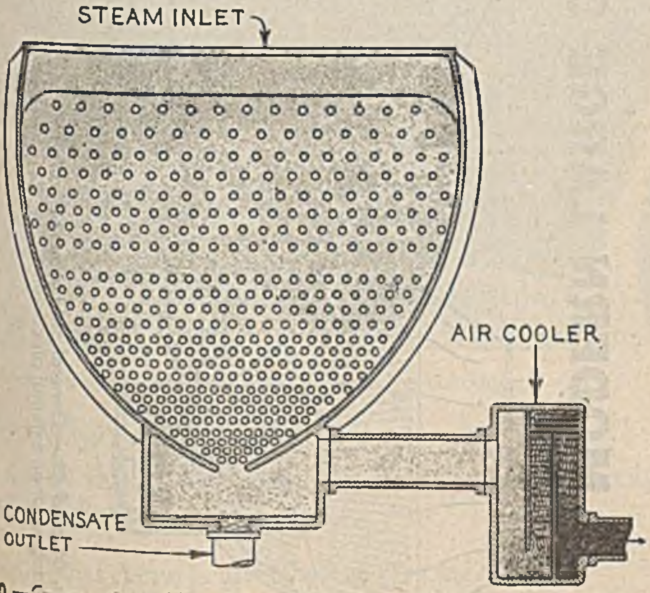
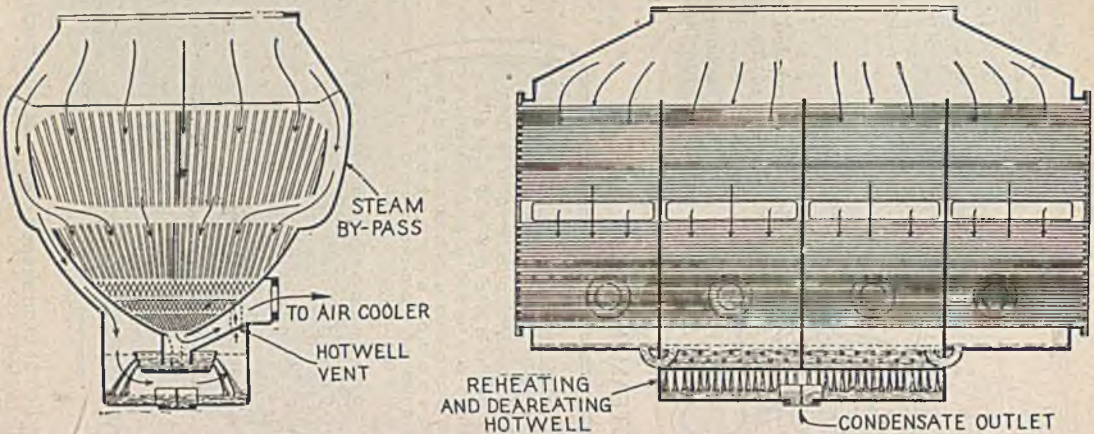


Fig. 40.—Cross section of heart shaped condenser with graduated tube spacing and multipass air cooler.

MODERN LARGE CONDENSER



Figs. 41 and 42.—General arrangement of a large condenser with steam by passes and re-heating hot well arranged for an external air cooler.

The proper proportioning of these lanes is very important to obtain efficient working of the condenser.

Ques. How should the tubes be spaced?

Ans. The wider tube spacing in the banks should be adjacent to the steam inlet with narrower spacing in the banks adjacent to the air outlet.

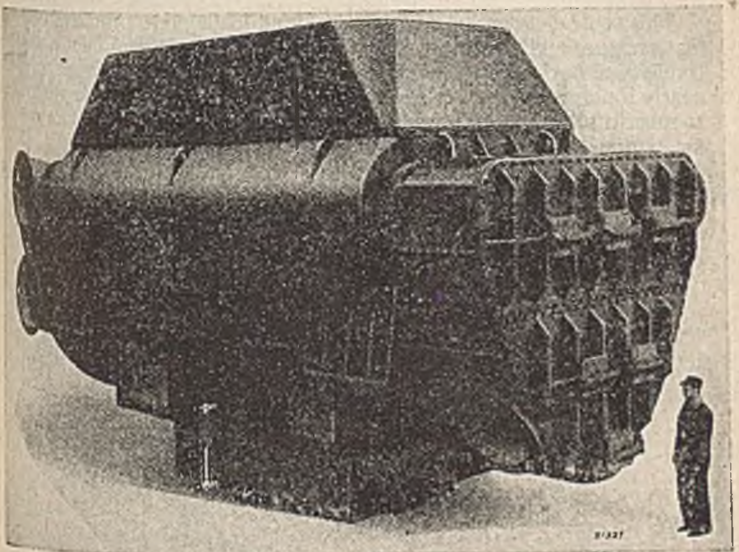


Fig. 43.—Exterior view of typical two pass marine condenser.

Ques. Why?

Ans. The variation should correspond to the reduction in the vapor volume.

Ques. What name is given to the spacing?

Ans. Differential spacing.

Radial Flow Dual Bank Condensers.—The essential features of this arrangement are shown in fig. 34, a type suitable for large installations.

The tubes of the first or colder pass consist of the two lower sections of tubes, and the tubes of the second or warmer pass being those of the two upper sections. The symmetrical division of the tubes into two distinct banks, in the manner indicated and by means of the wide central steam lane as shown, serves to produce several very valuable results.

This central steam lane at practically all loads gives practically equivalent pressure and temperatures both at the exhaust steam inlet and at the condensate outlet, and therefore provides a condensate temperature very nearly if not quite equal to the vacuum temperature. It also serves in effect to substitute two shallow condensers for one deep one and therefore provides for extremely good penetration of the steam among the tubes.

Further it furnishes a comparatively short and frictionless path between the exhaust steam inlet and the outlet to air pump.

CHAPTER 23

Condenser Auxiliaries

For the proper operation of a condenser various auxiliary equipment is required, varying in detail depending upon the type of main condenser and requirements.

Any installation (excepting nondescript bastard condensers) requires an air pump (**ignorantly** *called "vacuum pump"*).

The amount and kind of auxiliary apparatus required will depend upon the type of condenser and other considerations. Thus the outfit for a jet condenser is not the same as that for a surface condenser.

Jet Condenser Auxiliaries

In treating of auxiliaries for jet condensers the two types to be considered are:

1. Low level jet condensers, and
2. High level or barometric condensers.

Low Level Jet Condenser Auxiliaries.—In this type, the main item is a large pump which must remove:

1. Condensate
2. Injection water
3. Air and non-condensible vapors.

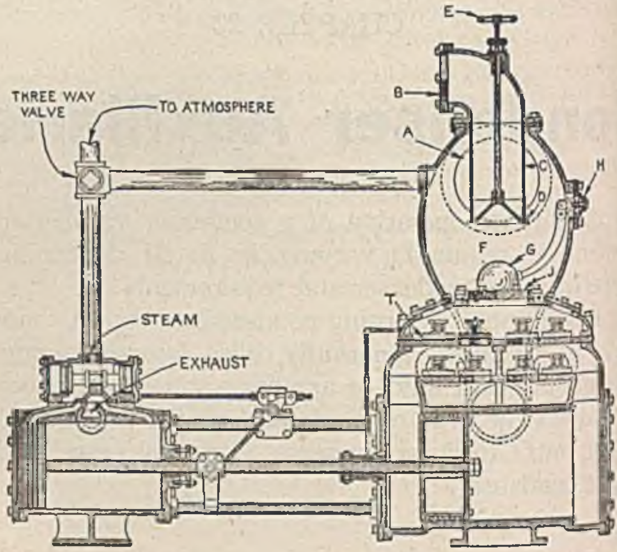


Fig. 1.—Low level jet condenser connected to removal pump. **In operation,** exhaust steam enters at **A,** and the condensing water at **B.** At **D,** there is a cone shaped spray nozzle connected with the tube **C.** The water issues from the nozzle **D,** in an umbrella shaped sheet or spray, which strikes the sides of the condensing chamber **F.** Thus the steam must pass through or into the spray on entering chamber **F,** where it is condensed. The mixture of condensing water and condensed steam descends through the contracted lower end of the condensing chamber **F,** in a solid stream, which insures any remaining vapor being condensed, thence into the inlet of the pump, which discharges the water through the valves **T** and opening **J,** into the hot well. In addition to discharging the mixture of condensed steam and water, the pump removes any air that may enter in the injection water or through leaks. The pump also raises the injection water used for condensing the steam, the greatest lift being generally twenty feet. At **E,** is a hand wheel with a long stem connected to the movable cone **D,** and by turning this wheel, the amount of injection water

The proper name for this pump to distinguish it from the wet air pump of a surface condenser is *removal pump*, in fact it has a *lot* to remove.

The air which must be removed from a jet condenser is much greater in quantity than that which must be removed from a surface condenser.

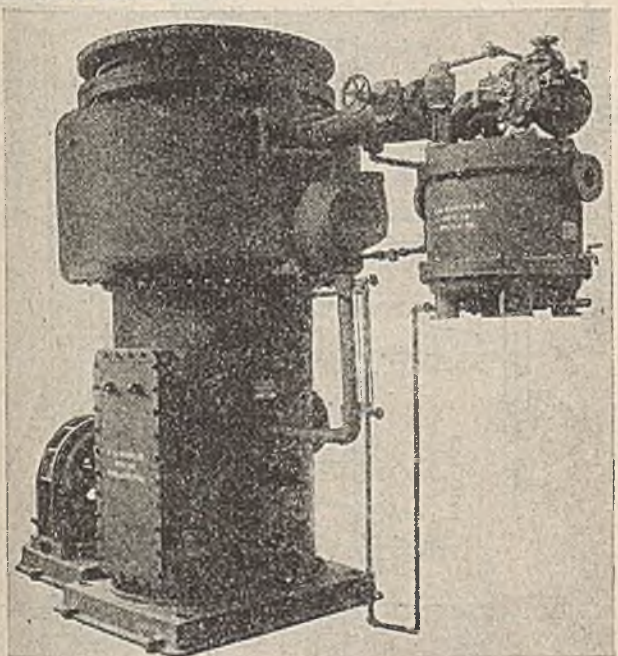


Fig. 2.—Exterior view of low level vertical cylinder jet condenser with motor driven removal pump and air ejector.

Fig. 1—*Text continued.*

may be regulated to suit the requirements. In case the pump slow down and stop, the water accumulating in the condensing chamber **F,** will gradually lift the float **G,** and as the float rises, it in turn opens the air valve **H,** admitting air to the exhaust pipe and engine cylinder, thus breaking the vacuum. This equalizes the pressure in the condensing chamber and stops the flow of the injection water.

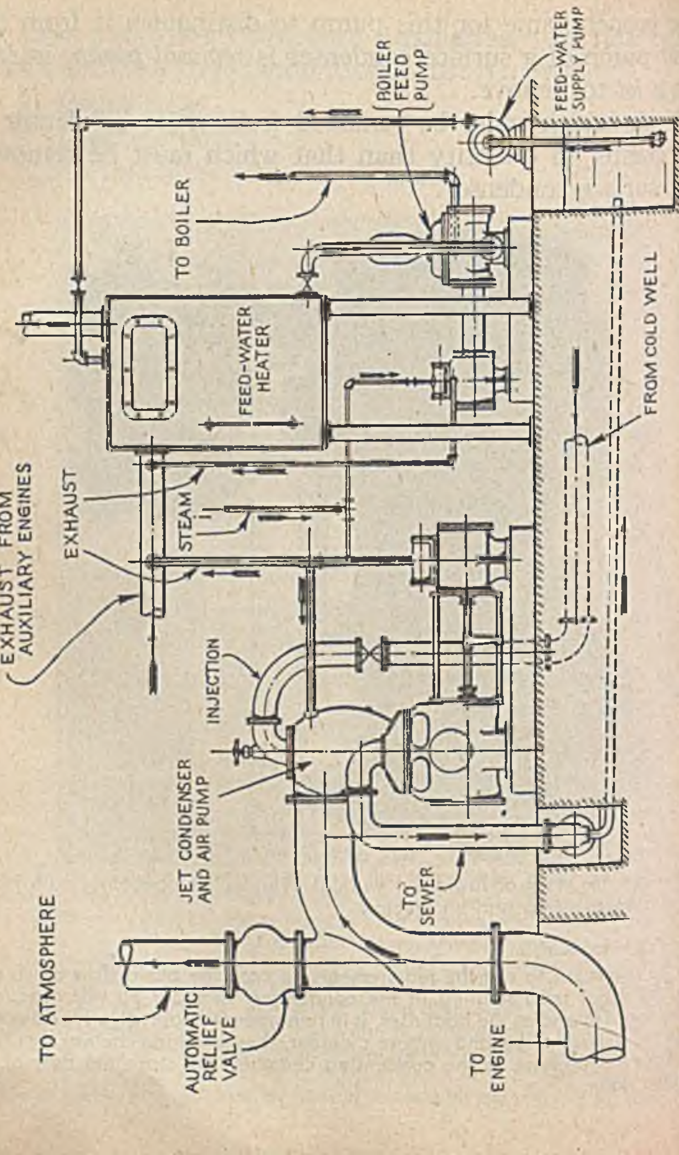


Fig. 3.—Steam engine installation with jet condenser equipment.

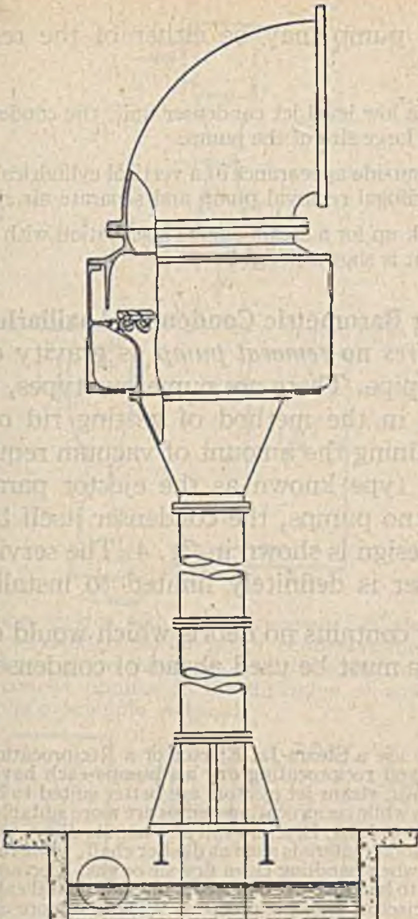


Fig. 4.—Flow diagram of ejector jet barometric condenser.

The removal pump may be either of the reciprocating or centrifugal type.

Fig. 1 shows a low level jet condenser unit, the condenser and removal pump. Note the large size of the pump.

Fig. 2 shows outside appearance of a vertical cylindrical design with electric driven centrifugal removal pump and separate air ejector.

A typical hook up for a steam engine installation with low level jet condenser equipment is shown in fig. 3.

High Level or Barometric Condenser Auxiliaries.—This type condenser *requires* **no** *removal pump* as gravity does the work in the long tail pipe. There are numerous types, the auxiliaries varying chiefly in the method of getting rid of the air and method of obtaining the amount of vacuum required.

The simplest type known as the ejector parallel flow condenser requires no pumps, the condenser itself being virtually a pump. This design is shown in fig. 4. The service application of this condenser is definitely limited to installations where:

1. The water contains no debris which would clog the water nozzles (strainer must be used ahead of condenser if the water be dirty).

NOTE.—When to use a Steam-Jet Ejector or a Reciprocating Dry Air Pump. Steam Jet Ejectors and reciprocating dry air pumps each have definite fields of application. *In general,* steam jet ejectors are better suited to handling large volumes at high vacuum while reciprocating pumps are more suitable for large volumes at low vacuum (28 in. or less). Ejectors will also handle dirty, wet, or dry mixtures containing sticky or solid materials such as dust or chaff, while reciprocating pumps should be used only when handling clean dry air or gas. Ejectors can also be made of suitable materials to handle corrosive gases, and accidental entrainment of liquid will not damage an ejector. In some processes it is often more economical to use a combination of both types. The cost and availability of steam and electricity also enters into the selection of the most suitable type of vacuum pump.

NOTE.—A steam jet ejector is a simple and reliable device for removing air and non-condensibles from condensers as well as from vacuum chambers in industrial processes.

NOTE.—When serving barometric condensers either single stage or two stage ejectors are used, depending upon the vacuum and operating requirements. Single stage units can be mounted anywhere between the top of the condenser and the hot well.

Fig. 5.—Single stage ejector. This type has a limited ratio of compression.

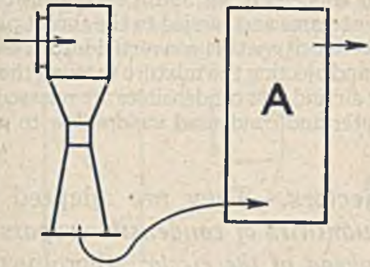


Fig. 6.—Single stage ejector with after condenser. A single-stage ejector discharging into an after-condenser which condenses the operating steam (and vapor if any) and allows the non-condensibles to escape to atmosphere. After-condensers can be applied to the discharge of any ejector, whether single- or multi-stage or multiple element.

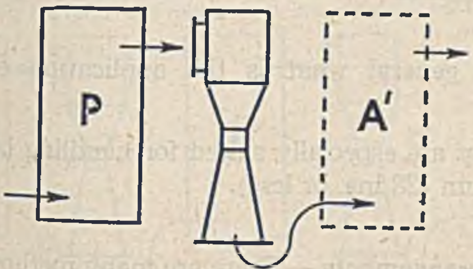


Fig. 7.—Single stage ejector with pre-cooler. When the mixture handled by the ejector contains vapors which can be condensed at operating vacuum and available water temperature, a pre-cooler can be used ahead of the ejector to reduce the weight of the mixture handled by the ejector. It reduces the size and steam consumption of the ejector.

2. The quantity of air or non-condensible gases going to the condenser with the steam is relatively low.

This condenser is admirably suited to serve turbines, vacuum pans, evaporators or steam engines where the air or non-condensibles are not excessive (nominally 20 to 40 lb./hr.)

Water at atmospheric pressure or slightly higher enters near the top of the condenser and is injected into the steam space through a number of efficient hydraulic nozzles, forming high-velocity streams or jets of water which converge in a throat piece at the bottom.

As the steam is condensed, air and non-condensible gases are entrained by the converging streams and carried to the throat piece, where the kinetic energy of the high-velocity jets is converted into pressure, compressing the non-condensibles and ejecting the mixture through the tail pipe into an open hot well. Here the air and non-condensibles are released to atmosphere, while the circulating water and condensed vapors flow to waste.

Steam Jet Ejectors.—They are adapted to service *where relatively* **large** *quantities of condensible vapors can be condensed in the precooler ahead of the ejector*, thereby reducing the load on the ejector.

Steam jet ejectors and reciprocating dry air pumps each have definite fields of application.

Ques. In general what is the application of steam jet ejectors?

Ans. They are especially suited for handling large volumes at low vacuum (28 ins. or less).

Ejector Arrangements.—There are many methods of arranging ejectors in combination with pre-coolers, inter-condenser, after-condensers, etc. The essentials of the various hook ups are shown in figs. 5 to 14.

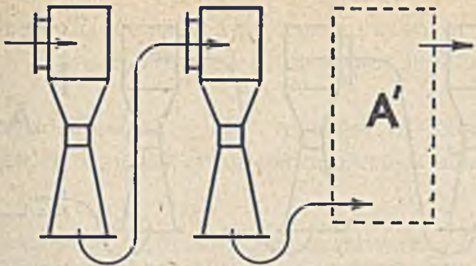


Fig. 8.—Two stage non-condensing ejector. For greater ratios of compression (higher vacuum) than are attainable with a single ejector, two ejector elements can be operated in series or stages. In its simplest form the primary discharges directly into the secondary. Such a two-stage non-condensing ejector is necessarily rather uneconomical in steam consumption because the secondary has to handle all the primary operating steam in addition to the non-condensibles and vapors.

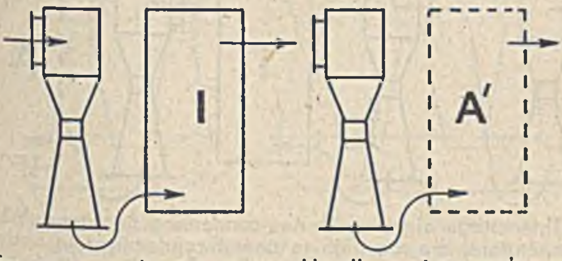


Fig. 9.—Two stage condensing ejector. Usually an intercondenser is inserted between stages to condense the operating steam used by the preceding stage, thereby reducing the load, size and steam consumption of the following stage.

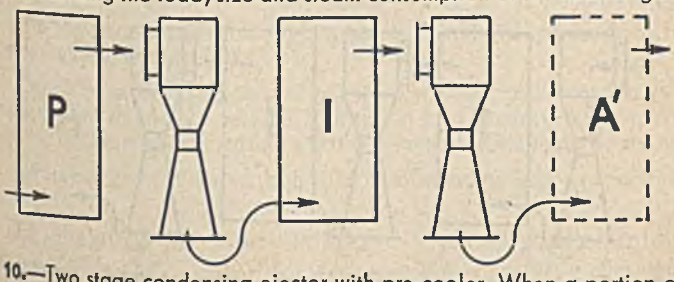


Fig. 10.—Two stage condensing ejector with pre-cooler. When a portion of the vapors entering the primary of a two-stage ejector can be condensed at operating vacuum and available water temperature, a pre-cooler can be applied ahead of the primary to reduce the size and steam consumption of the ejector.

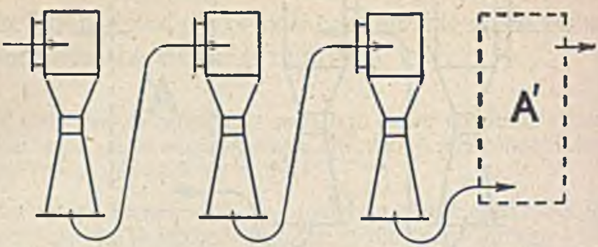


Fig. 11.—Three stage non-condensing ejector. For still higher vacuums (lower inlet pressures) than are economically attainable with a two-stage ejector, three ejector elements can be operated in series. This is the simplest but least economical three-stage ejector. Steam consumption is relatively high because operating steam is not condensed between stages.

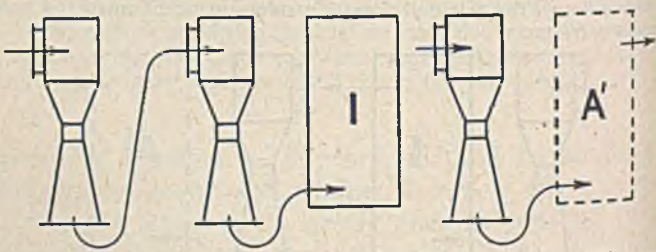


Fig. 12.—Three stage ejector with non-condensing first stage. When cooling water temperatures are too high to permit condensing out operating steam between booster and second stages of a three-stage ejector, the booster stage operates non-condensing and discharges directly into a standard two-stage condensing unit.

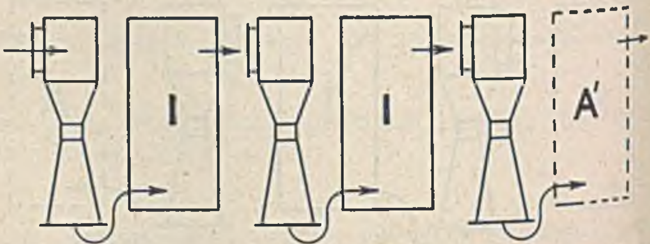


Fig. 13.—Three stage condensing ejector. When operating conditions and cooling water temperature permit intercondensers between all stages of a three-stage ejector, the unit is more economical in steam consumption than either of the arrangements shown in figs. 11 and 12.

Single Stage Ejectors.—The combination of a single stage ejector with a pre-cooler and after-cooler is shown in fig. 15.

Two Stage Ejectors.—The method of connecting these ejectors with barometric inter-condensers is shown in figs. 18 and 19.

The type shown in fig. 18 uses a small quantity of cooling water in multi-pass, counter-current flow, which comes in direct contact with, and condenses, the operating steam from the primary ejector.

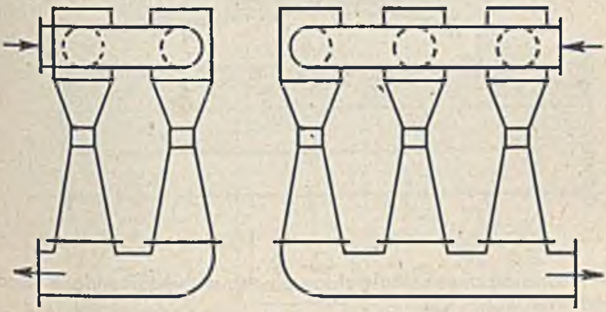


Fig. 14.—Multiple element ejectors. For greater flexibility in operation, the ejector elements making up single- or multiple-stage ejectors can be paralleled. Such paralleling is known as multiple elements (i.e. single-element, twin-element, triple-element, etc.)

The inter-condenser is of non-clogging type and will operate on dirty cooling water without strainers.

Operating steam from the second-stage ejector can be condensed under the overflow in the hot well as shown, or in a separate after condenser, similar to the inter-condenser, when non-condensibles must be recovered.

The arrangement shown in fig. 19 is suitable for larger capacities.

The design of the inter-condenser is multi-pass counter-flow similar to fig. 18, except for the method of water distribution which is of the disc flow used in counter-current barometric condensers.

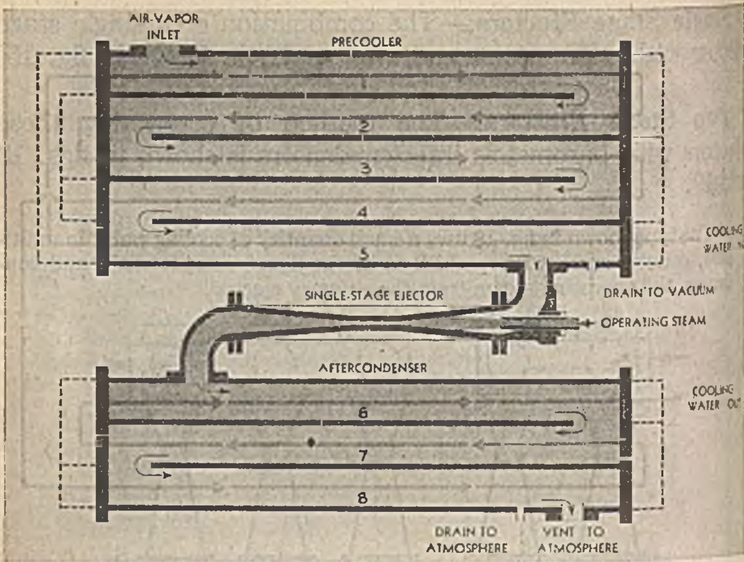


Fig. 15.—Flow diagram of single-stage ejector with MSP surface precooler and aftercondenser unit.

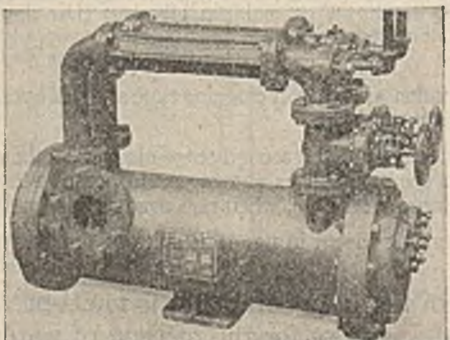


Fig. 16.—Exterior view of twin element ejector on precooler and after condenser unit.

An application of two stage ejectors with surface *inter-* and *after-* condensers is shown in figs. 20 and 21.

This hook up is suitable for installations where the mixing of the cooling water and condensate is undesirable. Water and vapor flow is multi-pass to insure minimum steam and cooling water consumption.

A combination of two stage ejectors with surface pre-cooler *inter-* and *after*-condensers is shown in figs. 22 and 23.

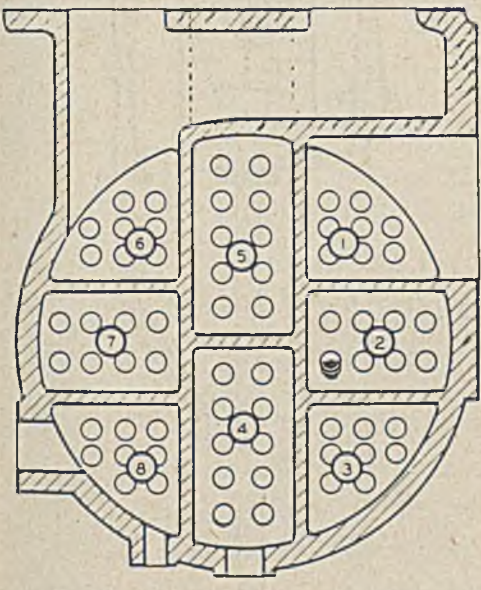


Fig. 17.—Cross section of surface pre-cooler and after condenser unit. The numbers correspond to like numbers in fig. 15.

Vapor-condensing capacity of the precooler increases the effective air-vapor-removal capacity of the two-stage ejector several hundred per cent, thus making it much more economical in steam consumption than a two-stage ejector having the same total air vapor capacity, but without the pre-cooler. In other words, the cooling water in the pre-cooler condenses practically all the vapors in the air vapor mixture so that the ejector removes only non-condensibles.

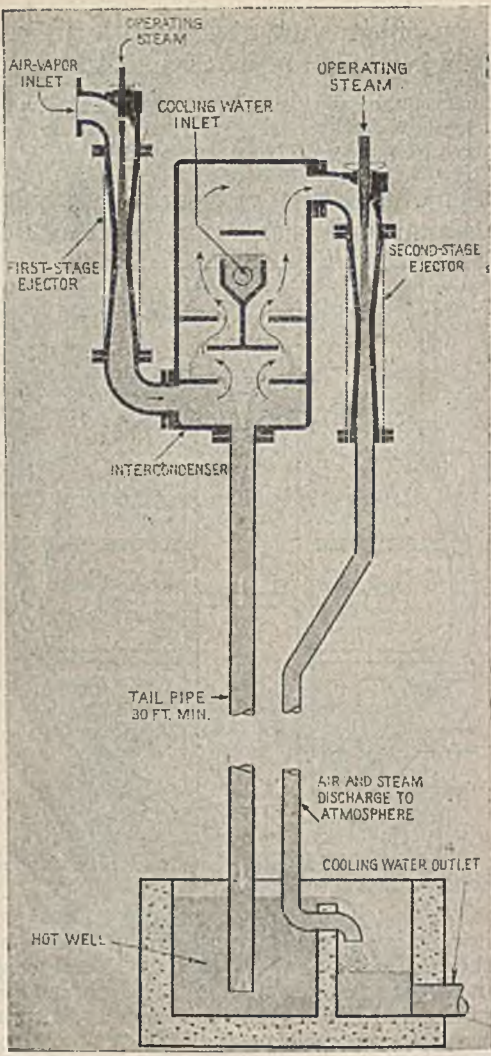


Fig. 18.—Flow diagram of two stage ejector with direct contact barometric inter condenser.

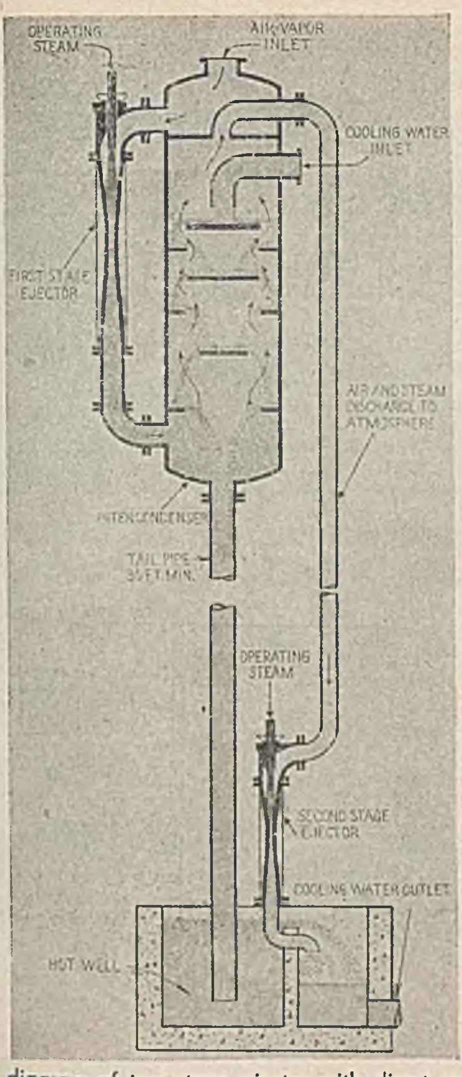


Fig. 19.—Flow diagram of two stage ejector with direct contact barometric inter condenser.

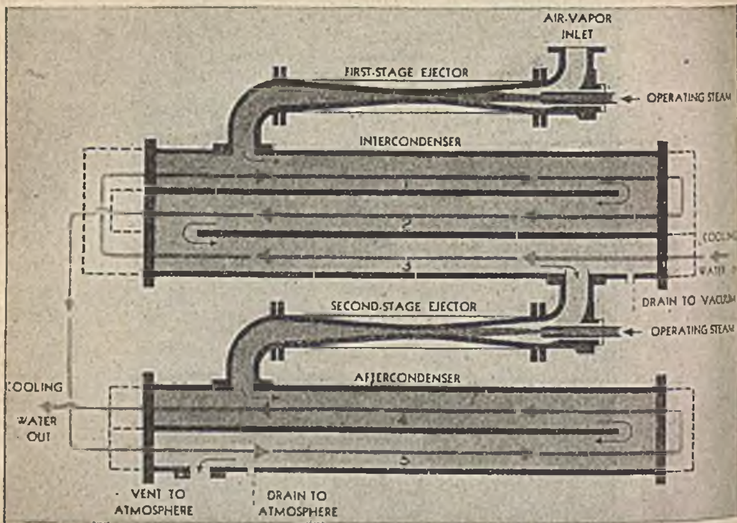


Fig. 20.—Flow diagram of two stage ejector and surface inter- and after condenser unit.

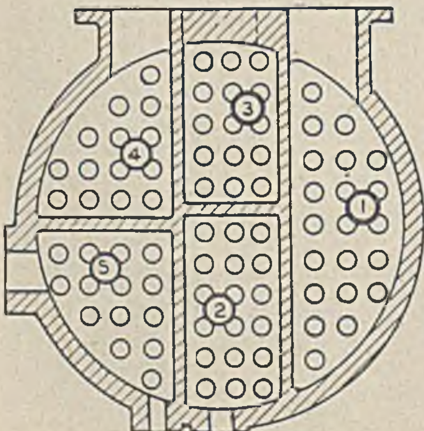


Fig. 21.—Cross section of surface inter and after condenser unit. The numbers correspond to similar numbers in fig. 20.

The pre-cooler, inter-condenser and after-condenser are built in a single shell. Inter-compartment leakage is prevented by bolting tube sheets to wide flanges on the partitions with collar studs and bolts.

High temperature gasket material is used throughout.

Water boxes in all sizes are provided with removable covers for easy access to tube ends.

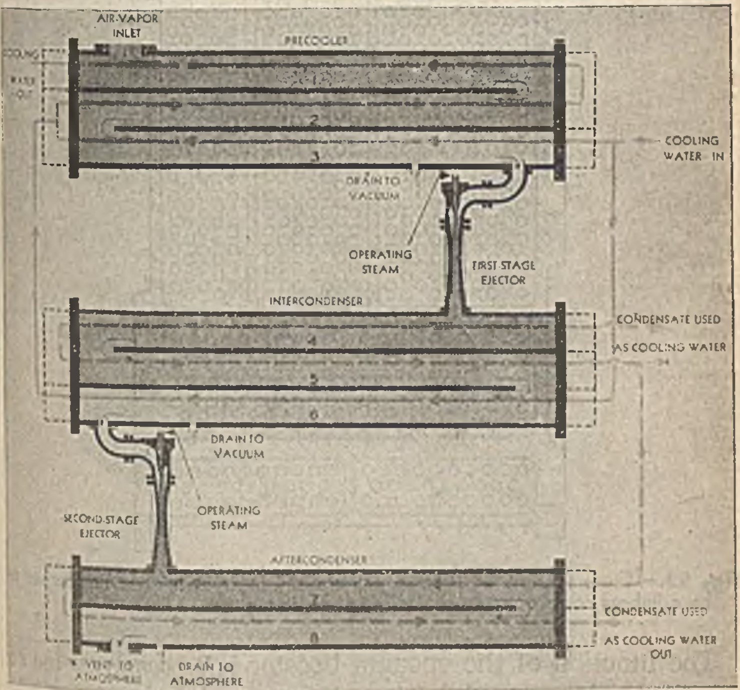


Fig. 22.—Flow diagram of two stage ejector and single section surface pre-cooler inter and after condenser unit.

Three Stage Ejectors.—For higher vacuums than are economically attainable with a two stage ejector, a third or booster stage can be added. The temperature of cooling water limits the vacuum obtainable in any condenser. Therefore processes where large quantities of vapors must be condensed at high vacuums necessitate the use of a vacuum booster.

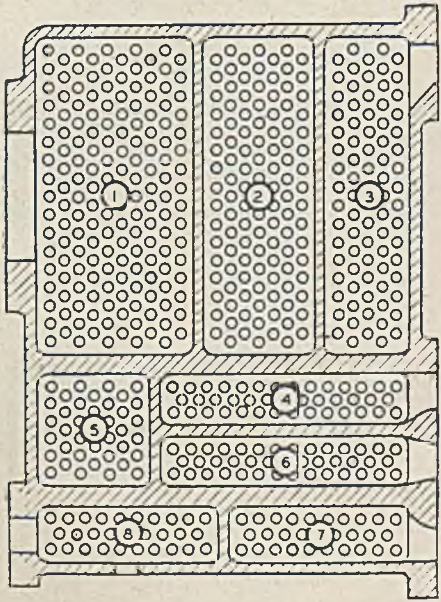


Fig. 23.—Cross section of surface pre-cooler, inter and after condenser unit. The numbers correspond to similar numbers in fig. 22.

The function of the vacuum booster is *to compress the condensible and non-condensible vapors to a lower vacuum where the vapors can be condensed with the water temperature available.*

In other words, the condensible and non-condensible vapors plus the booster operating steam are delivered to the booster condenser where all condensible vapors are removed. The non-condensibles are removed from the booster condenser by a standard two-stage ejector.

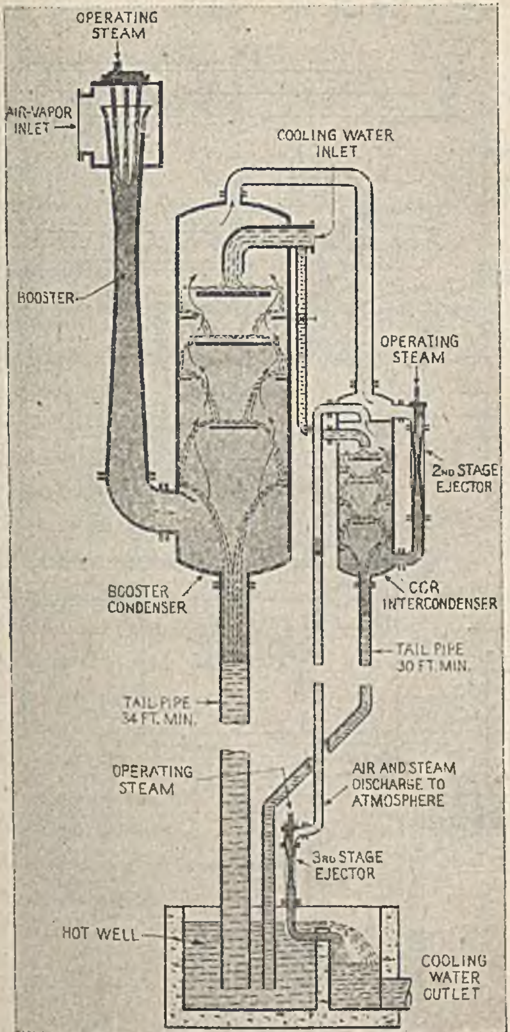


Fig. 24.—Flow diagram of three stage ejector with counter current barometric booster condenser and inter-condenser.

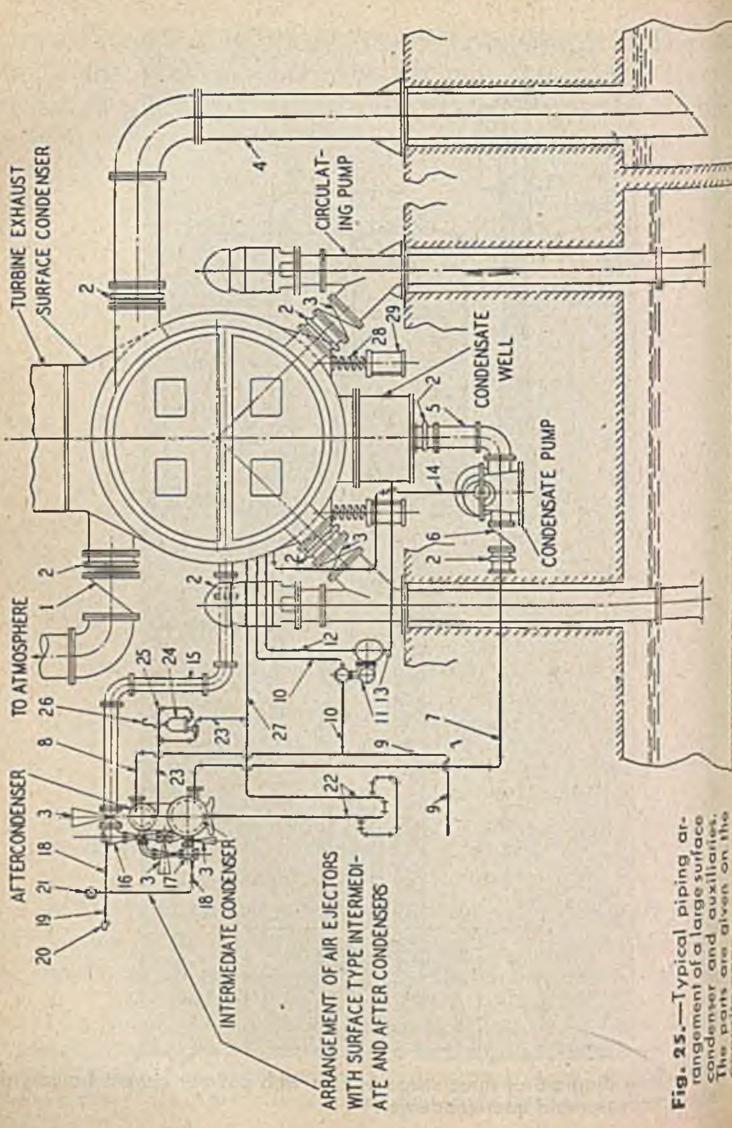


Fig. 25.—Typical piping arrangement of a large surface condenser and auxiliaries. The parts are given on the opposite page.

LIST OF PARTS
(FOR FIG. 25)

1. Atmospheric relief valve
2. Expansion joint
3. Gate valve
4. Circulating water discharge from main condenser
5. Condensate from condenser to condensate pump
6. Check valve
7. Condensate from condensate pump to intermediate and after condenser
8. Condensate from intermediate and after condensers
9. Condensate line to heaters
10. Condensate recirculating line to main condenser
11. Condensate control for recirculating line
12. Vent from control valve float chamber
13. Condensate line to control valve float chamber
14. Vent from condensate pump to main condenser
15. Air removal line from main condenser to first stage ejectors
16. Air ejectors (first stage)
17. Air ejectors (second stage)
18. Stop valve (steam)
19. Throttle valve (steam)
20. Steam Strainer
21. Steam pressure gauge
22. Intermediate condenser condensate drain loop
23. After condenser condensate drain dripping
24. After condenser drainer
25. After condenser drainer float chamber piping
26. After condenser drainer float chamber vent
27. Intermediate and after condensate return to main condenser
28. Main condenser support springs.
29. Main condenser spring support.

In some high vacuum processes where only non-condensibles are removed, the function of the vacuum booster is the same, but the booster condenser removes only the booster operating steam.

An arrangement of three stage ejector is shown in fig. 24.

CHAPTER 24

Condensers; Operation

In the operation of condensers, certain rules must be followed in starting an engine or turbine. The proper procedure will depend upon the type of condenser, whether independent or direct connected engine driven pumps are used; the conditions being quite different depending upon how the air pump is driven.

Keel Condenser; Engine Driven Pump.—This combination although only used on small boats, requires the greatest care in starting with cold engine. The keel condenser requires the least amount of cooling water because of the unlimited supply. This because the steam enters on the inside of the tubes there is no shell. Hence this type of condenser has the least volume and *is most liable to be flooded*.

Since the air pump works only when the main engine runs, in starting after warming up the engine with by pass steam it should be worked back and forth very carefully because if the exhaust and condenser be flooded with condensate, serious damage may result if the engine be turned over too fast. This relates especially to some faulty installations not having a relief valve. Even with a relief valve too much dependence should not be placed on same to clear condenser of condensate quick enough to prevent damage.

It will take some time of careful control of engine for the air pump to clear the condenser of condensate.

During this procedure watch the vacuum gauge, and do not bring engine up to speed until a full steady vacuum is obtained.

If an independent pump be used, no such extreme precaution is necessary because an independent pump is started before the engine, and pumps all the condensate out of the condenser making a vacuum available in starting the main engine.

Jet Condensers

Starting; Jet Condensing.—In the jet condenser, the air pump removes not only the air but the circulating water and condensate, hence it must be evident that if the pump do not remove the water from the condenser as fast as it comes in, the apparatus will quickly flood, unless the vacuum be broken by the proper working of an automatic device provided for that purpose, and back up into the engine cylinder causing serious damage. Hence in starting jet condensing, great precautions should be taken that the water does not reach the cylinder.

In starting a jet condensing engine, care should be taken to follow certain rules, to prevent water entering the cylinder with its attendant dangers.

There are two methds of procedure, depending upon whether the engine have:

1. An independent air pump; or,
2. A direct driven air pump

Ques. What is the procedure in starting with an independent air pump?

Ans. The injection valve is opened slightly and the air pump started to its normal speed.

Ques. What is done when the vacuum is established as indicated by the gauge?

Ans. The engine is warmed up in the normal manner and started.

Ques. How is the vacuum controlled as the engine is brought up to speed?

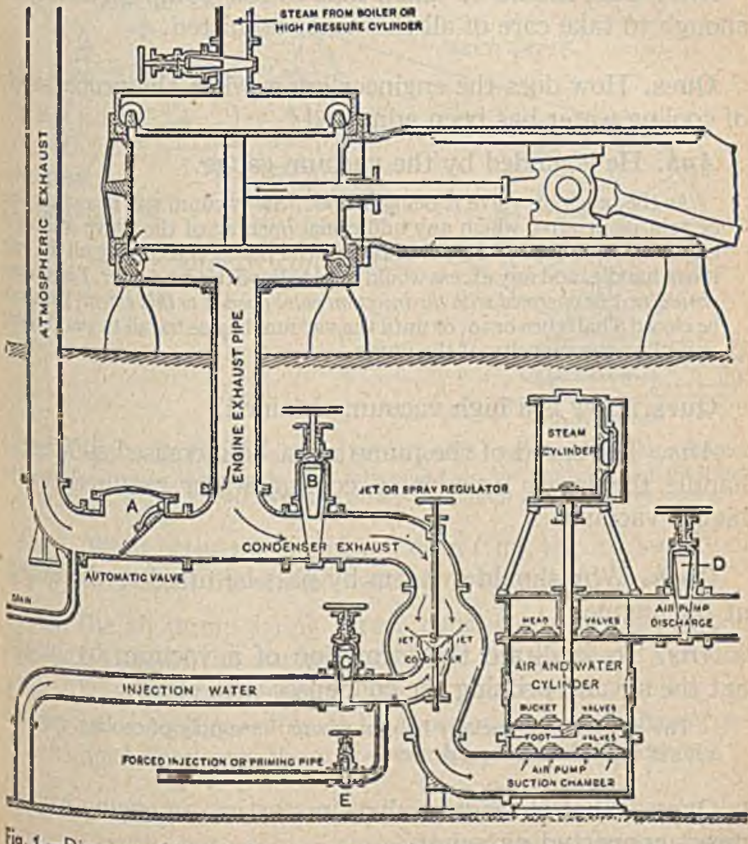


Fig. 1.—Diagram of jet condenser, pump and connections.

Ans. The injection valve is regulated so that the supply of cooling water will be sufficient to condense the steam otherwise the vacuum will fall.

Ques. What precaution should be taken when increasing the amount of cooling water?

Ans. Care should be taken that the air pump is running fast enough to take care of all the water admitted.

Ques. How does the engineer know when the proper supply of cooling water has been admitted?

Ans. He is guided by the vacuum gauge.

As the injection valve is being opened, the vacuum will increase up to a certain point after which any additional opening of the valve will not increase the vacuum. This indicates that the pump is receiving all the water it can handle, and any excess would tend to flood the condenser. *The condenser should not be operated with the injection valve opened to this extent*, but should be closed a half turn or so, or until the vacuum begins to fall to guard against exceeding the capacity of the pump.

Ques. How is a high vacuum obtained?

Ans. The speed of the pump must be increased sufficient to handle the larger amount of cooling water required for the higher vacuum.

Ques. Why should a steam by pass be fitted to the exhaust at the engine?

Ans. To facilitate the formation of a vacuum by blowing out the air and priming the condenser with steam.

This method is especially helpful where the supply of cooling water is at a lower level than the condenser.

Ques. What is the procedure in starting an enging having a direct connected air pump?

Ans. The cylinder is first warmed, and the engine set in motion before opening the injection valve. This allows the condenser to fill with steam which displaces the air. As soon as the engine is in motion the injection valve is slightly opened, the full supply of cooling water being not admitted until the normal speed has been reached.

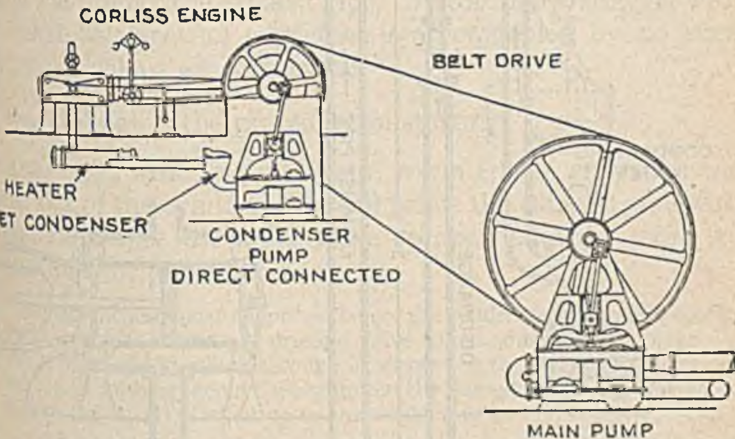


Fig. 2.—Engine with direct connected power pump. As piped, a water heater is placed between the engine and jet condenser, the latter being attached to the pump. Main pumps as designed by the author for city water works.

Ques. What is the reason for not admitting the full supply of cooling water until the engine has been brought up to speed?

Ans. The air pump being direct connected, the speed will vary with that of the engine and while the engine is running slowly, the pump displacement would not be sufficient for the full supply of cooling water. The condenser under these conditions might flood and the water back up into the cylinder.

Starting with Barometric or Siphon Condenser.—Since in this arrangement the hot well is located 34 feet below the condenser

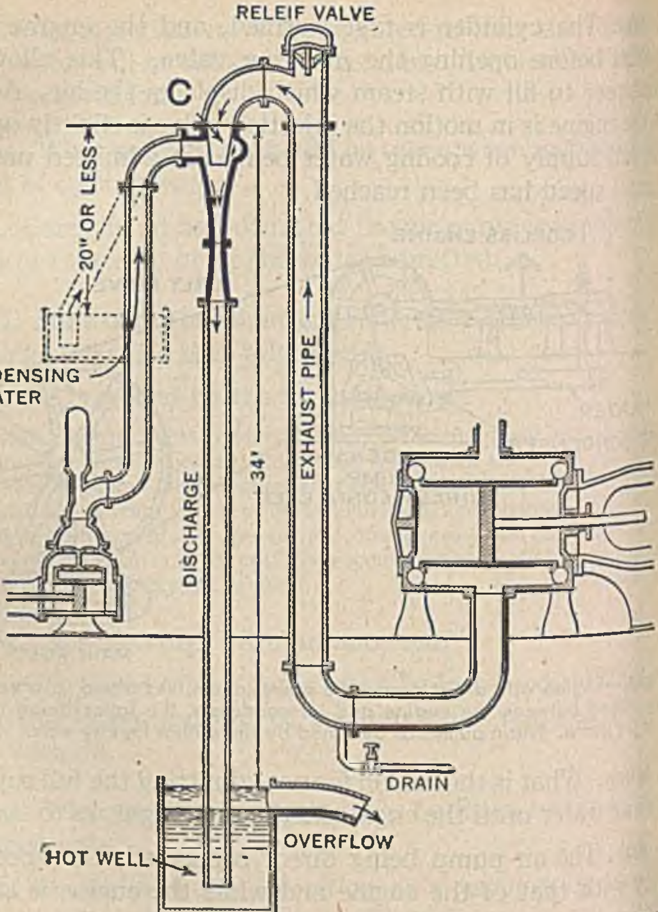


Fig. 3.—Barometric condenser installation with independent air pump. If the level of the injection water be not more than say 20 feet before the condenser inlet, the condenser will siphon the water over as soon as a vacuum is formed in it and the pump may be dispensed with. As 20 feet is about the limit to which water may be continuously lifted by the siphoning action it follows that when the water supply is more than 20 feet below the condenser, a pump must be used as shown. This arrangement is sometimes modified by the insertion of a tank, shown in dotted lines, at about the lower limit of the siphon.

no air pump is required to remove the condensed steam and cooling water. Hence, there is no way in which water can get into the engine cylinder unless it be allowed to accumulate in the pocket formed by the exhaust pipe, and not even then unless atmospheric pressure be admitted to the exhaust pipes through the uncovering of the water supply or discharge pipes; a proper drain will protect the exhaust pipe. Accordingly, starting with a siphon (barometric) condenser is accompanied by no such danger, as with a jet condenser.

Ques. What is the procedure in starting?

Ans. Open exhaust pipe drain, warm engine and work the water out of the cylinder; this will prime the exhaust pipe with steam and cause the relief valve to open allowing steam to escape into the atmosphere.

Since a vacuum must be formed before the condenser will siphon cooling water, open the starting or priming valve which admits water to the discharge pipe, and in falling through it, draws out the air, closing the relief valve and forming enough vacuum in the upper pipes and condenser to draw the injection water up to the condenser.

The starting valve would now be closed and the water supply adjusted by the injection valve, being guided by the vacuum gauge.

The barometric condenser, pump and connections are shown in fig. 3.

Starting with Exhaust Steam Induction Condenser.—When the condensing water is under a head, turn on the cooling water and when a vacuum is formed, start the engine.

Fig. 3.—*text continued.*

This is convenient where a single action or single cylinder tank pump is used to lift the water, since such pump gives a more or less intermittent flow, whereas a practically constant flow is required by the condenser. In the tank arrangement, the pump discharges intermittently into the tank and the condenser siphons continuously from the tank.

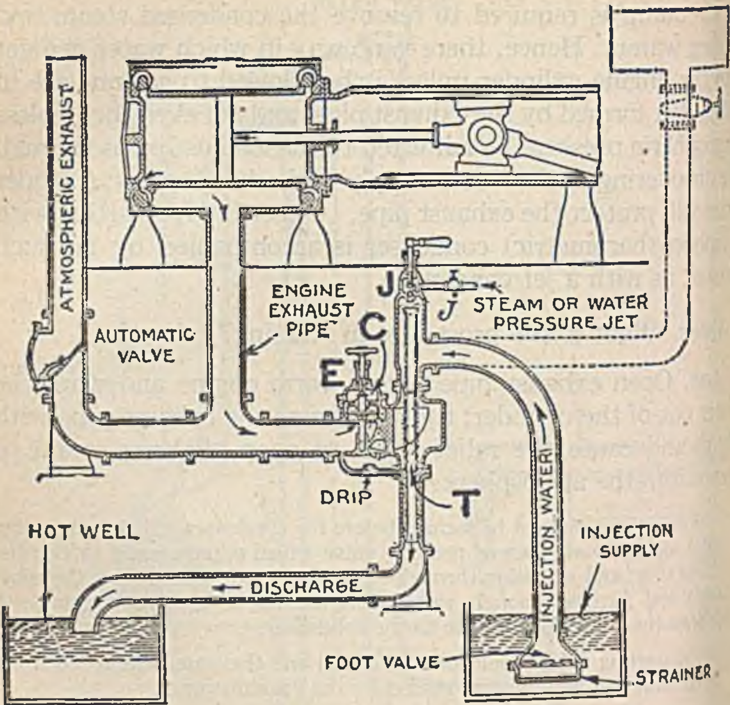


Fig. 4.—Exhaust steam induction condenser. **In operation,** exhaust steam enters through the valve E, and passes through the inclined perforations into the central tube T, as shown by the arrows. Owing to the velocity of its movement the air in the condenser and the injection pipe is drawn out with it, and the atmospheric pressure on the injection supply forces the condensing water up through the pipe and into the tube T, as shown. The exhaust steam is condensed by this water and a vacuum is left in the condenser and exhaust pipe. The original velocity with which the water entered the condenser and the added velocity due to the exhaust steam enable the mingled steam and water to overcome the atmosheric pressure on the discharge end and pass out into the hot well, just as the water from the injector overcomes the resistance due to friction and pressure and passes into the boiler. Evidently then the velocity of the discharge is sufficient to draw out the air and to get rid of the condensing water and condensed steam; so that no air pump is required as in the case of a jet or surface condenser, nor a 34-foot "tail" column, as in the injector or siphon condenser. **To start engine:** When the condensing water is under

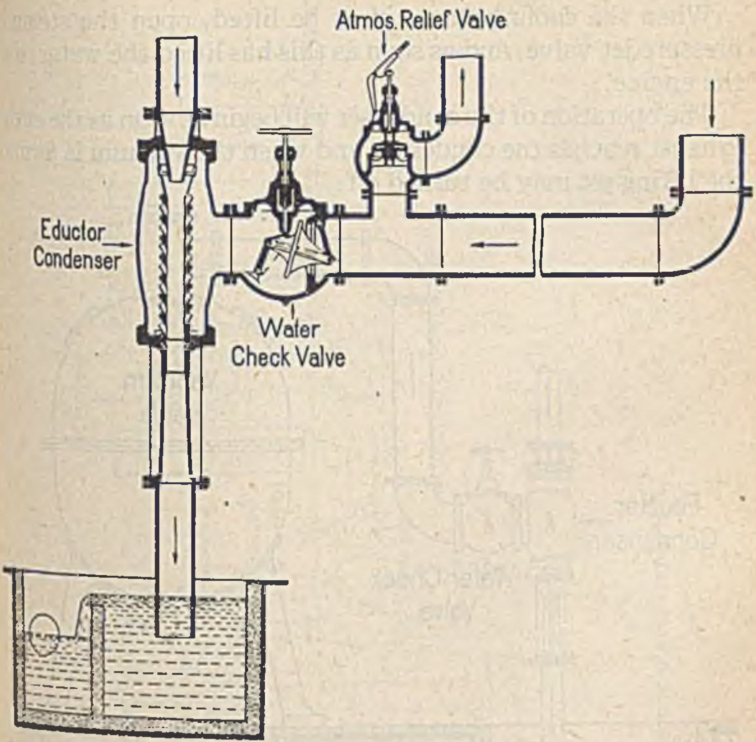


Fig. 5.—Standard arrangement of an eductor condenser and valves.

Fig. 4.—*text continued.*

a head, turn on the condensing water and when a vacuum is formed, start up the engine. When the condensing water must be lifted, open the steam or pressure jet valve **i**, and as soon as this has lifted the water start the engine. The operation of the condenser will begin as soon as the engine exhaust reaches the condenser and when the vacuum is formed the lifting jet may be turned off. In shutting down, stop the engine first, when the operation of the condenser will cease if the condensing water be under a lift; if the water supply be under a head, stop the engine first and then shut the valve in the water supply pipe.

When the cooling water must be lifted, open the steam or pressure jet valve, and as soon as this has lifted the water, start the engine.

The operation of the condenser will begin as soon as the engine exhaust reaches the condenser and when the vacuum is formed, the lifting jet may be turned off.

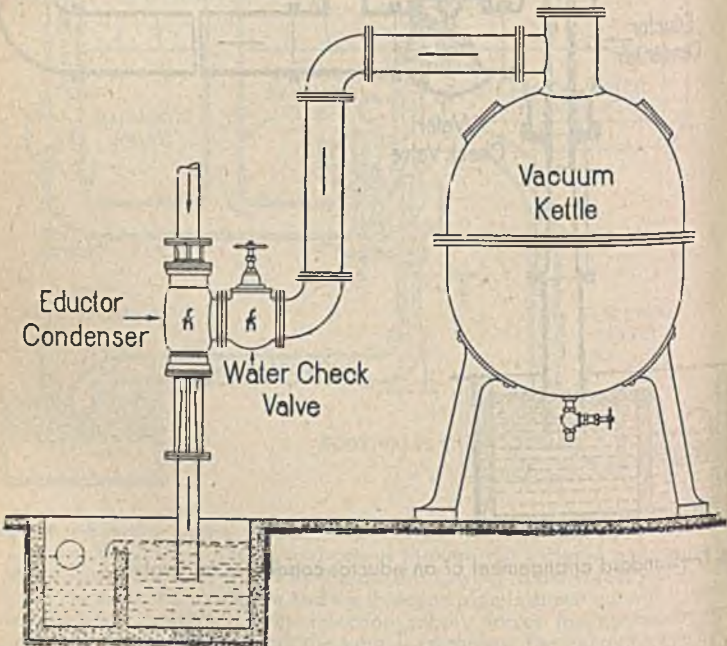


Fig. 6.—Eductor condenser connected to vacuum kettle.

In shutting down, stop the engine first, when the operation of the condenser will cease if the condensing water be under lift; if the water supply be under a head, stop the engine first and then shut the valve in the water supply pipe.

Fig. 4 shows an exhaust steam induction condenser and connections.

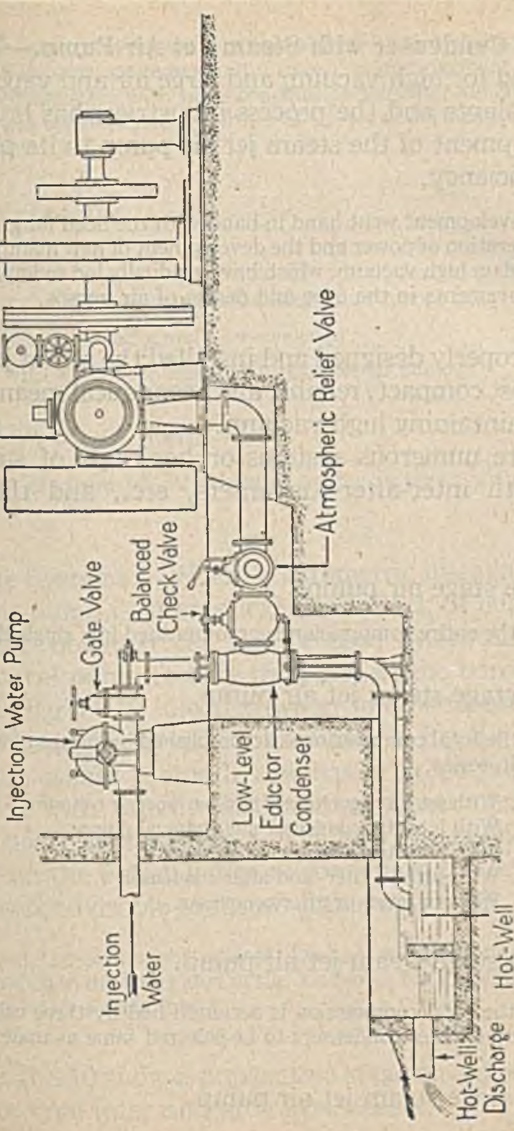


Fig. 7.—Typical eductor condenser and engine installation.

Surface Condenser with Steam Jet Air Pump.—The increasing demand for high vacuum and large air and vapor capacities in power plants and the process industries, has brought about the development of the steam jet air pump to its present state of high efficiency.

This development went hand in hand with the need for greater economy in the generation of power and the development of new manufacturing processes based on high vacuum, which have gradually led to important changes and improvements in the type and design of air pumps.

When properly designed and installed the steam jet air pump offers a most compact, reliable and economical means of producing and maintaining high vacuum.

There are numerous systems or hook-ups of steam jet air pumps with inter-after-condensers, etc., and they may be classed as:

1. Single stage air pump.

   Where the entire compression is accomplished in a single diffuser.

2. Two stage steam jet air pump.

   Where the total compression is accomplished by means of two exhausters operating in series.

   a. With jet inter-condenser and barometric column.
   b. With jet-inter-condenser and removal pump.
   c. With jet inter- and after-condenser.
   d. With surface inter- and after-condenser.
   e. Without inter- or after-condenser.

3. Three stage steam jet air pump.

   Where the total compression is accomplished by three exhausters operating in series. Inter-condensers to be selected same as under 2.

4. Four stage steam jet air pump.

Where the total compression is accomplished by two steam jet boosters in series with booster condenser and a two-stage arrangement of high and low vacuum exhausters with inter-condensers and after-condensers where specified. These condensers to be either of the jet or surface type according to requirements.

5. Vacuum booster.

Operating with:

a. Barometric condenser and steam jet air pump.

b. With barometric multi-jet condenser.

c. With surface condenser and steam jet air pump.

6. Hydro-steam air pump.

Consisting of steam jet exhauster operating in series with a water jet exhauster.

Two Stage Systems.—Where a barometric installation of the two stage air pump is impractical due to lack of head room, a jet type inter-condenser can still be used by installing a centrifugal removal pump to take the place of the barometric tail pipe and discharge the injection water and condensate against atmospheric or higher pressures as shown in fig. 8.

This pump, usually motor driven, has to be specially designed and provided with water sealed stuffing boxes and vents for operating against high vacuum. A submergence of not less than 4 ft. (between the outlet flange of condenser and center line of pump shaft) is advisable to insure stable operation.

To protect the equipment against flooding, a liquid level controller is usually installed in the lower part of the condenser body operating a suitable valve in the water supply line.

Figs. 9 and 10 show a duplex two stage steam jet air pump with surface type inter and after condensers.

In this arrangement there are two primary steam jets and two secondary steam jets each connected to a common steam supply line and each discharging into the combined inter and after condenser. This duplex arrangement of primary and secondary steam jets, permits operating both halves in parallel when large quantities of air or vapors are to be handled.

When operating at reduced capacity one of the primary jets and one of the secondary jets can be closed down, the remaining set handling all the air and working as a standard two stage machine.

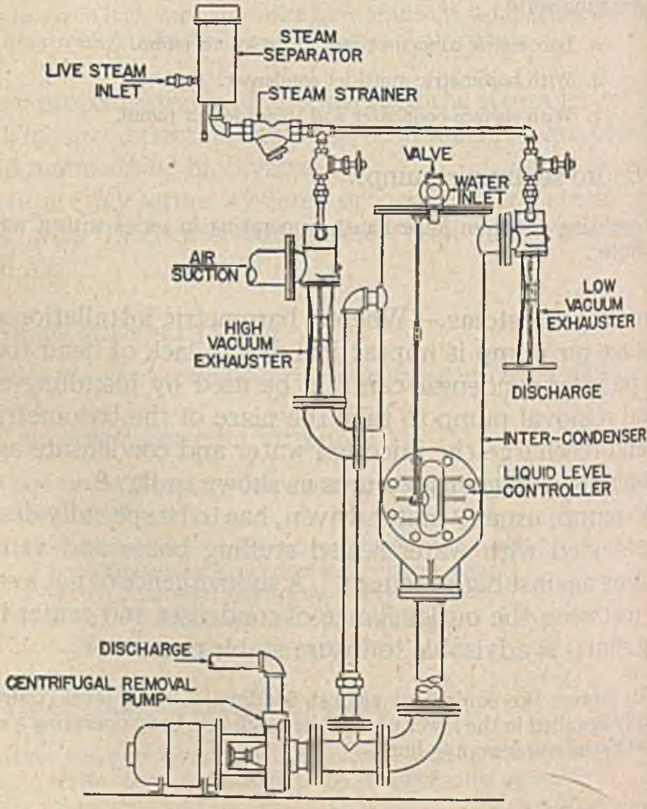
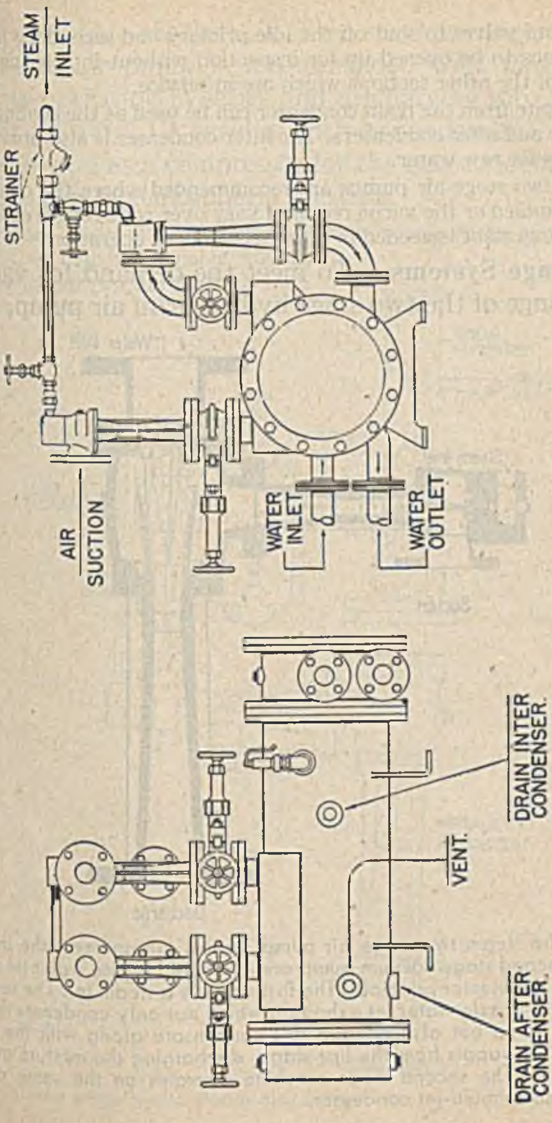


Fig. 8.—Two stage steam jet air pumps with jet inter condenser and removal pump.



Figs. 9 and 10.—Duplex two stage steam jet air pumps with surface inter and after condenser.

The use of valves to shut off the idle primary and secondary jets allows these sections to be opened up for inspection without interfering with the operation of the other sections which are in service.

Condensate from the main condenser can be used as the cooling medium in the inter and after condensers. The inter condenser is also provided with connections for raw water.

Duplex two stage air pumps are recommended where the quantities of air to be handled or the vacua required vary over wide ranges or where the duplex arrangement is needed to insure continuous operation.

Three Stage Systems.—To meet the demand for vacua beyond the range of the two stage hydro steam air pump, that is,

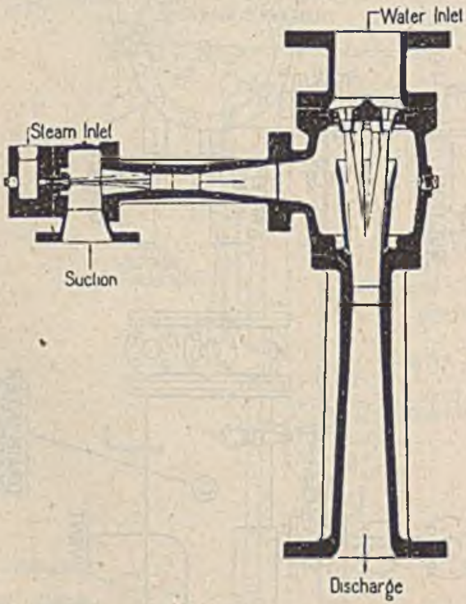


Fig. 11.—Hydro steam two stage air pump. In this arrangement the inter condenser and second stage vacuum pump are combined in one water jet machine operating on the ejector principle. The first stage is a steam jet. The secondary stage is a multi-nozzle water jet exhauster which not only condenses the steam from the first stage but also entrains this condensate along with the air and non-condensible vapors from the first stage, discharging the mixture at atmospheric pressure. The second stage machine operates on the same principle as the well known multi-jet condenser.

above 5 m/m Hg. absolute, a three stage arrangement of the hydro steam air pump is recommended with two steam jet condensers operating in series and discharging into a water jet exhauster as in fig. 12, serving both as condenser for the actuating steam and as a compressor for the non-condensible gases, operating against atmospheric pressure.

There are many manufacturing processes where steam mixed with large quantity of non-condensible vapors must be withdrawn at extremely high vacuum.

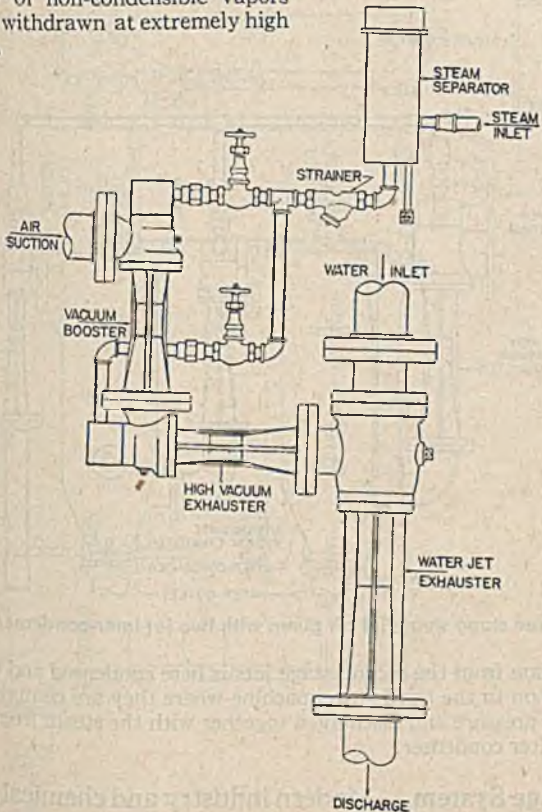


Fig. 12.—Three stage hydro steam air pump.

In these installations the steam must be handled as vapor along with the other gases, as no pre-condensers are used. For this service the three stage steam jet evacuator shown in fig. 13 is recommended.

The primary stage operates at very high vacuum and discharges into a No. 1 inter condenser of the jet type. The steam from process equipment and primary jets is condensed and the air and non-condensible vapors pass over to the second stage machine where they undergo a second compression and are discharged into the No. 2 inter condenser.

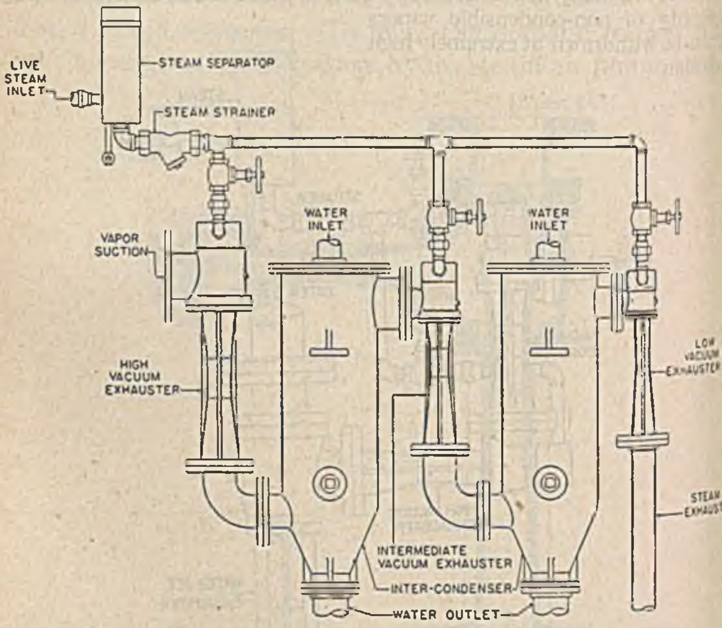


Fig. 13.—Three stage steam jet air pump with two jet inter-condensers.

The steam from the second stage jets is here condensed and the air and gases pass on to the third stage machine where they are compressed to atmospheric pressure and discharged together with the steam from this stage into the after condenser.

Four Stage System.—Modern industry and chemical developments are frequently calling for higher vacuum than can be

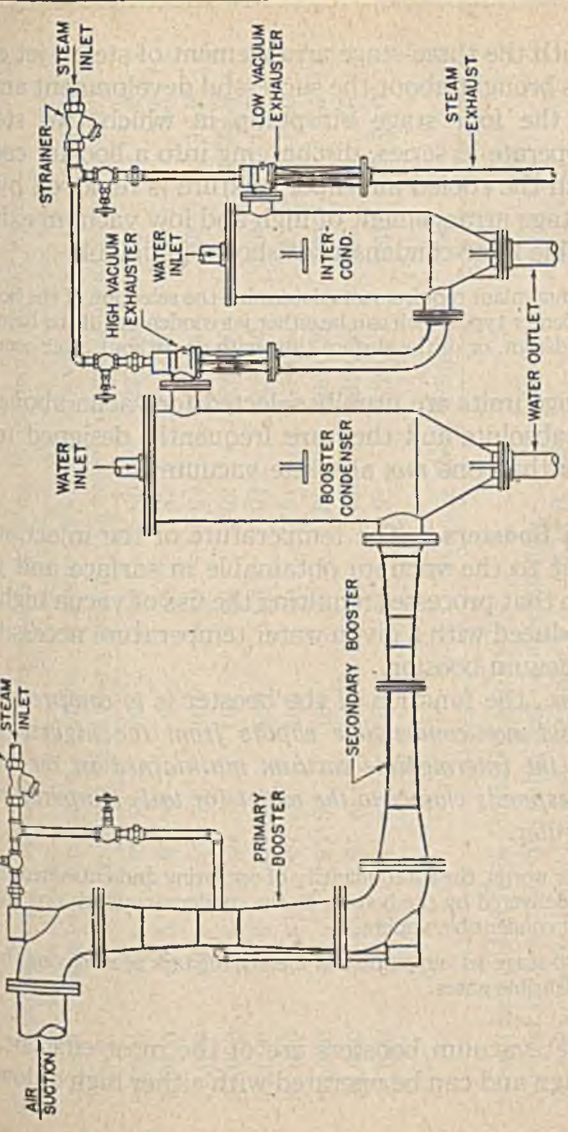


Fig. 14.—Four stage steam jet air pumps with two jet inter condensers.

secured with the three-stage arrangement of steam jet exhausters. This brought about the successful development and application of the four stage air pump in which two steam jet boosters operate in series, discharging into a booster condenser from which the cooled air vapor mixture is removed by means of a two stage arrangement of high and low vacuum exhausters with suitable inter-condenser, as shown in fig. 14.

Individual plant requirements determine the selection of the booster and inter condenser type which can be either jet condensers of the barometric or low level design, or of the surface type with or without after condenser.

Four stage units are usually selected for vacua above two or three *mm* absolute and they are frequently designed to maintain better than one *mm* absolute vacuum.

Vacuum Boosters.—The temperature of the injection water sets a limit to the vacuum obtainable in surface and jet condensers, so that processes requiring the use of vacua higher than can be produced with a given water temperature necessitate the use of a vacuum booster.

Therefore, the function of the booster is *to compress the condensible and non-condensible vapors from the highest (process) vacuum to the intermediate vacuum maintained in the condenser which corresponds closely to the outlet (or tail) temperature of the injection water.*

In other words, the total quantity of operating and entrained steam and vapors is delivered by the booster to the condenser, which precipitates the steam and condensible vapors.

The two-stage jet air pump has merely the task of removing the air and non-condensible gases.

Steam jet vacuum boosters are of the most efficient, single nozzle design and can be operated with either high or low steam pressures.

TEST OF CONDENSER AT CABIN CREEK, VA.

25,000 sq. ft. ⅞-inch—23 ft. long plus 1,500 sq. ft. cooler. 52,000 g.p.m.

LOAD KW.	BAROMETER	OBSERVED VACUUM	CONDENSATE TEMP.	WATER INLET	WATER OUTLET	STEAM CONDENSED	HEAT TRANSFER	TEMP. RISE
23,000	29.6	27.75	96	80	89	232,000	625	9.0°
25,000	29.6	27.70	96	80	89.5	250,000	646	9.5°
32,000	29.6	27.52	101	80	92	328,000	746	12.0°
32,000	29.6	27.53	101	80	92	328,000	750	12.0°
31,000	29.6	27.60	100	80	91	318,000	770	11.0°
31,000	29.61	27.60	100	80	91	318,000	765	11.0°
30,000	29.61	27.62	100	81	90.5	306,000	760	9.5°
29,000	29.61	27.65	99	81	90	295,000	750	9.0°
32,000	29.61	27.54	101	81	92	328,000	780	11.0°
						Average Heat Transfer	732	

Steam being bled from fourth stage only.
Air leakage less than five cubic feet on all readings.
Water velocity through tubes—52,000 g.p.m.=7.12 ft. sec.

Illustrating the application of jet air pump to condenser in actual service. Where conditions of water temperature permit the steam jet air pump is capable of maintaining extremely high vacua; in fact, within a fraction of an inch of the barometer.

Their construction permits the nozzle to be easily removed for examination or cleaning without dismantling any part of the booster body or pipe connections.

The three stage unit shown in fig. 13 is a typical application of a steam jet booster exhausting into a barometric condenser.

Similar arrangements are used extensively for vacuum distillation in oil refineries and for other chemical processes as well as for concentrating and crystallizing of liquids where vacua of ½ in. absolute and less must be maintained.

The non-condensible gases delivered to the barometric condenser are removed and compressed to atmospheric pressure by means of the two-stage air pump with barometric inter-condenser, as shown.

Steam jet boosters are likewise used for deodorizing vegetable oils under high vacuum.

CHAPTER 25

Condensers; Calculations

In the design of a condenser various calculations are necessary to properly proportion it to meet the requirements. These include such calculations as those for:

1. The vacuum that can be obtained.
2. Quantity of cooling water required.
3. Cooling surface.
4. Size of removal pump.
5. Size of wet air pump.
6. Size of dry air pump.
7. Size of condensate pump.

The Vacuum That Can Be Obtained.—By an inspection of any steam table it will be seen that the pressure of the vapor of water depends upon its temperature. That is, any enclosed space partly filled with water and exhausted of air, will be filled with the vapor of the water whose pressure depends upon the temperature of the water. Thus consider the following items from the steam tables:

Vacuum Ins. Mercury	*Absolute Pressure*	*Temperature* Fahrenheit
28	.946	100°
25.85	2.	126.15
23.81	3.	141.52

By inspection of the values for a 25.86 inch vacuum in the condenser the condensate would have to be cooled to 126.15° Fahr. For either an increase or lowering of the temperature of the condensate, the vacuum would decrease or increase respectively. That is, if the temperature increase to 141.52°, the vacuum would fall to 23.81 ins.; again if the condensate be cooled to 100° the vacuum would rise to 28°.

Ques. Could a 28° vacuum be obtained in a condenser with the cooling water at 100°?

Ans. No.

Ques. Why?

Ans. This is impossible in practice as the final temperature of the cooling water which is being heated by the incoming steam would result in a temperature of the condensate higher than that of the vacuum corresponding to that of the initial temperature of the cooling water.

Evidently the final temperature of the cooling water must be somewhat lower than the higher temperature of the steam. This *terminal difference* as it is called is necessary because the heat transfer process is not 100% efficient due to the time element.

Ques. To obtain a given vacuum what then is necessary?

Ans. The temperature of the cooling water should be somewhat lower than the temperature corresponding to the vacuum.

Quantity of Cooling Water Required.—The quantity of cooling water required for the condensing process (per lb. of steam condensed) depends upon numerous conditions as:

1. Required vacuum
2. Initial temperature of the cooling water.
3. Final " " " "

The quantity of cooling water required per lb. of steam to be condensed may be expressed as follows:

$$Q = \frac{H - h}{hf - hi} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

in which

Q = number of lbs. of cooling water to condense one lb. steam
H = total heat above 32° in the steam
h = " " " " " " condensate
hf = " " " " " " cooling water at *final* temperature
hi = " " " " " " cooling water at *initial* temperature

Since H − h is the latent heat of steam (symbol L), the formula may be written:

$$Q = \frac{L}{hf - hi} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

The value of H − h or L, being obtained direct from the steam table making subtraction unnecessary.

Example.—How many pounds of cooling water is required to condense one pound of steam of 90° temperature if the initial and final temperatures of the cooling water be 60° and 90°?

From the steam table:

L, for steam at 90°	= 1041.2	*B.t.u.*
hf, for cooling water at 90° =	58	"
hi, " " " " 60° =	28.08	"

Substituting in formula (2)

$$Q = \frac{1041.2}{58 - 28.08} = \frac{1041.2}{29.92} = 34.8 \text{ lbs.}$$

That is under the conditions of the example it would require 34.9 lbs. of cooling water to condense one lb. of steam. This is the *theoretical* quantity and is not possible in practice, as already pointed out, the final temperature of the cooling water (*hf*) cannot be as high as the temperature of the steam to be condensed.

Approximate Method of Calculating the Cooling Surface.—For an approximation (very nearly, but not exactly) instead of taking the total heats *hi* and *hf* of the cooling water, the temperatures corresponding are used and formula (2) becomes,

$$Q = \frac{L}{T - t} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

in which

T = ***final*** temperature of condensing water in deg. F.
t = ***initial*** " " " " " "

Solving the same example with formula (3)

$$T = 90° \text{ Fahr.}; \ t = 60°$$

Substituting

$$Q = \frac{L}{90 - 60} = \frac{1041.2}{30} = 34.7 \text{ lbs.}$$

Comparing the two methods the results are 34.9 (real value) and 34.7 (approximate value), the approximation being short by:

$$34.8 - 34.7 = .1 \text{ lb.}$$

Terminal Difference.—By definition, the terminal difference is: *The difference between the higher temperature of the steam entering the condenser and the always somewhat lower final temperature of the cooling water* (that is, the injection water of a jet condenser, or the circulating water of a surface condenser).

Ques. Why in practice is this terminal difference necessary?

Ans. As before stated, in practice it is practically impossible to render the heat transfer (from the steam to be condensed to the cooling water) 100% efficient because of the time element.

As can be realized the value of Q in the example just given is considerably less than would obtain in practice.

Ques. What would be a reasonable terminal difference in practice for ordinary conditions?

Ans. Say 10°.

Example.—How many pounds of cooling water are required to condense one pound of steam 90° temperature if the initial temperature of the cooling water be 60° and the terminal difference be 10°?

For a terminal difference of 10°, final temperature of cooling water 90° − 10° = 80° L = 1041.2 and T − t = 80 − 60 = 20

From which, applying formula (3)

$$Q = \frac{1041.2}{20} = 52.1 \text{ lbs. (approx.)}$$

Applying formula (2) $hf = 48.03$ $hi = 28.08$

$$Q = \frac{1041.2}{48.03 - 28.08} = \frac{1041.4}{19.95} \; 52.2.$$

Ques. In modern practice what value is given to the final temperature of the cooling water?

Ans. It is customary to make it 10° to 15° *less* than the temperature of the steam to be condensed.

This factor is dependent upon the design of the condenser.

Range of Design Vacuums.—According to Cameron with various circulating water temperatures, the usual range for surface condensers serving steam turbines is as follows: (For engine service the vacuum is lower, usually 26 ins. or 26.5 ins.)

DESIGN DATA

FOR CONDENSERS FOR STEAM TURBINES

Inlet Water—T_1.	Design Vacuum	Steam Temp.—Ts	Temp. Diff. $TD_1 = (Ts - T_1)$
50° F.	28.5″ to 29.0″	91.7° F. 79.0	41.7° F. 29.0
60	28.25″ to 28.75″	96.7 85.9	36.7 25.9
70	28.0″ to 28.5″	101.1 91.7	31.1 21.7
75	28.0″ to 28.25″	101.1 96.7	26.1 21.7
80	27.75″ to 28.0″	105.0 101.1	25.0 21.1
85	27.5″ to 27.75″	108.6 105.0	23.6 20.0

For the usual design or operating range, the capacity of any condenser with a given temperature and quantity of circulating water is directly proportional to the *temperature difference* (TD).

With efficient design it is possible to handle 10 to 15 or 20 lbs. of steam per sq. ft. of tube surface, depending on the desired vacuum and the quantity of circulating water.

The quantity of circulating water required for a given steam load is determined by the possible *temperature rise* (TR) which in turn is a fraction of the temperature difference (TD) (approx. 40% to 60% for single pass condensers, and 55% to 75% or 80% for two pass condensers, depending principally on the tube size and tube length).

The relation between quantity of circulating water in gallons per min. (Q) and TR irrespective of condenser size, type or design is as follows:

Steam condensed in lbs. per hr. × *B.t.u.* per lb. = Q × 500 × TR

The quantity of heat absorbed by the circulating water is generally assumed as 950 *B.t.u.* per lb. for turbine service or 1,000 *B.t.u.* per lb. for engine service. In special cases where the steam to be condensed is dry or superheated it will be higher.—*Cameron.*

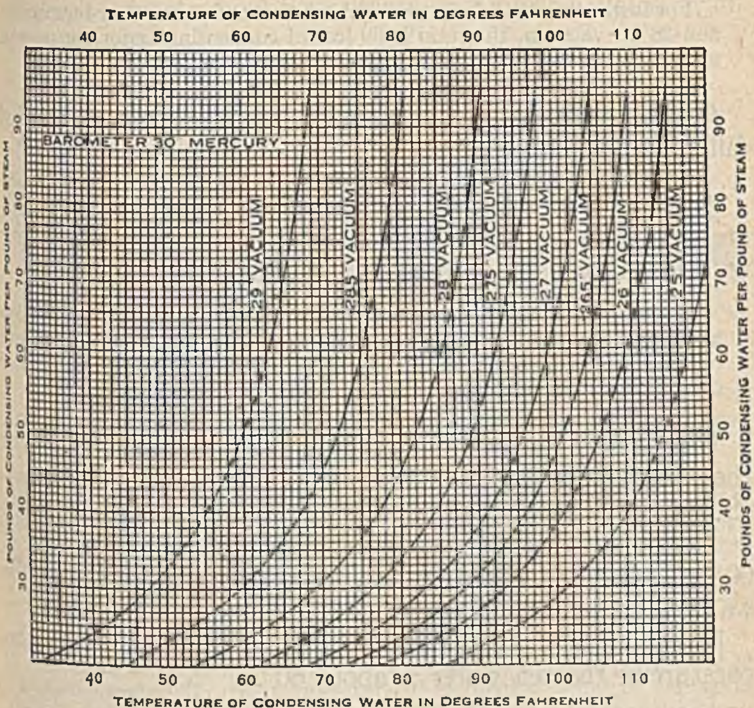


Fig. 1.—Curves indicating the ratio by weight of condensing water to steam. for various vacuum and water temperatures.

The diagram fig. 1 (according to Wheeler) gives curves indicating the ratio by weight of condensing water to steam for various vacua and water temperatures.

To use the curves, add the terminal difference to the water temperature and from the resulting temperature follow vertically to the intersection with the diagonal vacuum curve.

At the point of intersection follow horizontally to the ratio indicated.

For instance, with 15° F. terminal difference, 70° F. water temperature, and 28 in. vacuum, the ratio is 59 lbs. of condensing water per pound of steam.

Ques. In design, what should be noted about terminal differences?

Ans. In general, the terminal difference is necessarily greater with cold water than with hot, and necessarily greater with lower absolute pressures than with higher.

It should be clearly understood that, once all the operating conditions and the design of the condenser are fixed, then the actual terminal difference obtained in practice is the result of physical laws and not of any arbitrary decision to be made at will.

The Cooling Surface.—According to Seaton, in practice with the compound engines, brass condenser tubes 18 *B.w.g.* (Stubs) thick, a condensation of 13 lbs. of steam per sq. ft. per hour, with the cooling water at an initial temperature of 60° is considered fair work when the temperature of the feed water is to be maintained at 120°.

In general practice the following holds good when the temperature of the sea water is about 60°.

Terminal pressure, lbs. abs....	30	20	15	12⅜	10	8	6
Sq. ft. cooling surface per I.H.P.	3	2.5	2.25	2	1.8	1.6	1.5

For ships stationed in the tropics, the allowance should be increased 20%; for ships stationed in cold climates 10% less suffices (*Seaton*).

Cooling Surface for Water Works Condensers.—According to Worthington the basis of heat transfer for water works condensers is **250 B.t.u.** *per sq. ft. per hour of mean temperature difference.*

The amount of cooling surface in square feet necessary to condense a given amount of steam can be calculated from the formula

$$F = \frac{WH}{TU}$$

in which

F = sq. ft. cooling surface.
W = weight (lbs.) of exhaust steam per hour.
H = total heat (*B.t.u.*) per lb. of exhaust steam.
T = mean temperature difference between the circulating water and the exhaust steam.
U = rate of heat transfer in *B.t.u.* per 1 degree mean difference per sq. ft. of condensing surface per hour.

For ordinary calculations H is frequently assumed to be: 930 at 4 in. absolute; 935 at 3 in. absolute; 940 at 2 in. absolute; 945 at 1 in. absolute.

In commercial calculations, a value of 950 *B.t.u.* is taken as approximately correct for H. The value of T is found from the formula

$$T = T_s - \frac{T_0 + T_1}{2}$$

Where T_s is the temperature in steam space (assumed to correspond to absolute pressure); T_1, the temperature of circulating water at condenser *inlet* in deg. F. and T_0, the temperature of circulating water at condenser *outlet* in deg. F.

The Grashof Formula.—According to Grashof a simple arithmetic mean for the temperature difference is not correct, but the following formula developed mathematically by Grashof has been proven in practice to be very accurate, and is used very extensively:

$$D = \frac{T_2 - T_1}{\text{Hyp. Log.}\ \frac{T_s - T_1}{T_s - T_2}}$$

Where D = Mean temperature difference.
T_1 = The lowest temperature of the fluid.
T_2 = The highest temperature of the fluid.
T_s = The temperature of the gas or steam.

The transfer of heat through a unit of condenser tube area per unit of mean temperature difference was early recognized as varying greatly under different conditions. The most apparent variation being an increase with an increase in the velocity of the cooling water.

Many experimenters have carried out exhaustive tests along this line to determine the most practical value, but the results obtained vary greatly owing to the fact that in practice there are encountered certain resistances, which are in addition to the resistance offered by the metallic walls of the tubes.

Ques. In practice with surface condensers what opposes the transference of heat?

Ans. It is opposed by the resistance of the metallic walls of the tubes, the resistance of the steam side of the tube due to oil coating, or air entrained steam, and the resistance on the water side of the tube due to the formation of scale.

Ques. What tends to prevent the formation of a coating of oil on the condenser tubes?

Ans. A high steam velocity over the tubes, and there must be no dead ends or stagnant places in the condenser.

Ques. How are surface condensers generally arranged to avoid the resistance due to air entrained steam?

Ans. They are generally arranged so that the steam sweeps the air ahead to the point of removal.

Ques. What should be noted with respect to the water sides of the tubes?

Ans. Considerable attention should be given to keeping the tubes clean.

A high circulating water velocity will accomplish this to a marked degree, and is a more important reason for using small tubes, and several passes, than is generally recognized.

Ques. What values are given to the co-efficient of heat transmission or *B.t.u.* per sq. ft. per degree difference per hour?

Ans. It is generally taken in practice at 300 to 500, depending upon the degree of vacuum, condenser design and velocity of cooling water.

Velocity of Flow.—According to Whitham, the velocity of flow of the circulating water through the tubes should be between the limits 400 and 700 ft. per minute. As given by Marks, the mimimum allowable spacing of tubes is as follows:

Outside diameter of tube, ins......	⅝	¾	⅞	1	1¼
Pitch of tubes, ins................	15/16	1 1/16	1¼	1⅜	1⅝
Number of tubes per sq. ft. of plate	189	147	106	88	63

Jet Condenser Air Pump Calculations.—Now since the jet condenser pump must handle everything that must be pumped out of the condenser, in order to determine its size it is necessary to calculate:

1. Amount of steam to be condensed.
2. Amount of water required to condense the steam.
3. Amount of air and other non-condensable vapors to be removed from the condenser.

Each pound of injection water will absorb from the steam to be condensed, a number of heat units *approximately* equal to its rise in temperature in passing through the condenser, and the number of heat units to be taken out of each pound of steam to cause condensation will be equal to its total heat less the heat in the resulting condensate, that is,

$$\text{Quantity of injection water} = \frac{\text{total heat of steam} - \text{heat in condensate}}{\text{rise in temperature of injection water}}$$

or using the usual symbols,

$$Q = \frac{H - h}{T - t} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

in which

H = total heat in one pound of the ***steam.***

h = heat in one pound of the ***condensate.***

t = temperature at which the injection water ***enters*** condenser.

T = temperature at which the injection water ***leaves*** condenser.

Now, evidently, since the pump must handle both the injection water and the condensate, the total amount of water to be handled is

$$\underset{\text{total weight}}{Q'} = \Bigl(\underset{\text{condensate}}{1} + \underset{\text{injection water}}{\frac{H - h}{T - t}}\Bigr) \times \underset{\text{weight of steam}}{W} \quad \ldots\ldots (5)$$

in which

Q′ = total *weight* of water entering condenser.

W = Weight of steam to be condensed in lbs.

Water contains mechanically mixed with it $^1/_{20}$ or 5% of its volume of air at atmospheric pressure. If P = atmospheric pressure and p = absolute pressure in condenser, then a cu. ft. of water when it has entered the condenser is represented by .95 of a cu. ft. of water and $.05 \times P \div p$ of a cu. ft. of air.

Now if Q″ = the total *volume* of water entering the condenser per minute, T_1, temperature of the condenser, T_2, temperature of the cooling water (*before* entering condenser), then (according to Seaton) .95 × Q″ = volume of water in cu. ft. to be pumped from condenser per min. and

$$\text{the quantity of air} = .05 \times \frac{P}{p} \times \frac{T_2 + 461^\circ}{T_1 + 461^\circ}$$

hence the total volume to be abstracted per minute that is,

$$\text{total water and air} = .95\,Q'' \times .5\,\frac{P}{p} \times \frac{T_2 + 461^\circ}{T_1 + 461^\circ} \quad \ldots\ldots\ldots (6)$$

Example.—A 100-horse power marine engine requires 30 lbs. of feed water per horse power per hour. If the pressure in the jet condenser be 2 lbs. absolute (25.85 ins. vacuum), how much injection water is required and what size pump if the initial temperature of the injection water be 60° and the final temperature 110°?

Total steam to be condensed per minute, or

$$W = \frac{100 \times 30}{60} = 50 \text{ lbs.}$$

Since from the steam table the total heat in 1 lb. of steam at 2 lbs. absolute pressure is 1,115 *B.t.u.*, and the heat in the condensate 94 *B.t.u.*; substituting in (5)

$$Q' = \left(1 + \frac{1{,}115 - 94}{110 - 60}\right) 50 = 1{,}071 \text{ lbs. approx.}$$

The weight of water at 110° being 61.89 lbs. per cu. ft., then its volume at 110° or

$$Q'' = 1{,}071 \div 61.89 = 17.3 \text{ cu. ft.}$$

95% of which is water and 5% air. Hence the total volume to be abstracted from the condenser per minute is, taking the temperature of the condenser at 120°:

$$95\% \times 17.3 + 5\% \times \frac{14.7}{2} \times \frac{60 + 461}{120 + 461} = 16.56 \text{ cu. ft.}$$

Now the usual practice is to use a pump having a displacement of twice the volume to be pumped, that is, to let the pump fill half full of water, the remainder being occupied by the expanded air. Accordingly, given displacement is

$$16.56 \times 2 = 33.12 \text{ cu. ft. per minute}$$

The normal piston speed should be about 100 ft. per minute, hence area of piston = 33.12 ÷ 100 = .331 sq. ft., or .331 × 144 = 147.7 sq. ins.

$$\text{diam. of piston corresponding} = \sqrt{\frac{\text{area}}{.7854}} = \sqrt{\frac{47.7}{.7854}} = 8 \text{ ins. approx.}$$

and the length of stroke will depend on the number of strokes per minute, which, for say 60 strokes per minute, is

$$33.12 \div 60 = .663 \text{ ft. or } .663 \times 12 = 7.95, \text{ say } 8 \text{ ins.}$$

Ques. For accuracy what kind of heats should be used in condenser calculations?

Ans. Total heats instead of sensible heats or temperatures.

It should be understood that standard *B.t.u.* used is *mean B.t.u.* which is defined as: $\frac{1}{180}$ *part of the heat required to raise the temperature of one pound of water from* 32° *to* 212° *F*. In fact the heat required to raise the temperature of water 1° depends upon the temperature. Thus from the steam table:

Heat in 1 lb. water at 80° Fahr......	= 48.03 *B.t.u.*
Heat in 1 lb. water at 70° Fahr......	= 38.06
Heat to raise temperature of 1 lb. of water 10° between 70° and 80°	9.97 *B.t.u.*

This is the actual heat which is a little different from the result obtained by subtracting temperatures, that is 9.97 *B.t.u.* and not 80–70 or 10 *B.t.u.*

Surface Condenser Wet Air Pump.—In a surface condenser the pump has to handle only the condensate and air, the cooling or "circulating" water being pumped by the circulating pump. As compared with a jet condenser, the pump handles a much smaller quantity of water and a relatively large volume of air. The latter item is important because the air volume to be displaced is much larger than the water volume. The air entering by leakage is uncertain and may be 3 or more times as much as was liberated by the water.

Since the pressure in the *l.p.* cylinder of the engine is most of the time below atmospheric pressure, considerable air may leak in through the stuffing boxes unless they be tight.

The practice of some pump companies is to give the air pump a displacement equal to 20 times the volume of the condensate, if it be a horizontal double acting pump, and 12 times if vertical single acting.

According to Whitham the usual practice is to make the air pump for a surface condenser one half the capacity of one required for a jet condenser. This will enable the surface condenser to be used as a jet condenser in case of emergency. The air pump should *always* be placed below the condenser for best results (though this is not always possible with keel condensers), and the delivery valves should be water sealed.

Ques. Why are the proportions 20 and 12 given for horizontal and vertical pumps respectively?

Ans. The vertical pump is more efficient than the horizontal pump.

Formula for Wet Air Pumps.—According to Worthington it is considered good practice to make the capacity of the wet air pump for a surface condenser equal to 20 or 30 times the pounds of steam condensed per unit of time. The following formula may serve as a guide to the multiple to use:

$$DV = S\,1.5\left(\frac{54}{P_a} + 1\right)$$

in which

DV = Displacement of air pump in lbs. per hour

S = Pounds of steam per hour

P_a = Absolute pressure in inches of mercury

$\frac{54}{P_a}$ = Amount of vapor arising from the steam.

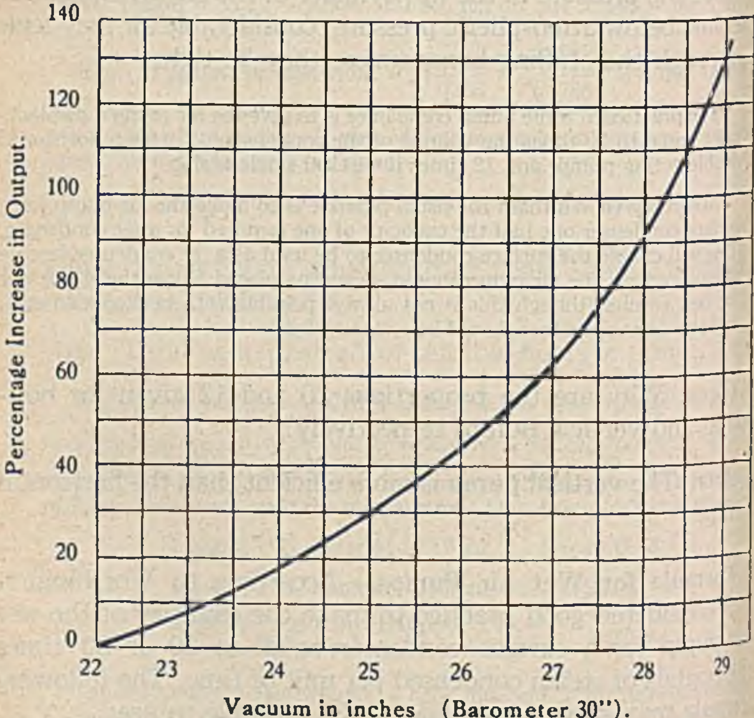


Fig. 2.—Economy at high vacuum. The curve, according to Manistee shows the output from an exhaust steam turbine as the vacuum increases from 22″ to 29″ from which will be seen that at 28½″ the increase in output is over 100 per cent of the power of the turbine with only 22″ of vacuum, and that the rate of increase rises rapidly as the vacuum approaches the higher limits. Thus;—the addition of 1″ of vacuum from 23″ to 24″ adds 10 per cent, but addition of 1″ vacuum from 28″ to 29″ adds 40 per cent.

Dry Air Pumps.—For high vacuum dry air pumps are necessary. The displacement of the dry air pump is, according to Worthington,

$$DV = \frac{54}{P_a} \times S, \text{ approximately.}$$

Condensate Pump.—This pump handles the condensate and nothing else and its displacement per minute is accordingly equal to the pounds of condensate per minute, that is

$$\text{Displacement} = \frac{\text{lbs. of steam per hour}}{\text{lbs. per gallon} \times 60}$$

taking 8.33 lbs. as approximate weight of one gallon

Displacement per minute in gallons

$$= \frac{\text{lbs. of steam per hour}}{8.33 \times 60}$$

$$= \frac{\text{lbs. of steam per hour}}{500}$$

Fig. 673.—*Text continued.*

to the power of the turbine on the same percentage scale for economy. High vacuum condensing plants therefore become essential. It must be borne in mind that the volume occupied by exhaust steam increases rapidly as the vacuum approaches the barometer reading, that is, the volume varies inversely as the absolute pressure. For instance, the volume is doubled by an increase of vacuum from 28″ to 29″ with a 30″ barometer. For this reason the Condenser vacuum for reciprocating engines is usually limited to 26″ or 27″ because the exhaust ports of a reciprocating engine are not generally large enough to take advantage of any further expansion of the steam. In this case the important point in a condenser is the maintenance of a steady vacuum under fluctuating air leakages or under any overload of the engine, also immunity from break-down if the steam is suddenly reduced, causing the vacuum to rise

The condensate of steam exhausting into a 25.8 inch vacuum will have a temperature of 126° Fahr. At this temperature one cu. ft. of the condensate weighs 61.61 lbs. and one cu. in.

weighs $\frac{61.61 \times}{1728} = .0357$ lb.

Example.—An engine exhausts 1500 lbs. of steam per hour. With condensate at a temperature of 126° corresponding to a weight of 0.357 lb. per cu. in., find displacement per minute of condensate pump.

Volume of condensate to be handled per minute

$$= \frac{1500}{0.357 \times 60} = 70 \text{ cu. ins.}$$

A condensate pump used in connection with a surface condenser (with a dry air pump) according to one authority is generally given a displacement of 2 to 3 times the volume of the condensate.

CHAPTER 26

Cooling Ponds

By definition, a cooling pond is *a shallow reservoir for cooling the water used for condensing steam, the large amount of surface area provided cools the water as it returns from the circulating pump discharge.**

Ques. Where are cooling ponds used?

Ans. In places where water for cooling is scarce or expensive or where it is rendered unfit for use in condensers by pollution from waste products of manufacturers.

Ques. What happens when the water is used over and over again?

Ans. There is a small loss by evaporation which must be made up from an outside source.

Ques. In general how much cooling water is required to condense steam at ordinary degrees of vacuum?

*NOTE.—Another name for cooling pond is *lodge*, but this name is seldom used.

Ans. About 25 to 35 times the feed water, depending upon the temperature of the cooling water.

Example.—A 100 horse power engine runs on 30 lbs. of feed water per hour per horse power. If the cooling water for the condenser be 30 times the feed water, how many gallons of cooling water are required for a 10 hour run?

$$\text{Total feed water} = 30 \times 100 \times 10 = 30{,}000 \text{ lbs. per day.}$$

$$\text{Total cooling water} = \frac{30{,}000 \times 30}{8\frac{1}{3}} = 108{,}043 \text{ gals. per day.}$$

Evidently at the usual City rates for water, this consumption would be prohibited, or at least the expense would largely offset the saving by condensing.

Ques. Where are cooling ponds used?

Ans. In sparsely settled districts where land is cheap.

Ques. What is used in place of cooling ponds where land is expensive or the space required is not available?

Ans. Cooling towers.*

Classification of Cooling Ponds.—There are several types of cooling ponds which may be classed:

1. With respect to the cooling method used, as

a. Natural.

b. Spray $\begin{cases} \text{single deck} \\ \text{double deck} \end{cases}$

*NOTE.—Cooling towers are presented in the Chapter following.

2. With respect to circulation, as

   a. Natural flow.

   b. Directed flow.

3. With respect to water capacity, as

   a. Shallow.

   b. Deep.

4. With respect to provision for preventing drift loss, as

   a. open.

   b. Louvre fence.

Natural Ponds.—By definition a natural pond is *one which the circulation or flow is* 1, *non-directed*, 2, *directed but not forced*.

Ques. What is required for a natural pond?

Ans. Considerable area.

Ques. What is the advantage?

Ans. The cost of a forced circulation pump is avoided.

Non-directed Flow Natural Pond.—The essentials of this type are shown in figs. 1 and 2. As shown the hot water discharge is located at the remote end and the intake at the bottom of the near end.

Ques. Is the cooling effect efficient in this type of pond?

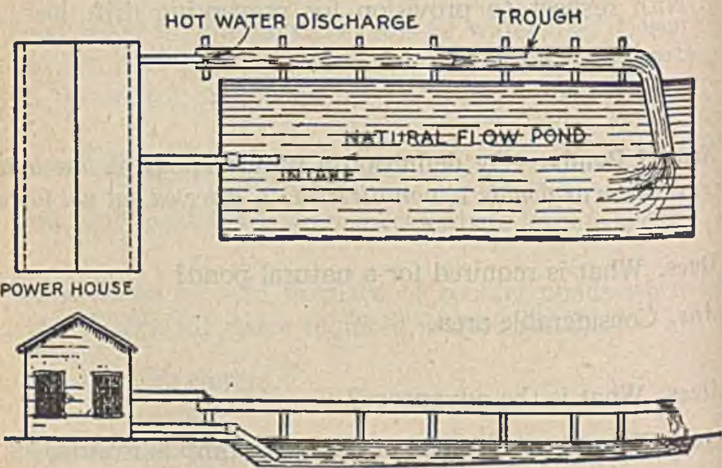
Ans. No.

That is to say, the *B.t.u.* given off per sq. ft. of cooling surface is small compared to other types. Accordingly in design great care should be taken to have the pond large enough for severest conditions.

Ques. How does cooling take place?

Ans. By radiation, convection and *evaporation.*

Ques. What is the relative importance of these means of cooling?



Figs. 1 and 2.—Non-directed flow natural cooling pond suitable for long and narrow lots.

Ans. In cold weather cooling takes place principally by radiation and convection, but in warm weather chiefly by evaporation.

Ques. For conditions prevailing in northwestern United States, how much natural cooling pond surface is required?

Ans. 2,500 sq. ft. per boiler horse power* at 26 inch vacuum.†

Ques. In general how much heat is absorbed by the evaporation of one pound of water?

Ans. About 1,000 *B.t.u.*

Ques. What determines the rapidity of evaporation?

Ans. 1, the temperature of the water and 2, the vapor tension in the air in immediate contact with the water.

In ordinary air, the vapor present is generally in a condition corresponding to superheated steam, that is, the air is not saturated. If saturated air be brought into contact with colder water, the cooling of the vapor will cause some of it to be precipitated out of the air. Again, if saturated air be brought into contact with warmer water, some of the latter will pass into the form of vapor.

The rise in temperature of the air from contact with hot water will greatly increase the water carrying capacity of the air, enabling a large amount of heat to be absorbed through evaporation of the water.

Directed Flow Natural Ponds.—In this type, the circulation or flow of the cooling water is directed by a series of baffles, similar to the method used in directing the flow of steam in a condenser. The essentials of this type are shown in figs. 3 and 4.

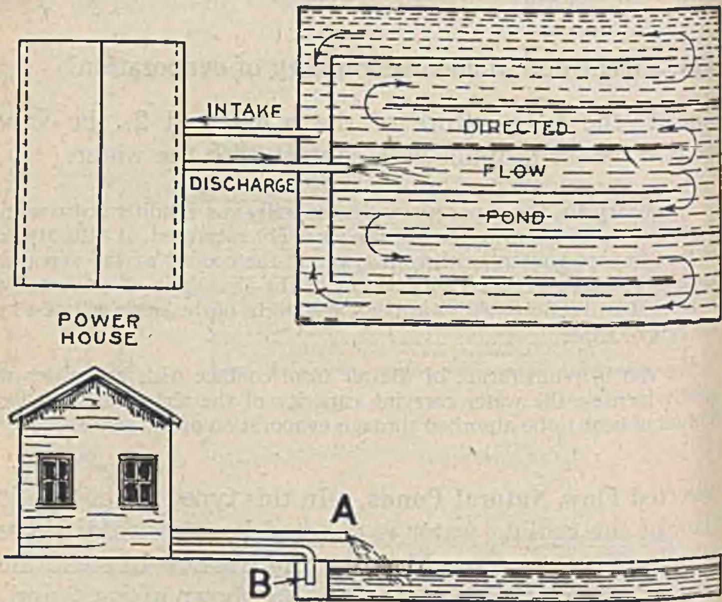
As shown, the baffles which direct the flow are plainly seen in fig. 3. Here the discharge enters at a central point at one side of the pond, flows to the opposite side, there divides into two streams, each of which traverses the length of the pond three times before uniting in the intake passage.

*NOTE.—By definition one boiler horse power is *the evaporation of* 34½ *lbs. of feed water per hour from and at* 212°, that is, from a feed water temperature of 212° Fahr. into steam at the same temperature.

†NOTE.—According to Fernall and Oriak.

Ques. What is the effect of this arrangement?

Ans. The cooling water travels a much larger distance from the discharge outlet to the intake and consequently the velocity of flow is much greater than with non-directed flow, resulting in more efficient cooling.



Figs. 3 and 4.—Directed flow natural cooling pond. **In operation,** the hot water enters the middle channel at A, and on reaching the far end divides into two currents, being directed by the baffle walls so as to traverse the pond several times before uniting at the intake point B.

Ques. What is the comparison between the non-directed and directed flow types?

Ans. Where a long lot is available, the non-directed type avoids the expense of baffle walls, but this is partially offset by the cost of the long trough.

Spray Ponds.—Where the space available is inadequate for a natural cooling pond the spray pond is used. Here the hot water from the condenser is sprayed over a pond (or, where space is very limited, over a roof), through a multiplicity of jets. As it passes through the air in a finely divided state, its surface area is greatly increased, thus intensifying the cooling by evaporation; hence, for a given cooling capacity the size of pond is much less than that of a natural pond.

The essentials of the single deck and double deck spray ponds are shown in figs. 5 and 6 respectively.

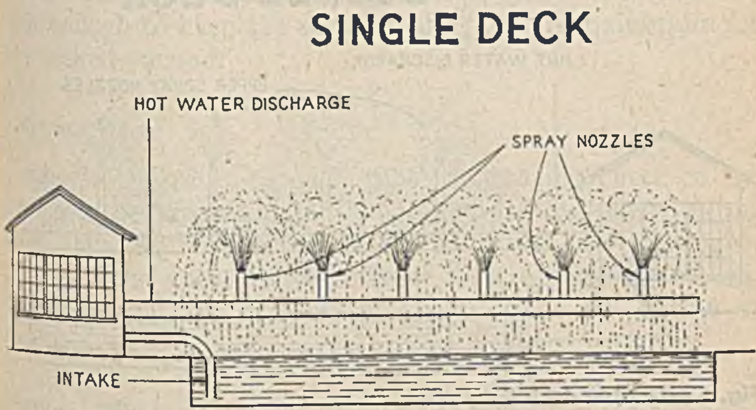


Fig. 5.—**Single deck** spray pond.

Ques. How are spray ponds proportioned as to cooling surface?

Ans. About 4 *sq. ft. of surface are required for a boiler horse power to condense steam at* 26 *ins. vacuum.*

Ques. What is the usual range of cooling?

Ans. It varies from 20° to 40° Fahr. depending upon conditions.

Ques. How may the water be cooled over a large range?

Ans. By spraying more water than passes through the condenser or heating medium.

This is accomplished by the double deck system (fig. 6). By this method it is possible to cool water within a few degrees of the wet bulb temperature of the surrounding air.*

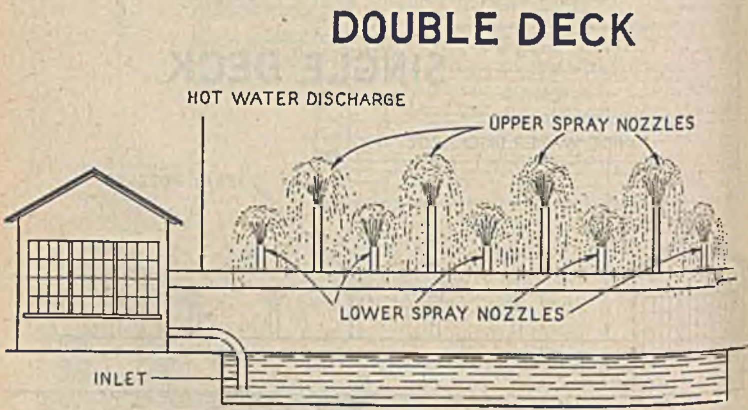


Fig. 6.—Double deck spray pond.

Ques. With a cooling pond where is the natural place to take the feed water?

*NOTE.—According to one authority a series of careful tests extending over a period of several weeks showed that the average amount of heat dissipated from the surface of a natural cooling pond with directed flow is 3.5 *B.t.u. per sq. ft. per hour per one degree difference*, and that the average heat dissipated by a spray cooling pond is 127 *B.t.u. per sq. ft. per hour per one degree difference* or approximately 36 times as much. This shows that a *natural* cooling pond capable of taking care of a 100 horse power plant, can be increased in capacity to take care of a 3,600 horse power plant by the addition of a spray cooling system.

Ans. Not from the pond but from the discharge pipe line between the condenser and the pond.

Ques. Why?

Ans. To save the heat in that portion of the condensing water delivered back to the boilers.

Ques. What precaution should be taken with respect to the temperature of the cooling water?

Ans. The ratio of condensing water to feed water should be large enough to keep the water in the pond in equilibrium for any desired vacuum.

Ques. Why?

Ans. If the ratio be such that the heating effect of the condenser be greater than the desired cooling effect of the pond, thus the temperature of the pond will rise to a higher point and the vacuum will decrease a corresponding amount, thus impairing the economical operation of the engine or turbine.

Ques. What creates air circulation when there is no breeze?

Ans. An effective current of air is created in an upward direction around each nozzle, due to the movement of the spray as well as to the heating effect which the spray has on the air which comes in contact with the water, thus rapidly carrying away the warm, moist air produced, and replacing it with cool, dry air brought in from all sides over the surface of the pond.

Ques. When are spray cooling ponds most efficient?

Ans. In extremely hot weather when high humidity prevails than in cool weather with low humidity.

Ques. How are cooling ponds usually constructed?

Ans. Earthen ponds usually answer every requirement.

Fig. 7.—Turbine type spray nozzle, showing removable turbine center for imparting the rotary motion to the liquid.

When provided with a grassed bank they make a neat water-tight. economical installation. Unless greater depth is needed for storage purposes, the water is rarely more than 3 feet in depth. From a cooling standpoint a pond 6 inches deep will usually give as good results as one 10 feet deep. Ponds are made deeper to provide storage for fire protection purposes.

An important feature in any spray pond is the spray arrangement over the pond or basin, and in connection with that the number and size of nozzles adopted. Many a spray system is giving poor cooling results because of the faulty engineering in connection with the pond layout.

The satisfactory handling of the above features of the design and the ability to use to the best advantage existing conditions, such as old ponds, canals, rivers and roofs, is dependent on the experience and foresight of the engineers to whom the work is intrusted. Where ground space is restricted the sprays can be double decked or arranged on the roof.

Water Loss.—Water is lost from a cooling pond by evaporation and drift.

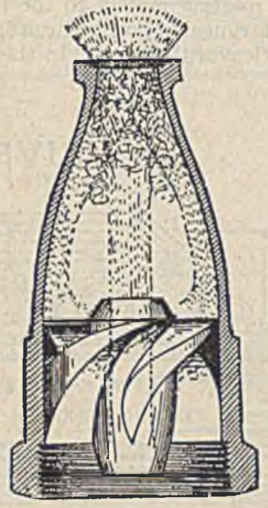


Fig. 8.—Turbine type spray nozzle with center jet nozzle. **In operation,** some of the water to be sprayed passes through the outer turbinated passages and is gradually given a rapid rotating motion. The non-rotary straight central jet strikes this rotating mass of water at a point just below the orifice in the space called the mixing chamber, resulting in a mixing or blending of the rotary and non-rotary jets and compelling issuance of the water from the orifice in a fine flaring spray.

Ques. What is the average combined loss?

Ans. Between 1 and 2 per cent of the amount of water sprayed when the spray particles are sufficiently heavy to settle within the pond limits.

Spray Nozzles.—An important feature of a spray pond is the spray nozzles. The object of these nozzles is to produce a fine, uniform spray at low operating pressure without clogging. They are provided with removable turbine centers having large passages.

A typical nozzle is shown in fig. 7. The turbine center is of such shape as to impart a rapid rotating motion to the liquid passing through it, producing a strong centrifugal action and causing the liquid to break up into a fine spray as it leaves the nozzle. In this nozzle at 5 lbs. pressure

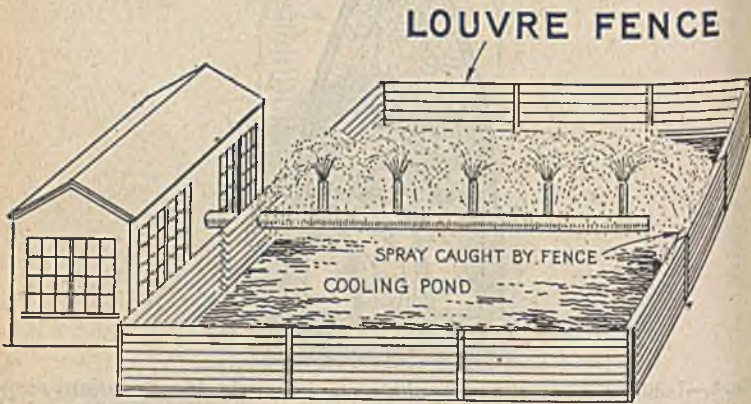


Fig. 9.—Louvre fence for preventing loss of spray water by air currents.

the spray issues forth in the form of an inverted cone composed of fine particles that will settle within the spray pond limit. At 7 lbs. pressure the liquid is broken up into a mist producing maximum cooling effect at low pressure.

Another type of spray nozzle, as shown in fig. 8, the turbine center has a central driving jet which impinges on the rotating water at the orifice, causing it to be ejected as a fine, dense uniform spray.

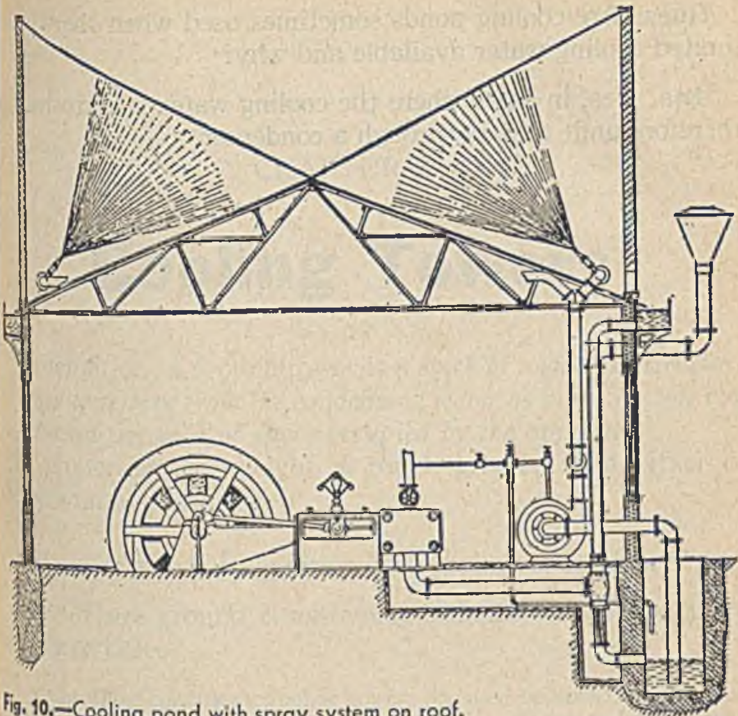


Fig. 10.—Cooling pond with spray system on roof.

Louvre Fence.—In order to prevent loss of water by the spray being carried away by air currents, the sides of spray ponds are sometimes extended or protected by a fence of a type called *louvre*. The essentials are shown in fig. 9.

Ques. What other duty does the louvre fence sometimes perform besides saving water?

Ans. Sometimes the fence is necessary to protect a power plant when located very near the pond.

Ques. Are cooling ponds sometimes used when there is unlimited cooling water available and why?

Ans. Yes, in cases where the cooling water is corrosive and therefore unfit to pass through a condenser.

CHAPTER 27

Cooling Towers

By definition, a cooling tower is *a stack or tower-like structure designed to remove from the condensing water as much heat as can be extracted per unit of space occupied by the apparatus.*

In proportion the height is much greater than either of the horizontal dimensions.

Ques. Under what conditions are cooling towers used?

Ans. Where ground is extremely valuable or not available as in large cities.

Under these conditions, cooling towers are used because in the first instance they occupy the least amount of space for a given cooling capacity and where ground is not available they may be placed on the roof.

Ques. Is the cooling tower the most economical method of cooling the condensing water?

Ans. No.

It must be regarded as an evil necessary to meet the exigencies of the case.

Ques. What are the conditions of cooling upon which the design of cooling towers is based?

Ans. The construction should be such as to expose the water to the cooling influence of the atmosphere while in a film or fine rain state.

Ques. In what way is the process mentioned in the last answer assisted?

Ans. By evaporation of part of the condensing water.

Ques. How is this effect augmented?

Ans. By counter currents of air maintained by side ventilation natural draught (using the tower as a chimney) or by forced draught with a fan.

Ques. What is the path of the condensing water in traversing the tower?

Ans. It is sprayed from the top, falling by gravity into a sump or reservoir at the bottom of the tower.

Ques. What provision is made to maintain the water in state of film or spray during its descent?

Ans. Various methods are used to secure a cellular construction of brush wood, earthenware pipes, wire mats, diaphragms or other baffles.

Ques. How is the loss of condensing water made up?

Ans. By admitting a supply from the public water mains or wells.

Classification of Cooling Towers.—Cooling towers may be classified:

1. With respect to the nature of the water baffles, as

a. Brush wood.
b. Earthenware.
c. Wire mat.
d. Checker work, etc.

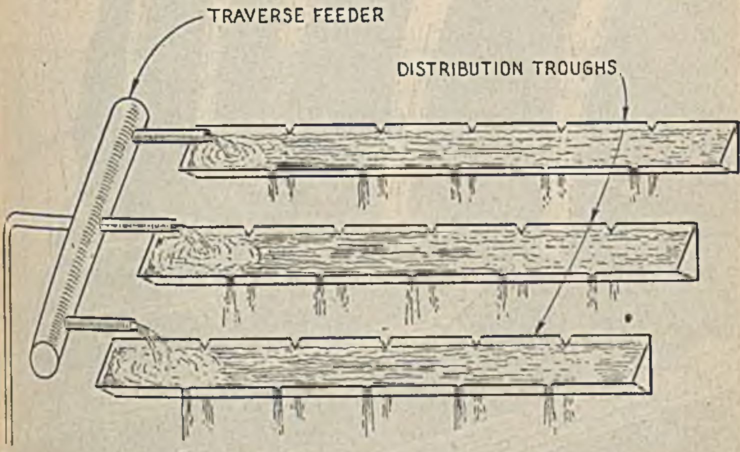


Fig. 1.—Detail showing **distribution deck consisting of transverse feeder and distribution troughs.**

2. With respect to ventilation, as

a. Natural draught (open).
b. Induced draught (chimney).
c. Forced draught (fan).
d. Combined forced and induced draught.

Distributors.—Since a number of sprinklers at the top of a cooling tower would result in a loss of water due to lateral wind currents, usually a distribution deck (variously designed) is used. The essentials are shown in fig. 1.

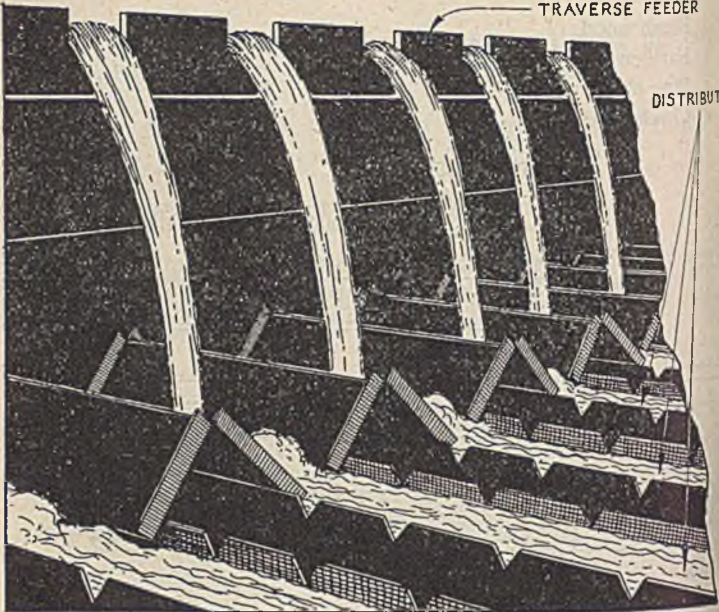


Fig. 2.—Typical distribution deck. In construction and operation, the transverse feeder consists of a steel box divided into two parts by means of a longitudinal position. The water enters at either end through a tapped flange. One of these ends is to be plugged and will serve in cleaning out the lower compartment. A series of vertical pipes is furnished in the partition, and the water issues from these pipes, which are adjustable, so that the amount can be regulated to distribute uniformly throughout the length of the trough. The two sides of the trough are notched at intervals. Through these notches the water overflows into the distributing troughs. The feeder is made of heavy steel plate and thoroughly painted at the shop before shipment. A steel walk-way is provided running the full length of the tower alongside of the feeder, giving access to the regulating pipes in the feeder, and to all portions of the tower at the top. A steel ladder is provided to give access to the walk-way from the base of the tower. The distributors are notched on the sides the full length, at intervals, and the water overflows through these notches to the trough deck below.

As shown it consists of a transverse feeder with outlets feeding into a series of parallel troughs. These troughs have notches at close intervals through which the water flows, being divided into a multiplicity of fine streams and therefore distributed over a considerable area.

An example of actual construction is shown in fig. 2. At the top of the tower is provided a feeder placed in the center of the tower, and running nearly the full length of the tower on its longest dimension.

At right angles to the main trough and extending at both sides of the trough are a series of distributors into which the water from the feeder discharges, as shown.

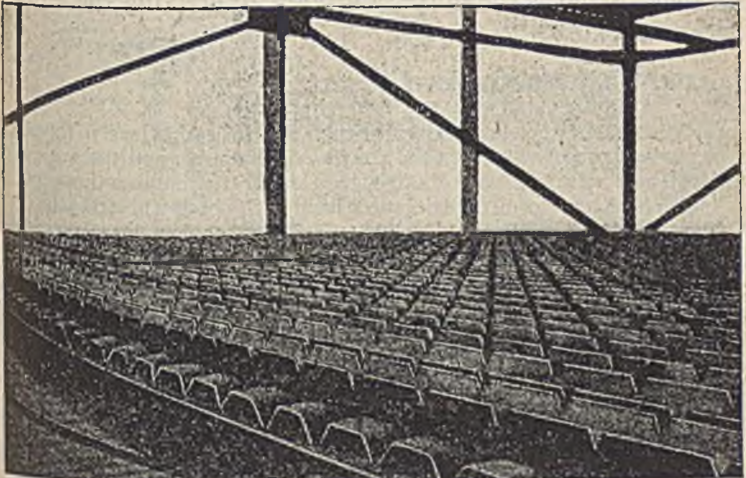


Fig. 3.—Typical distribution troughs showing construction.

The arrangement consists of:

1. A transverse distribution deck
2. Trough deck

Fig. 2 shows both decks and fig. 3 the distribution deck more in detail.

The trough deck system is made up of units of proper size and placed side by side so as to give the proper area to suit the capacity of the tower.

Each unit consists of a series of parallel troughs notched on the two sides. These parallel troughs rest on a similar series of parallel angle pieces running at right angles and bolted together with ⅜ in. bolts and double washers. The angle pieces stiffen the troughs so that they will maintain their level when full of water, and will also bear the weight of a man without deflection.

Ques. What is the important point in erecting these trough decks?

Ans. They *must* be level.

The system when level will cause the water to seek its level in the troughs regardless of how it enters the troughs from above, and the water being level, will naturally leave the troughs through the notches throughout its entire length uniformly. The water from each notch will strike the top of the supporting angle at the center and will tend to flow down both sides of the angle pieces in equal proportion.

The arrangement of troughs and angle pieces at right angles will prevent water passing directly through, but will enable a free passage of air up and around all parts of the angles and the troughs, thus exposing the water to the air under favorable conditions.

Ques. How are the trough decks arranged?

Ans. In series one directly below the other.

The troughs in one deck are arranged to run at right angles to these in the deck above and below.

Ques. What is the object of this arrangement?

Ans. This method will rectify the distribution of the water from deck to deck, as the tendency of the wind is to blow the water to the leeward side of the tower.

Ques. What would happen otherwise?

Ans. Without such rectification, the water concentrates at the leeward side of the tower and at the bottom of the tower, will splash in considerable volume. This will result in poor cooling effect and considerable loss of water by splashing out of the tower.

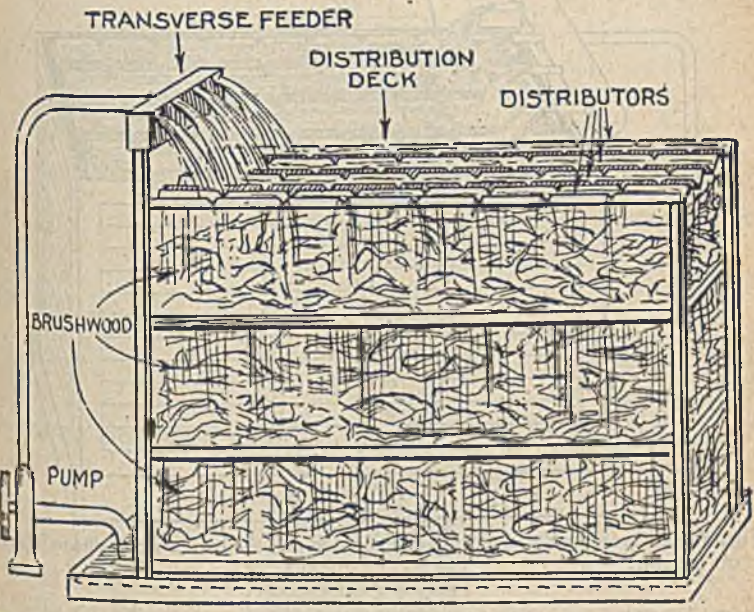


Fig. 4.—Open natural draught brushwood cooling tower. This is about the simplest form of tower, being of very ordinary construction.

With the trough deck system the distribution is equalized at each deck so that no matter in which direction the wind may blow, the water is always broken up into fine particles, and exposes the maximum water surface to the surrounding air.

Open Natural Draught Tower.—Fig. 4 shows an open natural draught tower of ordinary construction. Evidently this is a makeshift one, as no protection is provided to avoid loss of water, especially in windy places where the water is liable to be blown out through the sides of the tower.

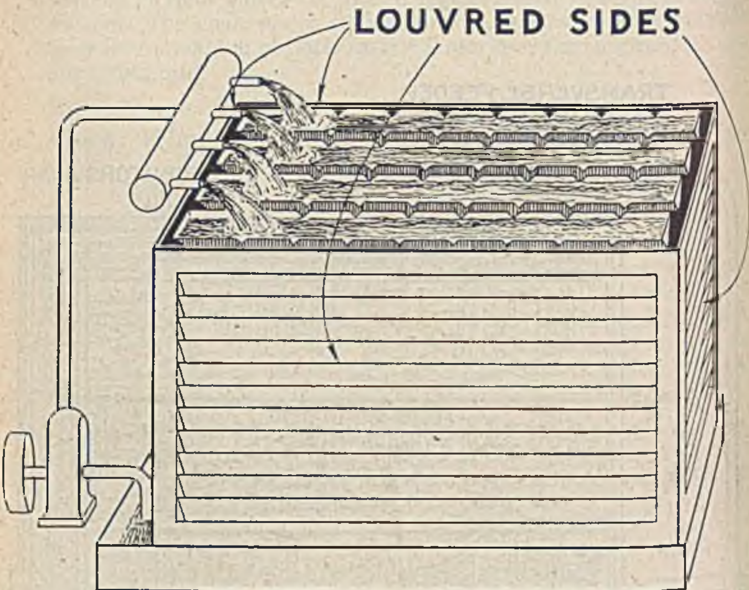


Fig. 5.—Louvered sides or slatted enclosure to prevent loss of water due to lateral air currents.

Ques. How could this loss of water be avoided?

Ans. By surrounding the tower with louvered sides as shown in fig. 5.

Ques. Mention an inherent defect of the natural draught tower.

Ans. It must of necessity be of larger dimensions than the forced draught type.

Ques. What may be said in favor of natural draught?

Ans. The expense of fan operations is avoided.

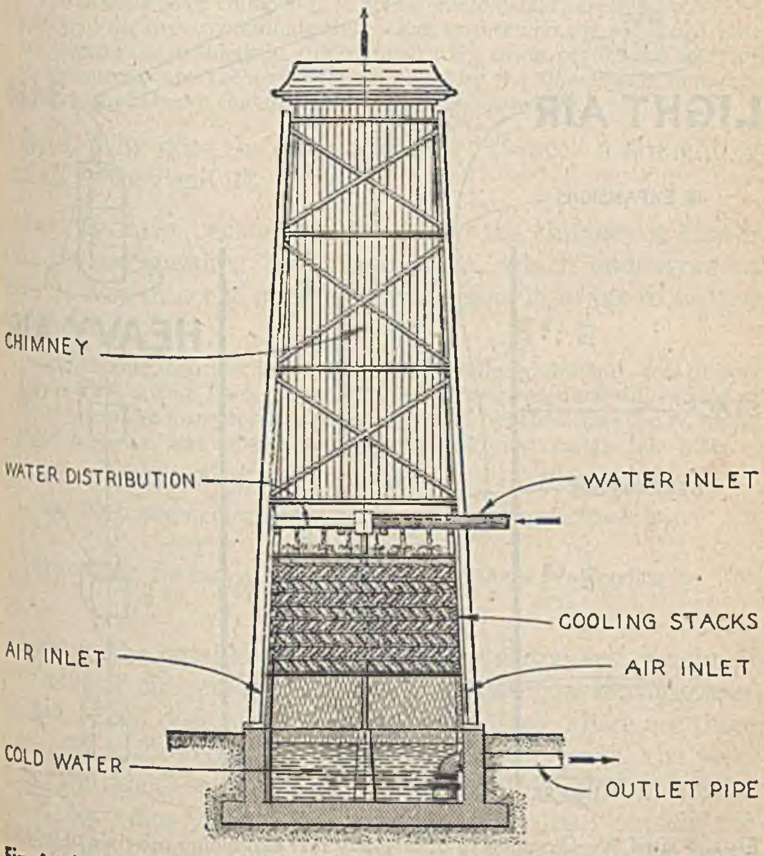
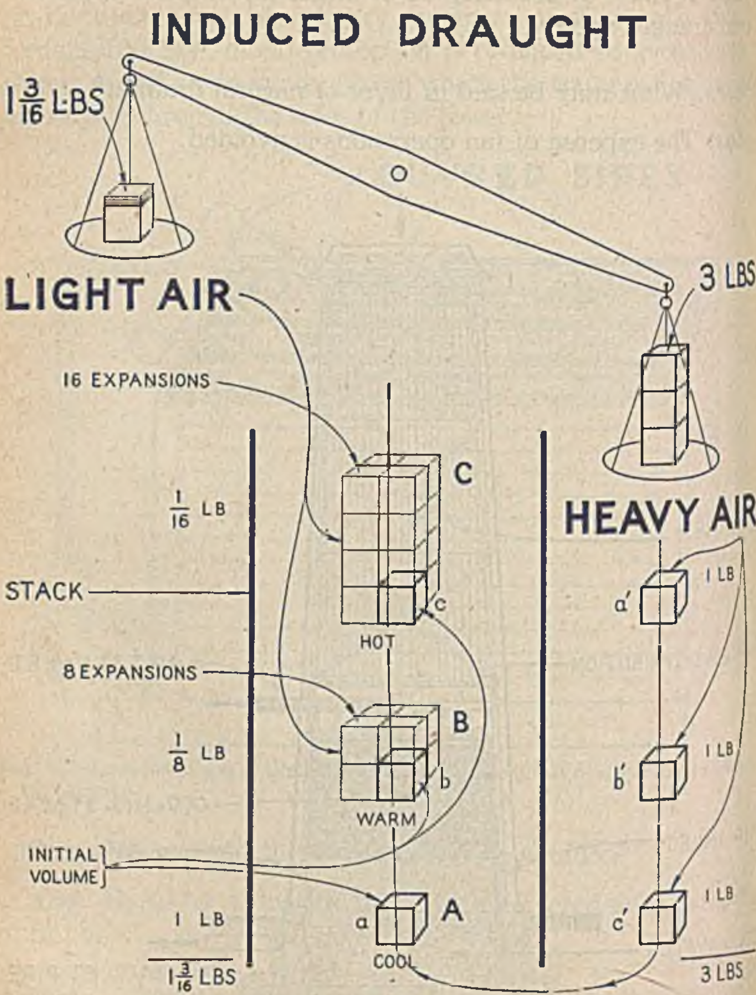


Fig. 6.—Induced draught cooling tower with zigzag cooling stacks.



Figs. 7 and 8.—Diagrams to accompany text explaining principle of induced draught.

Not only the expense of operation, but also upkeep of the fan and its driver.

Induced Draught Cooling Towers.—By definition, induced draught is *a form of draught accelerated by the effect of an enclosing structure such as a chimney or stack.* A typical induced draught chimney tower is shown in fig. 6.

The closed flue or chimney is of considerable height, erected above the portion of the tower containing the cooling surface and the water distribution system; the openings at the bottom of the tower permit the entry of the air currents and the air flow is produced by the difference in temperature existing between the top and bottom of the structure.

Ques. Why does the chimney tower "induce" a draught or "draw" as they call it?

Ans. It draws because the hot air in the chimney is lighter than the surrounding cold atmosphere, which endeavors to force its way into the chimney from below in order to restore equilibrium.

To illustrate, consider in fig. 8 a stack for induced draught, cool at bottom and hot at top. Each unit of air in traversing the stack will expand as the temperature increases and *becomes lighter*. Let the little cube **A**, represent 1 lb. of air, and suppose as it rises the following changes take place:

initial volume	8 expansions	16 expansions

then the corresponding weight of the initial volume decreases to

1	$\frac{1}{8}$	$\frac{1}{16}$

Accordingly the sum of the weights of unit volume in ascending is

$$1 + \tfrac{1}{8} + \tfrac{1}{16} = 1\tfrac{3}{16} \text{ lbs.}$$

Now on the outside of the stack the volume and weight of each unit of air remain the same so that considering three units *a*, *b*, *c* of decreasing weights in the stack, there are three units of *a′ b′ c′* of constant weight outside the stack, the total weight outside the stack being 3 lbs. and only $1\frac{3}{16}$ lbs. inside the stack. Accordingly, the downward force (3 lbs.) outside the stack being greater than that in the stack, the heavy units *a' b' c'* push the lighter units *a, b, c* up and out of the stack, thus inducing a draught as indicated by the lever scales, fig. 7.

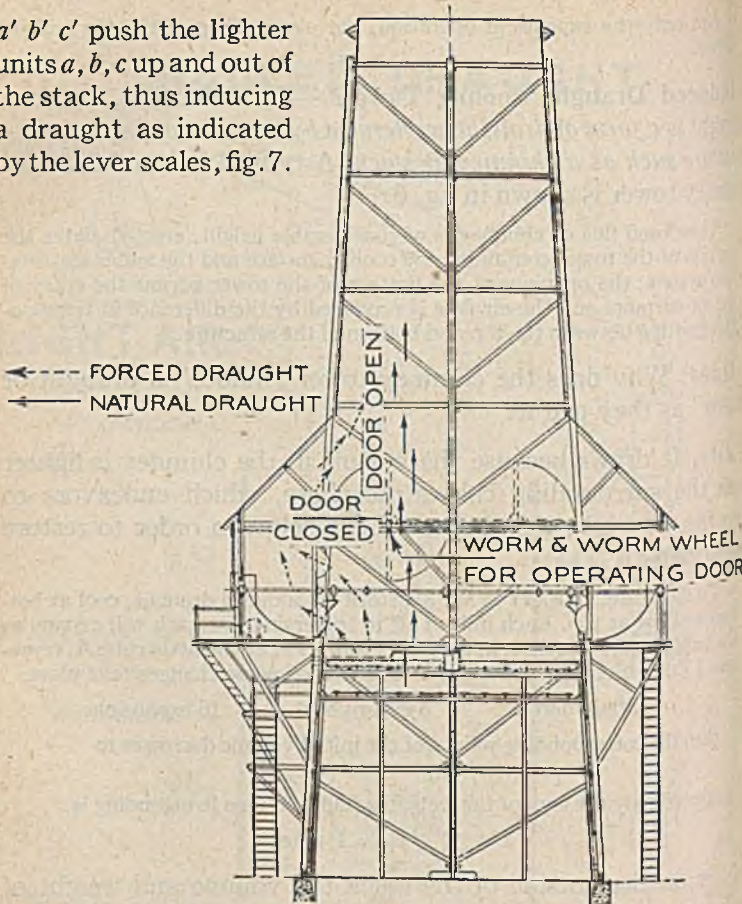


Fig. 9.—Induced natural draught tower as installed at Waco, Texas.

Ques. Considering the foregoing explanations define draught.

Ans. By definition, draught is *the difference in pressure available for producing a flow of air.*

Forced Draught Towers.—By definition, forced draught is mechanical draught, that is, the method of accelerating air current in an enclosed structure by means of a fan. The essentials are shown in fig. 10.

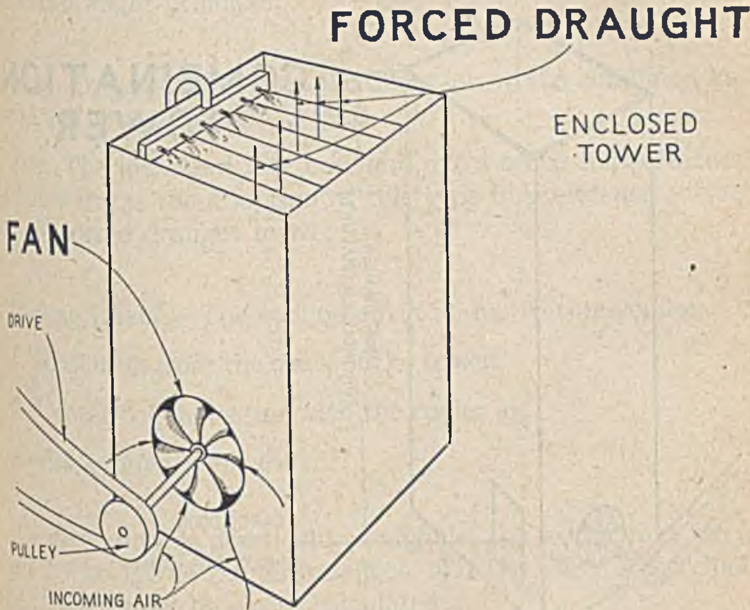


Fig. 10.—Elementary **forced draught** cooling tower showing fan.

The sides of the tower are closed and the air is delivered to the interior of and forced through the tower by fans which are usually located on opposite sides.

Ques. What provision is important on metal towers?

Ans. Access for painting all exposed surfaces of the housing and supporting frame work.

The inner surfaces of the housing in a well designed tower should be made accessible by removable strips inside of the tower, these being laid loosely on the horizontal racks so that a painter can start at the top of the tower and work downward, by simply removing and replacing the wood strips, checker-work, or trays, using them as a platform on which to stand.

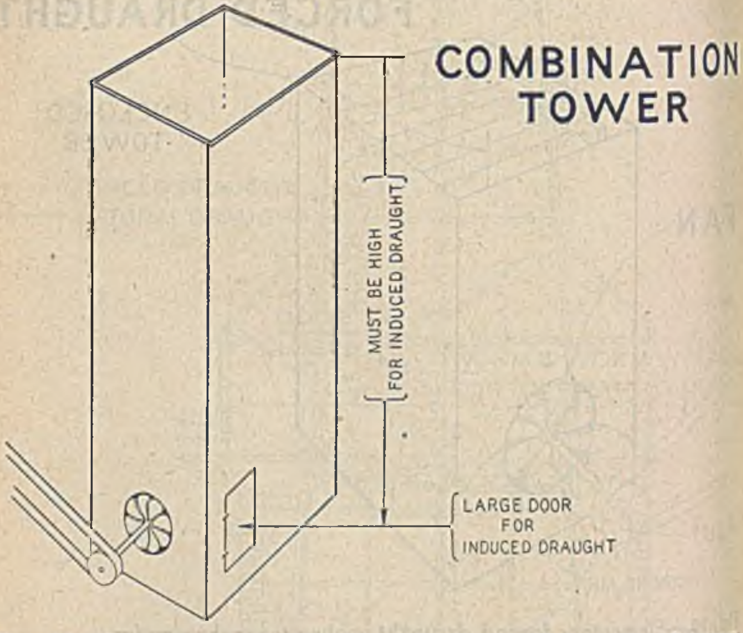


Fig. 11.—Elementary combination forced and induced draught cooling tower. Compare with fig. 10.

The importance of this feature, facilitating proper maintenance at small expense, will be appreciated when it is considered that with some types it is practically impossible to paint the inner surfaces after the tower is once put in operation.

Combined Natural and Forced Draught Towers.—A tower of this kind is larger and more expensive than the regular forced draught type, but has the advantages that on light loads or under very favorable atmospheric conditions, the fan can be shut down and large doors in the base of the tower may be opened, when a very fair degree of cooling is obtained on the induced draught principle.

Ques. In general, what is the objection to the combined induced and forced draught tower?

Ans. The increased first cost and extra space requirements do not warrant the adoption of this type in preference to the open or forced draught tower.

Cooling Effect.—The cooling effect is due to three causes:

1. Radiation from the sides of the tower.

2. Contact of the water with the cooler air.

3. *Evaporation of the water.*

The first cause is practically negligible, the second may vary from 1/5 to 1/3 of the entire effect, and the third is the chief effect which may be easily calculated.

Example.—A certain condenser requires 100 lbs. of water per minute, which is discharged at 110°, and it is desired to cool it to 70° F. What will be the evaporation in the cooling tower?

The total heat to be abstracted from the water per minute is

$$100 \times (110 - 70) = 4{,}000\ B.t.u.\ \text{(approx.)}$$

Now, if say 20% of the cooling effect be due to radiation and convection or contact of the water with the cooler air, then heat to be removed by evaporation is

$$4{,}000 \times (100\% - 20\%) = 3{,}200\ B.t.u.$$

At 110° the latent heat of evaporation is 1,030 *B.t.u.*, hence

evaporation = 3,200 ÷ 1,030 = 3.1 lbs. per minute*

Each pound of free air absorbs 2.375 *B.t.u.* while its temperature is raised 10 degrees. Thus the temperature difference between the water and the entering air limits the heat transfer by convection. For every 1,000 *B.t.u.* of heat transferred in this manner, 422 pounds, or about 5,600 cubic feet, of air must be brought in contact with the water and warmed 10 degrees, or 2,800 cubic feet 20 degrees, etc. The same volume of air will absorb an additional and much larger quantity of heat through evaporation. Each pound of air entering the cooling tower at 72° with 70% saturation, and leaving saturated at 102° will absorb only 7.2 *B.t.u.* by its rise in temperature, but 28.7 *B.t.u.* by the water it evaporates.

As the cooling capacity of the air is limited, it is clear that an economical installation must provide means by which a large quantity of air can be brought in contact with the water spray and quickly removed, after having been warmed and saturated, to give place to a fresh supply of air.

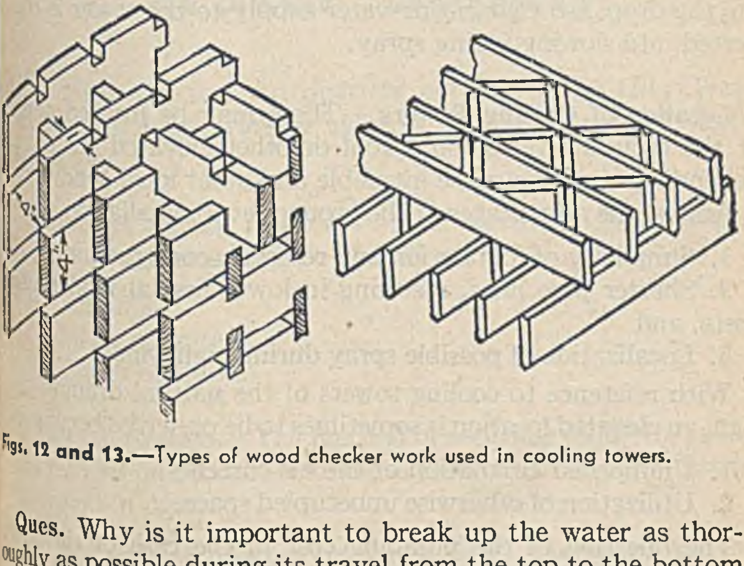
The foregoing discloses the relation of the heat to be extracted from the water and the amount of air required to absorb that heat, both directly and by evaporation.

Ques. What are the considerations in the design of a cooling tower?

*NOTE.—Bearing in mind that the latent heat absorbed by the cooling water while condensing one pound of steam in the condenser, must equal the latent heat extracted in the tower when evaporating one pound of water, the quantity of water evaporated will equal the quantity condensed, less the percentage of heat removed by convection and direct radiation. In other words, the cooling tower has to evaporate a quantity of water equaling 75 to 85 per cent of the weight of steam (corresponding to the feed water) passing through the turbine or engine. This loss must be replaced by a fresh supply.

Ans. Air will evaporate water until saturated, and the amount of moisture absorbed depends upon its initial humidity and temperature together with that of the water.

A cooling tower, therefore, should be proportioned for average summer conditions of the atmosphere, as in the winter it will cool the water considerably more and consequently a higher vacuum will be produced in the condenser.



Figs. 12 and 13.—Types of wood checker work used in cooling towers.

Ques. Why is it important to break up the water as thoroughly as possible during its travel from the top to the bottom of a cooling tower?

Ans. Evaporation and convection take place on the surface of the water only.

Ques. Why should the process of breaking up the water during its descent in the tower be repeated many times?

Ans. The individual drops or streams should not remain long undisturbed as this would permit their surfaces to cool without refrigerating the inner portions.

Ques. How is this accomplished?

Ans. By allowing the water to drip over a series of obstacles (such as shown in figs. 12, 13) rapidly breaking up and reforming the drops, so that entire water supply to the tower is converted into slowing falling spray.

Location of Cooling Towers.—These may be located either at the ground level or on a roof or other elevated structure, depending upon the space available and other local conditions.

Among the advantages of the ground level installation are:

1. Simplicity of foundation and reservoir construction.
2. Shorter pipe lines, resulting in lower first and operating costs, and
3. Localization of possible spray during high winds.

With reference to cooling towers of the natural draught design, an elevated location is sometimes to be preferred because of

1. Unimpeded circulation of the air currents.
2. Utilization of otherwise unoccupied space.

The question of the pumping cost in the case of elevated towers must be given consideration. With a surface condenser the ascending and descending water columns balance each other as far as the reservoir under the tower, and the additional pumping cost is only that occasioned by the friction in the increased length of pipe lines, which, with pipe lines of ample size, is a relatively small item. With the jet type of condenser, however, an elevated location of the tower is undesirable, for obvious reasons.

CHAPTER 28

Water Supply

In rural and isolated districts not served by City Water Works pipe lines, various systems of domestic water supply are in use.

Domestic Supply Systems.—In a classification of isolated plants, all systems may be classed as:

1. Rainfall
2. Hand operated
3. Power operated

Rainfall System.—The method of securing water by collecting rainfall on roofs is sometimes resorted to in rural districts as shown in fig. 1.

It is the most primitive and worst system. The water flowing from roof to the tank carries the dirt with it. Evidently such water is not fit for drinking and the tank must be frequently cleaned. When this system is installed drinking water is obtained from a well via bucket or pump.

Ques. Name some other objections to the rainfall system.

Ans. The supply is uncertain and also due to the considerable amount of dirt and other foreign matter which collects the pipes frequently become clogged.

Water Sources.—Aside from the method of collecting rainfall on roofs, water for domestic installation is obtained from:

1. Wells
2. Springs

There are various kinds of wells and they may be classed:

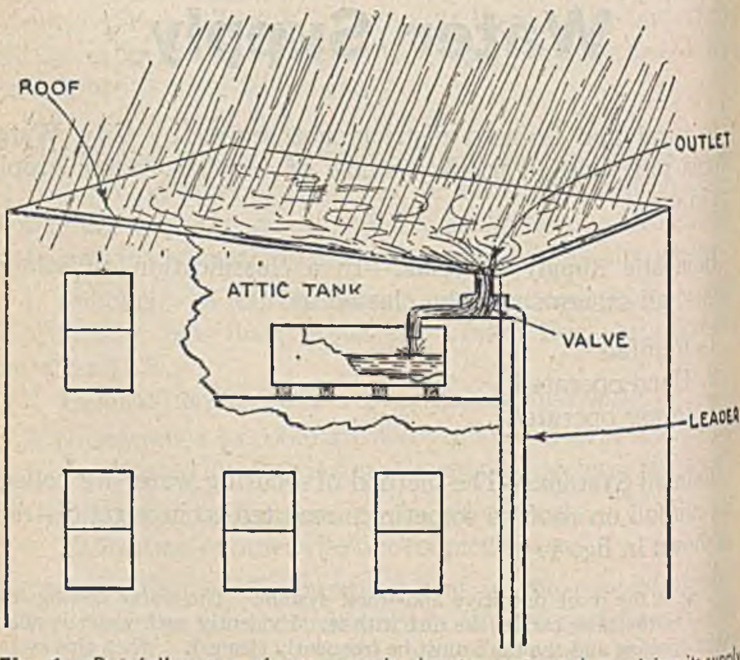


Fig. 1.—Rainfall system of water supply showing attic tank receiving its supply of water from roof, a valve being interposed as shown so that when the tank is full, excess water may be carried off through a leader.

1. With respect to the method of sinking, as:
   a. Dug
   b. Driven

2. With respect to depth, as:

   a. Shallow
   b. Deep
   c. Artesian

Dug Wells.—The chief advantage of the large diameter of a dug well is that it affords storing a considerable quantity of water and the possibility of placing the pump at a low level to secure a *lift* within limits.

Large wells are useful especially where the ground water flows through fine material with low velocity.

Ques. What trouble is not encountered in dug wells?

Ans. Dug wells avoid the clogging that occurs in driven wells located in iron bearing sands.

Driven Wells.—An ordinary driven well consists of a wrought iron pipe or steel tube 2 to 8 ins. in diameter with a strainer near the bottom.

Ques. How is a driven well driven?

Ans. It is forced into the ground by a heavy hammer or by the use of a falling weight or again with the end of a jet of water carried through a small pipe to loosen the material in advance of the point.

Fig. 2 shows a driven well and fig. 3 the method of driving.

Ques. How are strainers constructed?

Ans. They may be merely holes or slots in pieces of brass pipe, or larger holes in the pipe covered with brass gauze.

Ques. What is objectionable in strainer construction and why?

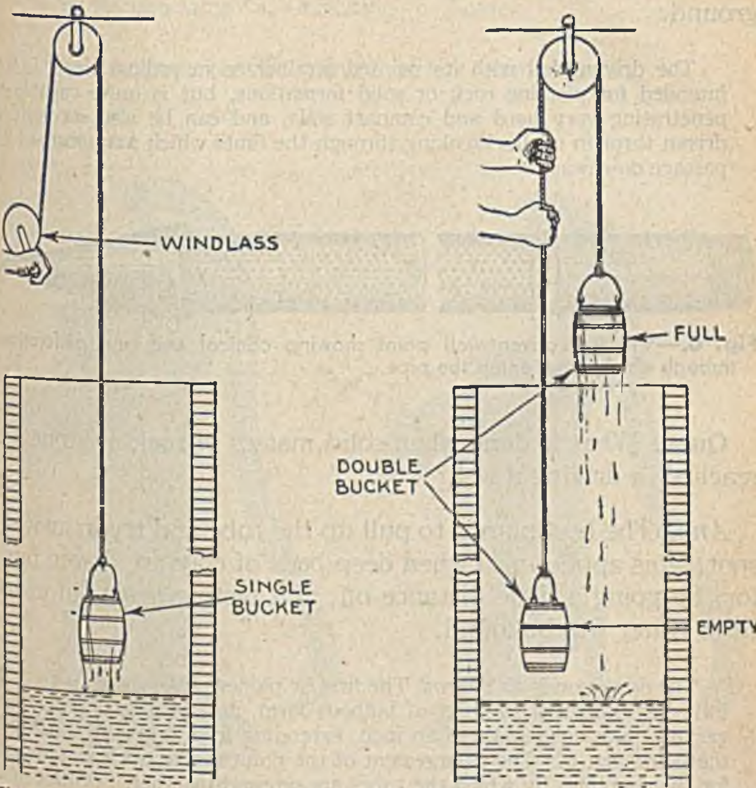
Ans. The use of different metals as it tends to give galvanic action causing corrosion and clogging.

Fig. 2.—Driven well showing pipes and perforated well point.
Fig. 3.—Method of driving pipe and perforated point with monkey.

Ques. What governs the size of the opening in a strainer?

Ans. The texture of the soil. They must be small enough to prevent the entrance of any large quantity of sand, but large enough to reduce the entrance velocity to a point where the friction will not be excessive.

Fig. 6 shows a typical strainer or driven well point as it is called.



Figs. 4 and 5.—Ordinary dug wells illustrating methods of raising water with: 1, windlass and single bucket, and 2, by means of two buckets. Here the weight of the empty bucket descending reduces somewhat the exertion necessary to raise the full bucket.

Ques. Describe the process of driving a well.

Ans. It resembles pile driving, but with the distinction, that while piles receive the blows of the "monkey" on their heads, the tubes are not struck at all, the blow being communicated by the clamp, which receives the blow near the ground.

The driven well with its pointed strainer as in ordinary use, is not intended for piercing rock or solid formations, but is quite capable of penetrating very hard and compact soils, and can be also successfully driven through chalk, breaking through the flints which may obstruct its passage downward.

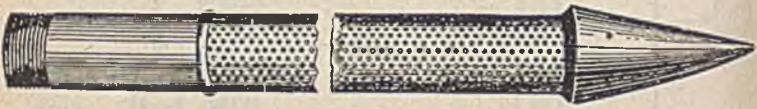


Fig. 6.—View of driven well point showing conical end and perforations through which water enters the pipe.

Ques. What is done when solid masses of rock or stone are reached in driving a well?

Ans. The best plan is to pull up the tube and try in another spot. This applies also when deep beds of clay are driven into; for, by going a little distance off, and testing again, in many cases water will be found.

The operation is as follows: The first or pioneer tube shown in fig. 2 is furnished with a steel point of bulbous form, and perforated with holes varying from one-eighth to an inch, extending from 15 in. to 3 ft. from the point, fig. 6. The enlargement of the point serves to clear a passage for the couplings by which the tubes are screwed together. On this tube the clamp fig. 3 is held about 3 ft. from the point by two bolts; the clamp is of wrought iron with steel bushing screwed internally so as to form teeth to grip the tube.

Next, the cast iron driving weight or monkey is slipped on to the tube above the clamp. The monkey is operated with ropes.

House Hand Pumps.—Various types of pumps are used to supply water in the house. They may be classed:

1. With respect to the nature of the stroke or movement, as:

   a. Reciprocating *b.* Oscillating *c.* Rotary

2. With respect to the nature of the pumping, as:

   a. Lift *b.* Force

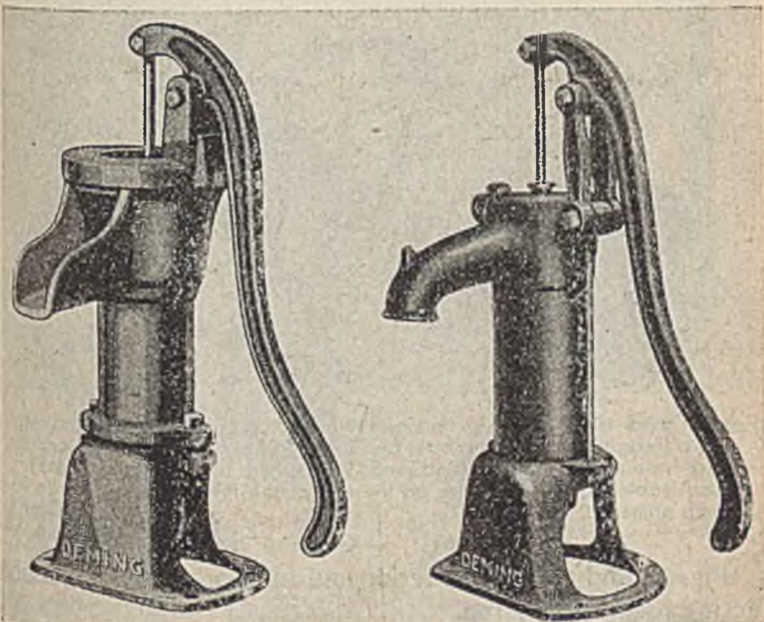


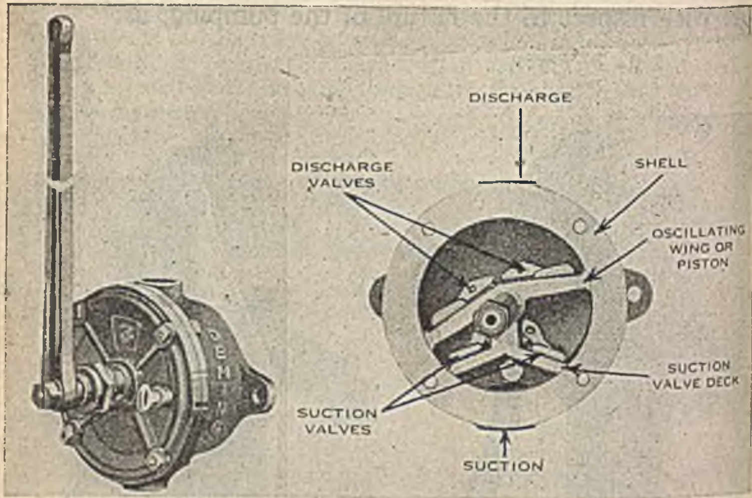
Fig. 7.—House lift pump. Cylinder may be drained by raising handle to extreme height.

Fig. 8.—House force pump. Made with closed top and brass stuffing box nut to conform with Board of Health requirements. It is impossible for dirt or foreign matter to drop into the pump from the top and contaminate the water supply.

3. With respect to the frequency of the cycle, as:

a. Single acting *b.* Double acting

Figs. 7 and 8 show familiar patterns of reciprocating lift and force pumps respectively.



Figs. 9 and 10.—Double acting oscillating force pump; exterior and interior views. Except in the stuffing box, and for a gasket between the lid and shell, there is no packing of any kind used. When there is no danger of freezing, it is advisable to use foot valve on the end of the inlet pipe in order to keep pump primed.

Figs. 9 and 10 show exterior and interior views of a double acting oscillating pump.

While this type pump is intended for pumping hot liquids, foods, wines, etc., it is equally well adapted for hot or cold water.

The rotary type pump shown in fig. 11 is sometimes used. The inlet may be either at the side (as shown) or at the bottom.

Installation of Hand Pumps.—The following suggestions relate to house hand pumps either lift or force, as usually installed in the kitchen.

House lift pumps and house force pumps are complete units, each having its cylinder, plunger and valve located in the stock and base of pump ready for connection to inlet or inlet and discharge pipes.

Water may be drawn horizontally any reasonable distance if the inlet pipe used be of sufficient diameter. The inlet piping should be as direct

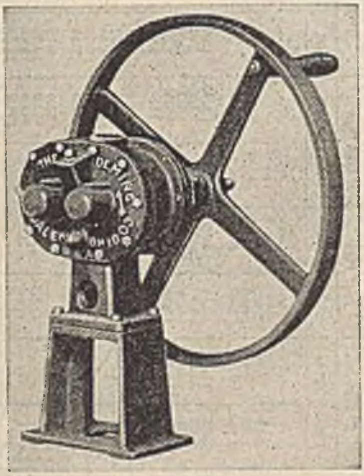


Fig. 11.—Rotary hand pump. It is made with a hand crank and is provided with brackets for attaching to post or wall.

and short as possible. Pipe should be laid out so that air pockets or "humps" are eliminated.

The pump may be installed away from the well as in fig. 12 or directly over it whichever be the more convenient. However, the vertical distance from surface of water to bottom of pump cylinder, plus the additional head in feet resulting from pipe friction, must not be more than the maximum inlet lift for which the pump is recommended. See accompanying Pipe Friction Table, pages 716 and 717.

Loss of Head in Feet Due to Friction, per 100 Feet of *15-Year-Old* Ordinary Iron Pipe

Pipe Friction

For *New and Smooth* Iron Pipe, the Loss of Head will be .7 of that Shown in Table

Gallons per Minute	½" Pipe		¾" Pipe		1" Pipe		1¼" Pipe		1½" Pipe		2" Pipe		2½" Pipe		3" Pipe	
	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.
1	1.05	2.1	....	....	....	....	....	....	....	....	....	....	....	....	....	....
2	2.10	7.4	1.20	1.9	....	....	....	....	....	....	....	....	....	....	....	....
3	3.16	15.8	1.80	4.1	1.12	1.26	....	....	....	....	....	....	....	....	....	....
4	4.21	27.0	2.41	7.0	1.49	2.14	0.86	0.57	0.63	0.26	....	....	....	....	....	....
5	5.26	41.0	3.01	10.5	1.86	3.25	1.07	0.84	0.79	0.39	....	....	....	....	....	....
10	10.52	147.0	6.02	38.0	3.72	11.7	2.14	3.05	1.57	1.43	1.02	0.5	0.65	0.17	0.45	0.07
15	....	....	9.02	80.0	5.60	25.0	3.2	6.5	2.36	3.0	1.53	1.0	0.98	0.36	0.68	0.15
20	....	....	12.03	136.0	7.44	42.0	4.29	11.1	3.15	5.2	2.04	1.82	1.31	0.61	0.91	0.25
25	....	....	....	....	9.30	64.0	5.36	16.6	3.94	7.8	2.55	2.73	1.63	0.92	1.13	0.38
30	....	....	....	....	11.15	89.0	6.43	23.5	4.72	11.0	3.06	3.84	1.96	1.29	1.36	0.54
35	....	....	....	....	13.02	119.0	7.51	31.2	5.51	14.7	3.57	5.1	2.20	1.72	1.59	0.71
40	....	....	....	....	14.88	152.0	8.58	40.0	6.3	18.8	4.08	6.6	2.61	2.20	1.82	0.91
45	....	....	....	....	....	....	9.65	50.	7.08	23.2	4.60	8.2	2.94	2.80	2.05	1.15
50	....	....	....	....	....	....	10.72	60.	7.87	28.4	5.11	9.9	3.27	3.32	2.27	1.38
70	....	....	....	....	....	....	15.01	113.	11.02	53.0	7.15	18.4	4.58	6.2	3.18	2.57
90	....	....	....	....	....	....	....	....	14.17	84.0	9.19	29.4	5.88	9.8	4.09	4.08
100	....	....	....	....	....	....	....	....	15.74	102.0	10.21	35.8	6.54	12.0	4.54	4.96
120	....	....	....	....	....	....	....	....	18.89	143.0	12.25	50.0	7.84	16.8	5.45	7.0
140	....	....	....	....	....	....	....	....	22.04	190.0	14.30	67.0	9.15	22.3	6.35	9.2
160	....	....	....	....	....	....	....	....	....	....	16.34	86.0	10.46	29.0	7.26	11.8
180	....	....	....	....	....	....	....	....	....	....	18.38	107.0	11.76	35.7	8.17	14.8
200	....	....	....	....	....	....	....	....	....	....	20.42	129.0	13.07	43.1	9.08	17.8
220	....	....	....	....	....	....	....	....	....	....	22.47	154.0	11.38	52.0	9.99	21.3
240	....	....	....	....	....	....	....	....	....	....	24.51	182.0	15.69	61.0	10.89	25.1
260	....	....	....	....	....	....	....	....	....	....	26.55	211.0	16.90	70.0	11.80	29.1
280	....	....	....	....	....	....	....	....	....	....	....	....	18.30	81.0	12.71	33.4
300	....	....	....	....	....	....	....	....	....	....	....	....	10.61	92.0	13.62	38.6

Gallons per Minute	4" Pipe		5" Pipe		6" Pipe		8" Pipe		10" Pipe		12" Pipe		14" Pipe		15" Pipe		16" Pipe		20" Pipe	
	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.	Vel.	Fric.
40	1.02	0.22	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
45	1.17	0.28	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
50	1.28	0.34	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
70	1.79	0.63	1.14	0.21	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
75	1.92	0.73	1.22	0.24	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
100	2.55	1.23	1.63	0.39	1.14	0.14	....	....	....	....	....	....	....	....	....	....	....	....	....	....
120	3.06	1.71	1.96	0.57	1.42	0.25	....	....	....	....	....	....	....	....	....	....	....	....	....	....
125	3.19	1.86	2.04	0.64	1.48	0.28	....	....	....	....	....	....	....	....	....	....	....	....	....	....
150	3.84	2.55	2.45	0.88	1.71	0.32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
175	4.45	3.36	2.86	1.18	2.00	0.48	....	....	....	....	....	....	....	....	....	....	....	....	....	....
200	5.11	4.37	3.27	1.48	2.28	0.62	....	....	....	....	....	....	....	....	....	....	....	....	....	....
225	6.32	6.61	3.67	1.86	2.57	0.74	....	....	....	....	....	....	....	....	....	....	....	....	....	....
250	6.40	6.72	4.08	2.24	2.80	0.92	1.60	0.22	....	....	....	....	....	....	....	....	....	....	....	....
275	7.03	7.99	4.50	2.72	3.06	1.15	1.73	0.27	....	....	....	....	....	....	....	....	....	....	....	....
300	7.66	9.38	4.90	3.15	3.40	1.29	1.90	0.36	....	....	....	....	....	....	....	....	....	....	....	....
350	8.90	12.32	5.72	4.19	3.08	1.69	2.20	0.41	....	....	....	....	....	....	....	....	....	....	....	....
400	10.20	15.82	6.54	5.33	4.54	2.21	2.60	0.56	....	....	....	....	....	....	....	....	....	....	....	....
450	11.50	19.74	7.35	6.65	5.12	2.74	2.92	0.64	1.80	0.21	....	....	....	....	....	....	....	....	....	....
475	12.30	22.96	7.88	7.22	5.55	3.21	3.10	0.79	1.94	0.25	....	....	....	....	....	....	....	....	....	....
500	12.77	24.08	8.17	8.12	5.60	3.26	3.20	0.81	2.04	0.28	1.42	0.11	....	....	....	....	....	....	....	....
550	....	....	8.00	9.66	6.16	3.93	3.52	0.98	2.25	0.33	1.57	0.14	....	....	....	....	....	....	....	....
600	....	....	9.80	11.34	6.72	4.70	3.84	1.16	2.46	0.39	1.71	0.15	....	....	....	....	....	....	....	....
650	....	....	10.62	13.16	7.26	5.50	4.16	1.34	2.66	0.46	1.85	0.19	1.37	0.09	....	....	....	....	....	....
700	....	....	11.44	15.12	7.84	6.38	4.46	1.54	2.86	0.52	2.00	0.22	1.47	0.10	....	....	....	....	....	....
750	....	....	12.26	17.22	8.50	7.00	4.80	1.74	3.06	0.59	2.13	0.24	1.58	0.11	....	....	....	....	....	....
800	....	....	....	....	9.08	7.90	5.12	1.97	3.28	0.67	2.27	0.27	1.68	0.13	....	....	....	....	....	....
850	....	....	....	....	9.58	8.75	5.48	2.28	3.48	0.75	2.41	0.31	1.79	0.14	....	....	....	....	....	....
900	....	....	....	....	10.30	10.11	5.75	2.46	3.68	0.83	2.56	0.34	1.89	0.16	....	....	....	....	....	....
950	....	....	....	....	10.72	10.71	6.06	2.87	3.88	0.91	2.70	0.35	2.00	0.16	1.73	0.12	....	....	....	....
1000	....	....	....	....	11.32	12.04	6.40	3.02	4.08	1.01	2.84	0.41	2.10	0.19	1.82	0.14	....	....	....	....
1100	....	....	....	....	12.50	14.31	7.03	3.51	4.50	1.20	3.13	0.49	2.31	0.23	2.00	0.16	....	....	....	....
1200	....	....	....	....	13.52	16.69	7.67	4.26	4.91	1.46	3.41	0.57	2.52	0.26	2.18	0.19	....	....	....	....
1500	....	....	....	....	....	....	9.60	6.27	6.10	2.09	4.20	0.85	3.15	0.39	2.73	0.28	2.39	0.24	....	....
2000	....	....	....	....	....	....	12.70	10.71	8.10	3.50	5.60	1.43	4.20	0.66	3.64	0.47	3.19	0.39	....	....
2500	....	....	....	....	....	....	....	....	10.10	5.33	7.00	2.18	5.25	1.01	4.55	0.72	3.99	0.56	....	....
3000	....	....	....	....	....	....	....	....	12.10	7.42	8.40	3.39	6.30	1.57	5.46	1.12	4.79	0.80	3.08	0.27

Vel.—Velocity feet per second. Fric.—Friction head in feet.

Contents of Round Tanks in U. S. Gallons for Each Foot in Depth

Inside Diameter Ft.	In.	Gallons One Foot In Depth	Inside Diameter Ft.	In.	Gallons One Foot In Depth	Inside Diameter Ft.	In.	Gallons One Foot In Depth	Inside Diameter Ft.	In.	Gallons One Foot In Depth
1	0	5.87	5	9	194.19	10	6	653.69	15	3	1365.96
1	3	9.17	6	0	211.44	10	9	678.88	15	6	1407.51
1	6	13.21	6	3	229.43	11	0	710.69	15	9	1457.00
1	9	17.98	6	6	248.15	11	3	743.36	16	0	1503.62
2	0	23.49	6	9	267.61	11	6	776.77	16	3	1550.97
2	3	29.73	7	0	287.80	11	9	810.91	16	6	1599.06
2	6	36.70	7	3	308.72	12	0	848.18	16	9	1647.89
2	9	44.41	7	6	330,38	12	3	881.39	17	0	1697.45
3	0	52.86	7	9	352.76	12	6	917.73	17	3	1747.74
3	3	62.03	8	0	375.90	12	9	954.81	17	6	1798.76
3	6	73.15	8	3	399,76	13	0	992.62	17	9	1850.53
3	9	82.59	8	6	424.36	13	3	1031.17	18	0	1903.02
4	0	93.97	8	9	449.21	13	6	1070.45	18	3	1956.25
4	3	103.03	9	0	475.80	13	9	1108.06	18	6	2010.21
4	6	118.93	9	3	502.65	14	0	1151.21	18	9	2064.91
4	9	132.52	9	6	530.18	14	3	1192.69	19	0	2121.58
5	0	146.83	9	9	558.45	14	6	1234.91	19	3	2176.68
5	3	161.88	10	0	587.47	14	9	1277.86	19	6	2233.52
5	6	177.67	10	3	617.17	15	0	1321.54	20	0	2349.46

Water Required per Minute to Feed Boilers

Using the "Centennial Standard"—30 pounds or 3.6 gallons of water per horsepower per hour, evaporated from 100° F. to 70 pounds steam pressure per square inch.

H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons	H.P. Boiler	Feed Water Gallons
20	1.2	60	3.6	110	6.6	190	11.4	400	24.0
25	1.5	65	3.9	120	7.2	200	12.0	450	27.0
30	1.8	70	4.2	130	7.8	225	13.5	500	30.0
35	2.1	75	4.5	140	8.4	250	15.0	600	30.0
40	2.4	80	4.8	150	9.0	275	16.5	700	42.0
45	2.7	85	5.1	160	9.6	300	18.0	800	48.0
50	3.0	90	5.4	170	10.2	325	19.5	900	54.0
55	3.3	100	6.0	180	10.8	350	21.0	1000	60.0

Capacities of Water Piping in Buildings — Length 100 Ft. in U. S. Gallons per Minute

Size Pipe	½	¾	1	1¼	1½	2	2½	3	4
Pressure									
17 lbs.	3.2	9.1	18.7	33.5	51.6	106	200	290	589
30 lbs.	5	14	28	52	78	160	308	436	885
40 lbs.	6	16	33	60	90	184	350	504	1023
50 lbs.	6.5	17.5	37	70	101	206	390	564	1143
60 lbs.	7	19.5	40	76	110	226	430	617	1252
75 lbs.	7.5	22	45	85	123	253	480	690	1400
100 lbs.	9	25	52	99	142	292	558	797	1607

Ques. What precautions should be taken with lower end of inlet pipe?

Ans. It should not touch the bottom of the well.

A strainer and foot valve should be placed at the lower end of the inlet pipe.

Ques. What precaution should be taken in pipe fitting?

Ans. All joints should be tight.

Ques. What should be done before placing the pump in position?

Ans. It is advisable to submerge the piston in water for an hour or so.

Ques. Why?

Ans. It will expand the piston leather and cause it to fit the cylinder more snugly.

Ques. How should the inlet pipe be placed when laid horizontally?

Ans. It should be placed three feet underground to prevent freezing.

Ques. How about the pipe leading vertically to the pump?

Ans. It should also be protected against freezing by ample thickness of insulation.

NOTE.—To determine the capacity of house lift or force pumps, multiply the gallons per stroke by 50, which is an average rate of pumping, and refer to accompanying table of cylinder capacity.

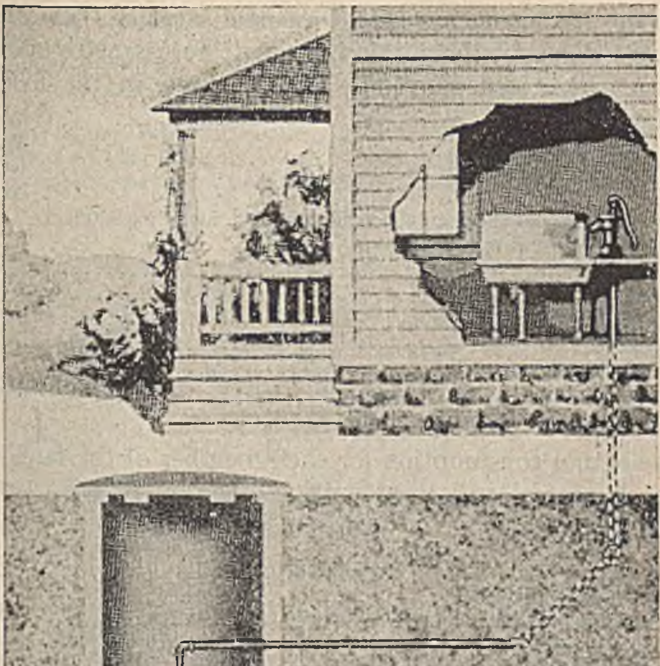


Fig. 12.—Typical installation of a house pump in the kitchen at a distance from the well.

Water Requirements for Domestic Service.—In proportioning a domestic installation it is first necessary to know the average water consumption. This is given in the following tabulation:

	Approx. Gallons Per Day
For each member of the family, for all purposes, including kitchen, laundry, bath and toilet purposes	25
Each horse	10
Each cow	15
Each hog	3
Each sheep	1¼
Each 100 chickens	4
Drinking fountains	50 to 100

The actual consumption for each member of the family and for each animal will vary with the season of the year and conditions.

Home Fixtures

Filling ordinary lavatory	1½
Filling average bathtub	30
Flushing a water closet	6
Each shower bath	30

Yard Fixtures

	Gallons Per Hour
½ inch hose with nozzle	200
¾ inch hose with nozzle	275 to 300
Lawn sprinkler	120

Ques. What should be noted in selecting a pump as to size?

Ans. Its capacity should be in excess of the actual requirements for any given period.

Ques. Why?

Ans. The reason is apparent in the example which follows.

Example.—If ½ inch hose with nozzle is to be used for sprinkling, this will consume water at the rate of 200 gallons per hour, and it is therefore essential, to permit use of water for other purposes at the same time, to have a pump capacity in excess of 200 gallons per hour.

Where ½ inch hose with nozzle is to be used, the pump should have a capacity of at least 220 gallons per hour which leaves available for other uses of water at the rate of 20 gallons per hour when the hose is being used.

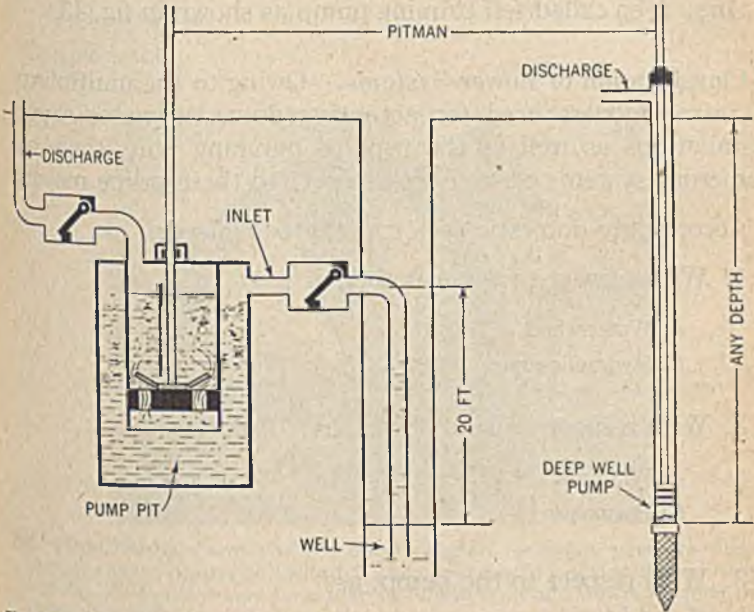


Fig. 13.—Elementary so-called syphon pump showing pump barrel projecting down into a concentric or outside cylinder providing an annular space which traps water so that the pump is always primed. Anyone having any experience with pumps, especially old and worn pumps, will appreciate this feature. If the left were greater than 20 ft. the pump should be lowered in a pit or a deep well pump used as shown at the right.

Ques. What other conditions should be noted in selecting a pump as to size?

Ans. The pump capacity even for ordinary requirements should be such as will not necessitate the pump operating more than a few hours per day at the most.

Ques. What kind of pump should be used to avoid the annoyance of frequent priming especially when worn?

Ans. A so called self priming pump as shown in fig. 13.

Classification of Power Systems.—Owing to the multiplicity of prime movers used for actuating domestic water supply installations as well as the type of pumping unit, there are numerous systems classed with respect to these prime movers.

Accordingly domestic systems may be classed:

1. With respect to water power, as

   a. Water wheel
   b. Hydraulic ram

2. With respect to wind power, as

   a. Wind mill
   b. Pneumatic

3. With respect to the pump, as

   a. Shallow well
   b. Deep well
   c. Jet (air lift)
   d. Centrifugal

4. With respect to special features, as

   a. Two tank feed in

Water Wheels.—Where the terrain is suitable, that is on country estates, where there is a running stream with even a few feet fall and a sufficient flow of water, a water wheel properly geared to a power pump will supply ample water at sufficient head to supply a country estate.

The wheel may be either:

1. Overshot, or
2. Undershot

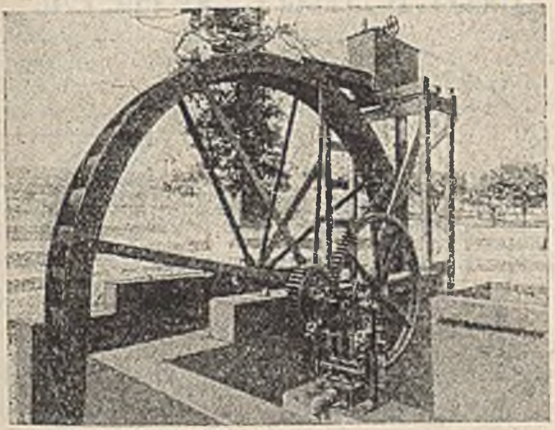


Fig. 14.—Overshoot water wheel direct coupled to triplex pump to supply water to country estates and villages. The pump should be housed over in cold climates. A dynamo may be added to furnish light and power for farm purposes. The water to drive the water wheel may be conveyed from the dam by an underground pipe or through an overhead flume, as preferred.

depending upon the head available of the water supply and its volume. Fig. 14 shows a water wheel of the overshot variety geared up to a triplex pump.

Note here the flexibility of the system. The gearing in fig. 14 is about 2 to 1, that is, the wheel makes two revolutions to one of the pump.

Of course if the head pumped against were greater the velocity reduction could be greater, that is, say 3 to 1, the wheel making three revolutions to one of the pump.

Where the head is not sufficient for an overshot wheel, the undershot wheel may be used as in fig. 15.

From fig. 14, it is seen that these wheels are of considerable diameter and evidently require a water supply of sufficient head for a tangential flow at the elevation of the top of the wheel.

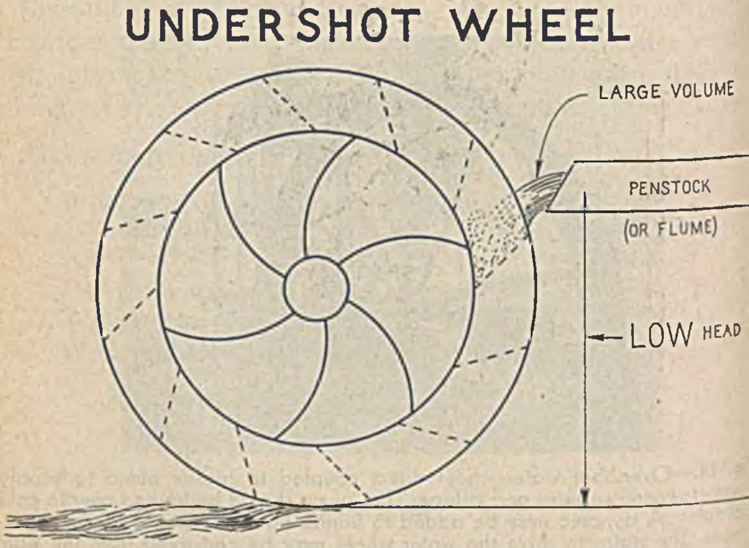


Fig. 15.—Breast water wheel.

The number of buckets in the undershot wheel that are full of water at one time is considerably less than in the overshot wheel, evidently more water is required, that is, assuming adequate supply the capacity of the buckets must be greater than in the overshot wheel. That is to say, for equal diameters, the buckets of the undershot wheel must hold more than those of the overshot wheel. In other words they must be wider.

Hydraulic Ram.—The *impulse* type of pump has been presented at length in the chapter following and no amplification of this system is here necessary.

Wind Mills.—In a recent trip by the author out West, including New Mexico and Arizona, he was surprised to see the large number of wind mills, especially in the rural districts, in contrast to the small number in the Eastern Seaboard where numerous installations have been replaced by City Water Works service.

In the opinion of the author, where the terrain is favorable, that is, where the wind is not blanketed by trees or hills, the windmill is the cheapest method of isolated domestic water supply. Evidently the power costs nothing and the maintenance is nil, except where the owner is too lazy to keep the mill properly lubricated.*

As a source of energy wind power is of importance, especially for pumping water in suburban or rural districts which have no city water supply—this energy as before stated costs nothing.

Mechanics of the Wind Mill.—The operation of a wind mill is due to the pressure of the wind acting on numerous vanes inclined to the direction of the wind and rotating in a plane perpendicular to the direction of the wind as shown in fig. 16.

Ques. What is the requirement for maximum effect?

Ans. The plane of the wheel must be kept perpendicular to the direction of the wind.

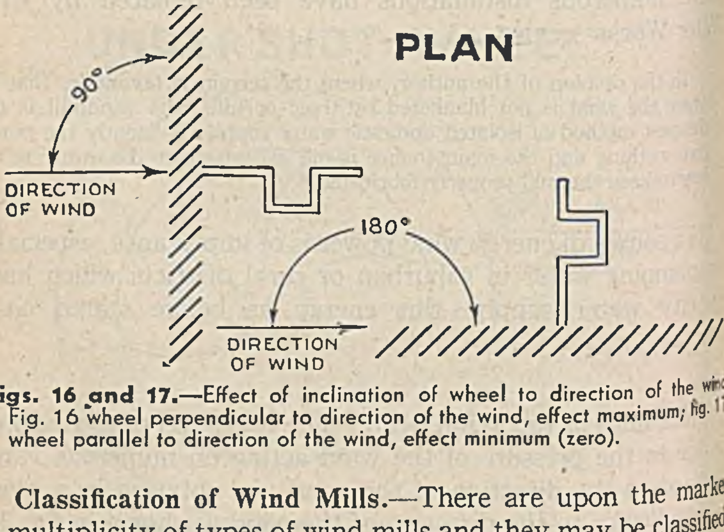
*NOTE.—The familiar squeak of some cheap installations is not only the fault of the owner, but of the design of the tower with rather hazardous climbing to the top to lubricate.

Ques. How is this accomplished?

Ans. The main casting on which the wheel is pivoted is arranged to turn in a stationary collar on turntable as shown in fig. 18.

Ques. How is the turning controlled?

Ans. By a tail attached to the main casting and upon which the wind acts.



Figs. 16 and 17.—Effect of inclination of wheel to direction of the wind. Fig. 16 wheel perpendicular to direction of the wind, effect maximum; fig. 17, wheel parallel to direction of the wind, effect minimum (zero).

Classification of Wind Mills.—There are upon the market a multiplicity of types of wind mills and they may be classified:

1. With respect to the power transmission, as

   a. Direct stroke
   b. Augmented stroke
   c. Geared

2. With respect to direction control, as

   a. Tail or vane
   b. Vaneless

3. With respect to the material of construction, as
    - *a.* Wooden
    - *b.* Metal

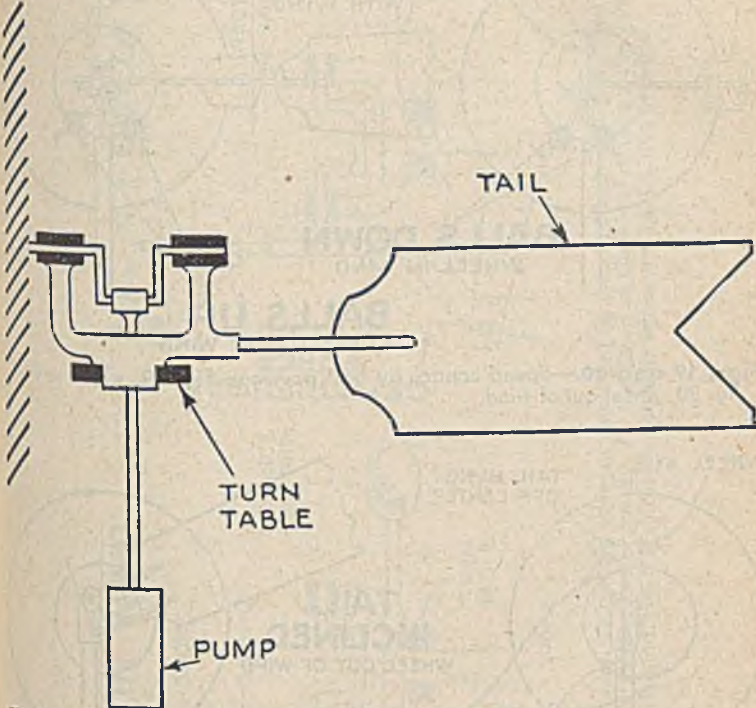
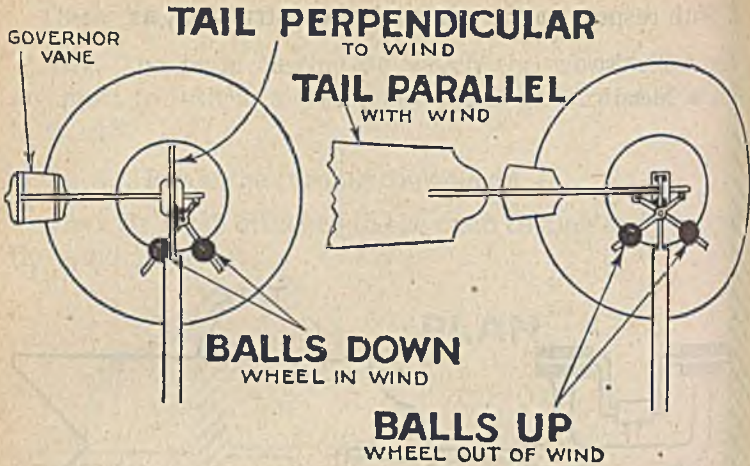


Fig. 18.—Elementary wind mill showing turn table and tail.

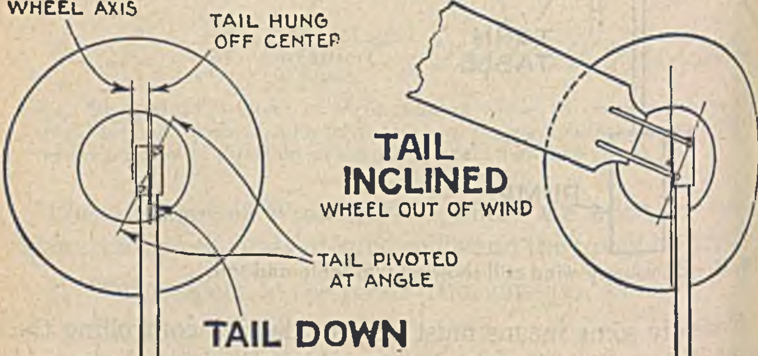
Evidently some means must be provided for controlling the speed of the mill, otherwise in case of very high wind, damage may result.

Numerous methods of speed control are employed as by use of:

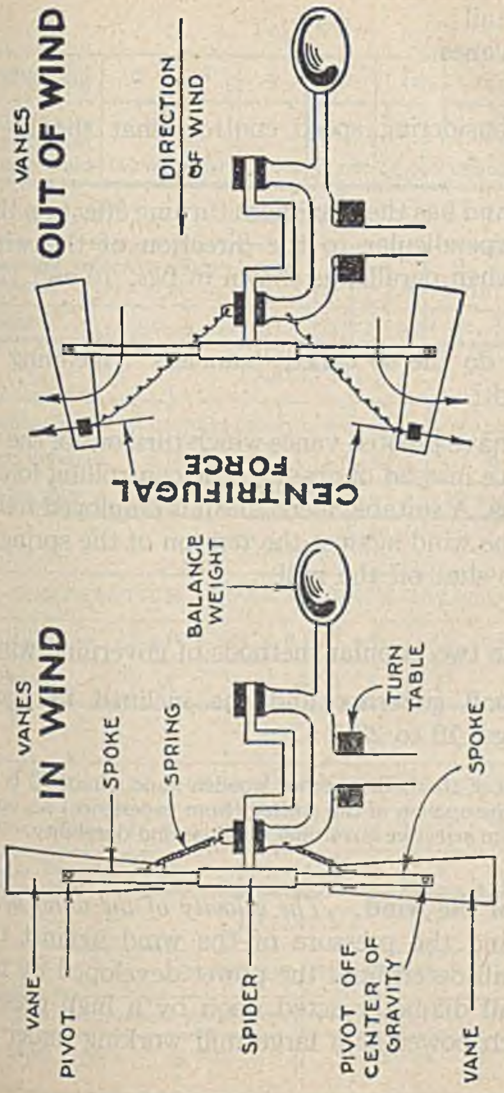
1. Weight governor



Figs. 19 and 20.—Speed control by ball governor. Fig. 19, wheel in wind; fig. 20, wheel out of wind.



Figs. 21 and 22.—Speed control by inclined tail, wheel pivoted off center. Fig. 21, wheel in wind; fig. 22, wheel out of wind. **In operation,** the wheel being pivoted off center with respect to the turntable, tends to cause it to turn out of the wind, and in so doing the tail is lifted into an inclined position, on account of being pivoted at an angle with the vertical. Gravity acts to bring the wheel back into the wind.



Figs. 23 and 24.—Speed control by pivoted vanes as employed on the so called vaneless (tailless) mills. The vane sections are hinged off their center of gravity. As the speed of the wheel increases, the weight of the sections turns them out of the wind. Fig. 23, shows the vanes **in wind** and fig. 24, **out of wind.**

2. Inclined tail
3. Pivoted vanes

Ques. In considering speed control what should be considered?

Ans. The wind has the maximum turning effect on the wheel when it is perpendicular to the direction of the wind, and has no effect when parallel as shown in figs. 16 and 17.

Ques. How do the so called "vaneless" (meaning tailless) wind mills work?

Ans. They have pivoted vanes which turn out of the wind by centrifugal force instead of gravity, the controlling force of the other two types. A suitable mechanism is employed to hold the vanes out of the wind against the tension of the springs, when it is desired to shut off the mill.

Ques. Name two popular methods of governing wind mills.

Ans. The ball governor and the inclined tail governor, as shown in figs. 19 to 22.

An example of the ball governor wooden vane wind mill is shown on page 739. In the opinion of the author (from experience) an outstanding mill judged from sensitive governing qualities and durability.

Properties of the wind.—*The velocity of the wind determines its pressure*, and the pressure of the wind against the sails of the wind mill determines the power developed by the mill. A mill of small diameter acted upon by a high pressure develops as much power as a large mill working under a lower pressure.

TABLE I.

Size of Pumping Mill	6	8	9	10	12	14	16	20
No. Gals. water raised 1 ft. hourly, 15-mile wind	10,000	20,000	24,000	35,000	68,000	110,000	160,000	300,000

TABLE II.

Average wind velocity, miles per hour....................	4	5	6	7	8	9	10	11	12	13	14	15
Co-efficient	16	8	5	3	2	1.4	1.	.85	.70	.60	.54	.50

TABLE III.

Gallons hourly	.35	170	220	260	300	360	420	550	850	1200	2200	3400	5000
Cylinder, diam. in.	2	2¼	2½	2¾	3	3¼	3½	4	5	6	8	10	12
Discharge pipe, diam. in.	1½	1¼	1¼	1½	1½	2	2	2	2½	2½	3½	4	5

TABLE IV.

COMPARATIVE POWER OF BACK-GEARED MILLS.								
Size of Mill.......	4-ft.	6-ft.	8-ft.	9-ft.	10-ft.	12-ft.	14-ft.	16-ft.
Horse-Power	$\frac{1}{12}$	$\frac{1}{6}$	$\frac{3}{10}$	$\frac{2}{5}$	$\frac{3}{5}$	1	1 [illegible]/5	$2\frac{3}{5}$

TABLE V.

FORCE OF THE WIND IN POUNDS PRESSURE.								
Velocity.....Miles	8	10	12	15	20	25	30	40
Force......Pounds	⅓	½	¼	1	2	3	4½	8

TABLE VI.

POWER OF THE WIND.			
Velocity per Hour.	Pressure per Sq. Foot.	Velocity per Hour.	Pressure per Sq. Foot.
10 Miles	½ Lb.	20 Miles	2 Lbs.
15 "	1 "	25 "	3 "

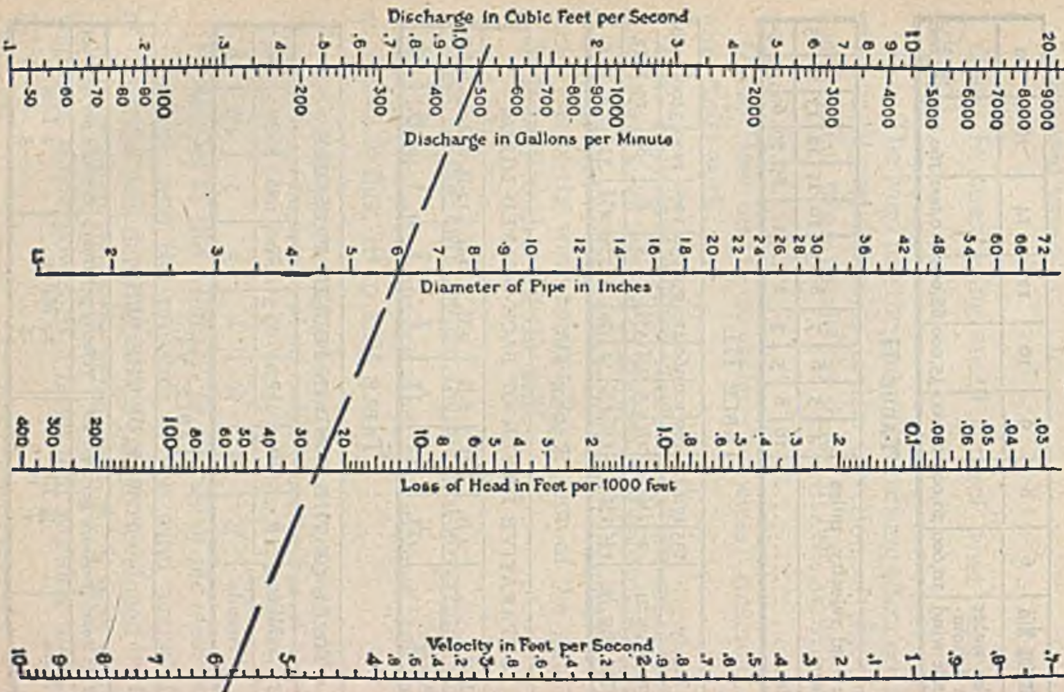


Fig. 25.—Diagram for calculating pipe sizes, discharge velocities and loss of head in water pipe. Lay a straight edge on scales at the points for any two known quantities and the unknown quantities will lie at the intersection of the straight edge with the other scales. **Example.**—To discharge 500 gals. per minute through a 6 in. pipe following the dotted line would show a loss of head 1,000 ft. of approximately 25 ft. head and a velocity of 5.8 ft. per second.

The mean average velocity of the wind for the entire United States is very nearly eight miles per hour. However, for large areas such as the great plains east of the Rocky Mountains, the mean average is about eleven miles per hour, and yet in certain small areas situated in the mountainous districts the mean average velocity is as low as five miles per hour. Therefore, in selecting and loading a mill, reference should be had to the wind velocity prevailing in that particular locality. In general, wind mills suffice to do a large quantity of work and no storage capacity would be required, but when it does blow it is "free" and experience dictates that a mill shall be erected sufficiently large to pump enough water, when the wind does blow, to last over, with the assistance of ample storage capacity.

Power of Wind Mills.—The power of a wind mill depends on: 1, the diameter of the wheel; 2, area and number of vanes; and 3, velocity of the wind. The following tables give some useful data relating to the sizes and capacities of pumping wind mills.

Table IV gives the horse power of several sizes of mills working in a fifteen mile wind: if the wind velocity be increased or diminished, the power of the wind mill will increase or decrease in the ratio of the squares of the velocity. Table V will show the comparative power or force of the wind in velocities from eight to forty miles per hour for each square foot of surface.

Rules for approximately determining size of wind mill to use.—In approximately determining the size of wind mills for a given installation, the daily water consumption must be given as a basis for calculation. *Divide daily consumption by* 8 *to find hourly capacity of the mill, and if properly loaded the mill will pump on an average eight hours daily.*

Table I gives the maker's number of the pumping mill, and the number of gallons each will raise one foot high per hour, with a wind having a velocity of fifteen miles per hour.

Rule.—Multiply the quotient by total water lift in feet and with the coefficient given in Table II.

The product will in Table I show what mill to use. The size of the cylinder and discharge pipe will be found in Table III.

Example.—No. 9 pump will raise 24,000 gallons of water one foot high in one hour. Now if the water is to be raised 50 feet then by dividing 24,000 by 50 the quantity raised becomes 480 gallons per hour.

From Table V it will be seen also that a wind velocity of fifteen miles per hour develops a power three times as great as an eight mile wind, and a twenty mile wind is twice as powerful as a fifteen mile, or six times that of an eight mile wind.

Hence, *a small increase in velocity greatly increases the power of the wind mill*, while a low velocity gives but little working force.

From Table VI it is seen that a twenty-five mile wind gives six times as much power as a ten mile wind, but really gives twenty-six times *the net efficient power of the ten mile wind*, therefore it will not be proper to calculate on using a power wind mill in as low a velocity as ten miles.

Example.—A person in Atlanta, Ga., uses 2,600 gallons of water daily. He has a well in which the water stands 30 feet from ground level. To obtain pressure, the water is to be elevated into a tank 50 feet above ground. $2,600 \div 8 = 325$ gallons to be pumped hourly when wind mill works.

Average wind velocity at Atlanta is 9 miles per hour, answering to coefficient 1.4 in Table II, and total water lift is $30 + 50 = 80$ feet. $325 \times 1.4 \times 80 = 36,400$ gallons.

If first estimate of 2,600 gallons daily were liberal, so that for instance 2,400 gallons would be sufficient, Table I shows that a 10 foot mill can be used, but to keep on the safe side, choose a 12 foot mill. 325 gallons hourly corresponds in Table III, to 3¼ inches cylinder with 2 inches discharge pipe as proper sizes.

If the 10 foot mill be chosen, take the 3 inch cylinder.

A 14 foot wind mill working in a fifteen mile wind will do more work than two average horses, and when working in a twenty mile wind will do more work than four good horses, while in a twenty-five mile wind it will do more work than six good horses.

Wind Mill Pumps.—This type of pump has an extension of the piston rod above the upper guide with a hole for connection with the pump rod from the wind mill. Such a pump with a *pitman* extending from the pump upward into the tower to the mill is shown in fig. 26.

This figure is introduced to show the tank connections with a regulator on the base of a four post tower. The float in the water tank throws the mill in or out of gear according as the water rises or falls in the tank.

When the tank is filled with water it pulls the mill out of gear and stops the pump; as a result there can be no overflow or waste. The tank is thus not allowed to become empty and permit its drying apart, inducing leakage. Through the medium of a float in the tank, when the water has been

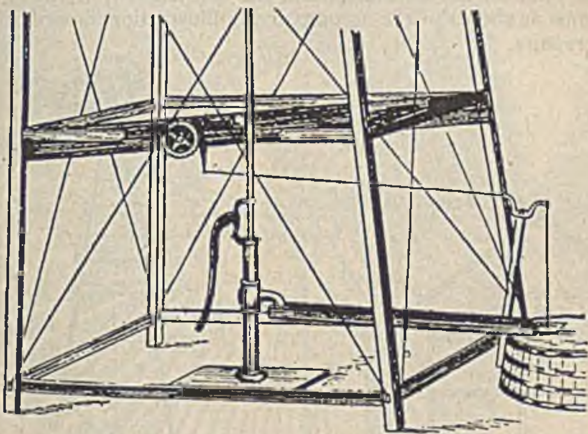


Fig. 26.—Base of wind mill showing connection with pump and automatic cutoff.

lowered but a few inches, the mill is again put in gear and the tank refilled to the desired height, at which the float is set.

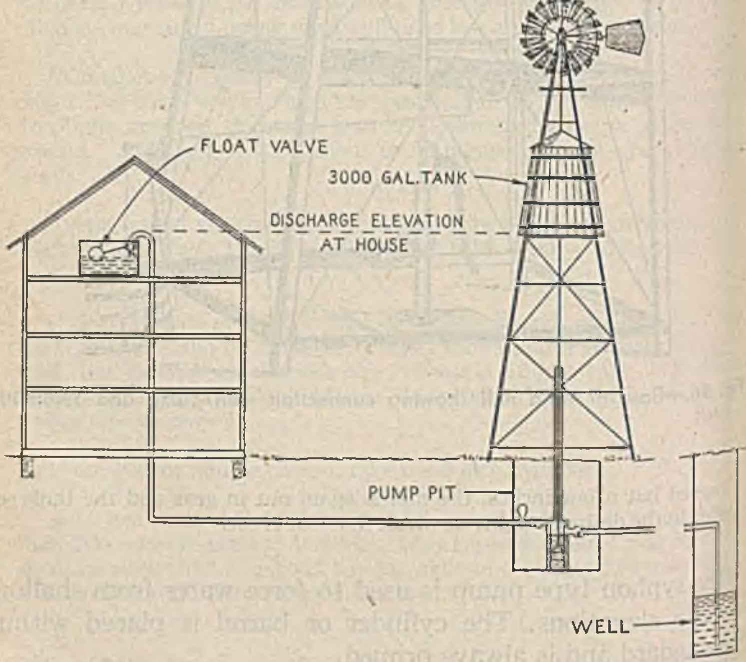
The syphon type pump is used to force water from shallow wells to elevations. The cylinder or barrel is placed within the standard and is always primed.

Where the distance from water level in well to ground level is over say 20 ft., the pump should be lowered to bring the lift within this limit; that is, the inlet valve on pump should not be over 20 ft. elevation from the water level in the well.

In cases where pump cannot be lowered conveniently, a deep well pump should be used.

Figs. 27 and 28 is an example where pump was lowered to bring lift within limit. In this case a deep well pump could not be used as the wind mill tower could not be placed over well. Where a deep well is located directly under wind mill tower a deep well pump is used, as in fig. 26, thus avoiding the necessity of digging a separate pump pit.

The plumber should understand the construction of the various types of pumps as shown in the accompanying illustrations in order to properly make repairs.



Figs. 27 and 28.—Pump connections at residence of the author at **Stornoway,** near Sea Bright, N. J., illustrating location of pump in pit to reduce lift to practical limit. As shown in fig. 27 the well was about 50 ft. from the tower and the arrangement of buildings prevented placing wind mill tower directly over well, otherwise a deep well pump would have been installed. Mill used for this installation shown on opposite page.

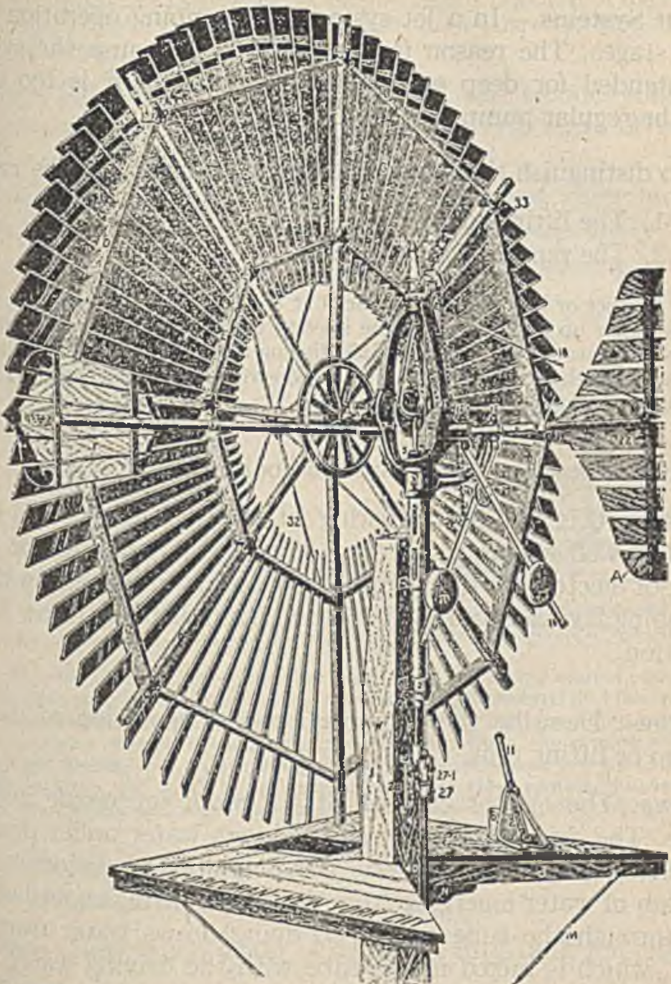


Fig. 29.—Direct stroke wind mill with wooden vanes. **In construction,** the mill is made mostly of steel, wrought and malleable iron, the whole central iron work being in the main frame. The vanes are attached to stayed oak ribs making a very substantial construction.

Jet Systems.—In a jet system the pumping operation is in two stages. The reason for two stages is because the system is intended for deep well service where the lift is too great for the regular pump.

To distinguish them the two pumps required may be called:

1. The lifting unit
2. The pumping unit

The jet or lifting pump is placed at the bottom of the well and forces the water up to the level of the inlet to the other pump. On deep well installations the jet system avoids digging a deep pit at the well site and also keeps the pumping unit in a location where it is more likely to receive proper attention.

Ques. Describe the jet system hook up.

Ans. It is a unit consisting of either a centrifugal or a shallow well reciprocating type of pump, in combination with a jet or ejector, for the purpose of pumping from wells in which pumping level is more than 25 ft. vertical distance below pump location.

Ques. Describe the construction and operation of the jet pump or lifting unit.

Ans. The ejector consists of jet body, jet nozzle and jet tube. The drive or supply pipe delivers water under pressure to jet nozzle which converts the pressure into velocity. The stream of water emerging from the nozzle at high speed shoots up through the tube and in so doing draws water from the well, which is mixed in the tube with the driving water. The high velocity of the water while passing through the tube is transformed into pressure and this pressure delivers both the driving water and the water drawn from well, to a higher point.

The elementary diagram fig. 31 shows the basic principles of the system and figs. 32 and 33 two examples of construction—the single pipe and two pipe patterns.

Ques. How much of the output of the second stage or pumping unit does it take to operate the first stage or lifting unit?

Ans. It depends upon the lift, that is, from water level in well to level of inlet valve in pump. Approximately from

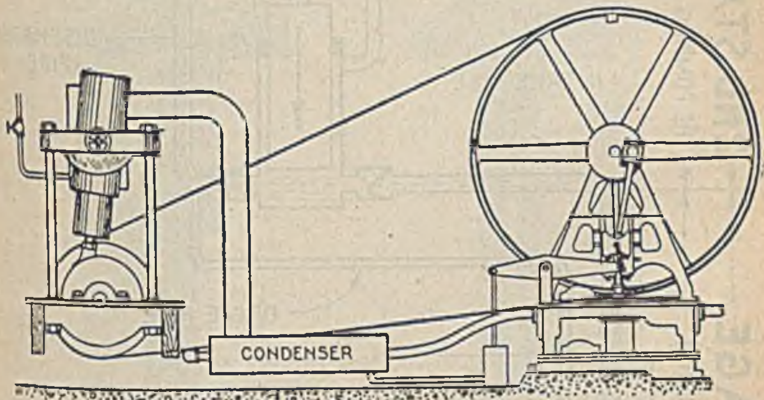


Fig. 30.—Graham jacketed, transfer expansion oscillating engine, and Dunham power pump with belt transmission. **It must be evident** that this arrangement permits the engine and pump to work under the most favorable conditions for economy. Hence by proper proportion of pulleys, the engine may run at **high speed,** thus reducing the size of the cylinder and loss by condensation, leakage, etc., and the pump may be run at **slow speed,** thus reducing the loss by slip, and eliminating water hammer, slamming of valves, etc.

50 ft. lift, about ½ and for say 100 ft. below about ¾ of the output of the pumping unit.

Ques. How much water is delivered into tank for different lifts?

Ans. Referring to the previous answer for 50 ft. lift about ½ and for 100 ft. ¼ of the output of the pumping unit.

Ques. What becomes of the water used to operate the jet pump?

Ans. It is circulated, going down in the drive pipe and coming up in the intake pipe.

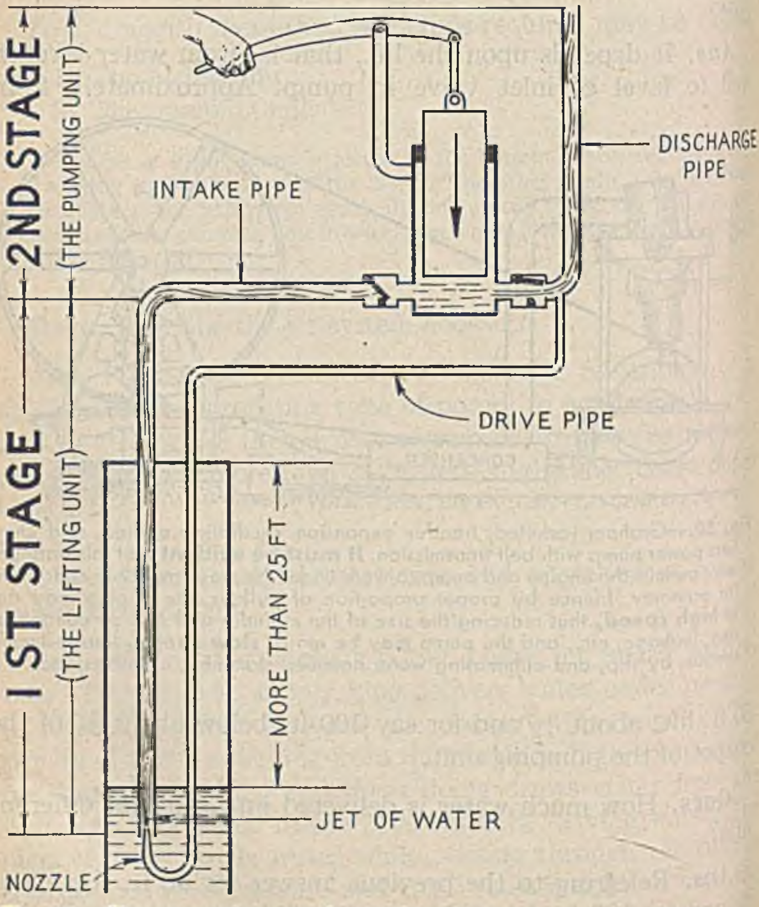
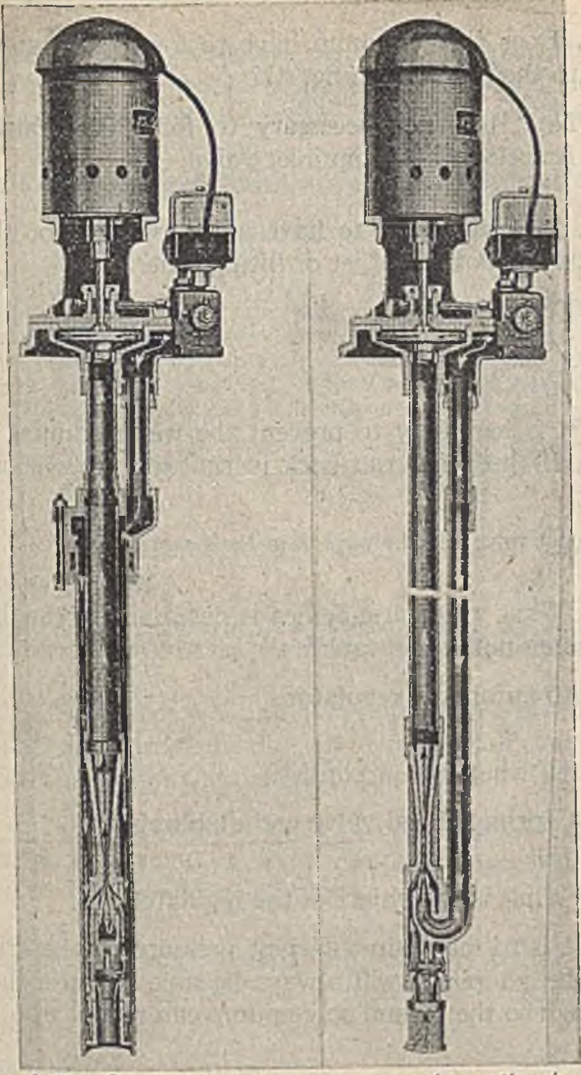


Fig. 31.—Elementary diagram of water jet system.



Figs. 32 and 33.—One and two pipe jet systems with centrifugal pump in the second stage.

Ques. Does the jet pump have to force the water all the way up to the pump as in fig. 31?

Ans. No. It is not necessary to force the water higher than 25 ft. below the pump inlet valve.

Ques. Is it necessary to have a foot valve (not shown in fig. 28) on the bottom of jet or lifting unit?

Ans. Yes.

Ques. Why?

Ans. It is necessary to prevent the water running back in the well. If the water ran back in the well the system would not start.

The foot valve should always be at the lowest point.

Ques. What automatic device is necessary in the discharge pipe between pump and tank?

Ans. An automatic regulator.

Ques. Of what does it consist?

Ans. A spring loaded valve and diaphragm.

Ques. What is the object of the regulator?

Ans. It is to maintain sufficient pressure on the drive line, so that the jet pump will always be able to raise the water high enough so the second stage pump can pick it up.

Ques. What would happen if the regulator were not used?

Ans. The pressure in the system might go down far enough so that the jet pump would be unable to lift water up to the second stage pump and the system would cease to work.

Ques. Why is the regulator valve spring adjustable?

Ans. To regulate the amount of water going through the valve.

Suppose valve be set so that enough water goes down to the jet pump to lift say 50 ft. Now, if the water level in the well should go down to a point where the same jet pump can still lift, but require more driving water, the spring has to be reset, so that more water goes down to the well, and of course, less water will be delivered to the tank.

Ques. Why is a diaphragm used in connection with regulator valve?

Ans. In order to preserve the balance so not to add back pressure on the pump.

Valve stem and diaphragm are connected together and once the valve is off the seat, it is held off the seat by diaphragm under pressure.

Ques. Why cannot a simple hand operated throttling valve or a spring loaded valve be used to maintain sufficient pressure on the jet, instead of the balanced diaphragm valve?

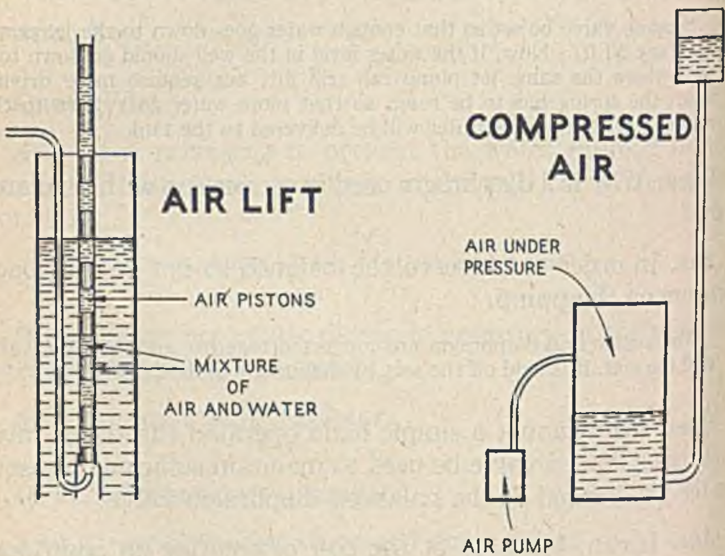
Ans. It can, but only at the cost of creating an additional head on the pump and thus decreasing the amount of water delivered by the pump into the tank.

Pneumatic Systems.—There are two systems in which compressed air is utilized as a medium to elevate water, known as:

1. Compressed air system
2. Air lift system

Although compressed air is used in each system they differ fundamentally as indicated in the elementary diagrams figs. 34 and 35.

Evidently in fig. 35 air is introduced into an enclosed tank at a pressure sufficient to force the water into an elevated tank or to maintain a pressure sufficient to supply the outlets at the top story of a building.



Figs. 34 and 35.—Elementary diagrams of pneumatic and air lift systems.

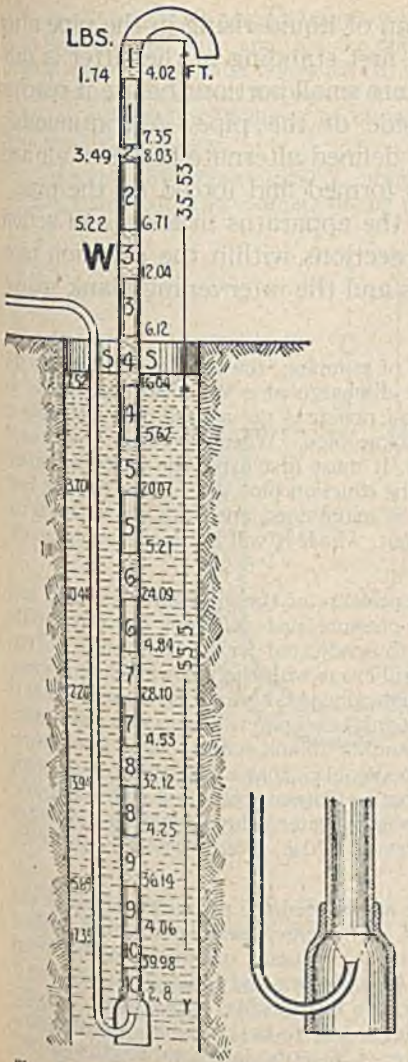
The air lift system consists of a simple and effective method of raising liquids from wells and bore holes as in fig. 34.

A pipe for compressed air is inserted down the middle of the pump tubing or rising main and provided with a nozzle or ejector at its lower end which directs an annular jet upward through the space between the air pipes and the rising main.

The depth to which the ejector must be placed is found by experiment and depends upon the flow of the water or oil through the strata, the height to be lifted and the available pressure.

Many water works petroleum companies, etc., operate a large number of wells by this means, from a central air compression station.

Air Lift.—Fig 34 is an introduction to this system. The Pohle air lift is one of the simplest methods of raising water. The process consists *in submerging a portion of an open-ended eduction pipe in a body of the liquid to be raised and continuously introducing into the liquid within the lower part of the pipe a series of bubbles of air.* The bubbles contain enough air to expand immediately across the pipe and fill the same from side to side, forming pipe-fitting piston-like layers at or just above the point of their entrance into



Figs. 36 and 37.—Diagram of air lift system showing principle of operation.

the pipe, whereby the column of liquid rising in the pipe after the forcing out of the liquid first standing in the latter is subdivided by the gaseous fluid into small portions before it reaches the level of the liquid outside of the pipe. A continuously upward-flowing series of well defined alternate layers of air and short layers of liquid is thus formed and forced up the pipe.

Figs. 36 and 37 represent the apparatus in a state of action pumping water, the shaded sections within the eduction pipe **W**, representing water layers and the intervening blank spaces air layers.

At and before the beginning of pumping, the level of the water is the same outside and inside of the discharge pipe **W**, incidentally, also, in the air pipe. Hence the vertical pressures per square inch are equal at the submerged end of the discharge pipe. When, therefore, compressed air is admitted into the air pipe, it must first expel the incidental standing water before air can enter the eduction pipe **W**. When this has been accomplished, the air pressure is maintained until the water within the eduction pipe has been forced out, which it will be in one unbroken column, free from air bubbles.

When this has occurred the pressure of the air is lowered or its bulk diminished and adjusted to a pressure just sufficient to overcome the external water pressure. It is thus adjusted for the performance of regular and uniform work, which will ensue with the inflowing air and water, which adjust themselves automatically in alternate layers or sections of definite lengths and weights. It will be seen in the figures that the lengths of the water columns (shaded) and air (blank spaces) 1 and 1 are entered at the right of the discharge pipe **W**; also, that under the pressure of two layers of water 1 and 2, the length of the air column 2 is 6.71 feet long, and so on. The lengths of aggregate water columns and the air columns which they respectively compress are also entered on the right of the water pipe.

On the left of the water pipe are entered the pressures per square inch of these water columns or layers. Thus the pressure per sq. in. of column 1 is seen to be 1.74 lbs.; that of 2, consisting of two columns or layers 1 and 2 each 4.02 feet long, to be 3.49 lbs., and that of 10, consisting of nine columns or layers of water 1 to 9, inclusive, each 4.02 ft. long, and one of 3.80 ft. in length (viz., layer 10), to be 17.35 lbs., and the aggregate length of the layers of water is 39.98 ft. in a total length of 91 feet of pipe.

It will be noted that the length of pipe below the surface of the water in the well is 55.5 ft., and that the difference between this and the aggregate length of the water layers (39.98) is 15.52 ft. That is, on equal areas the pressure outside of the pipe is greater than the pressure on the inside by the weight due this difference of level, which is 47.65 lbs. for the end of the discharge pipe.

It is this difference of 15.52 ft., acting as a head that supplies the water pipe, which puts the contents of the pipe in motion, and overcomes the

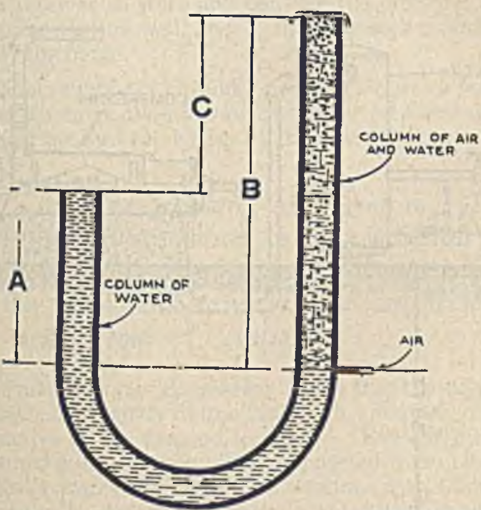


Fig. 38.—Principle of air lift. The injection of air at the point indicated produces a column of mixed air and water B, which has less weight than the solid column of water A. Thus the weight of the column A, causes the water to rise the distance C, and overflow.

resistance in the pipe. In general the water layers are equal each to each, and the pressure upon any layer of air is due to the number of water layers above it.

Thus the pressure upon the bottom layer of air 10 in the figure is due to all the layers of water in the pipe (17.35 lbs.), and the pressure upon the uppermost layer of air 1 is due to the single layer of water 1, at the moment of its discharges beginning, viz., 1.74 lbs. per sq. in. As this discharge progresses this is lessened, until at the completion of the discharge of the water layer the air layer is of the same tension as the normal atmosphere.

The air pipe is connected with an air receiver on the surface, which is at or near the engine room, in which there is *an air compressor*. This air pipe is provided with a valve on the surface. Before turning on the air the conditions in the

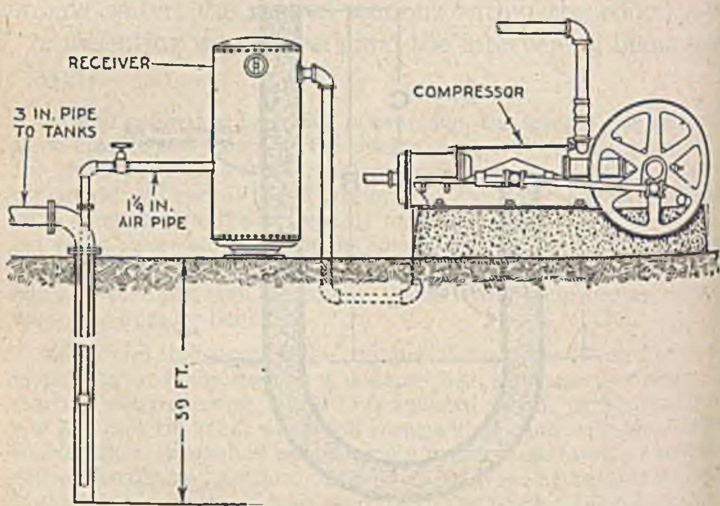


Fig. 39.—Air lift plant showing compressor air tank and connections.

well show water at the same level on the outside and inside of the eduction pipe. At the first operation there must be sufficient air pressure to discharge the column of water which stands in the eduction pipe.

This goes out *en masse*, after which the pump assumes a normal condition, the air pressure being lowered and standing at such a point as corresponds with the normal conditions in the well. This is determined by the volume of water which the well will yield in a certain time and the elevation to which the water is discharged.

After the standing water column has been thrown off by the pressure the air rising through the water reduces its weight, with the result that the water is expelled as fast as the well supplies it, *the water outside the pipe acting as a head, flows into the discharge pipe by the force of gravity.*

The machinery necessary for a system of pumping comprises: 1, *an air compressor;* 2, a receiver to store and equalize the pressure; 3, the head piece and foot piece for the well; and 4, the necessary piping for the air supply and water discharge.

With an available supply of air under pressure *the pump proper consists of simply a water discharge and air pipe,* the latter arranged and properly controlled to inject air into the former at the point of proper submersion.

"Pneumatic" or Compressed Air System.—The basic essentials of this system are shown in the elementary drawing fig. 35. In brief, *water is pumped into a closed tank and air compressed in the same tank to a pressure sufficient to force the water to the highest required elevation.*

Moreover the mechanism of the system is such that the supply of water and air will be such as to partly fill the tank with water and air (at proper pressure) irrespective of the demand for water. Evidently this involves an automatic power pump, usually direct connected to an electric motor, which automatically starts and stops in accordance with the intermittent withdrawals of water to maintain a constant water level in the tank.

At the same time the air pump or compressor functions similarly to maintain a constant pressure of compressed air.

Fig. 40 shows the pneumatic system in elementary form illustrating principal elements. An outfit embodying these elements and various refinements as actually constructed is shown in fig. 41.

The air pump as usually constructed, instead of being of the piston type (shown in fig. 40 simply for clearness) is of the diaphragm type, as shown in fig. 42.

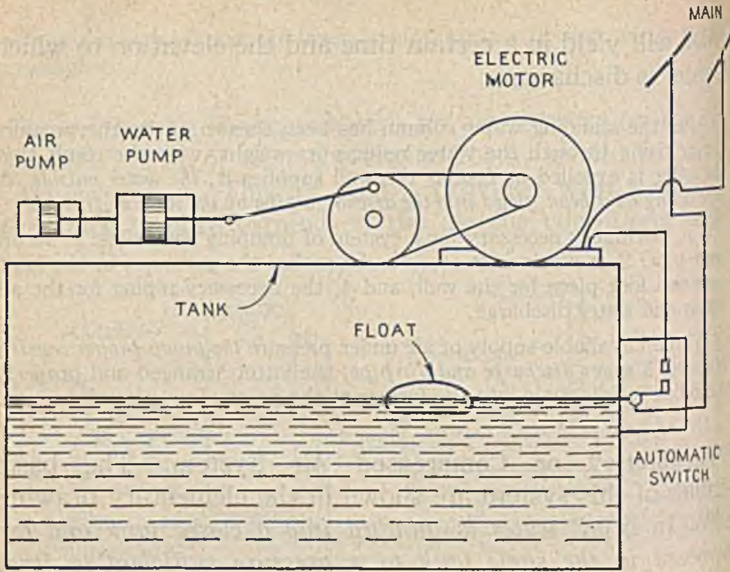


Fig. 40.—Elementary diagram of pneumatic system.

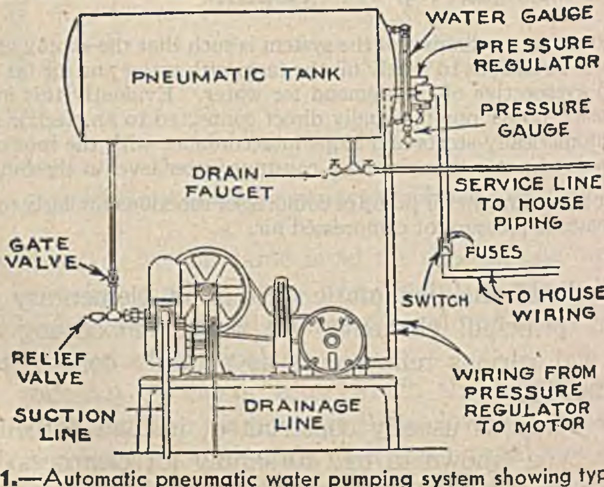


Fig. 41.—Automatic pneumatic water pumping system showing typical piping and wiring connections for use with pneumatic tanks when installed either

A pneumatic outfit with names of parts is shown in fig. 41.

Fig. 44 is typical of the construction of power pumps suitable for domestic service.

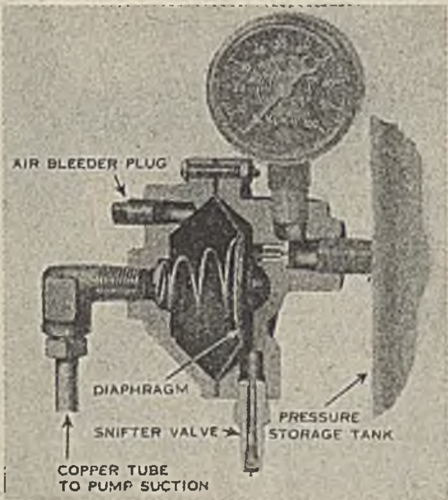


Fig. 42.—Automatic air pump of the system shown in fig. 40. This air pump is screwed into the tank and connected through copper tubing to the inlet side of the centrifugal pump. The inlet pull of the centrifugal pump moves the diaphragm of the air pump, thus drawing air through the snifter valve on the tank side of the diaphragm. When the centrifugal pump stops, the pressure is equalized on both sides of the diaphragm and the trapped air is forced into the tank. This action occurs with each start and stop of the centrifugal pump until the proper amount of air has been forced into the tank, at which time the air pump automatically stops delivering air.

Fig. 41—*Text continued.*

on the floor back of pumping outfit, along side or overhead. Wherever the construction of the tank allows, it is recommended that it be set vertically. The pressure regulator and pressure gauge are attached to the lower gauge fitting. This installation is recommended for locations where there is limited space. Both the tank and pump should be installed in the basement or other place where they will be protected from freezing temperature.

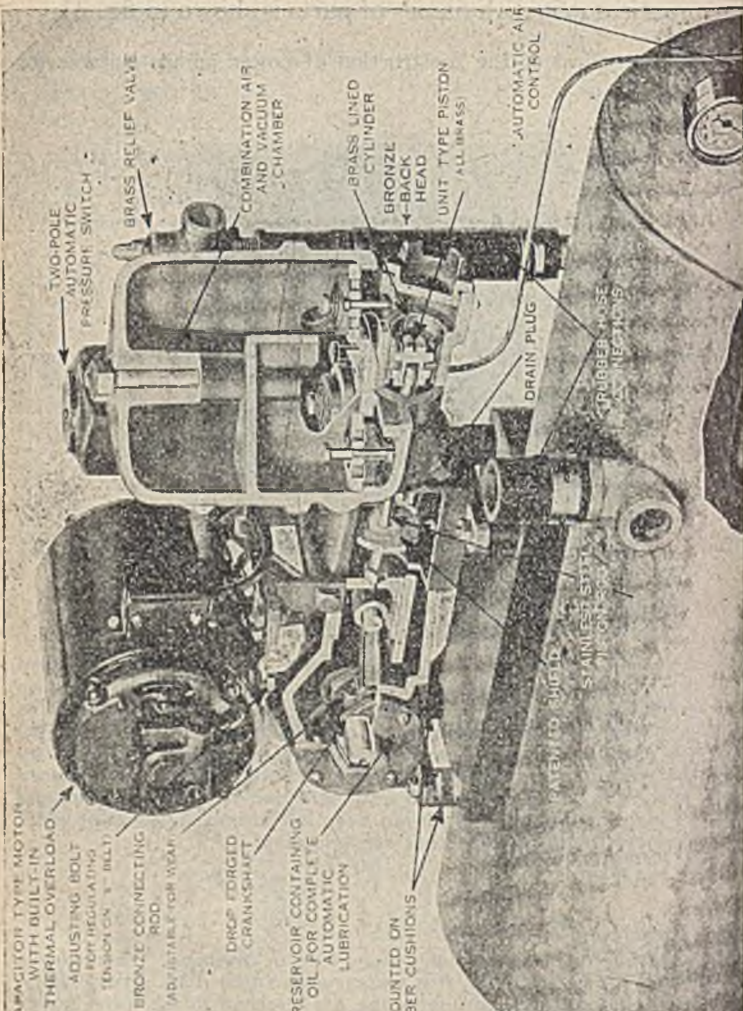


Fig. 38.—Typical pneumatic house water system. Sectional view showing mechanism.

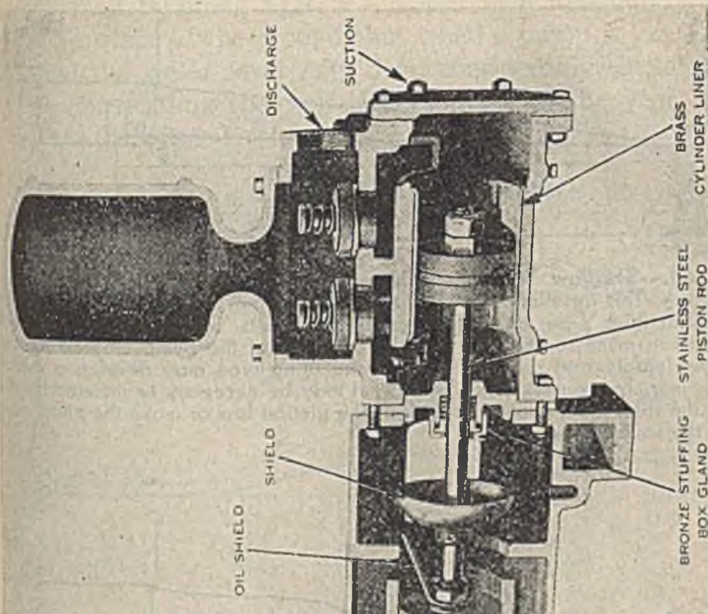


Fig. 44.—Horizontal double acting piston power house pump.

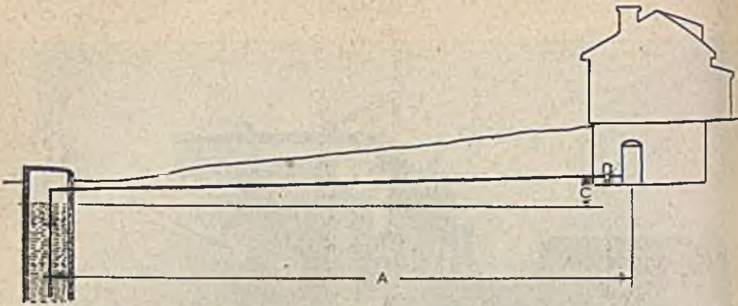


Fig. 45.—Shallow Well Installation.—1. Pump and tank at point of discharge. This installation is satisfactory where the vertical lift C, plus friction loss in inlet pipe does not exceed lift limits of the pump. If friction add too much to the total lift, the capacity of the pump may be reduced considerably and pumps of the reciprocating type may develop a vacuum hammer. To overcome this difficulty it may be necessary to increase the size of the inlet pipe in order to reduce the friction loss or move the pump closer to the source of supply.

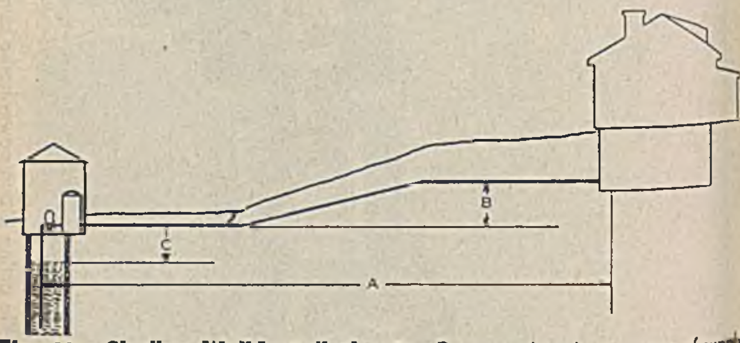


Fig. 46.—Shallow Well Installation 2.—Pump and tank at source of supply. With this type of installation, no trouble should be encountered on the inlet side of the pump provided the total lift do not exceed the lift limit of the pump. However, a long discharge A, from pump and tank to point of discharge may set up pipe friction losses sufficient to reduce the flow of water. Careful consideration must also be given to vertical discharge elevation. Pressure in the tank must be greater than at point of discharge in order to force a given amount of water per minute through the discharge pipe and maintain a minimum pressure on the plumbing fixtures at the higher level. The extra pressure required is determined by the actual discharge elevation B, plus discharge pipe friction losses when delivering a given quantity of water to the house fixtures.

Deep Well Pumping System.—Water in a well may stand near the surface of the ground, but when a pump is placed in operation the water level may drop considerably, depending on the vein feeding the well. Dry seasons of the year may also affect the water level.

If this water level drop below 25 feet from the pump inlet valve, a deep well pump must be installed

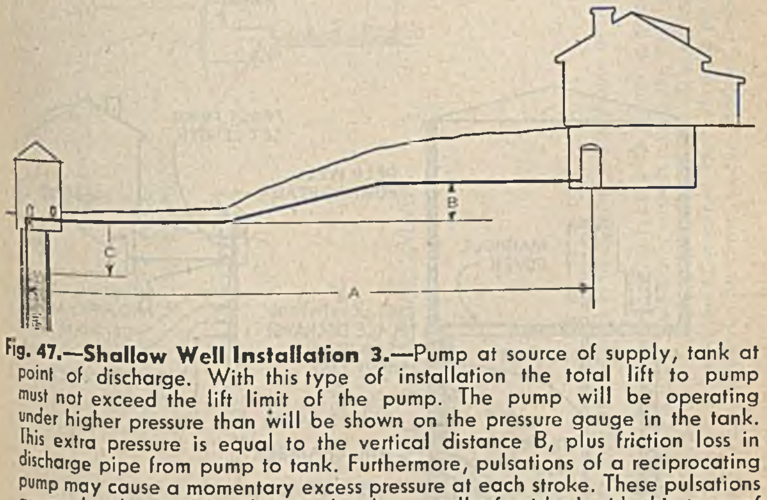


Fig. 47.—Shallow Well Installation 3.—Pump at source of supply, tank at point of discharge. With this type of installation the total lift to pump must not exceed the lift limit of the pump. The pump will be operating under higher pressure than will be shown on the pressure gauge in the tank. This extra pressure is equal to the vertical distance B, plus friction loss in discharge pipe from pump to tank. Furthermore, pulsations of a reciprocating pump may cause a momentary excess pressure at each stroke. These pulsations are partly absorbed by the air chamber usually furnished with this type of pump, but under these conditions, installing in the discharge line near the pump, an additional air chamber with check valve is recommended.

Ques. How are reciprocating deep well pumps installed?

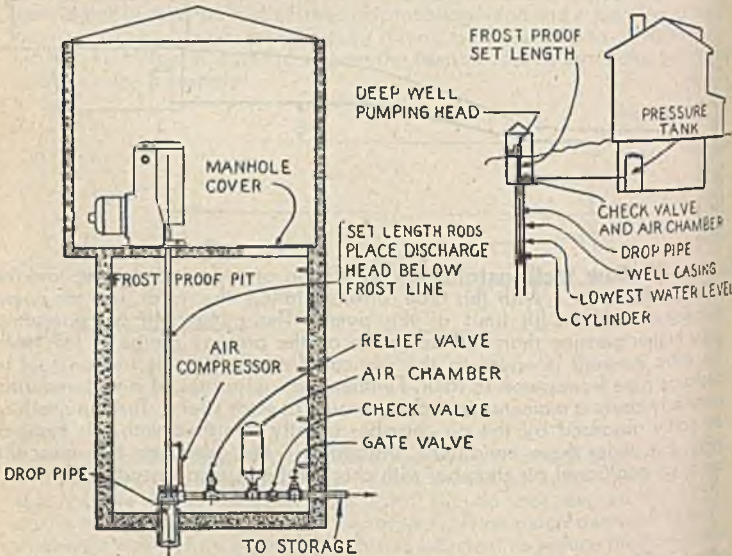
Ans. Directly over the well.

Ques. Are deep well jet pumps similarly restricted?

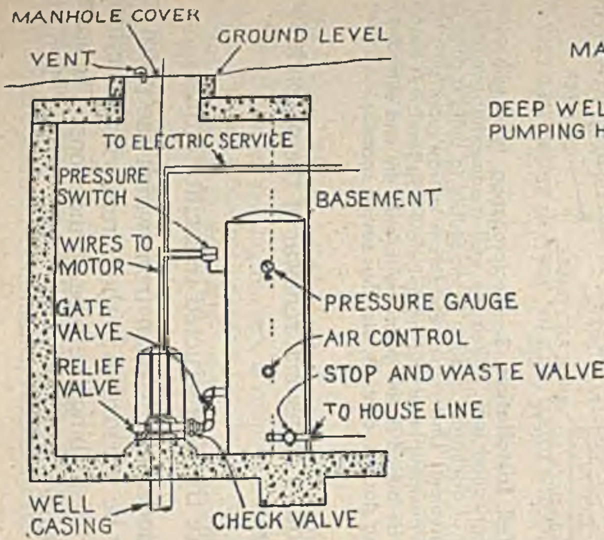
Ans. No, they may be placed some distance away from the well.

Ques. How does a deep well pump work?

Ans. The actual work of pumping water is done by a cylinder or working barrel, located in the well below the lower water level.



Figs. 48 and 49.—Deep Well Installation 1.—Reciprocating type with pumping head with frost proof set. Where it is not possible to have a well conveniently located at the house nor to provide a waterproof pit below the frost line, the method shown here should be used. In such an installation a frost proof set length may be attached to the pump which puts the discharge head and piping below the frost level.



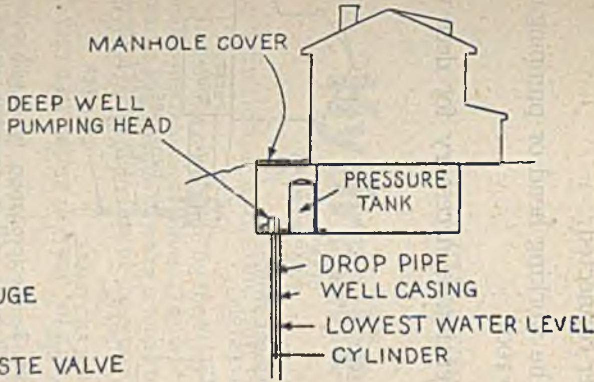


Fig. 50.—Arrangement of deep well pump head in frost proof pit at home as indicated in fig. 51.

Fig. 51.—Deep Well Installation 2.—Reciprocating type with pumping head and tank at house. One of the most satisfactory installations of a deep well water system is with the well located just outside the foundation wall of the house, so that the pump room really becomes a part of the basement, giving easy access to the pumping head, tank and all controls in any kind of weather. Note that the plunger rod can easily be pulled up through the manhole for renewal of the cylinder crimps or cup leathers, inspection of valves, etc.

Ques. How is this cylinder connected?

Ans. It is connected to the working head or pumping unit by means of pipe and pump rod.

Ques. What periodic attention is necessary for deep well pumps?

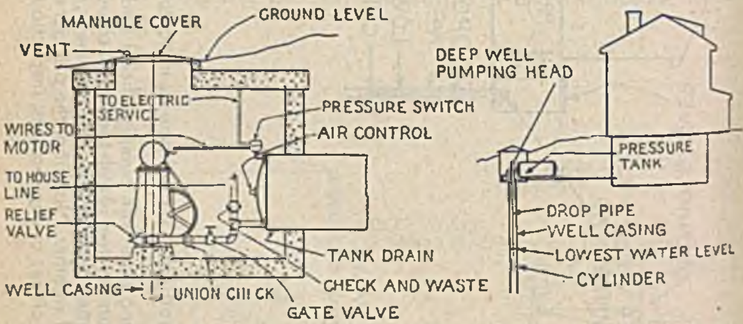


Fig. 52—Deep Well Installation 3.—Reciprocating type with pumping head in pit with horizontal tank. Often the well is somewhat removed from the house or point of discharge, making it impossible to build a pump room connecting the basement. Under such conditions a frost-proof pit may be arranged to set the pumping head and motor below frost level. A horizontal type of tank may be buried in the ground with only the end extending into the pit for attaching gauges, controls, etc. thus saving space.

Ans. They require occasional renewal of the piston leathers and valve facings.

Ques. How are they constructed to facilitate this?

Ans. The cylinder top is open from which the piston and lower valve may be withdrawn without removing the cylinder from the weli or disturbing the pipe connections. The accompanying illustrations, figs. 48 to 52, show various deep well installations.

CHAPTER 29

Hydraulic Rams

By definition, a hydraulic ram is *an impulse pump.*

The impulse takes place by the expenditure of dynamic inertia possessed by a moving column of water. That is to say, the energy due to the momentum of a long column of water is made to force a portion of the water to an elevation higher than that of the source of supply.

A ram can never deliver all the water supplied to it.

A portion of it is wasted to allow the ram to "breathe". That is, after the ram has delivered a charge of water by impulse and the water in the supply pipe has come to rest, a certain quantity of water escapes to waste through the opening of an automatic impulse valve during each interval when the water starts moving again and accelerates up to the speed at which the impulse takes place.

The general hook up of a ram is shown in fig. 1.

Here the water working the ram is supplied through a sloping pipe as shown. The ram should not be placed too close to the source of the supply, or the impulse will be weak, which reduces the height to which the ram can deliver water.

With a long supply pipe the ram becomes more energetic forcing water to higher elevations.

Essential Parts of a Ram.—The various parts necessary for a ram are shown in elementary form in fig. 2.

In addition to the enclosed box like base and air chamber with their pipe connections. Three valves are neçessary:

1. Impulse valve.

2. Air chamber valve.

3. Snifting or air valve.

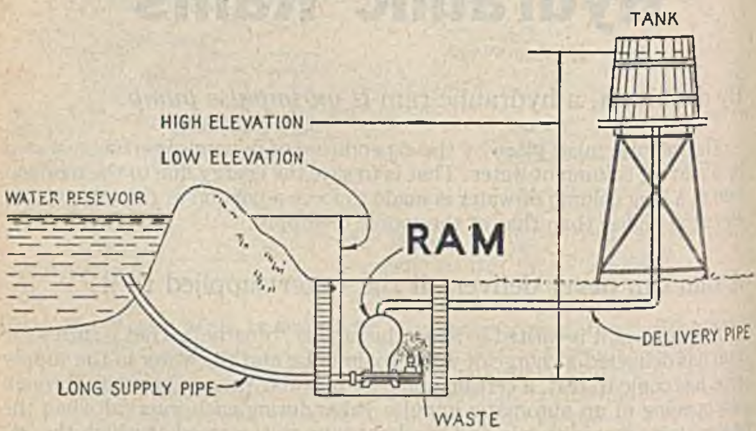


Fig. 1.—Hook up of ram to water supply and tank showing low elevation of supply and high elevation of tank.

All these valves are check valves.

The impulse valve is the heart of the machine as upon it operation depends.

The air chamber valve prevents a back flow of water after impulse, except when closing.

The snifting or air valve admits a supply of air for the air chamber to make up for that absorbed by the water.

How a Ram Works.—At the beginning of the cycle, when the air chamber valve closes, the column of water in the supply pipe rebounds a short distance which removes the pressure from the impulse valve and permits it to open of its own weight.

The vacuum thus created causes the snifting or air valve to open admitting a small amount of air into the base of the ram.

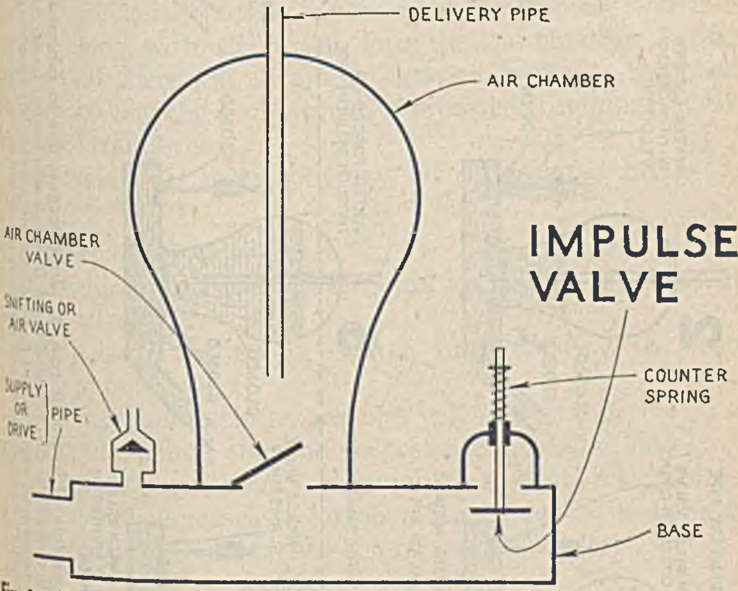
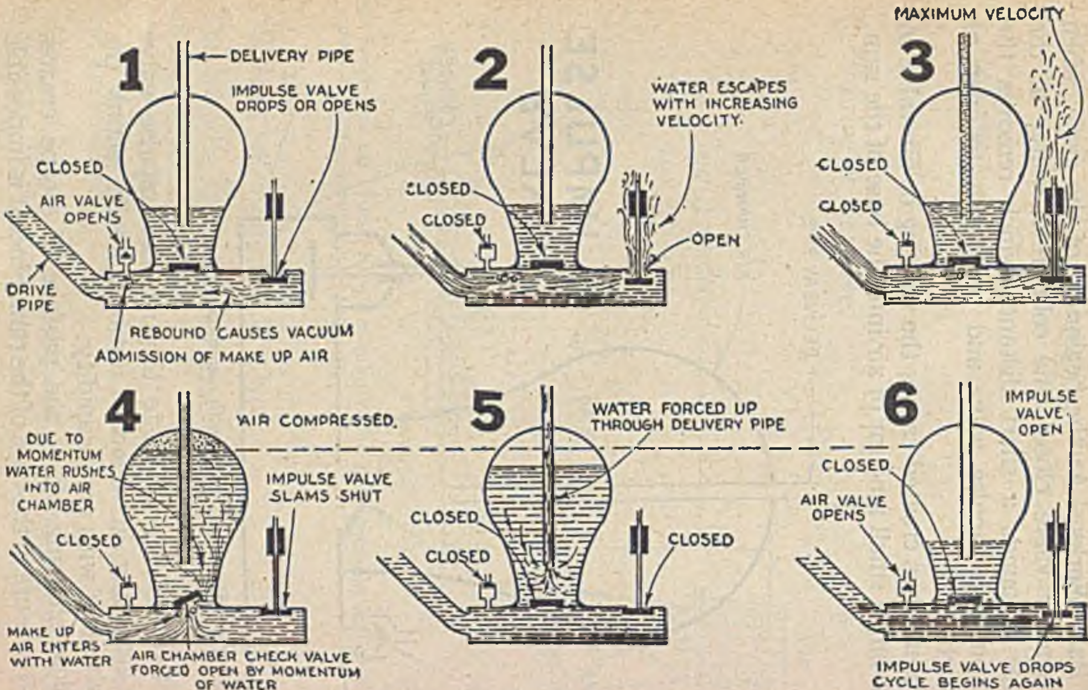


Fig. 2.—Elementary drawings of ram with names of parts.

Water starts flowing to waste through the impulse valve gradually increasing in velocity until it finally slams the impulse valve shut with great rapidity.

The sudden shutting off of the passage of escape creates tremendous pressure in the base of the ram. Since it is impossible



Figs. 3 to 8.—Working cycle of hydraulic ram. **1,** slight vacuum due to rebound causes air check valve to open and admit small amount of air; impulse valve drops, water begins to escape; **2,** velocity of escaping water increases; **3,** maximum velocity or full flow of escaping water; **4,** escaping water slams impulse valve shut, causing impulse which forces water (and air) into air chamber; **5,** inflow of water compresses air in air chamber which cushions impulse and forces water out through delivery pipe; **6,** impulse valve drops, cycle begins again. The positions of the various valves at each instant depicted are plainly shown in the diagrams.

to stop the flow of water instantly (because of its dynamic inertia) it immediately finds another path through the air chamber valve gradually coming to rest.

During the interval it is opposed by a gradual increase of air pressure in the air chamber due to the rising level of the incoming water.

While the charge of water is flowing into the air chamber, the air previously admitted through the snifting valve is carried along with the water into the air chamber, which insures that there will always be some air in the air chamber, thus completing the cycle, which is repeated, beginning at the instant of equilibrium.

The cycle thus described may be summarized briefly as follows:

1. Air chamber valve closes.
2. Rebound creates vacuum and draws in air through snifting valve.
3. Impulse valve opens. Water starts flowing to waste through impulse valve.
4. At maximum velocity impulse valve slams shut.
5. Impulse causes tremendous pressure in boxlike base.
6. Flow detours through air chamber valve into air chamber.
7. Pressure increases in air chamber gradually bringing to rest the inflowing water, thus completing the cycle.

Ques. What is the supply pipe usually objectionably called?

Ans. The drive pipe.

Ques. Why objectionably called?

Ans. The word drive does not define the function of the pipe.

Ques. What is the impulse valve sometimes objectionably called?

Ans. The impetus valve.

Two syllables are better than three.

The Impulse Valve.—This is the most important and sensitive part of a ram and upon its proper design and adjustment depends the satisfactory functioning of this impulse type of pump.

In the design of an impulse valve there are several requirements that are essential for proper operation. They are:

1. Correct valve area
2. Light weight of valve and stem
3. Adjustable counter balance
4. Adjustment for length of stroke
5. No frictional resistance in stem bearings.

Evidently if the valve be too large or too small the correct flow will not be obtained.

A valve too small will introduce added frictional resistance to flow, retarding acceleration of flow during the "waste" period, that is, while the valve is open.

Again if too large the velocity of flow past the valve will be reduced and the rapidity with which the valve "slams shut" will be reduced*.

The valve area, that is, area of port opening should be such that the maximum velocity of flow will be 200 ft. per minute.

In order to avoid too much static and dynamic inertia the valve element should be light and partially counter-balanced by a spring—not a counter weight. This results in very quick closing without lagging due to static inertia of the assembly.

*NOTE.—This rapid closing is just as important as a rapid break of the electrodes of make and break ignition.

With an adjustable spring counter-balance the effect of weight may be reduced.

Means should be provided for adjusting the length of stroke(that is, "lift" of an ordinary valve) as no ram will work satisfactorily unless the valve be limited to the correct stroke to suit the pumping conditions.

Fig. 9 shows an impulse valve as designed by the author embodying the foregoing features.

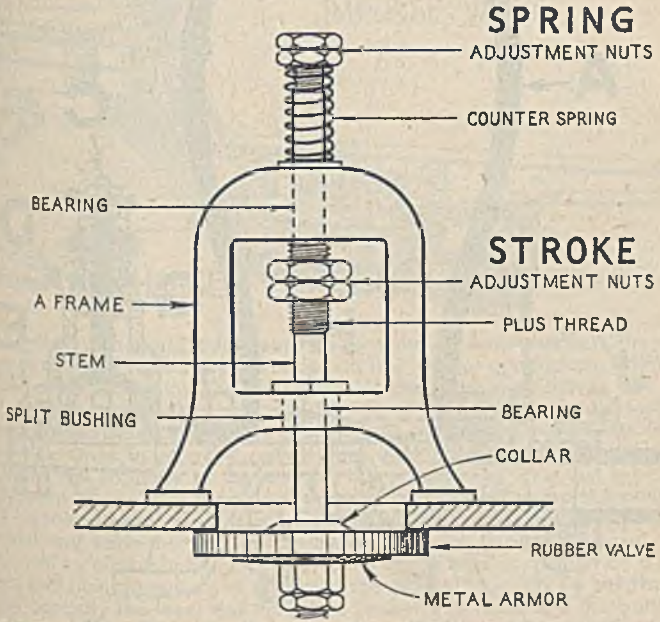


Fig. 9.—Graham impulse valve fully adjustable.

The weight of the valve should be just enough to allow it to drop when the system is in equilibrium.

The accompanying illustrations figs. 10 to 14 show approved ram construction.

Ram Calculations.—After it has been determined how high the water delivered by the ram is to be raised, the following table by Humphreys will give the necessary number of feet fall

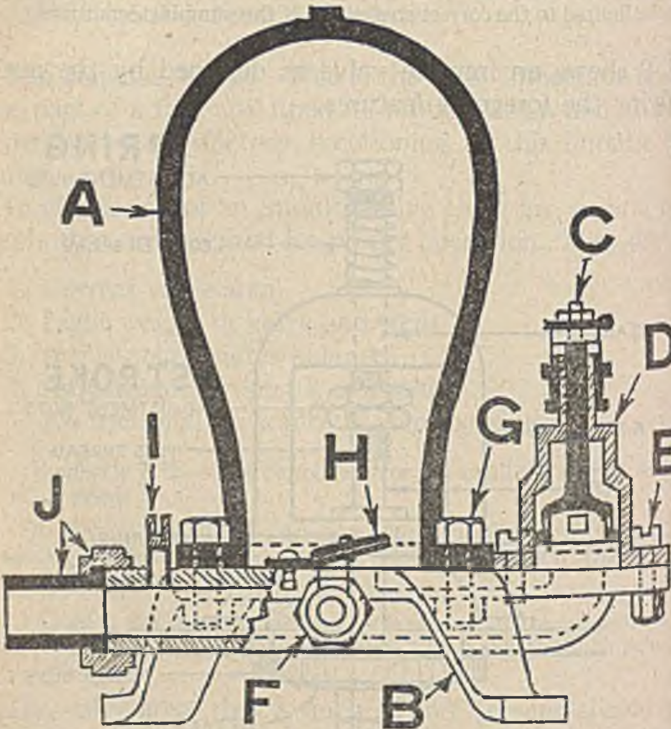


Fig. 10.—Sectional view of typical ram showing construction. The parts are: A, air chamber; B, base; C, impetus valve; D, valve case; E, case screw; F, discharge nut and tube; G, air chamber bolt; H, check valve; I, sniffing valve; J, drive nut and tube.

and the length of drive pipe required for the successful operation of the ram:

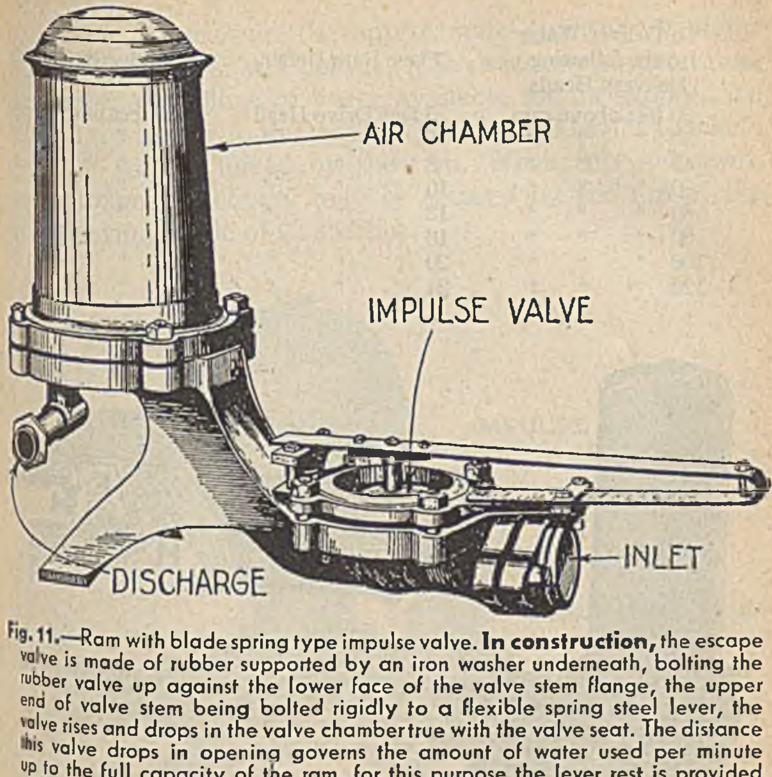
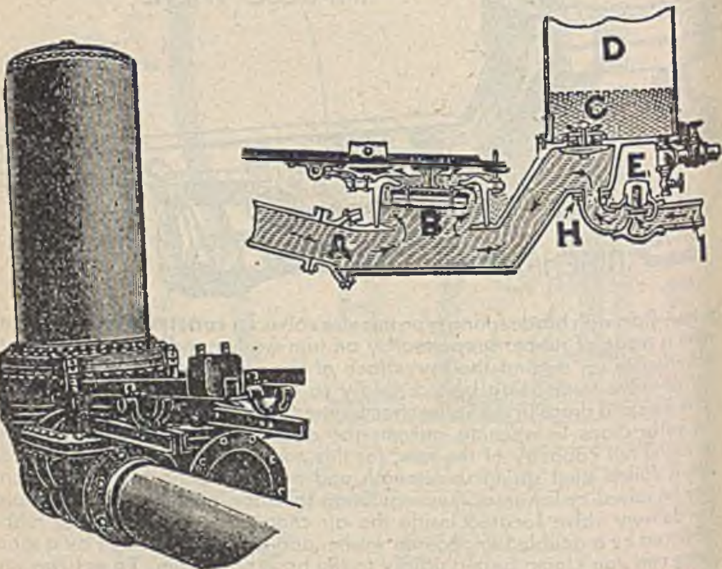


Fig. 11.—Ram with blade spring type impulse valve. **In construction,** the escape valve is made of rubber supported by an iron washer underneath, bolting the rubber valve up against the lower face of the valve stem flange, the upper end of valve stem being bolted rigidly to a flexible spring steel lever, the valve rises and drops in the valve chamber true with the valve seat. The distance this valve drops in opening governs the amount of water used per minute up to the full capacity of the ram, for this purpose the lever rest is provided with a coiled steel spring underneath and two regulating bolts by which it may be raised or lowered, thus regulating the drop or opening of the valve. The delivery valve located inside the air chamber is also made of rubber, reinforced by a double flanged iron washer and is held in position by a rubber spring and iron clamp bolted rigidly to the base of the ram. **To set the ram** at full capacity the lever rest should be properly lowered and the bolt in the end of the lever sufficiently drawn that the opposite end lies gently upon the lever rest. In setting the ram to use less water per minute the manipulation of these bolts should be reversed. At no time should the water escape through the valve with force spurting from the opening; the lever rest should be lowered or the tension bolt in the end of lever slackened. The brass air feeder pin which regulates the air supply must always be sufficiently open that a little escapes at each stroke of the ram. A metallic sound in the pipes or irregular flow at place of delivery indicates that the air pipe should be slightly opened; a quantity of air escaping at intervals, indicates it should be slightly closed.

To Deliver Water to the following Delivery Heads	Place Ram Under	Conducted Through
20 feet above ram	5 feet Drive Head	30 feet of drive pipe
30 “ “ “	6 “ “ “	30 “ “ “ “
40 “ “ “	8 “ “ “	40 “ “ “ “
50 “ “ “	10 “ “ “	50 “ “ “ “
60 “ “ “	12 “ “ “	60 “ “ “ “
80 “ “ “	16 “ “ “	80 “ “ “ “
100 “ “ “	20 “ “ “	100 “ “ “ “
120 “ “ “	24 “ “ “	125 “ “ “ “



Figs. 12 and 13.—Double acting ram. Fig. 12, exterior view; fig. 13, sectional view.

Example.—If the total vertical distance from level of the ram to point of delivery be 80 feet, from the above it will be seen that it is necessary to place ram 16 feet lower than the water which supplies the ram, and to use 80 feet of drive pipe. Any size of hydraulic ram can be used under the above conditions.

In order to ascertain the approximate quantity of water which a certain ram will deliver, it is necessary to determine the number of gallons of water available for its supply, the fall from the supply to the ram and the vertical height to which the water will be forced by the ram. When this is known, the approximate delivery may be secured by the use of the accompanying Table of Capacities.

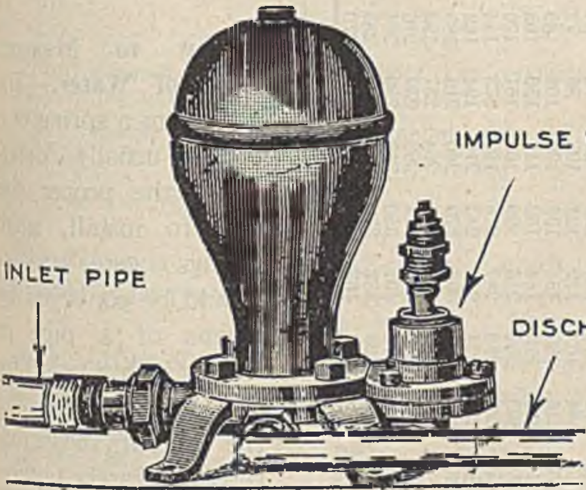


Fig. 14.—Familiar pattern of hydraulic ram made in sizes ½ to 14 gals. per minute.

Rule.—*Multiply the factor opposite drive head and under delivery head by the number of gallons used per minute by the ram*. The result will be the number of gallons delivered by the ram in twenty-four hours.

Example.—With a supply of 5 gals. per minute under a fall of 6 ft. and a delivery head of 20 ft., according to Humphreys, a No. 4 ram will deliver approximately 5 times 324 or 1,620 gals. of water in 24 hours.

TABLE OF CAPACITIES

Drive Head in feet	Delivery Head in Feet 10	15	20	30	40	50	60	70	80	90	100	120	140	160	180	200
4	432	288	216	144	108	76	64	55	48	43	38	32	....	....	....	....
5	540	360	270	180	135	108	80	69	60	51	48	40	34	....	....	....
6	....	432	324	216	162	130	108	82	72	64	58	48	41	36	32	....
7	....	505	378	252	189	151	126	108	84	75	68	56	48	42	37	....
8	....	....	432	288	216	173	144	123	108	86	78	64	55	48	42	38
9	....	....	486	324	243	194	162	139	121	108	86	72	62	54	48	43
10	....	....	540	360	270	216	180	154	135	120	108	80	69	60	53	48
12	....	....	....	432	324	259	216	184	162	144	130	108	83	72	64	57
14	....	....	....	505	378	303	252	216	189	168	151	126	108	84	75	66
16	....	....	....	....	432	346	288	247	216	192	173	144	123	108	86	77
18	....	....	....	....	486	389	324	278	243	216	194	162	138	121	108	86
20	....	....	....	....	540	432	360	308	270	240	216	180	154	135	120	108
22	....	....	....	....	....	475	396	339	296	264	237	198	169	148	132	118
24	....	....	....	....	....	519	432	370	324	288	259	216	184	161	144	130
26	....	....	....	....	....	....	468	401	350	312	280	234	200	175	156	140
28	....	....	....	....	....	....	505	432	378	336	303	253	216	188	168	151
30	....	....	....	....	....	....	540	463	405	360	324	270	231	202	180	161

For a small supply, the amount of water available per minute can easily be determined by the length of time required to fill an ordinary gallon pail.

How to Measure Flow of Water.—The flow from a spring or a brook usually determines the proper size ram to install, small springs or streams should be conveyed by means of a pipe or trough into a vessel for one minute or less, and the contents measured, thus accurately obtaining the flow of gallons per minute. If fall be insufficient to convey the water into a bucket or tub, sink the vessel in the ground (or water) until it passes under the end of pipe or trough.

Relative Head and Fall giving Maximum Efficiency

(According to Columbiana)

To deliver Water to Height of	Place Ram Under	Proper Length of Drive Pipe
20 feet above ram	3 feet head or fall	30 feet of drive pipe
30 " " "	4 " " " "	30 " " " "
40 " " "	5 " " " "	40 " " " "
50 " " "	7 " " " "	50 " " " "
60 " " "	8 " " " "	60 " " " "
80 " " "	10 " " " "	80 " " " "
100 " " "	14 " " " "	100 " " " "
120 " " "	17 " " " "	125 " " " "

*Any size ram may be operated under these conditions and will afford the following approximate delivery:

No. 2	require	2 to 3 gals. per min.	and del.	10 to 15 gals. per hour
" 3	"	2 " 4 " " "	" "	10 " 20 " " "
" 4	"	3 " 7 " " "	" "	15 " 35 " " "
" 5	"	6 " 12 " " "	" "	30 " 60 " " "

The following rule (according to Rife) gives the number of gallons of water that may be delivered per hour to a given point.

Rule.—*Multiply the number of gallons the spring or stream flows per minute by the feet in fall. Multiply this product by 40, then divide by the number of feet the water is to be elevated above the ram.* The result will be the number of gallons delivered per hour.

Example.—If the flow of water be 15 gals. per minute under 8 ft. head, how many gallons delivered per hour at 50 feet elevation?

Flow of water per minute.............15 gallons
Fall of same.......................... 8 feet

120
40

50)4800

96 gallons delivered per hour

Handy Table Showing How to Find Number of Gallons a Ram Will Deliver Per Day of 24 Hours

Fall in Feet	ELEVATION IN FEET																		
	4	6	8	10	12	14	16	18	20	25	30	35	40	50	60	70	80	90	100
	The Figures Below Represent (Theoretical) Number of Gallons Discharged in 24 hours for Each Gallon of Water Per Minute Supplied to the Ram																		
2	360	240	180	144	120	102	90	80	72	...	...		...	...	...	...	...	...	...
3	...	360	270	216	180	153	135	120	108	86	72	...	...	...	...	...	...	...	...
4	...	...	360	288	240	204	180	160	144	115	96	82	72	...	...	...	...	...	...
5	...	...	...	360	300	255	225	200	180	144	120	102	90	72	...	...	...	...	...
6	...	...	...	...	360	309	270	240	216	173	144	123	108	86	72	...	...	...	...
7	...	...	...	...	...	360	315	280	252	201	168	143	126	100	84	72	...	...	...
8	...	...	...	...	...	...	360	320	288	230	192	164	144	115	96	82	72	...	...
9	...	...	...	...	...	...	...	360	324	259	216	184	162	129	108	92	81	72	...
10	...	...	...	...	...	...	...	...	360	288	240	205	180	144	120	102	90	80	72

The figures in the above table are based on a Ram efficiency of 50 per cent.

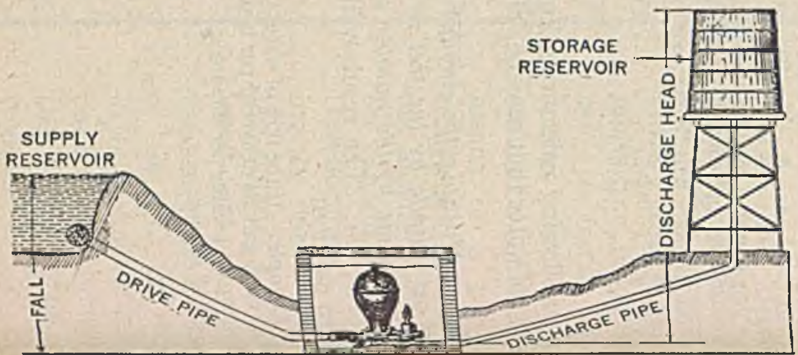


Fig. 15.—Ram installation to accompany table of proportionate head or fall giving highest efficiency in operation.

The following rule by Rumsey gives the number of gallons delivered per 24 hours:

Rule.—*Multiply the number of gallons furnished by the spring per minute, by 936; multiply this product by the height*

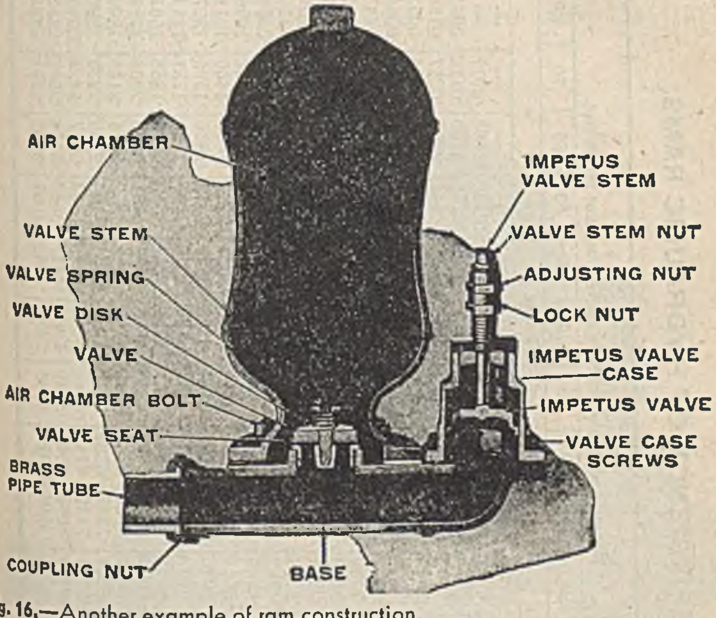


Fig. 16.—Another example of ram construction.

of the spring (in feet) above ram; then divide by the height (in ft.) between ram and point of delivery. The result will be the number of gallons delivered per day of 24 hrs.

How to Install a Ram.—For satisfactory operation a ram should not be installed in a haphazard way, but in accordance with certain conditions which experience has proven to be essential for best results.

EFFICIENCY OF HYDRAULIC RAMS

Height of fall in feet.	Elevation of discharge above delivery valve of ram in feet.														
	15	18	21	24	27	30	35	40	45	50	60	70	80	90	100
	Percentage.														
2	·0724	·0533	·0402	·0307	·0255	·0181	·0112	·0063	·0027	—	—	—	—	—	—
3	·1327	·1020	·0807	·0651	·0532	·0441	·0326	·0243	·0181	·0132	·0063	·0017	—	—	—
4	·1960	·1535	·1234	·1020	·0854	·0724	·0560	·0441	·0348	·0281	·0180	·0112	·0063	·0027	—
5	·2614	·2068	·1686	·1404	·1189	·1020	·0807	·0652	·0533	·0441	·0307	·0217	·0150	·0099	·0063
6	·3282	·2614	·2146	·1800	·1535	·1327	·1063	·0870	·0724	·0608	·0441	·0325	·0243	·0180	·0132
7	·3960	·3170	·2614	·2203	·1885	·1640	·1327	·1096	·0920	·0782	·0580	·0441	·0340	·0264	·0205
8	·4647	·3733	·3090	·2614	·2248	·1960	·1595	·1327	·1121	·0960	·0724	·0560	·0441	·0351	·0281
9	·5341	·4303	·3572	·3030	·2614	·2285	·1868	·1561	·1327	·1142	·0870	·0682	·0545	·0441	·0360
10	·6040	·4877	·4058	·3450	·2984	·2614	·2145	·1800	·1535	·1327	·1020	·0807	·0651	·0533	·0441
11	·6745	·5459	·4549	·3874	·3357	·2947	·2425	·2041	·1746	·1514	·1172	·0934	·0760	·0627	·0524
12	·7453	·6040	·5043	·4302	·3733	·3282	·2708	·2285	·1960	·1704	·1327	·1063	·0870	·0723	·0608
13	·8166	·6627	·5540	·4732	·4112	·3620	·2994	·2532	·2177	·1896	·1483	·1194	·0983	·0821	·0694
14	·8881	·7217	·6040	·5166	·4494	·3960	·3282	·2780	·2395	·2090	·1640	·1327	·1096	·0920	·0782
15	·9600	·7809	·6543	·5601	·4877	·4303	·3572	·3030	·2614	·2285	·1800	·1460	·1211	·1020	·0870
16	—	·8404	·7048	·6040	·5263	·4647	·3863	·3282	·2835	·2482	·1960	·1595	·1327	·1121	·0960
17	—	·9001	·7555	·6480	·5650	·4993	·4157	·3535	·3058	·2680	·2123	·1731	·1444	·1223	·1050
18	—	·9600	·8064	·6921	·6040	·5341	·4451	·3790	·3282	·2880	·2286	·1868	·1561	·1327	·1142
19	—	—	·8574	·7364	·6430	·5690	·4746	·4046	·3507	·3081	·2449	·2006	·1680	·1430	·1262
20	—	—	·9086	·7800	·6823	·6040	·5042	·4303	·3733	·3282	·2614	·2145	·1800	·1535	·1327
21	—	—	·9600	·8254	·7217	·6392	·5340	·4561	·3960	·3486	·2780	·2286	·1920	·1640	·1420
22	—	—	—	·8701	·7612	·6745	·5640	·4820	·4188	·3688	·2947	·2425	·2041	·1746	·1514
23	—	—	—	·9150	·8007	·7098	·5940	·5080	·4417	·3892	·3114	·2567	·2163	·1853	·1609
24	—	—	—	·9600	·8404	·7433	·6241	·5341	·4657	·4097	·3282	·2708	·2185	·1960	·1704

The following general information should be noted.

The fall or vertical distance from surface of water in the supply reservoir to supply opening in the ram, must not be less than two feet, as this is the lowest head under which a ram will work satisfactorily.

The discharge elevation must not be less than six times nor more than twelve times the fall.

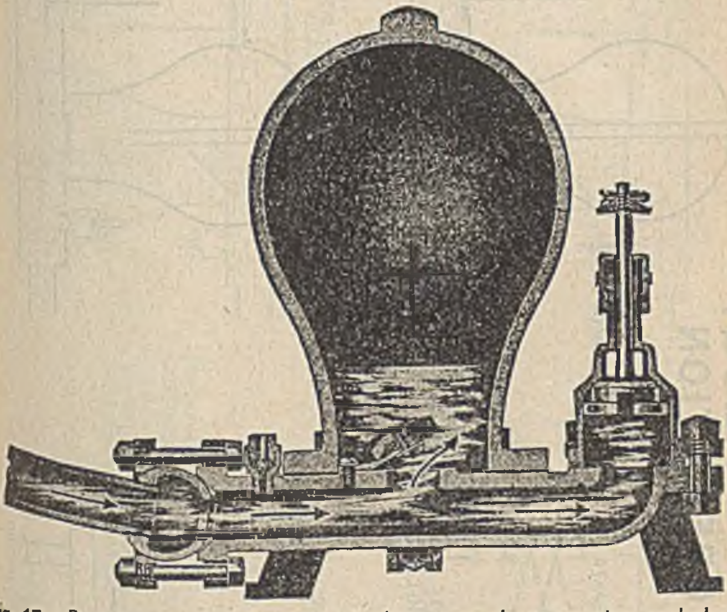
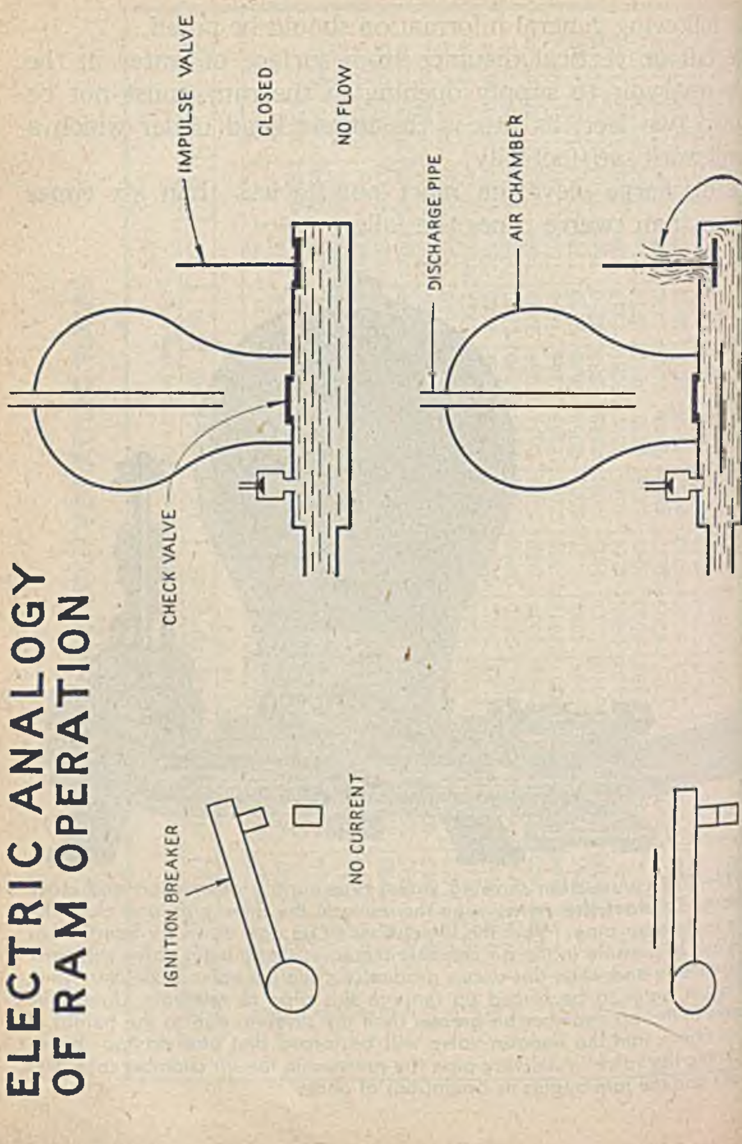
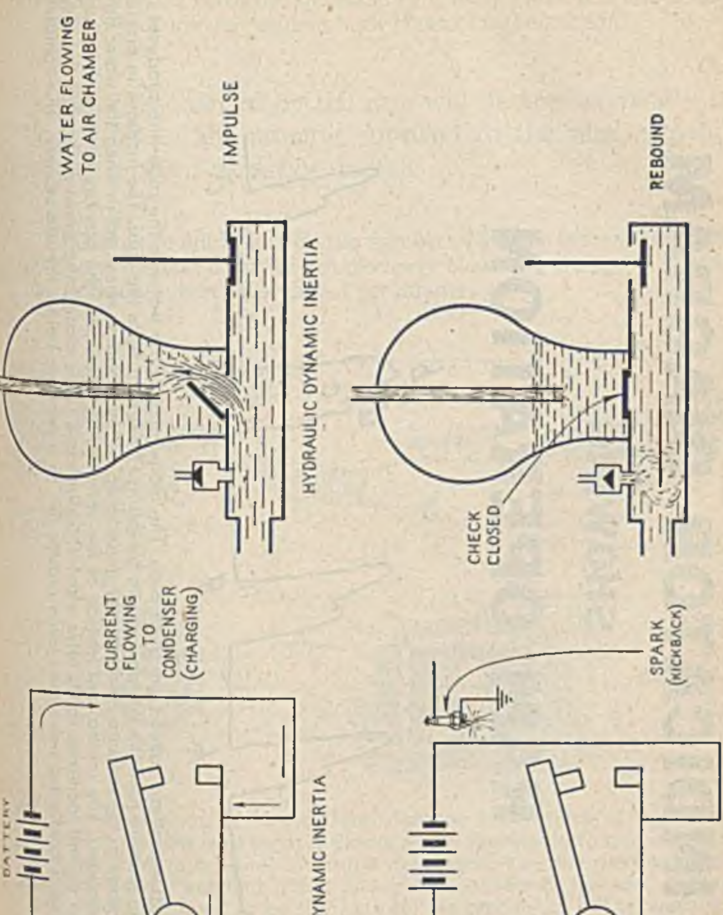


Fig. 17.—Ram construction showing swivel type supply connection and clock on valve. **To start the ram,** open the valve in the drive pipe and close the valve in delivery pipe. Work the impetus valve up and down by hand a few times and as pressure in the air chamber increases the impetus valve will work automatically and when this occurs gradually open the valve in delivery pipe and allow water to be forced up through this pipe to reservoir. Unless the pressure in the air chamber be greater than the pressure due to the height of source above ram the impetus valve will be forced shut and stay so; hence by closing the valve in delivery pipe the pressure in the air chamber increases rapidly and the ram begins its operations at once.



Figs. 18 to 21.—Electric analogy of ram operation. Continued on next page, figs. 22 to 25.



Figs. 22 to 25.—Electrical analogy of ram operation—*Continued.*

INDICATOR DIAGRAM

SHOWING

RAM OPERATION

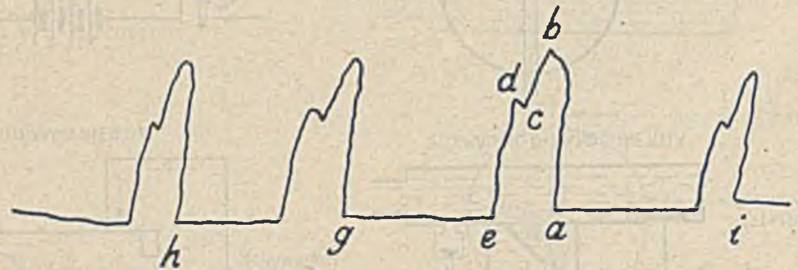


Fig. 26.—Indicator diagram taken from drive pipe. The momentum of the water in the pipe line produces air increase in the pressure near the valve as shown by the rise from **a** to **b,** in the card. The increase in pressure will cease as soon as the pressure is sufficiently great to open the discharge check valve when the water flows into the air chamber. The inertia and friction of the valve being overcome, the pressure drops with a backward wave to **c,** and increases to **d,** after which the discharge valve closes; the pressure drops in the space around **g,** and a backward wave may even reduce this to a pressure below the atmosphere. This completes one cycle.

Should the discharge elevation be less than six times the fall, this condition may be corrected by installing a valve in the line and throttling it so as to produce the required back pressure by restriction.

Quantity discharged by the ram will be approximately from 1/12 to 1 /24 of the amount supplied to the ram, depending upon the ratio of elevation to fall.

Multiply the fall in feet by the number of gallons per minute supplied; divide the product by twice the discharge elevation; the result will be the approximate quantity discharged per minute.

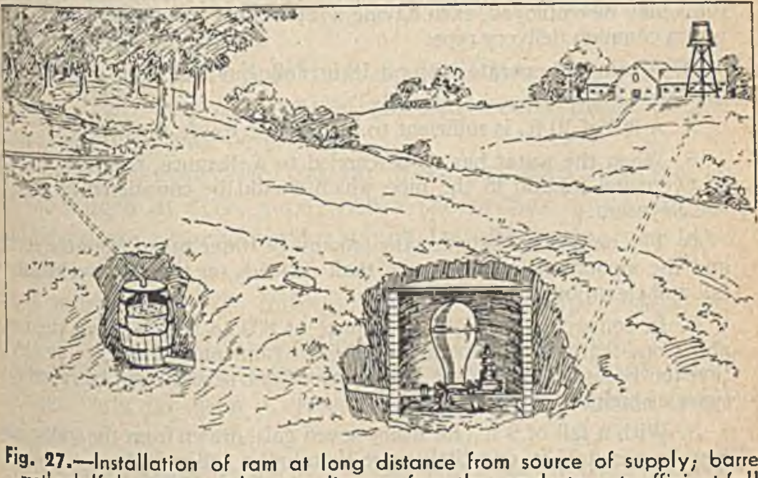


Fig. 27.—Installation of ram at long distance from source of supply; barrel method. If the ram must be at a distance from the supply to get sufficient fall, pipe the water to a barrel set at the right point. Lay the drive straight on a gradual incline, bending it just enough to connect to the ram. If the drive pipe should be found to be too long for the amount of fall secured then the water should be piped from the source of supply to a point within the required distance of the ram. At this point, an open barrel should be placed and the drive pipe connected to it, or should the fall be so great as to interfere with the operation of the ram, adoption of the same plan is advised. The pipe leading from the source of supply to the barrel should be at least a size larger than the drive pipe between the barrel and the ram.

Batteries of Rams.—In many cases larger capacities are desired than it is practical for one ram to handle to good advantage. For such conditions several rams may be used connected in parallel to a single discharge pipe.

Points Relating to Rams.—1. Where the water supply is limited, a ram must be used which the spring will furnish with at least the minimum quantity of water stated by the manufacturer of the ram selected.

2. Where the water supply is abundant, the selection of the proper size of ram is governed by the quantity of water desired per day. If more water be desired than a single ram can furnish, batteries of two or more rams may be employed, each having a separate drive pipe but discharging into a common delivery pipe.

3. A ram will operate with an 18 in. fall, but will pump further when the fall is greater.

4. A fall of 10 ft. is sufficient to raise water 150 ft. or more.

5. When the water has to be carried to a distance, the ram also has to overcome friction in the pipe which should be considered as part of the elevation.

6. The proportion between the amount of water taken from the spring and the amount delivered to the tank depends on the relative height of fall and elevation.

7. In conveying water to a distance of 800 to 900 ft. from the ram, about one-tenth of the amount driving the ram can be raised to an elevation ten times as high as the fall, or about one-seventh can be raised five times as high as the fall.

8. With a fall of 5 ft., for every seven gals. drawn from the spring, one may be raised 25 ft. or a little more than half a gallon 50 ft.

9. Make all joints tight.

10. Place the upper end at least a foot under water and protect it with a strainer coarse enough to permit a free flow.

11. A full way gate to shut off the water is convenient.

12. Avoid sharp turns or elbows as far as possible and make all joints tight.

13. Put a full way gate valve near the ram.

14. In setting, the ram should be bolted to a level foundation of timber or concrete. It is best located in a covered pit with a drain for the waste water.

Installing Instructions.—Place the ram in a masonry lined pit and bolt it on a level foundation. Provide drainage for waste water from the bottom of pit. Pipes should be laid without bends to minimize friction, and be placed below the frost line. Where turns are necessary, long bends are better than abrupt angles.

The top of the drive pipe should be fitted with a strainer and be placed a foot or more below the surface of the water and six or more inches above the bottom of reservoir. All joints should be air tight.

The length of a drive pipe must never be less than five times nor more than ten times the fall. Any convenient length within this range may be used.

If source of supply be too far from the ram location to keep the length of drive pipe within the proper limits, install an intermediate reservoir from which the drive pipe can take its supply.

Pipe at least one size larger than the drive pipe should be used to connect the source of supply to this reservoir.

Discharge should be piped into a reservoir or storage tank. If storage tank be open it should be fitted with a piped overflow.

If pneumatic tank be used, a relief valve set at the required pressure should be installed to prevent breakage from excessive pressure.

The length of the discharge pipe for any given elevation may be any length, provided the pipe be of sufficient size.

Connect all pipes before starting the ram and leave them uncovered until it is found there are no leaks.

How to Start a Ram.—To put a ram into operation simply open gate valve in supply pipe and press down the impulse valve which permits the water to discharge; then allow it to rise. The ram should continue to operate after this procedure is repeated a few times. If it should not, the impulse valve stroke is not properly adjusted. By changing the adjustment and trying again, the correct adjustment will be readily obtained. The nuts on the impulse valve govern this adjustment.

Should the fall be not sufficient, the quantity discharged will be very small. If the fall be too great, the impulse valve seat and valve will soon wear out due to the excessive hammering of these parts.

How to Stop a Ram.—The supply pipe should be provided with a gate valve by means of which to control the flow in the supply pipe and to stop the ram.

Ram Troubles.—1. Should the ram stop with the impetus valve up, there is a leak somewhere about the valve seat or flange which must be located and stopped. No weight is needed on the impetus valve to make it drop. It is heavy enough to drop of itself, if there be no pressure under it from leakage.

2. Should the ram stop with the impetus valve down, it is evidence of an insufficient supply of water. The strainer may be stopped up or the gate way not have opened wide enough. If the water level in the spring be drawn down, the trouble lies with the quantity of water furnished. The ram can then be adjusted to a shorter stroke to use less water and if necessary one or more of the valve ports can be closed up by forcing dry wood into the opening or soldering in a piece of brass.

3. If the ram operate and deliver no water, the air may be exhausted from the air chamber. To refill it, shut off the pipes and take off the cap on the opposite side from the discharge pipe. If the impetus valve be making an upward stroke and then fluttering at the top, there is air in the drive pipe. Either the pipe is leaking or else its end is not completely submerged in the spring. An obstruction lodged under the check valve so as to prevent its closing may also cause this trouble.

CHAPTER 30

Special Service Pumps

The proper selection of pumps must be based upon a thorough understanding of the characteristics of the fundamental types known as centrifugal, reciprocating, turbine, rotary, etc. The differences and essentials of all these types have been fully explained in the foregoing chapters.

It remains to present features of design that render any one of these fundamental pumps suitable for a specialized or unusual service.

By definition, a special service pump is *one of a type and design that renders it specially suited to unusual service conditions*, such as fire, sewage, deep well, oil pumping, etc. Evidently in the design of such pumps provision should be made for meeting all the requirements of the particular special service for which the pump is intended.

Special Service Conditions.—The multiplicity of special pumps which have been introduced are due to the great variety of unusual conditions of operations, met with in practice. These conditions may be classified or divided into different groups such as:

1. With respect to pressure
   - *a.* Low (tank)
   - *b.* Medium (tank)
   - *c.* High (*boiler feed*)
   - *d.* Extra high (*hydraulic*)

2. With respect to the nature of the liquid
   - *a.* Pure water
   - *b.* Sewage {rain, sludge}
   - *c.* Magma
   - *d.* Trash
   - *e.* Thick
   - *f.* Paper stock
   - *g.* Oil
   - *h.* Various chemicals
   - *i.* Milk, etc.

3. With respect to capacity-pressure conditions, as
   - *a.* Irrigation (propeller)
   - *b.* Hydraulic

4. With respect to installation conditions, as
   - *a.* Dry pit
   - *b.* Shallow well
   - *c.* Deep well
   - *d.* Automatic sump

5. With respect to miscellaneous applications, as
   - *a.* Fire
   - *b.* Railroad
   - *c.* Automobile
   - *d.* Diesel
   - *e.* Drainage
   - *f.* Contractors
   - *g.* Measuring
   - *h.* Mines
   - *i.* Long stroke
   - *j.* Sinking
   - *k.* Differential

Low Service or Tank Pumps.—Where water and other liquids are to be pumped to moderate heights at working pressures up to 75 lbs. per sq. in. light duty simplex or duplex piston pumps are generally used. They are adapted for use in connection with railroad water stations, gas plants, tanneries, sugar refineries, bleacheries, etc.

These pumps are similar in design to standard general service pumps, but by reason of their cylinder proportions they are more economical in steam consumption for this class of service.

An idea of the difference in cylinder dimensions is shown in the following tabulation of general service and low service duplex (tank) pumps:

General Service Pumps

Steam Cylinder Diameter, ins.	Water Cylinder Diameter, ins.	Stroke ins.
3	2	4
	ranging up to	
16	10	12

Low Service, Tank Pumps		
3	2½	4
10	10	12

In these tabulations note that there is greater difference between diameters of steam and water cylinders of the general service pumps than for the low service or tank pumps. This is because the general service pumps are designed to pump against greater pressures than the tank pumps. That is,

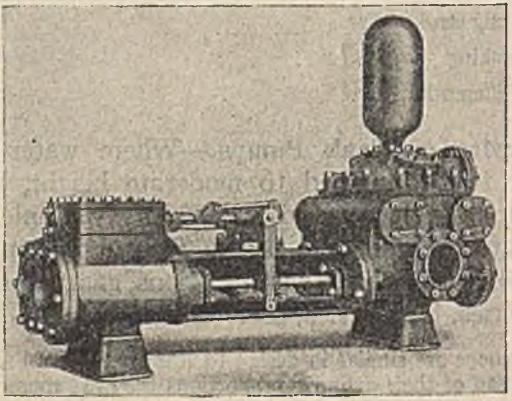


Fig. 1.—Low service or tank steam duplex pump for water working pressures up to 75 lbs. per sq. in. They are adapted for use in connection with railway water stations, gas plants, tanneries, oil works, sugar refineries, bleacheries, etc.

general service pumps are proportioned to pump against pressures up to 200 lbs. per sq. ins. as compared with a 75 lb. limit for low service or tank pumps.

Figs. 1 and 2 show typical steam duplex low service and general service pumps from which it is seen that they are similar in design, the chief difference being in the cylinder proportions.

For general and low service duplex pumps three kinds of packing are provided for the water end:

1. Soft canvas
2. Babbitt float rings
3. Ball and snap rings.

as shown in figs. 3 to 5.

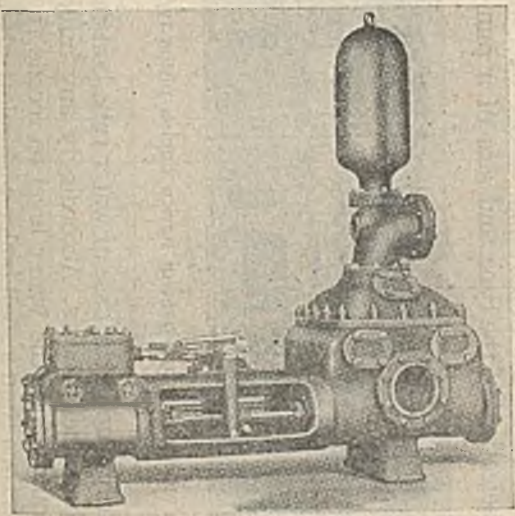


Fig. 2.—General service steam duplex pump for water working pressures up to 150 lbs. per sq. in. They are adapted to railroad water stations, large industrial plants and various other installations where a greater head than for low service pump is essential. The chief difference between the low and general service is the steam water cylinder ratio.

Ques. For what conditions is canvas packing used?

Ans. For pumping hot or cold water.

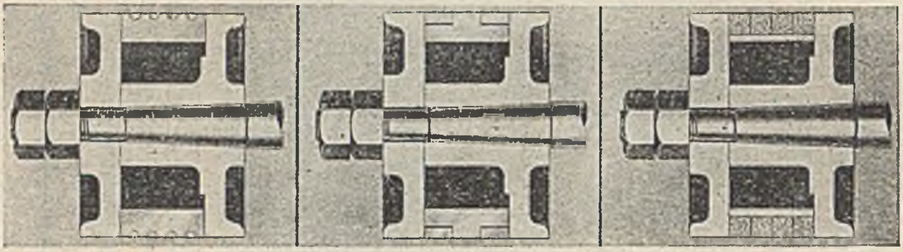
Ques. When are ball and snap rings used?

Ans. For pumping viscous fluids such as crude oil, tar, molasses, etc.

Ques. What other packing may be used for this purpose?

Ans. Babbitt float rings.

Boiler Feed Pumps.—Owing to the importance of the operation of feeding water into boilers, the proper selection of the type and size of a pump for this service is



Figs. 3 to 5.—Various packings used on low and general service duplex piston type pumps as explained in the text.

essential. This is particularly true in the case of water tube boilers which have a much smaller water capacity than shell boilers. Accordingly any interruption of the feed to a water tube boiler would cause the water level to recede to the lowest level for safe operation in much less time than a shell boiler with relatively large water capacity.

So important is boiler feeding for marine plants that two separate means of feeding are required.

Many types of pump are in use for boiler feeding, all of which have been treated in a general way.

The general requirements are:

1. Operation at variable speed.
2. Operation with hot feed water.

The large number of types of pumps used for boiler feeding may be divided into two classes:

1. Reciprocating
2. Centrifugal

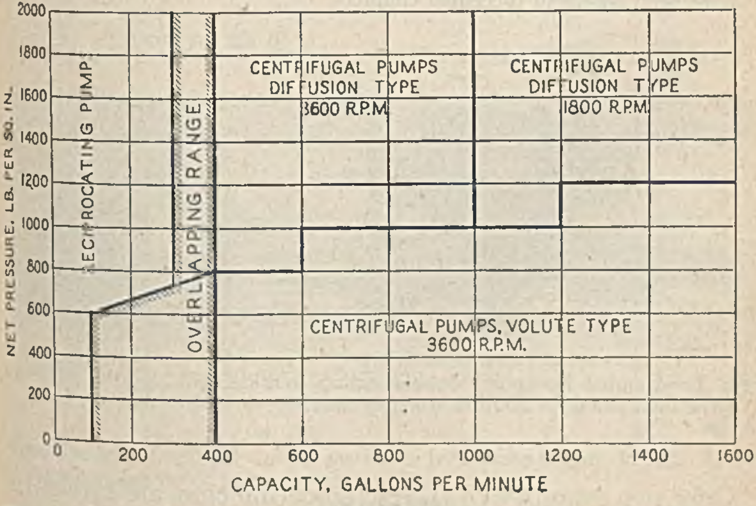


Fig. 6.—Chart indicating selection of boiler feed pump according to a noted authority.

Between these two classes, two factors which govern selection are:

1. Steam pressure
2. Size of plant

Suggested selection according to Worthington is given in the chart, fig. 6. Because of space requirements and cost, the upper limit of the reciprocating pump is given as 400 gallons per minute; other factors also enter into limiting capacity.

For reliability the duplex is to be preferred to the simplex.

A duplex pump will always start while a simplex pump may or may not start. Practically all forms of simplex and duplex pumps are used, such as piston, plunger, vertical, horizontal, pot valve, etc. All of these patterns have been shown in preceding chapters.

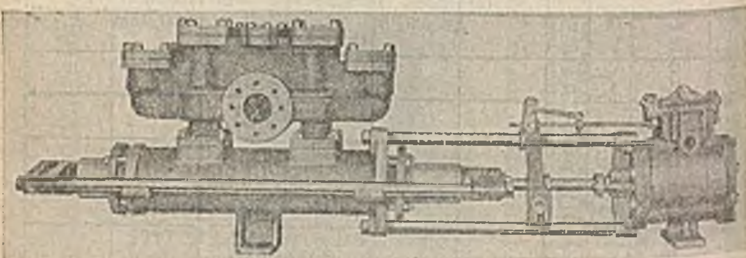


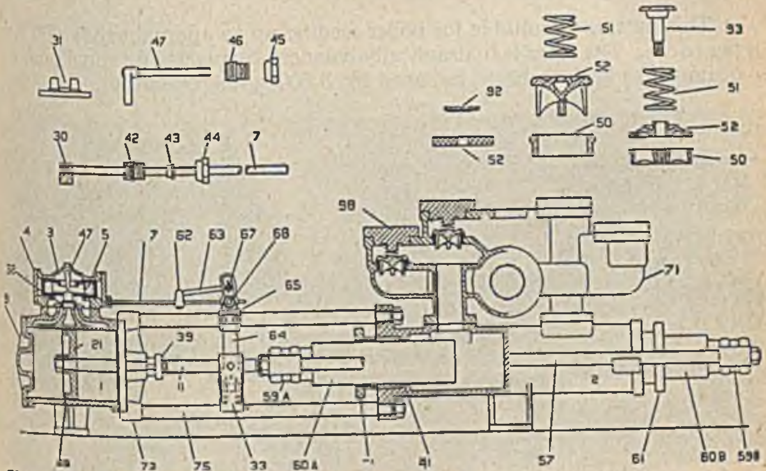
Fig. 7.—Simplex horizontal double acting, outside end packed, pot valve boiler feed pump for 300 lbs. working pressure.

Only two examples of the reciprocating class are here given. Fig. 7 shows a very rugged design of simplex double acting pump outside end packed pot valve pattern suitable for 300 lbs. working pressure.

A general idea of proportions and range of standard sizes is obtained from the accompanying tabulation. The sectional view and part list, figs. 8 to 22, show construction of the pump illustrated in fig. 7.

TABULATION OF SIZES OF PUMP FIGS. 7 TO 22

Diameter of Steam Cylinder, Inches	Diameter Plunger, Inches	Length of Stroke, Inches	Boiler Feed and Pressure Service: Gallons per Stroke	Strokes per Minute	Gallons per Minute	Boiler H. P. Pump Will Feed	General Service: Strokes per Minute	Gallons per Minute	PIPE SIZES: Steam	Exhaust	Suction	Discharge	Floor Space In Inches
5½	3¾	7	.33	45	14	200	80-100	26- 33	¾	1	2	1½	76 × 11
6	4	10	.55	40	22	318	60- 80	33- 44	1	1¼	3	2	94 × 11
8	5	12	1.02	40	40	580	50- 60	51- 60	1	1½	4	2½	110 × 14
10	6	12	1.47	40	58	840	50- 60	73- 88	1¼	2	4	3	114 × 16
12	7	12	2.00	40	80	1160	50- 60	100-120	2	2½	6	4	118 × 18
14	8	12	2.61	40	104	1500	50- 60	130-156	2	3	6	4	125 × 18
14	8	16	3.48	40	139	2000	40- 50	174-208	2	3	6	4	151 × 18
14	9	20	5.50	35	192	2780	40- 50	174-217	2	3	7	6	176 × 23
16	10	18	6.12	35	214	3100	40- 50	220-275	2½	3	8	7	166 × 23



Figs. 8 to 22.—Sectional view of the boiler feed pump shown in fig. 7, with accompanying list of parts.

LIQUID END

2. Pump Cylinder
41. Water Cyl. Throat Bush.
50. Water Valve Seat
51. Water Valve Spring
52. Water Valve
57. Side Rod
59A Plunger Crosshead, Inner
59B Plunger Crosshead, Outer
60A Plunger, Inner } Note
60B Plunger, Outer
61A Plun Gland, Inner } Note
61B Plun. Gland, Outer
71. Valve Pot. See note.
83. Discharge Connection
92. Plates for Valve
93. Valve Stem
94. Valve Guard
98. Valve Pot Cover

Fig. 23 shows a small duplex pump of the one piece pattern with pressed in tube liners. This is a very small pump, size 3 x 2 x 4 for 250 lbs. water pressure.

For medium and large size boiler feeding at high steam pressure, centrifugal pumps are used. With steam pressures above the limit for reciprocating pumps and ranging as high as 2,000 lbs. per sq. in. and over, the multi-stage centrifugal pump is used.

A typical six stage centrifugal pump direct coupled to an electric motor is shown in fig. 24.

This pattern is suitable for boiler feeding up to approximately 600 lbs. per sq. in. The rotor is hydraulically balanced by placing an equal number of impellers back to back, designed for 3,600 *r.p.m.* operation.

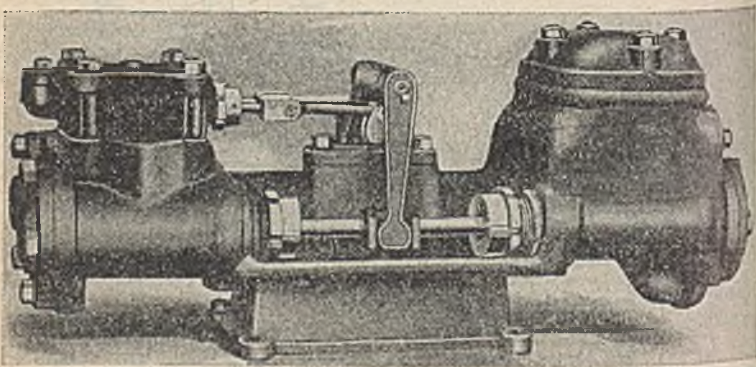


Fig. 23.—Small duplex boiler feed steam pump one piece pattern.

"Doctor" Boiler Feed Pumps.—In Western river steam boat practice, a unique type vertical beam engine with crank and fly wheel operating four pumps and having feed water heaters supported by the frame. Two single lift pumps draw water from the river and deliver it to the heaters, while the other two or feed pumps proper, pump from the heaters into the boilers. Each pump has sufficient capacity to supply all the boilers so that one of either kind may be disconnected for inspection or repair. This pump is shown on page 316.

Injectors as Boiler Feeders.—This is an efficient method as it delivers hot water to the boiler without pre-heating; that is, no feed water heater is required to do the heating. As a boiler feeder the efficiency of an injector is 100% less the very small

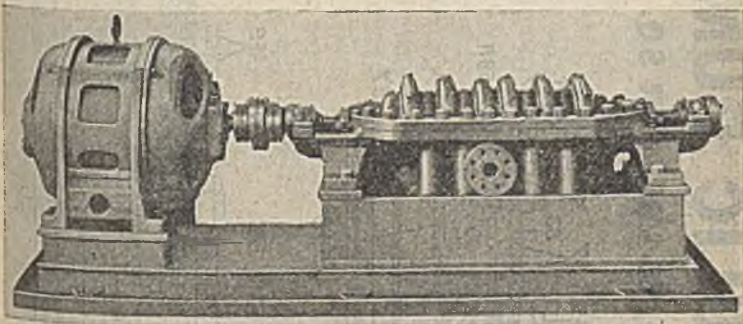


Fig. 24.—Six stage centrifugal boiler feed pump direct coupled to electric motor.

loss by radiation. The loss of work in the injector due to friction, reappears as heat which is carried into the boiler.

Ques. Is the injector the most economical means of feeding water to the boiler?

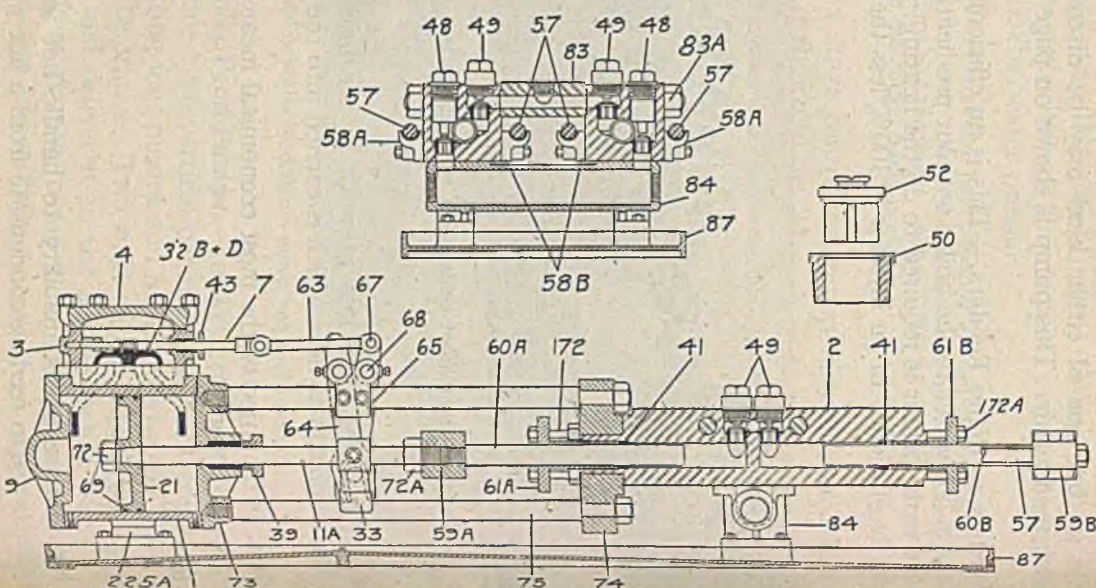
Ans. No.

Ques. Why?

Ans. Because of its inability to handle hot water which excludes its use in feeding condensate from a hot well.

HYDRAULIC PUMP

FOR PRESSURE UP TO 10,000 LBS. PER SQ. IN. OR MORE



Figs. 25 and 26.—Duplex outside end packed plunger hydraulic pump, sectional view and list of parts.

LIST OF PARTS

FOR HYDRAULIC PUMP—PAGE 796

1	Steam Cylinder	60A	Plunger Inner
2	Pump Cylinder	60B	Plunger Outer
3	Steam Chest	61A	Plunger Gland, Inner
4	Steam Chest Cover	61B	Plunger Gland, Outer
7	Valve Rod	63	Link (Long or Short)
9	Steam Cylinder Head	64	Lever
11A	Steam Piston Rod	65	Stand
21	Steam Piston	67	Link Stud
32	Steam Valves "B" & "D"—Note	68	Lever Stud
33	Crosshead	69	Steam Piston Ring
39	Piston Rod Gland	72	Piston Rod Nut
41	Pump Cylinder Throat Bush	72A	Piston Rod Nut Plunger End
43	Valve Rod Gland	73	Steam Cradle Head
48	Valve Block Plug, Suction	74	Pump Cradle Head
49	Valve Block Plug, Discharge	75	Cradle Bar
50	Valve Seat	83	Discharge Connection
52	Valve	83A	Discharge Connection Bolt
57	Side Rod	84	Suction Connection
58A	Side Rod Bearing Outer	87	Drain Plate
58B	Side Rod Bearing Inner	172	Pump Cradle Block Stud
59A	Plunger Head Inner	172A	Gland Stud Outer
59B	Plunger Head Outer	225A	Steam Cylinder Foot

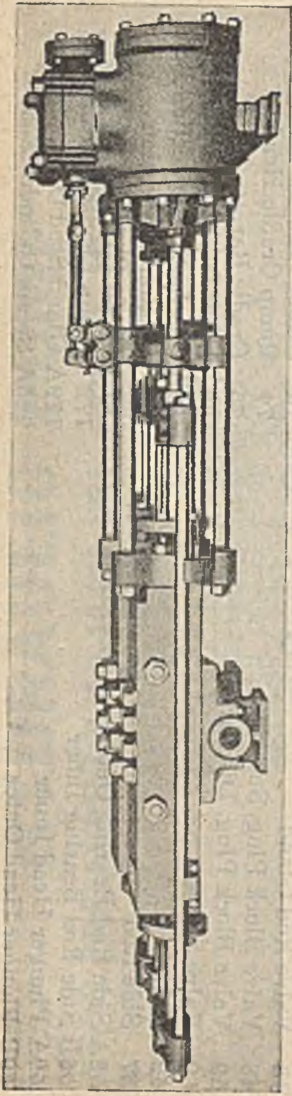


Fig. 27.—External view showing general appearance of the pump illustratedin figs. 25 and 26.

"Hydraulic" Pumps.—The word *hydraulic* used in connection with pumps means *extra high pressure*, as from say 1,000 to 5,000 lbs. per sq. in. or over. Evidently the chief characteristics of a pump of this type are:

1. Very substantial construction at the water end to withstand the excessive pressures.

2. Very large steam cylinder diameter in proportion to diameter of the water cylinder.

3. Heavy duty stuffing boxes at the water end.

Hydraulic pumps are made in various patterns, including horizontal, simplex and duplex designs.

A typical duplex design is shown in figs. 25 and 26, with accompanying list of parts. The external appearance of this pump is shown in fig. 27.

A feature characteristic of most hydraulic pumps is that the pump cylinders are solid forged steel blocks, all passages being bored out.

Note the small size of the water valves which suggest that the water capacity is very small.

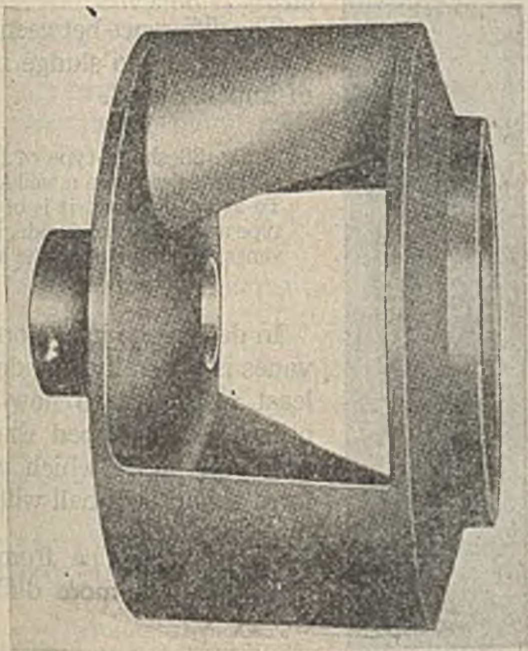
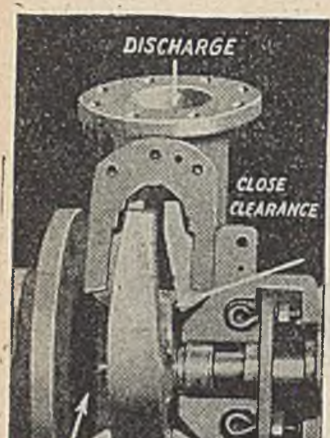


Fig. 28.—Impeller for raw sewage.

A characteristic feature is the ratio of steam, piston and water plunger diameters, The rating by manufacturers for maximum balanced water pressure is usually given at 90 lbs. steam pressure.

Sewage Pumps.—For this service there are two classes of centrifugal pump adapted to pump: 1. Raw sewage, or 2. Sludge (treated sewage).



By definition, sludge is the solid *precipitant which remains after raw sewage has been chemically or bacterially treated.*

The difference between pumps for raw sewage and sludge is the type of impeller used.

Fig. 28 shows type of impeller used for raw sewages on a well known pump. To avoid clogging, it is of the enclosed type and made extra wide, having 2 to 4 vanes depending upon the size of pump.

In design the inlet portion of the vanes is generally rounded to offer least resistance to flow and it is otherwise so shaped that strings, rags and paper which are apt to form a wad or a ball will not clog.

Handling sludge from a treatment plant is more difficult than

Fig. 29.—Type impeller with large inlet throat used for handling of high consistencies and uniform capacities. A quill completely shrouds the shaft and prevents the wrapping of strings and rags about it. The stock can only flow directly to the eye of the impeller where then, and not only then, it is subjected to the twisting motion of the shaft.

handling raw sewage because of the presence of larger quantities of various solids.

A typical sludge pump has in addition to the proper type impeller, a double threaded screw in the inlet connection to force the sludge into the impeller; each thread of the screw connects to a vane of the impeller.

Ques. What provision is made for cutting up the stringy material so as to prevent clogging of the pump?

Ans. There are four flutes and a ring in the screw housing and four flutes in the pump casing all of which have cutting edges.

Four of the latter are in front of the impeller and two behind it.

Ques. Where does the cutting take place?

Ans. Between the edges of the screw and the cutting edges in the screw housing and between the edges of the impeller blades, and the cutting edges on the pump casing.

Ques. Describe the travel of the sludge through the pump.

Ans. Sludge enters the pump inlet and is carried forward in a continuous stream by the screw conveyor to the impeller.

Solids or stringy substances that extend beyond the edges of the screw are cut up between the stellited edges of the screw and stellited flutes of the screw housing.

Magma Pumps.—By definition, the term magma includes *any crude mixture especially of organic matter in the form of a thin paste*. It also means *a confection*, hence the name given to *a pump belonging to a sugar house apparatus, designed for moving the various heavy mixtures and non-liquids occurring in the process of sugar making.*

The distinguishing feature of these pumps is the absence of inlet valves. The liquid handled must flow by gravity to the unit under negative lift which means head and these for the function of the inlet valves can be performed by the piston.

One pattern of magma pump is shown in fig. 30. The big opening in the liquid end is the inlet. A sectional view and list of parts of another make of magma pump is shown in figs. 31 to 36.

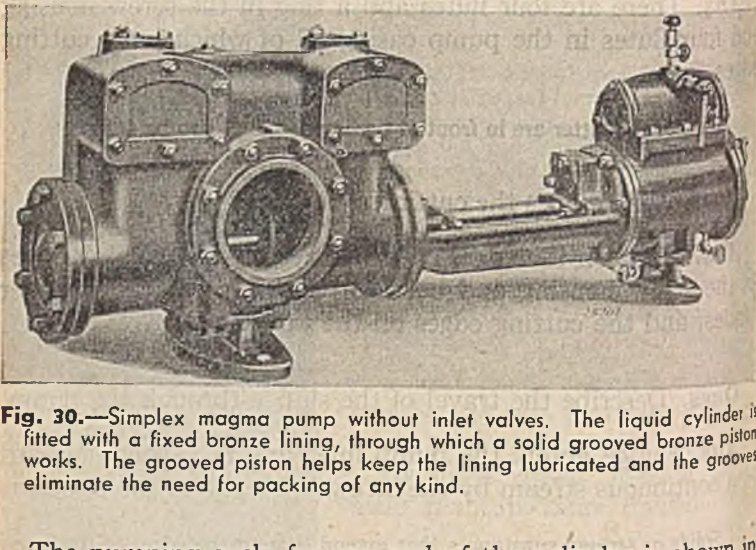


Fig. 30.—Simplex magma pump without inlet valves. The liquid cylinder is fitted with a fixed bronze lining, through which a solid grooved bronze piston works. The grooved piston helps keep the lining lubricated and the grooves eliminate the need for packing of any kind.

The pumping cycle for one end of the cylinder is shown in figs. 37 to 40.

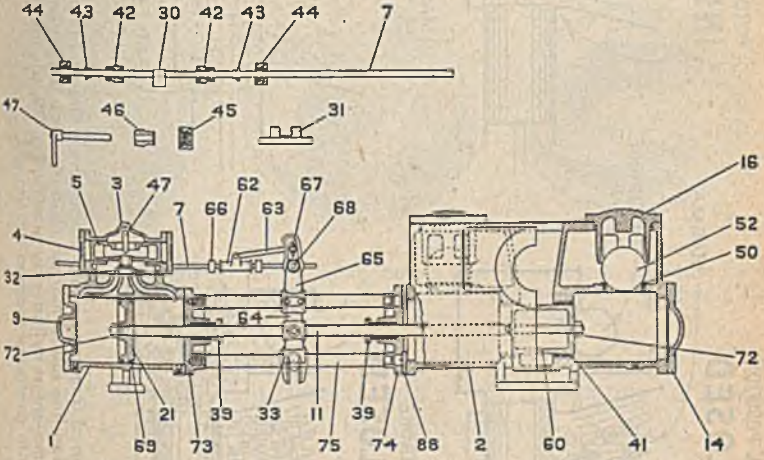
From this it is seen that the function of an inlet valve is performed by the piston similarly, as in the case of a two cycle gas engine.

Trash Pumps.—For this duty centrifugal pumps are used, designed especially for the economical handling of unscreened sewage and fluids containing large solids and much foreign matter. Adaptations are for pumping bagasse, beet pulp, tannery waste, food solids, sardines, storm water, draining under passes, low consistency paper pulp, etc.

The important feature is the shape of the impeller which is shown in figs. 41 to 43.

Fig. 41 shows general appearance of the impeller; figs. 42 and 43 are the same impeller sawed in half to show the shape of the blades which make it possible to pump the liquids just mentioned.

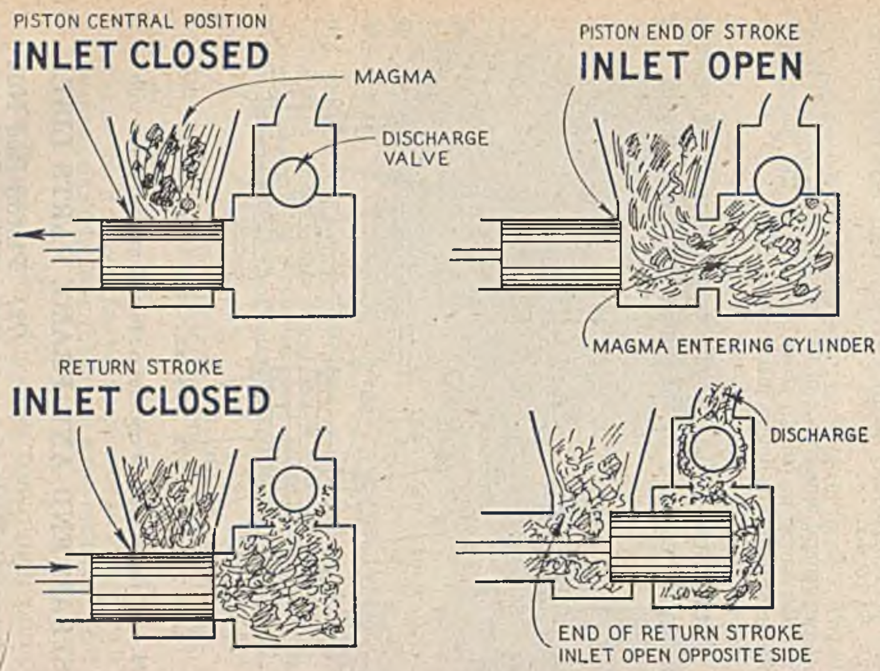
Dry Pit Installation.—Fig. 44 shows a dry pit for sewage and trash pumps designed to be placed in a dry place. The



Figs. 31 to 36.—Sectional view of another pattern of magma pump with list of parts.

STEAM END AND FRAME PARTS LIST.

2. Pump Cylinder
14. Pump Cylinder Head
16. Valve Pot Cover
41. Plunger wearing ring.
72. Piston Rod Nut
50. Water Valve Seat
52. Water Valve
60. Plunger
88. False Head



Figs. 37 to 40.—Operation of a magma pump progressively shown. In fig. 37 note plunger in central position, inlet closed, but connecting line full of magma ready to drop into cylinder; fig. 38, movement of plunger to left opens inlet and magma enters cylinder by gravity; fig. 39, plunger on return stroke, inlet closed; fig. 40, inlet stroke completed with charge of magma discharged.

driving unit as seen is located on a floor plate above the pump. The arrangement shows the usual method of connecting pump and drive unit also piping connections. With this arrangement the sewage enters the inlet reservoir at the left to which the pump inlet pipe is connected. The float switch on the motor floor is actuated by the float in the inlet reservoir.

Ques. What is the action of the float switch?

Figs. 41 to 43.—Impeller for handling trash. Fig. 41, general appearance; figs. 42 and 43, same impeller sawed in half to show more clearly shape of vanes.

Ans. It automatically starts and stops the pump to keep the liquid in the reservoir at the desired level.

In the layout shown in fig. 44 the pump is motor driven through single section flexible shaft.

Pumps for the Paper Industry.—The technical knowledge needed to ably serve the requirements of the paper industry is not acquired over night, nor by the acquisition of one or two engineers. It is rather the accumulated experience of years of contact with the actual operation of mills, the development of definite theories of engineering which have been tested and found good in the field. Centrifugal pumps are used, the important feature in design being the impeller, the shape of which to give best results is obtained by years of development rather than by any theoretical solution.

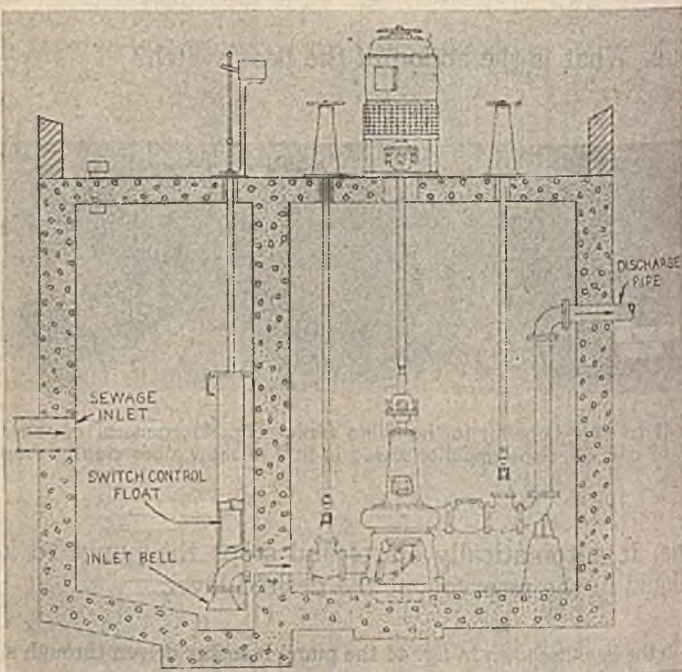
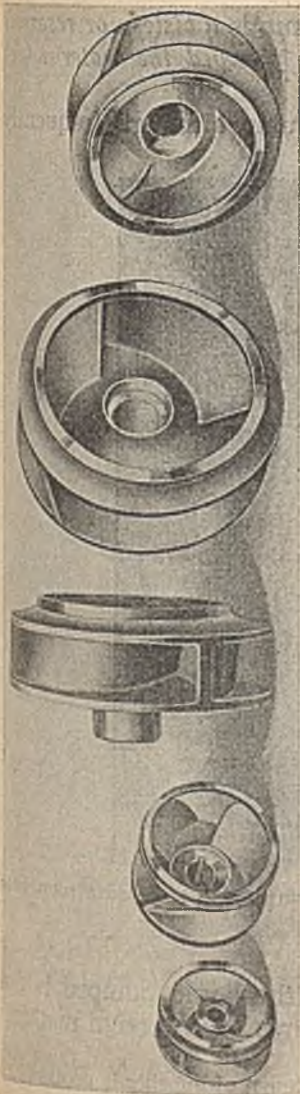


Fig. 44.—Dry pit construction with piping unit and connections installed.

Typical impellers for various paper stocks are shown in figs. 45 to 49.

The shape and number of vanes is governed by the service conditions. All are shrouded and do not require wearing plates.



Figs. 45 to 49.—Impellers for various paper stocks.

The general characteristics of these impellers are few vanes, large inlets, heavy rigid block plates and flanges, the large inlet passage and few vanes.

These features mean that stock flows through the wide impellers at low velocities.

Ques. What is the required relation between consistency and velocity?

Ans. The higher the consistency, the lower must be the velocity of the stock.

Ques. What construction is used on pumps for handling chlorinated paper stock, also concentrated hydrochloric or other corrosive liquids?

Ans. All internal parts exposed to stock are rubber covered with medium or hard rubber as may be required.

Sump Pumps.—By definition a sump is *a cistern or reservoir made at a low point from which is pumped the water which accumulates there by drainage.*

A sump pump is not a sewage pump although frequently

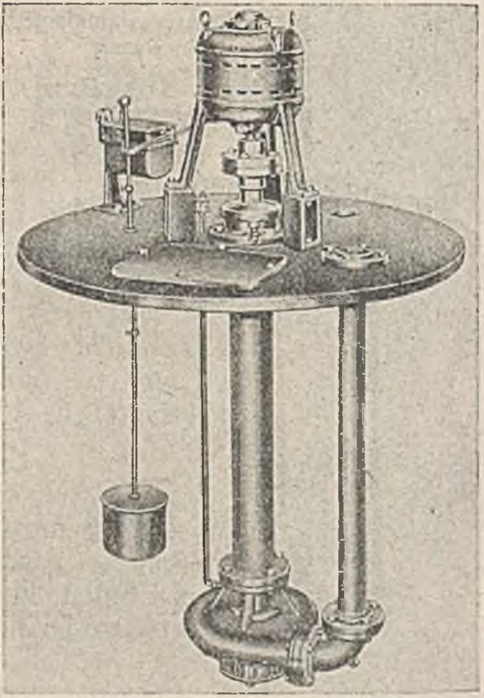
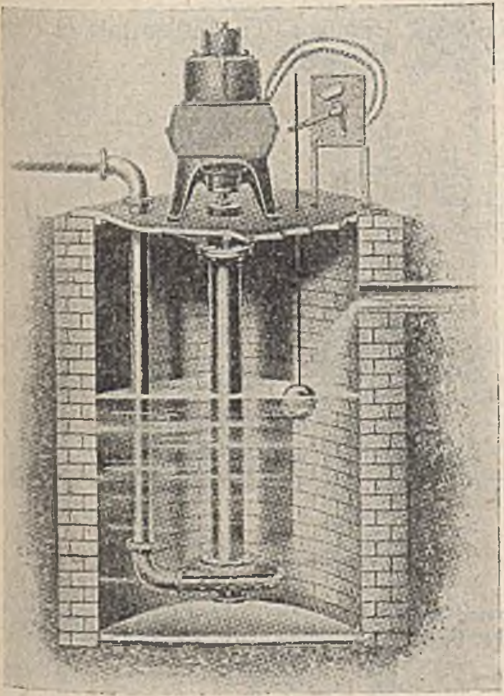


Fig. 50.—Sump pump unit showing foundation plate, pump, drive connection and automatic float control.

incorrectly so called. Accordingly the liquid pumped is not thick like sewage and is relatively free from foreign matter.

Because of this no highly specialized design of impeller is necessary.

An important service for sump pump is for draining non-water proof basements, which because of this defect become more or less flooded during each heavy rain.



The apparatus is entirely automatic in its action as the pump is submerged, thus requiring no priming.

The motor is controlled by a float and so arranged that the motor starts automatically when the liquid reaches a given height in the pit or cellar, stopping when it is empty.

Fig. 51.—Vertical electric motor driven sump pump for submerged operation with automatic float control.

One type shown in fig. 50 is arranged with the pump and discharge pipe supported from the sump cover so that all parts can be removed for inspection and repairs by lifting the cover.

Fig. 51 shows a brick wall sump with outfit arranged for disposal of accumulated storms and surface water on depressed areas such as underpasses.

In such cases frequently existing sewer systems are not low enough to permit connection to them, and when the cost of the construction of a new gravity system would be prohibited, the automatic sump here described is used.

Shallow and Deep Well Pumps.—The essentials of these installations are given in figs. 52 to 55.

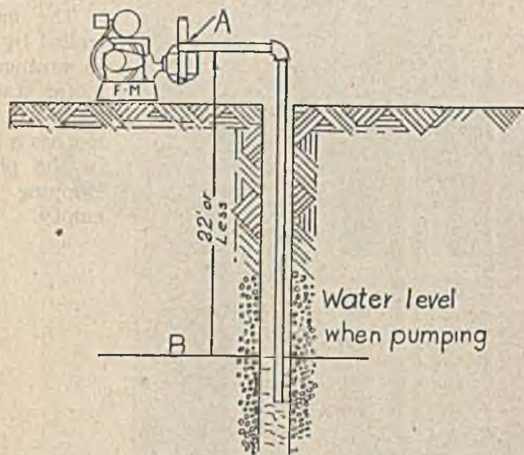


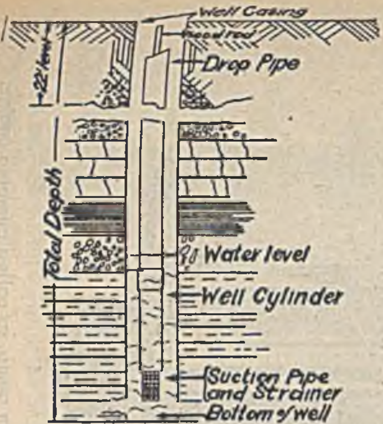
Fig. 52.—Shallow well pump installation. The working level of the water in the well must not be more than 22 feet below the pump inlet and should be preferably less.

Ques. What is a so called deep well "engine"?

Ans. A rather far fetched name for a vertical direct acting steam pump.

The general appearance of one of these units is shown in fig. 56 and construction in figs. 57 and 58.

Ques. In design name an important point to be considered.



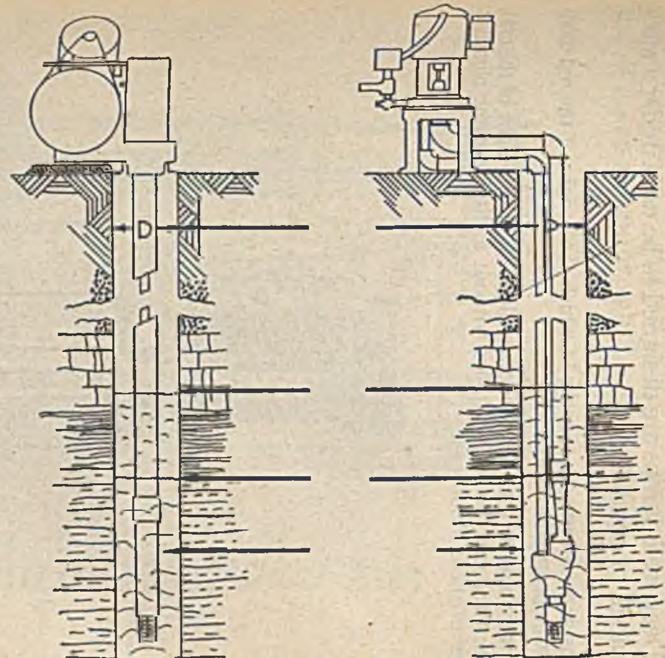


Fig. 53.—Deep well construction details. For successful deep well pump installations, the cylinder should be placed 5 ft. or more below the working level of the water and fitted with strainer on bottom of cylinder.

Figs. 54 and 55.—Deep well pump installation. Fig. 54, plunger type; fig. 55, ejector type. In fig. 54 diameter of well determines size of cylinder that could be admitted. Fig. 55, diameter of well determines size of ejector that can be admitted.

Ans. Deep well pumps are designed to be used under a widely varying range of conditions.

For instance, on very deep wells in which the weight of the rod constitutes a large proportion of the load.

They may also be used for shallow wells where the weight of the rod is negligible; they may be built with or without a displacement plunger to suit either of these conditions.

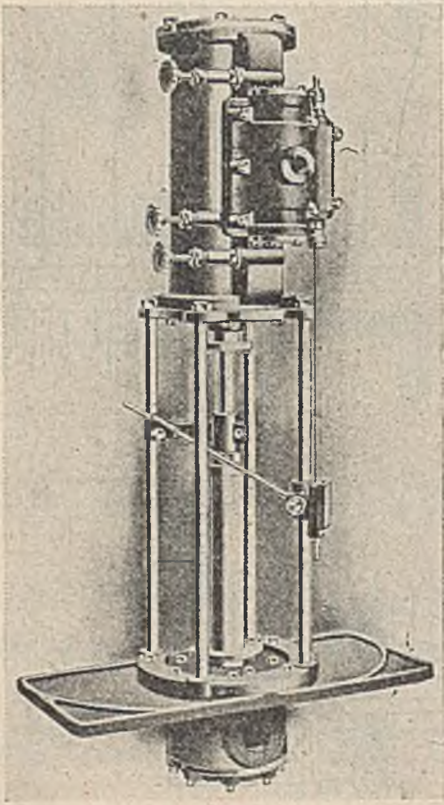
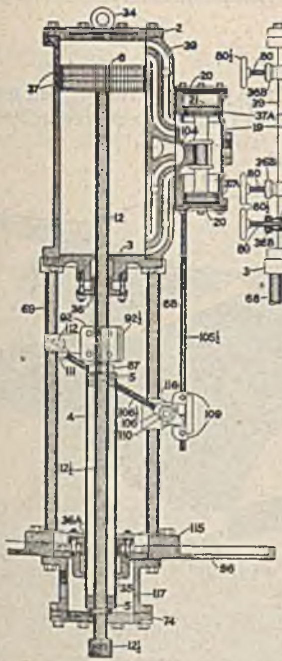
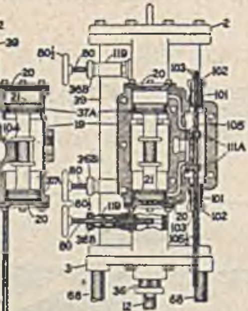


Fig. 56.—Vertical deep well steam pump, so called pumping engine.

Ques. What provision is made to accommodate the wide range of pumping rod weight?





2 Upper Steam Cylinder Head.
3 Lower Steam Cylinder Head.
4 Displacement Plunger.
5 Displacement Plunger Heads.
6 Steam Piston.
12 Steam Piston Rod.
12½ Lower Piston Rod.
19 Steam Valve Chest.
20 Steam Valve Chest Heads.
21 Steam Chest Piston.
34 Lifting Eye for Head.
35 Plunger Stuffing Box.
36 Steam Piston Rod Gland.
36A Plunger Stuffing Box Gland.
36B Gland for Regulating Valve.
37 Steam Cylinder Rings.
37A Steam Chest Rings.
39 Steam Cylinder.
68 Tie Rods.
74 Well Casing Flange.
80 Regulating Valve Stem.
80½ Hand Wheel for Regulating Valve.
87 Displacement Plunger Jam Nut.
92 Piston Rod Guide or Crosshead.
92½ Crosshead Clamp Cap.
96 Base Plate.
101 Auxiliary Valve Stem Stuffing Box.
102 Auxiliary Valve Stem Stuffing Box Nut.
103 Auxiliary Valve Stem Stuffing Box Glend.
104 Main Slide Valve.
105 Auxiliary Slide Valve.
105½ Auxiliary Valve Stem.
106 Actuating Lever.
106½ Rod Extension for Actuating Lever.
109 Right and Left Cam Blocks on Valve Stem.
110 Actuating Lever Stud.
111 Stem Collar on Piston Rod Guide to work Lever.
111A Collars on Valve Stem Rod.
112 Stud Bolt to hold Collar to Rod Guide.
115 Lower Head for Yoke Rods.
116 Actuating Lever Bracket.
117 Discharge Box.
119 Regulating Valve Sleeve.

Figs. 57 and 58.—Sectional views of the deep well pump shown in fig. 56 with accompanying list of parts.

Ans. Cushion valves are provided for the steam cylinder by means of which the steam may be throttled when entering the cylinder, and the exhaust may be throttled when leaving the cylinder, and this may be done independently on the up and down strokes.

This enables the engineer to adjust the pump so that it will make both up strokes and down strokes at uniform rate of speed regardless of conditions.

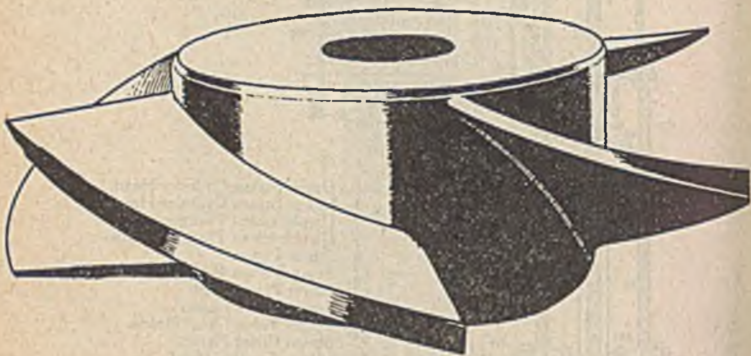


Fig. 59.—Propeller type impeller as used on irrigation pump.

Irrigation Pumps.—For this service, pumps are especially designed for large capacity and low head. Other applications are land drainage, flood control, storm water disposal, etc.

These pumps are also known as "propeller" pumps because the impeller resembles somewhat a marine propeller as may be seen in fig. 59.

Service requirements usually call for reasonably portable self-contained pumping units that can be suspended in a simple manner from a flow or structure over the water.

Long Stroke Piston Pumps.—This type of pump is suitable for rolling mills, blast furnaces, sugar or oil refineries and other situations where continuous pumping is required.

A typical long stroke piston pump is shown in fig. 60.

Ques. What are the distinguishing features of this pump?

Ans. The stroke is nearly twice as long as for ordinary service and the valve chests are placed at both ends of the water cylinder.

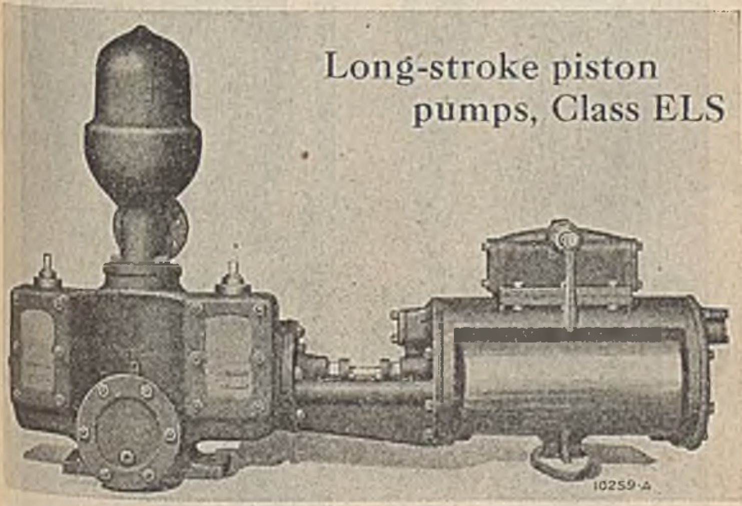


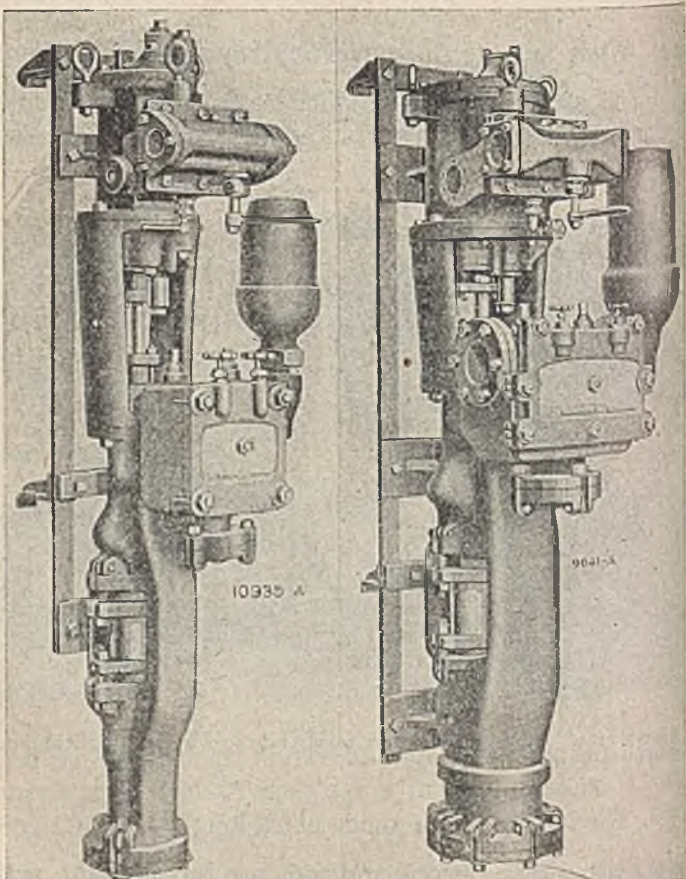
Fig. 60.—Simplex long stroke piston pump having various applications including oil refineries.

Ques. Why is the stroke made extra long?

Ans. Because the pistons and rods are brought into action only half the number of times for the same piston speed which reduces wear.

Ques. Why are the valve chests placed at the ends?

Ans. So that the water passages are short and direct.



Figs. 61 and 62.—Sinking pumps. Fig. 61, prospector's; fig. 62, regular mine shaft sinking pump.

Ques. Give some standard cylinder proportions illustrating long stroke that is diameter cylinder and length of stroke.

Ans. 10 x 5; 10 x 6; 12 x 7, etc.

Sinking Pumps.—The pumps illustrated here are direct connected double acting outside center packed plunger steam pumps. According to size they are classed as:

1. Prospector's
2. Mine shaft sinking.

They are of rugged construction and capable of handling gritty water.

Figs. 61 and 62 show general appearance of the prospector's and mine shaft sinking patterns respectively.

Ques. What is the peculiar placement of the water valves?

Ans. The admission and discharge valves are arranged in a chamber at the top of the water end of the pump.

Ques. How is the exhaust steam disposed of?

Ans. The pump can be arranged to condense the exhaust steam in a chamber connected in the inlet line.

Ques. What becomes of the condensate?

Ans. It enters the pump through the inlet and is discharged to the surface with the other water being pumped.

The water cylinders must be full of water before exhaust is turned into condenser. This is accomplished by exhausting into atmosphere until the water cylinders are primed.

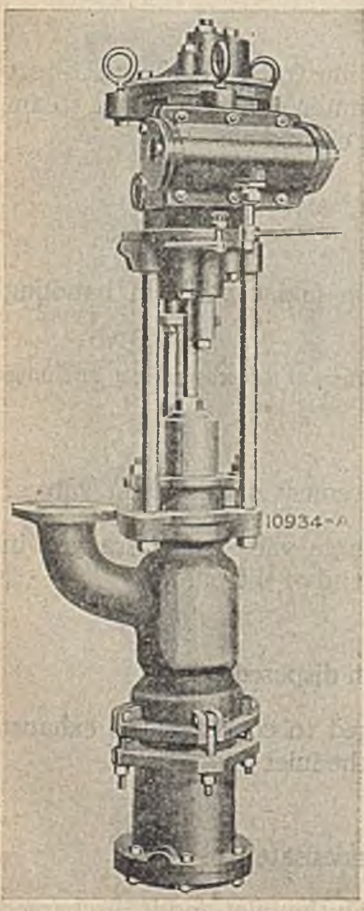


Fig. 63.—Contractor's differential plunger sinking pump adapted especially to light lift with water containing a good deal of sediment.

Contractor's Differential Sinking Pump. — This type pump is suited for situations where the lift is light and where the water contains a good deal of sediment. Its general appearance is shown in fig. 63.

Ques. Name an important feature of this pump?

Ans. Lightness.

Ques. How is lightness attained?

Ans. By discarding the valve chest and placing the valves in the lower cylinder and plunger.

Ques. What is the nature of the flow of water?

Ans. It is uni-directional.

Ques. What is the advantage of uni-directional flow?

Ans. It reduces frictional resistance and prevents the accumulation of sediment over the valves.

Fire Pumps.—For this service the construction must be substantial and according to the regulations of the National Board of Fire Underwriters: They must be tested and approved by both the Underwriter's Laboratories, Chicago, Ills., and the inspection department of the Associated Factory Fire Insurance Companies.

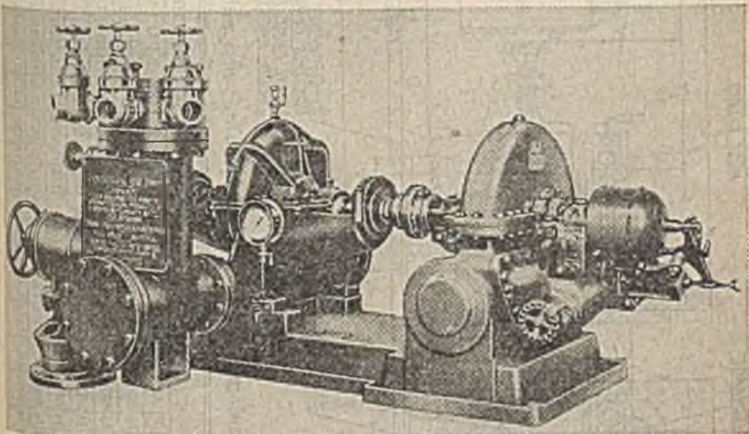


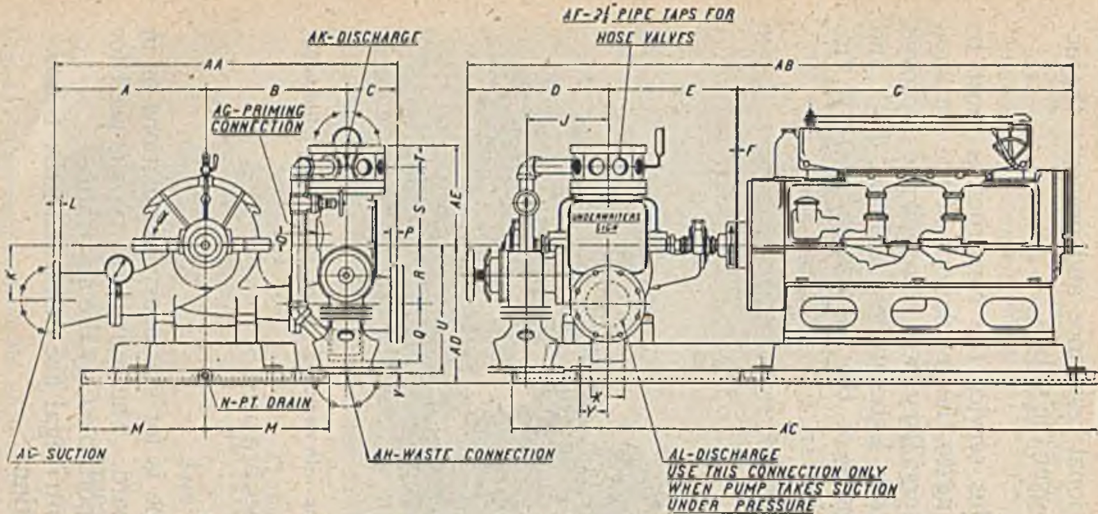
Fig. 64.—Centrifugal fire pump with complete fittings, but without relief valve.

Two fundamental types are used:

1. Reciprocating
2. Centrifugal

The reciprocating type is for stationary installations as in buildings and on shipboard. The centrifugal type is used for both stationary and on portable City fire apparatus.

Fig. 64—Shows a centrifugal fire pump with the following features of construction:



Figs. 65 and 66.—Dimensional drawings of gasoline engine driven fire pump, with accompanying dimension table.

Size Pump	GPM at 100 lb. Pressure	RPM	ALL DIMENSIONS ARE IN INCHES																											
			A	B	C	D	E	F	G	J	K	L	M	N	O	P	Q	R	S	T	U	V	X	Y	AA	AB	AC	AD	AE	AF	AG
5" AF-7	750	1750	27	25	9	23½	22¾	½	66¼	14	9½	1⅛	22	1	2	1⅛	10	10¼	13¾	3¾	21¾	1	6	4⅝	61	112⅝	100	22	17½	3	3
5" AF-10	750	*1600	27	25	9	23½	22¾	½	68½	14	9½	1⅛	22	1	2	1⅛	10	10¼	13¾	3¾	21¾	1	6	4⅝	61	114¾	104	24	17½	3	3
5" AF-10	1000	1750	27	25	9	25	22¾	½	66¼	14½	9½	1⅛	22	1	2	1⅛	10	10¼	13¾	3¾	22¼	1	6	4⅝	61	114½	100	22	17½	4	3½
5" AF-10	1000	*1600	27	25	9	25	22¾	½	68½	14½	9½	1⅛	22	1	2	1⅛	10	10¼	13¾	3¾	22¼	1	6	4⅝	61	116¾	104	24	17½	4	3½
6" AF-15	1500	1750	30	32	12	28	24¼	¾	68½	17	11½	1 5/16	20½	1	4½	1⅛	10	12	15¾	3¾	22¼	1¼	8	9	74	121¼	111	27	19¾	6	4
6" AF-15	1500	*1500	30	32	12	28	24¼	¾	70¼	17	11½	1 5/16	22½	1	4½	1⅛	10	12	15¾	3¾	22¼	1¼	8	4	74	123	115	31	19¾	6	4

*Maximum Speed permissible under the specifications of the Inspection Department of the Associated Factory Mutual Fire Insurance Companies.

1. Extra large valves and water passages to permit the complete and easy filling of the chambers when the pumps are operating at maximum speed.
2. Valve gears of special design to enable the pump to operate at a high rate of speed without jar or possible derangement of the valves.

Because of these features the pumps can be operated beyond their rated capacities in emergencies, when an additional quantity of water might prove the means of preventing heavy fire loss.

Ques. For what working pressure are the reciprocating fire pumps designed?

Ans. For a constant water working pressure not exceeding 150 lbs. per sq. ins.

Ques. What is the capacity range for standard sizes?

Ans. 204 to 660 gallons per minute at rated speed.

Ques. What is the usual capacity range of standard gasoline driven centrifugal fire pumps?

Ans. From 750 to 1500 gallons per minute.

Figs. 65 and 66 with the accompanying table show two views of a gasoline driven fire pump, various dimensions given in the table.

Chemical Pumps.—All iron or iron pumps with bronze fittings should be used where possible, due to low cost. For common chemicals the following alloys should prove most economical.

Sulphuric acid.—All iron for 95 to 100% concentration. All lead has the lowest rate of corrosion for any concentration up to about 55% hot or cold and has very good resistance from 55 to 75% hot or cold.

Monel metal has slightly lower corrosion rate between the latter values. For 75 to 95% concentration hot or cold, Monel metal appears to be best suited. Acid resisting bronze and also Ni-resist can be used.

Hydrochloric acid.—Medium and soft rubber pumps are recommenced for cold hydrochloric acid. For hot concentrated acid Hastelloy B.

Caustic soda.—All iron pumps are entirely satisfactory for cold solution. For high and boiling temperatures all nickel gives the lowest rate of corrosion.

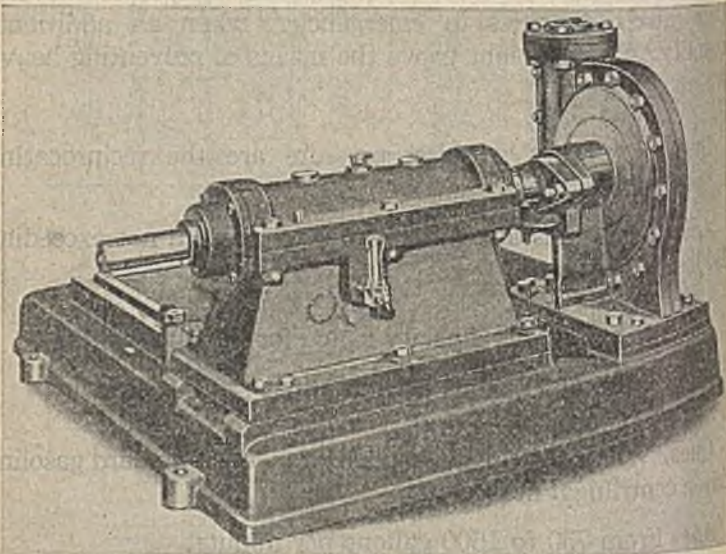


Fig. 67.—Lead chemical centrifugal pump arranged for tex rope drive.

Lead Acid Pumps.—Among the chemicals which may be handled successfully with lead pumps are hot or cold sulphuric acid of all strengths, sulphurous acid, alum, ammonium sulphate, and copper sulphate.

Designed primarily for all lead construction, these pumps are also available in all iron or special alloys, including aluminum alloy.

In construction impeller and pump casing are so designed that even when of all lead construction, relatively high impeller speeds and casing

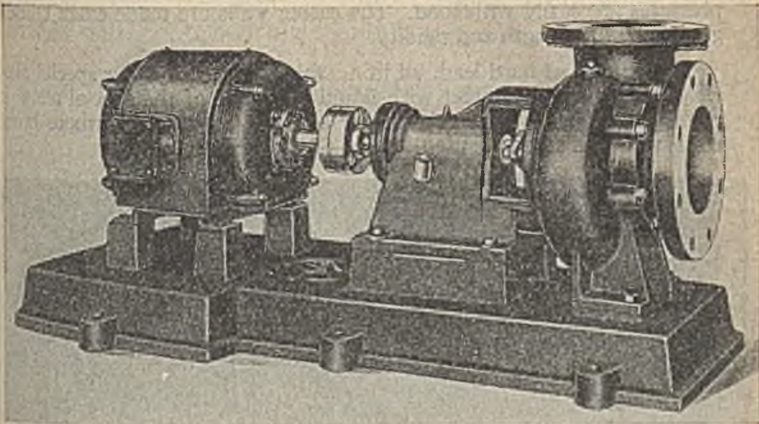


Fig. 68.—Stainless steel chemical centrifugal pump with solid bearing stand.

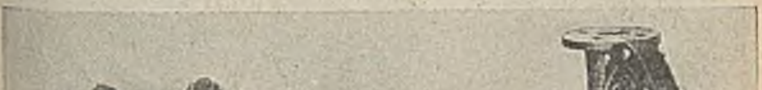
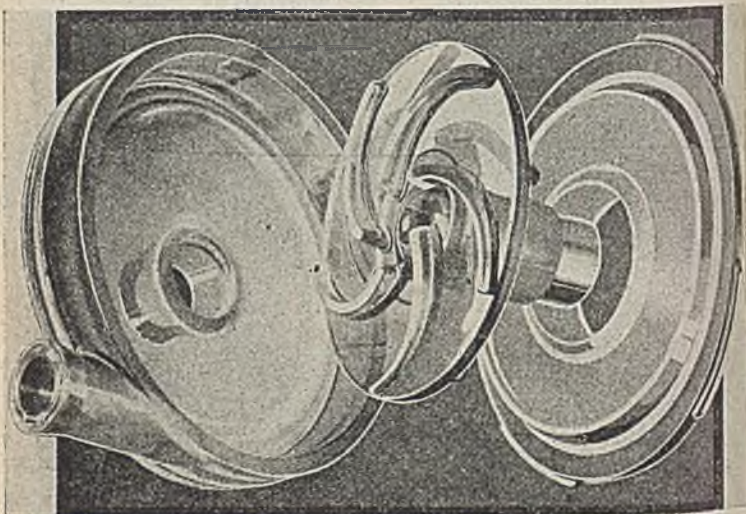


Fig. 69.—Rubber lined chemical centrifugal pump with split housing and split bearing stand.

pressures are easily withstood. The casing walls are made extra thick to give required strength and rigidity.

When made of hard lead, all iron, stainless steel or other special alloy, all parts coming in contact with liquid are made of the special alloy, including stud bolts, nuts (except for all lead construction where bronze bolts, nuts and studs are used). Fig. 67 shows a lead pump.



Figs. 70 to 72.—Glass pump impeller and casing.

Alloy Pump.—This design is suitable for construction in iron, bronze, stainless steel, nickel, Hastelloy **B** and **C**, and most other machineable alloys, but not in all lead. The particular metal or alloy depending upon the application. Fig. 68 shows a stainless steel pump.

Rubber Lined Pumps.—This construction adopts pumps for handling solutions containing abrasive solids. All internal parts exposed to liquid are rubber covered with medium or hard rubber as may be necessary for particular acid to be handled. Fig. 69 shows a rubber lined pump split housing pattern with solid bearing stand.

Glass Pumps.—The application of centrifugal pumps made of Pyrex glass are for pumping acids, milk, fruit, juices, and acid solutions without attack on the pump parts and without contaiminating the liquid being pumped by chemical acting between the liquid and pump material.

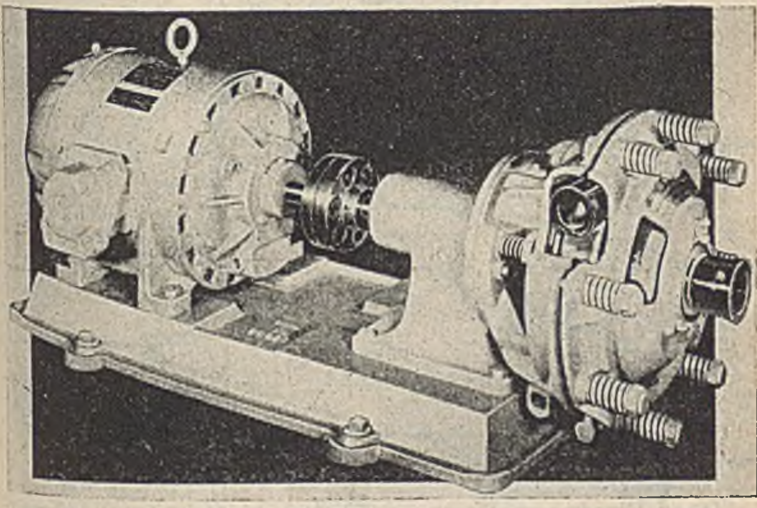


Fig. 73.—Glass centrifugal pump direct connected to electric motor.

With exceptions of hydrofluoric and glacial phosphoric glass is resistant to all acids in all strengths. Glass pumps should not be used with alkaline solutions. Figs. 70 to 72 show glass impeller and glass casing of a glass pump. The general appearance of one pattern electrically driven is shown in fig. 73.

Diaphragm Pumps.—By definition a diaphragm pump is: *One that employs a yielding substance such as rubber to perform* *a pumping operation in place of a piston or plunger.* There are two basic types:

1. Closed‘ 2. Open

For dewatering trenches, flooded foundations, coffer drains or other flooded depressions where the proportion of mud or sand to water is usually great, nothing has yet been proven as satisfactory as a properly designed diaphragm pump especially engine driven.

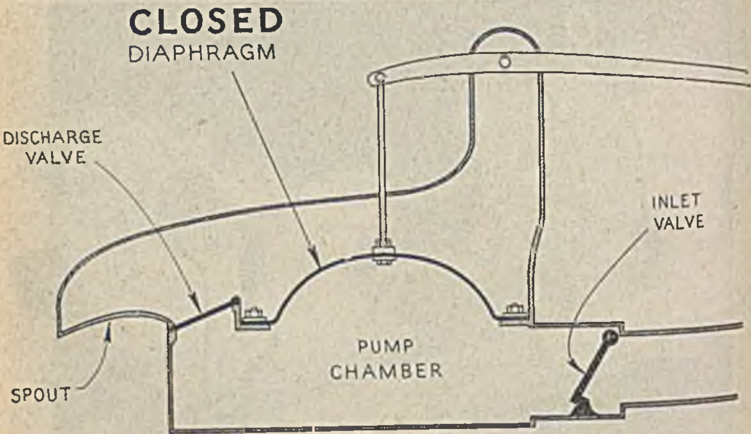
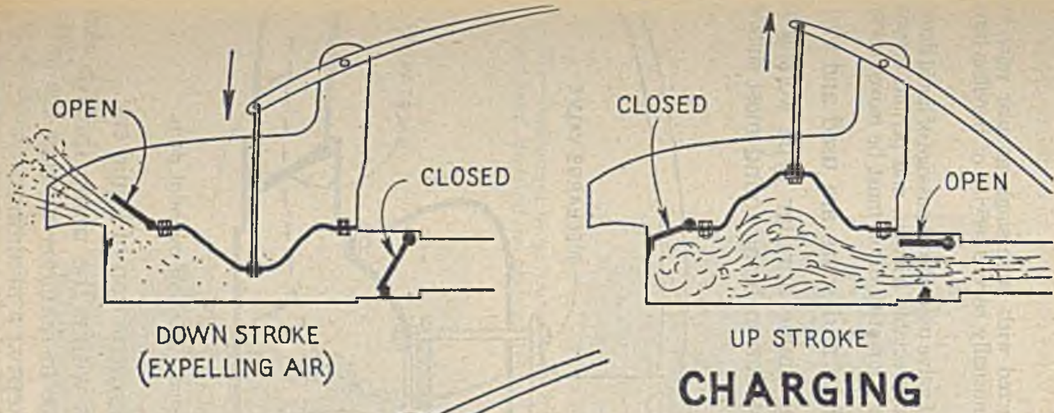


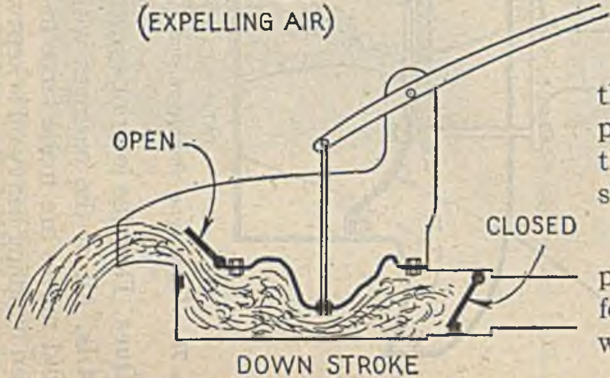
Fig. 74.—Closed type diaphragm pump showing essential parts.

Closed Diaphragm Pumps.—This type employs a diaphragm which does not carry a discharge valve, the two valves necessary for pump operation being in the base.

The essential parts are a pump chamber closed by a diaphragm and having connected to the chamber and inlet, a discharge valve as shown in fig. 74. The operation of the pump is shown in figs. 75 to 77.

Open Diaphragm Pumps.—This type of pump differs from the one just described in that there is an opening in the diaphragm which serves as a seat for the discharge valve. Thus





Figs. 75 to 77.—Operation of closed type diaphragm pump progressively shown.

the discharge valve is carried by the diaphragm instead of being an extension to the pump chamber. This difference is shown in the elementary diagram, fig. 78.

Duplex Oil Pumps.—The outside end packed pattern while used principally for handling heavy crude oil, is equally well adapted for pumping water and other liquids against pressures that range as high as 750 lbs. per sq. in.

They are especially constructed with large single annular valves and direct passages to meet the unusually severe service of handling heavy. viscous crude oil.

The single annular pattern of valve used allows passage of liquid through the seats and valves without appreciable friction, and permits operating the pumps at high speed when a quantity of oil must be moved quickly.

Ques. What kind of valves and seats are used and why?

Ans. Brass valves of the single annular type working on cast iron seats. This construction being found most suitable for handling oil.

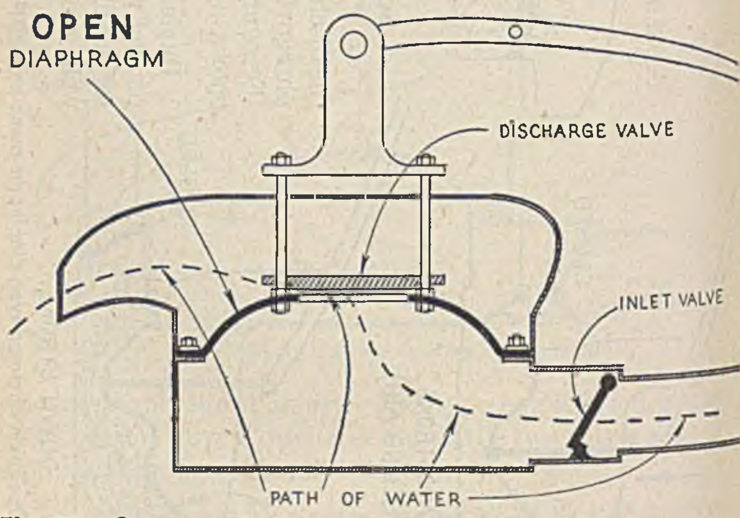


Fig. 78.—Open type diaphragm pump showing essential parts.

Ques. Describe interior construction of liquid end.

Ans. All of the passages within the pump through which liquid moves are made large in diameter so that the friction when handling heavy oil is kept at a minimum.

Frequently in oil field service the valves must be reached quickly. In the pump shown in fig. 88, each valve can be reached by removing a small plate.

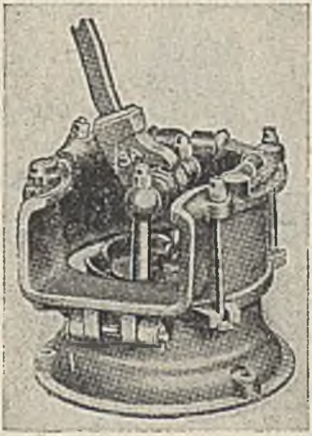


Fig. 79.—Diaphragm bilge pump with bottom inlet having such features as hinged head, beveled rim and groove, etc.

Fig. 80.—Diaphragm lift and force pump designed for yachts and small boats. It has a three way cock on inlet for washing decks aside from bilge use.

Hot Oil Power Pumps.—Competition and progress in the oil industry are forcing the refineries to install the most efficient of refining equipment. The hot oil pumps may be classified as:

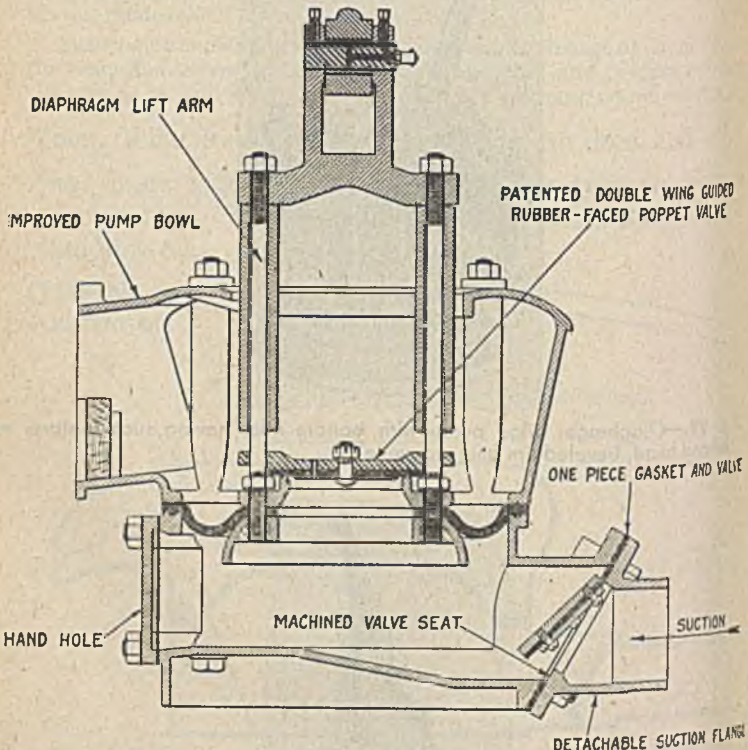


Fig. 81.—Open diaphragm pump. This pump delivers the water out of a "pitcher style" spout and is recommended for dewatering purposes where the pumped water can be carried away by gravity flow. The discharge valve is double wing outside guided and has automatic air vent. This baffle plate when properly used keeps the level of water in the bowl of the pump above the extreme lift of the double wing, outside guided discharge valve in it. This air vent prevents air locking on high lifts. If this air vent be not sealed with water the pump will either lose its prime or fail to deliver its full capacity. The baffle plate being an independent piece, can be removed from the pump bowl when it becomes necessary to clean out the bowl.

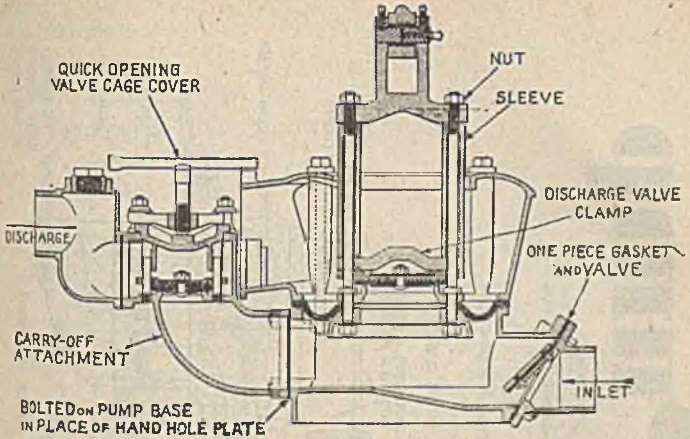


Fig. 82.—Diaphragm pump with carry off attachment.

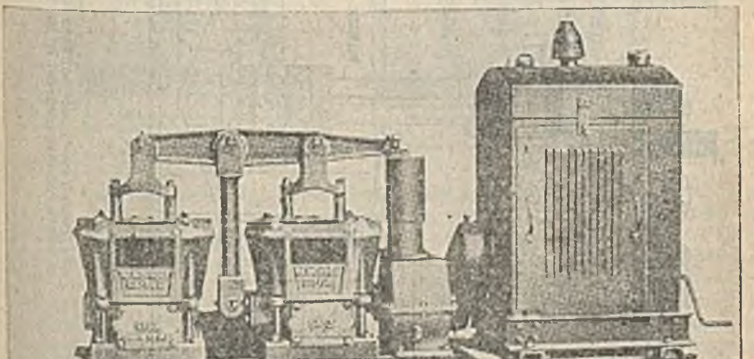


Fig. 83.—Double, walking beam type diaphragm pump with gas engine drive. The unit is mounted on a truck.

Trench Pump

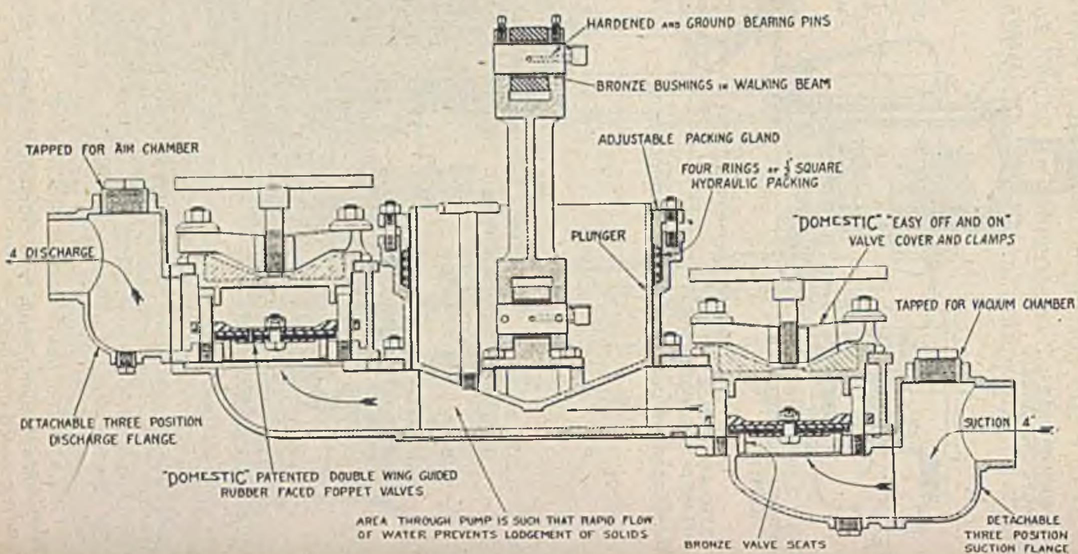
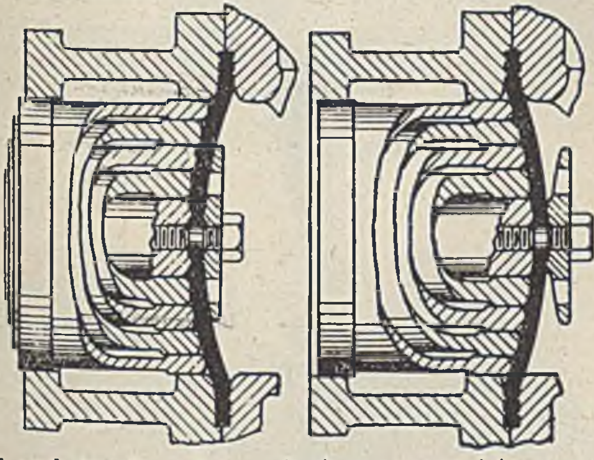


Fig. 84.—Vertical plunger trench pump.

1. Simplex
2. Duplex
3. Crank and fly wheel

} Steam driven

4. Gas engine driven power pumps.
   a. Direct connected
   b. Enclosed frame.



Figs. 85 and 86.—Special type diaphragm actuated by concentric ring pistons. There is a series of concentric rings, each mechanically actuated to produce a different length stroke. Heavy rubber (or artificial rubber) diaphragms separate the working mechanism from the two liquid ends. **In operation,** the maximum travel of the piston is in the innermost ring; the minimum in the outermost. At the end of the pressure stroke the piston rings shape the diaphragm to the form of a convex surface of a sphere. At the end of the inlet stroke the drawback disc on the face of the diaphragm to a concave surface of the same sphere. Thus the diaphragm is completely supported by the pistons at every stage of both inlet and pressure strokes.

Fig. 89 shows liquid end detail and fig. 90 sectional view. The liquid cylinders are made of open hearth steel forgings, machined from the solid.

Cooling water for the two plungers on each side enters the inboard end through flexible hose and the lower piping and is conducted to the inside of each plunger.

After the cooling water has reached the inner end (the hottest section of the plunger) it is piped back to the crosshead, through an internal pipe in the hollow plunger and discharged through upper piping and flexible hose into an open funnel on the outboard end, where the temperature can be determined.

Cooling water enters each stuffing box jacket on its bottom side, near the packing end and is discharged through the top of the jacket near the cylinder block.

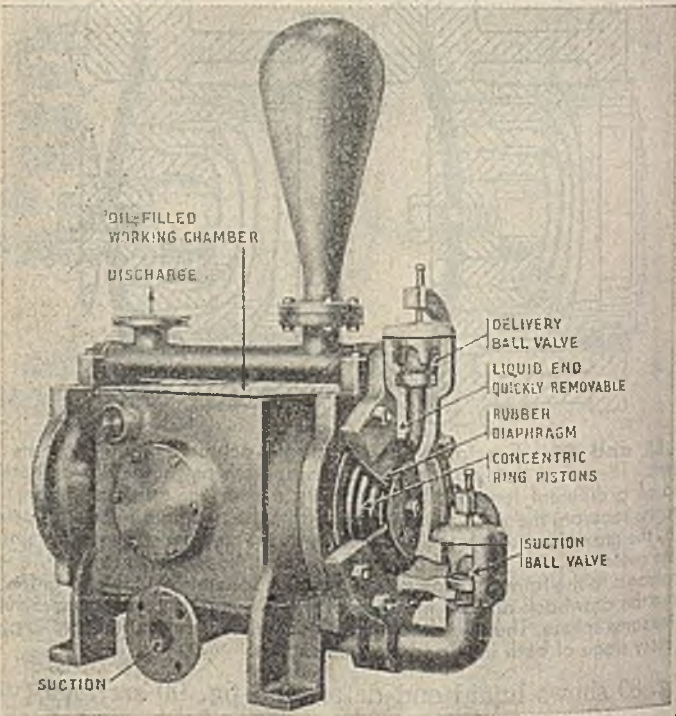


Fig. 87.—Combination diaphragm and piston pump. **In construction,** a ball inlet valve and a ball discharge valve are located in each head or liquid end of the pump. The liquid ends including the ball valves can be easily removed as a unit for inspection and cleaning. **In operation** only the liquid ends, the discharge manifold and the air chamber come in contact with the material pumped. This type pump is made in capacities from ¾ to 90 g.p.m. At pressures up to 100 lbs. per sq. in. and will operate on lifts up to 18 ft.

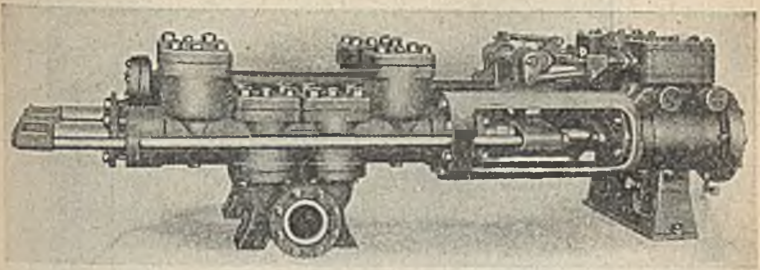


Fig. 88.—Duplex outside end packed plunger oil pump for pressures up to 750 lbs.

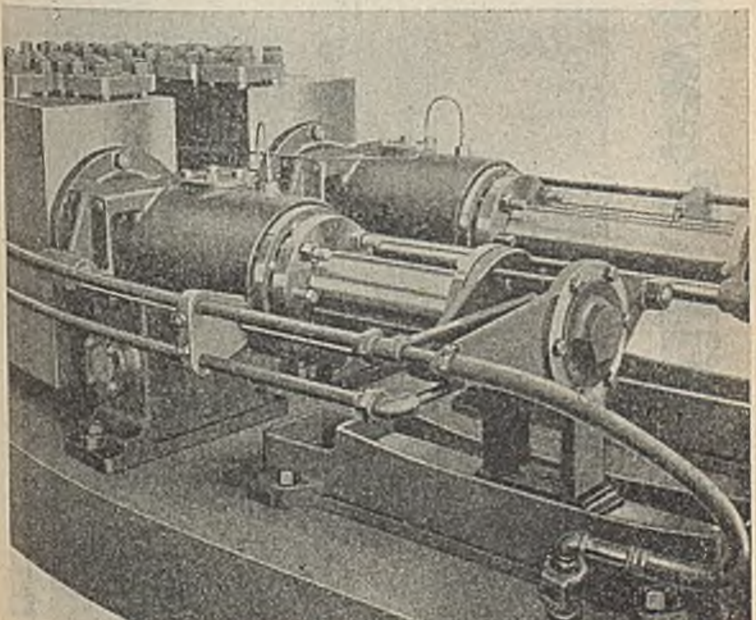


Fig. 89.—Liquid end of gas engine driven hot oil pump.

Hot Oil Power Pump

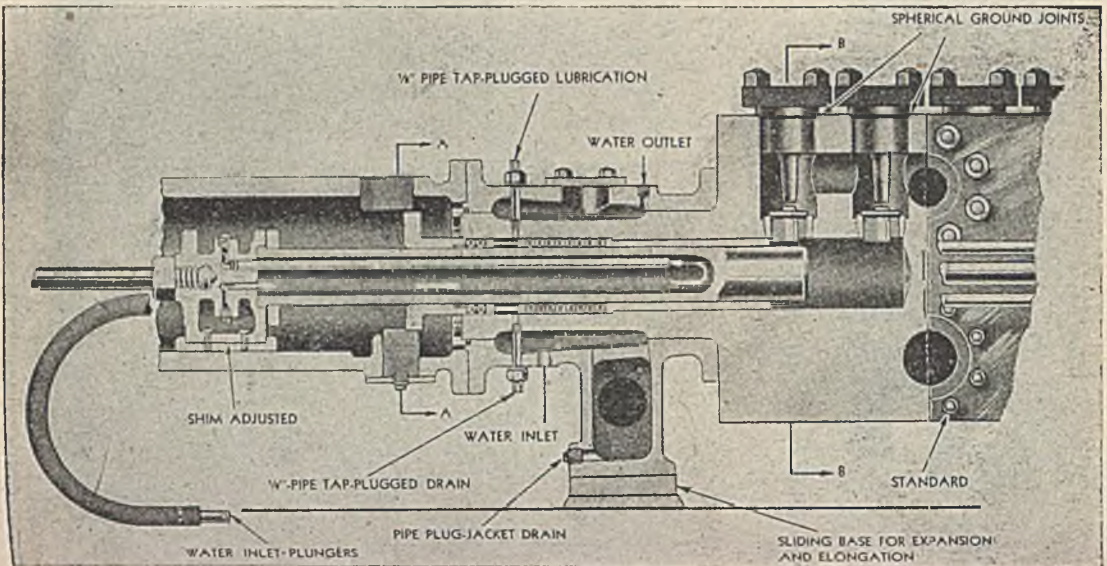
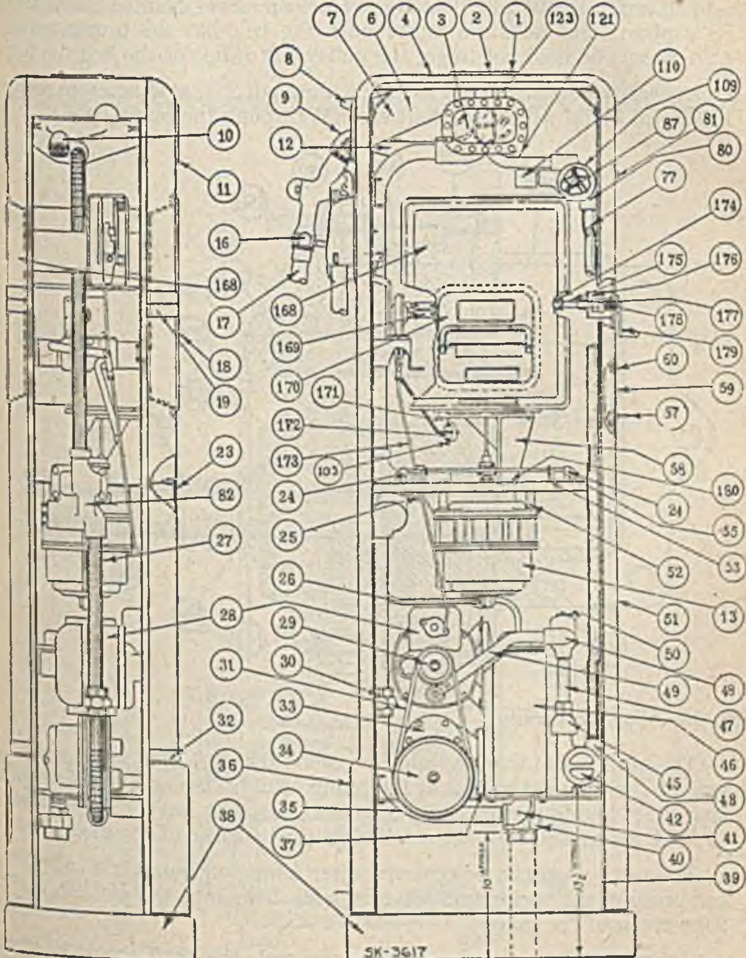


Fig. 90.—Assembly of power hot oil pump.

Gasoline Measuring and Recording Pump.—A typical pump of this type is shown in figs. 91 to 93. It is driven by a V belt 31,



Figs. 91 and 92.—Views of gasoline measuring and recording pump.

and is directly connected to the air release in which is located a strainer screen 79, and by-pass valve 78, fig. 94.

To tighten the belt 31, loosen the four cap screws holding the motor to its support, and raise the motor until the belt has the proper tension. Care should be used not to get the pulley out of line or the belt too tight.

A special rotary seal is used on the pump shaft 203, as in fig. 93 to prevent leaking of liquid where the shaft extends through the pump head.

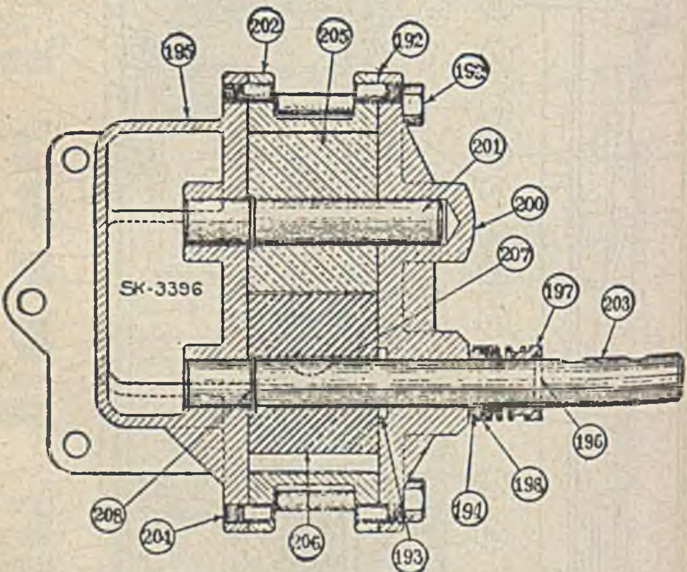


Fig. 93.—Pump assembly.

This rotary seal is made up of a carbon disc 194, and a seal ring held against the polished surface of the pump head by the spring and retaining ring 197. No adjustment is necessary as the spring has the proper tension to hold the carbon ring against the polished surface of the pump head.

The rotary seal can be removed after taking off pump belt and pulley, compressing the spring and retaining ring 197, until the pin 196, can be slipped out of the shaft.

Before starting to remove the rotary seal, the shaft should be wiped clean and a small amount of lubricating oil added.

Do not use any instrument between the carbon disc and polished surface that will mar the polish on the carbon disc or the pump head. Before replacing the rotary seal, the shaft should be wiped clean. If the pump do not deliver full capacity, it may be due to a leaky or restricted pipe line, foreign matter under the by-pass valve, a loose pump belt, clogged strainer, screens, etc.

By-Pass Air Release and Strainer.—The pump 33, as shown

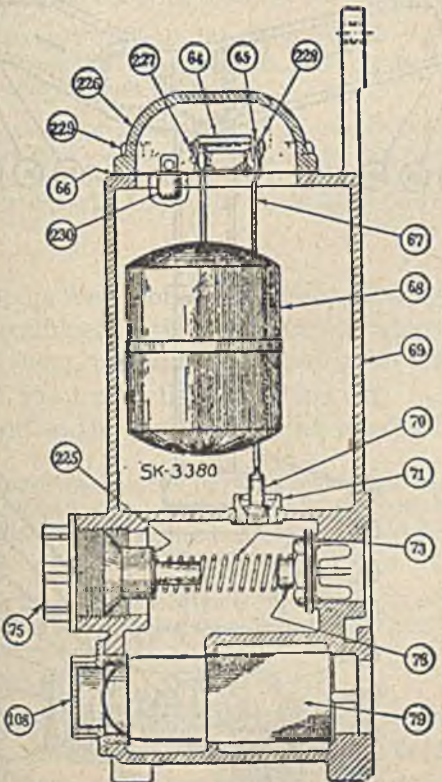


Fig. 94.—By-pass air release and strainer.

in fig. 94 is directly connected to the air release, in which is located the by-pass valve 78, and the strainer 79.

All liquid before entering the meter passes through the strainer and air release. Any air is separated from the liquid and passes out through the vent pipe 26, as shown in fig. 92.

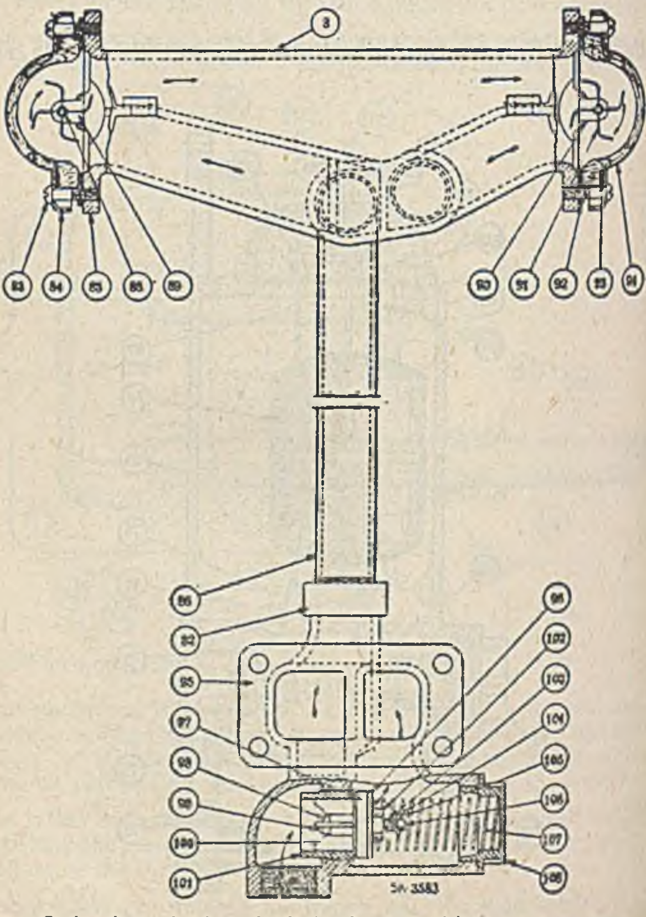


Fig. 95.—Sight glass, check and relief valve assembly.

Should liquid rise in the float chamber of the air release and raise the float 68, the valve 70 opens and the liquid is returned to the inlet side of the pump. By removing the strainer cap 108, the strainer screen can be removed for examination or cleaning. This should be done at regular intervals.

The by-pass when leaving the factory is set at 20 lbs.

This pressure can be changed by adding or removing washers 225, from between the by-pass valve cap 75, and spring 73, adding washers when more pressure is desired and removing them for less pressure.

If the by-pass valve 78 leak, the pump will not deliver full capacity. To examine, remove the by-pass cap 75, valve spring 73, and the valve 78 and clean valve seat and valve.

In replacing these parts make sure the valve spring centers on the valve cap 75 and valve 78. The strainer cap 108, and by-pass valve cap 75, must be tight to prevent air being admitted to the inlet side of the pump reducing its efficiency. In replacing these caps use shellac on the threads to make a tight joint.

Sight Discharge Assembly.—The sight discharge 94 as shown in fig. 95, is equipped with a spinner 91, which rotates when gasoline is flowing through the hose and indicates to the customer that gasoline is flowing to his car.

It is designed so the glass 94 can be removed for cleaning.

To clean remove the dials 168, fig. 92 by removing the four screws holding them in place. Remove the round head machine screws 83, fig. 95 and the glass can be lifted off and cleaned. Care should be taken not to damage the gaskets 92.

In replacing the glasses the round head machine screws should be pulled down evenly by gradually tightening them, criss-crossing around the glass until all screws are tight.

Check and Relief Valve Assembly.—The check and relief valve 100, fig. 95 functions in two ways. It holds the liquid in the sight discharge and also relieves any excess pressure due to the expansion of the liquid above the check valve.

Should the liquid drop down in the sight discharge glass 94, it indicates that one or both of the valve discs 97 or 98 are leaking. Remove the valve cap 108, fig. 95 slowly and catch the liquid that will drain out of the sight glasses and headers above the check valve.

Take out the poppet assembly and examine the check and relief valve seats 101 and the valve discs 97 and 98. Cleaning the seats and discs may overcome the trouble. If it do not, new discs should be installed.

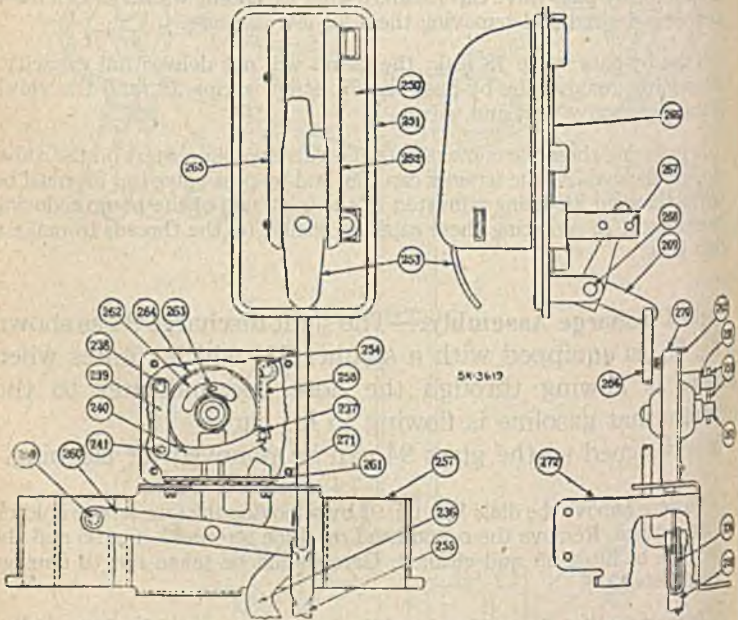


Fig. 96.—Hose hook and interlock assembly.

Light Bulbs.—No light bulbs are furnished standard with the pump. In order to insert bulbs, open the front and back housing.

To light the dial faces, use two 50 or 60 watt lamp bulbs, screwed into the lamp socket 110, fig. 92.

Electrical Wiring.—Before making any electrical connections, make sure the hose hook 253, as shown in fig. 96, is in as far as it will go shutting off the electric current to the pump.

In order that the wiring will meet the requirements of the local ordinance and the National Board of Fire Underwriters, an experienced electrician should be called to do this work.

The outfit is standard with a ⅓ *h.p.* vapor proof, 110-220 volt, single phase, 60 cycle motor.

On the pulley end of the motor is a voltage indicator. To change to 220 volts remove the screw holding voltage indicator and switch the indicator to 220 volts and replace the screw.

The wire used on the light circuit is black and white. In connecting the pump to light circuit, white wire should connect to white wire and black to black. The motor leads are tagged. Make sure current corresponds to the current specifications on tag.

Switches should be provided inside the building so that the current can be completely shut off from the pump when the station is not open for business.

The double conduit is used only when pumps are installed in series. For single pump installations, plug one outlet in conduit box 42, fig. 92.

In some localities conduit boxes are required with four outlets. Use No. 14 rubber covered wire run through ¾ in. conduit in connecting the pump to the electric circuit.

If desired to use single phase motor on a polyphase circuit, see that the voltage connection is correct and connect to any one phase of the polyphase circuit.

The lights in the pump are all controlled by the switch rod 81. To light, pull out switch button 81. The starting switch is located on the motor.

Every six months the oil cups on the motor should be filled with a good grade of medium auto oil.

Testing.—To insure efficient operation on newly installed pumps, all air in the outfit must be completely expelled by circulating 20 or more gallons through it. After this has once been done, the air release in the outfit will assure efficient, accurate operation.

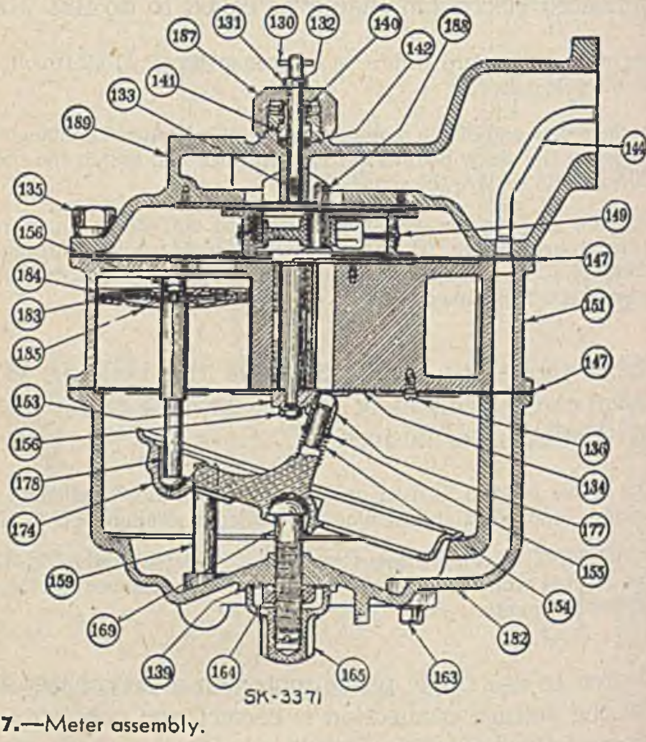


Fig. 97.—Meter assembly.

The meter is tested with liquid in measures approved by the Department of Weights and Measures before being shipped and is then sealed as correct. In ordinary use no adjustment should be necessary.

Five gallon measures that have been certified by the Department of Weights and Measures should be used in the test.

To adjust remove the seal cup 165, fig. 97. Loosen the jam nut 164, on the adjusting screw 169, and turn the adjusting screw to the left when pump is over-measuring and to the right when pump is under-measuring.

The adjusting screw should be given a part of a turn and the test repeated. One turn of the adjusting screw will change the reading approximately 69 cu. ins. in 5 gallons.

Tighten the jam nut 164, to prevent a leak at this joint. In tightening this nut, hold the adjusting screw 169, with a screw driver to keep it from turning with the jam nut. Replace the seal cap 165, and seal with wire and seal lead.

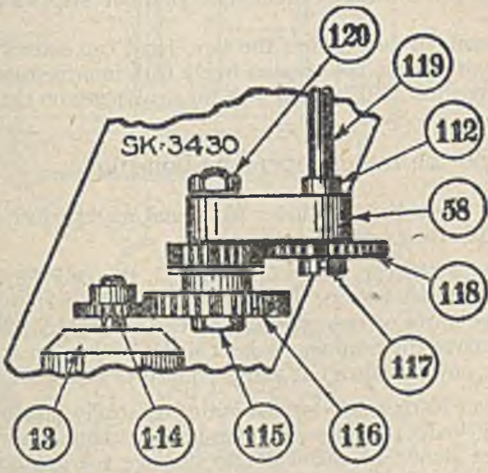


Fig. 98.—Gear bracket assembly.

Example.—Suppose when drawing ten gallons of liquid through the meter, the indicator tally nine and one-half gallons. Turn the adjusting screw 169, to the left until the quantity discharged agrees with the amount indicated by the indicator.

Care of Meter.—The drive shaft 133, fig. 97 is packed with a special packing 142, which is held in place by the gland 141, spring 140, and the meter cover nut 187. No adjustment of this packing is necessary or can be made.

To remove the packing the gear pin 115, fig. 98 should be removed and the gear 116 moved out of the way. Remove the groove pin and drive gear 114, fig. 98. Remove collar 131, and meter cover nut, fig. 97, and the old packing can be removed and replaced.

Before inserting new packing, see that the packing box is perfectly clean. Give the cork rings a light coating of oil soap, and see that they go into the packing box perfectly straight.

After tightening meter cover nut so it will not leak, re-assemble the parts and replace the printer support 58.

In re-assembling and before the hex. head cap screws are tightened. see that the drive gear 114, meshes freely with intermediate gear 116, and that the drive shaft 119, lines up with the drive shaft on the printer.

To examine the meter, open the housings.

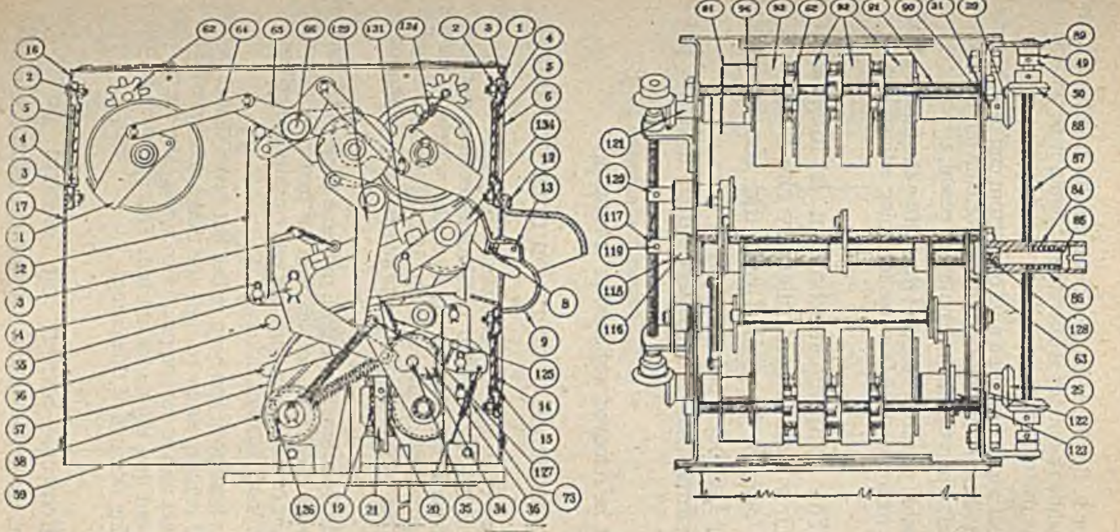
Mark the meter bowl, cylinder block and meter cover so they can be re-assembled in their original position.

By removing the cap screws 135, fig. 97, the cylinder block 151, and meter bowl 182, will become disengaged from the cover casting, therefore support these parts when removing the cap screws 135. To examine the plungers, remove the retaining plate 134, by taking out screws 136. The plungers can now be taken out and examined or replaced.

The plunger leathers are formed out of especially prepared leather and must fit the cylinder perfectly. Any new leathers should be ordered from the factory. Care should be taken not to damage the gaskets 147 and not change the adjusting screw 169 when dismantling and re-assembling the meter. If the adjusting screw be not changed the meter will not need re-testing.

When re-assembling make sure the drive arm block 177, engages the lower crank arm 153.

Hose Hook and Interlock Adjustment.—The printer, hose hook and motor are interlocked to prevent withdrawal of gasoline without issuing a ticket. If able to pull out the hose hook and start the motor without inserting a ticket and re-setting the printer to zero, make the following adjustment:



Figs. 99 and 100.—Printer assembly I.

Loosen the nut at upper end of switch rod 173, fig. 92, disconnect the rod from motor switch and turn the rod clockwise to lengthen its stroke. Make one turn and test.

If the motor cannot be started upon pulling out the hose hook after re-setting the printer to zero, shorten the stroke of the switch rod by turning the rod counter-clockwise.

The ticket door 9, fig. 99, is interlocked with the hose hook to prevent raising the door while the motor is operating. Should it be possible to raise the door while the motor is running, adjust as follows:

Remove the seal wire around the nuts on bottom end of locking rod 100. Adjust the nuts downward on the rod until motor cuts off, before the ticket door can be raised. Then tighten the nuts and reseal.

If it be impossible to raise the ticket door after the hose hook has been lowered and the motor stopped, reverse foregoing adjustment.

Operation.—The printing mechanism is interlocked with the motor switch which is controlled by the hose hook lever. The hose hook cannot be pulled out and the motor started until the printer has been set to zero.

The printer cannot be set to zero until a ticket has been inserted.

Ticket cannot be removed while motor is running.

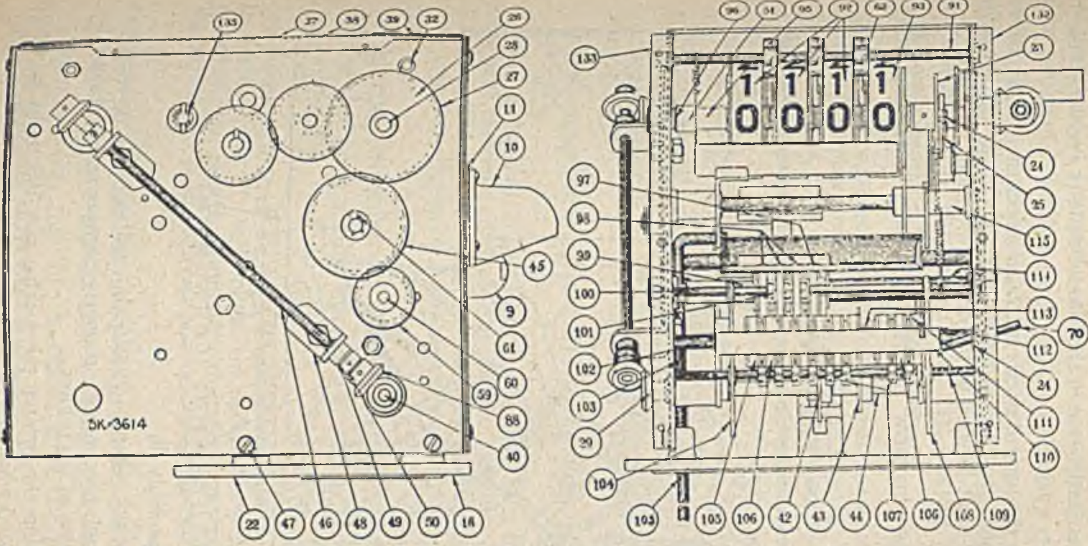
The following sequence of operation must be followed:

Raise ticket cover 9, fig. 99 and insert sales ticket then lower cover. Turn re-set crank 179, fig. 92, two revolutions counter-clockwise to the positive stop or until the shutter 51, over the wheels indicator lowers and uncovers the wheels as shown in fig. 102. This resets the gallon indicators and printing to zero, locks and prints the zero reading on the sales slip.

Remove nozzle from hose hook. Pull out hose hook which starts motor. Make delivery and close hose nozzle. Shut off motor by pushing in hose hook. Raise ticket cover 9. This automatically prints the gallons delivered on the sales slip. Remove sales ticket.

Each time the hose hook has been pulled out (even though no liquid has been discharged) it is necessary to raise the ticket door, remove the ticket, put in a new ticket and re-set the printer to zero before the hose hook can be again pulled out and the motor started. This operation is required due to the interlocking features of this pump.

Removing Printer.—Switch off electric power. Re-set printer to zero. Remove hair spring on shaft 175, and pu



Figs. 101 and 102.—Printer assembly II.

shaft out as far as it will go. Disconnect the interlock rod 103, fig. 92 by removing the seal wire and adjusting nuts, 172.

Remove the four fillister head machine screws 171, that hold the printer to the support 58. Shift the printer sideways to disengage the two drive couplings 169 and lift out of pump.

Do not change the position of the interlock shafts over which the drive couplings 169, go.

When moving the printer sideways to disconnect the printer, the lower shaft 169 will turn back about one quarter of a turn. In re-assembling printer turn this shaft forward to the original position.

A new printer can now be installed or the printer oiled or examined.

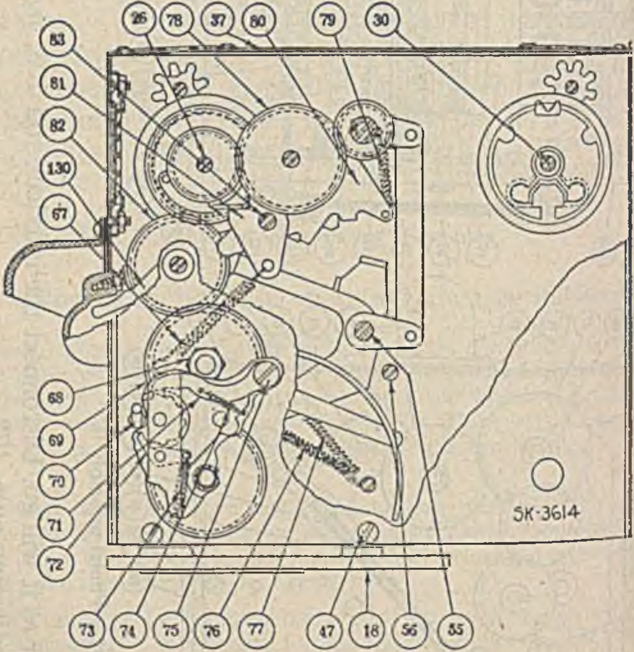


Fig. 103.—Printer assembly III.

In assembling the printer to the pump see that the drive shaft that extends up through the support, properly engages and is centered in the drive shaft coupling of the printer.

In re-assembling the printer the locking rod 103, must be adjusted. See instructions under heading Hose Hook and Interlock Adjustment.

Adjustment of Printer.—Before any adjustment can be made on the printer, it is first necessary to remove it from the pump as described in last paragraph. If when resetting, the counter wheels do not stop at zero, but continue to turn past, it would then indicate that the tension of the spring 79, fig. 103, has become weakened or the spring broken.

In the event after setting the dial numerals to zero, the sales slip is printed with numbers other than zero, it indicates that the printer is out of time. This cannot happen unless gear wheel 45, has been removed or some pin sheared off.

To re-time proceed as follows:

Using the reset crank 179 and shaft 175, fig. 92, turn the reset crank 179 until the zeros on the type wheels 97 and 98, are in alignment, facing directly upward or in the printing position, at which point stop the reset operation.

Then disengage the gear 45, which is the center gear on the outside of the left bracket.

After disengaging it from the other two gears turn the reset crank so as to complete the reset operation turning the indicator wheels Nos. 92 and 93 to zero. This synchronizes the indicator wheels with the type wheels. At the end of this operation, again engage the gear 45, with gears 27 and 59.

Should the "Number of Sales Counter" refuse to register, it may be due to a faulty spring 74, or to the eccentric screw 75 coming loose.

If the eccentric screw be loose, it should be adjusted so that on the reset the advance pawl 71, will engage the sprocket wheel, as shown in the illustration, and pull the counter to the next numeral. The screw should then be held with a screw driver and lock nut tightened.

Counters.—The gallon totalizer counter and the number of sales counter are inside the housing and can be seen only when the door 59, fig. 11, is open, and then only when uncovered by pushing down on the lever 70.

Tank Strainers on Gasoline Pipe Lines.—These are used for removing sediment and abrasive matter from the fluid which is transported.

Ques. Where are they placed?

Ans. At the inlet side of a pump station where the removal of the abrasive matter prevents it being carried into the working parts of the pumping equipment.

Ques. How are the strainers connected and why?

Ans. In parallel in order to make it possible for operators at the station to switch from one strainer to another without interrupting flow through the pipe line.

Preventing Air Pockets in Oil Gathering Lines.—Necessity for bleeding off the air or free gas pockets that might accumulate at high places in field gathering lines and cause knocking or locking has resulted in the design and use of many different kinds of traps, vents and relief valves.

Air or gas locks occurring in lines not employing adequate bleeders or relief valves can cause much trouble and even stop the flow of fluid building up back pressure sufficient to stall the pumping engine.

Ques. Where does most trouble occur?

Ans. In the line on the discharge side of the pump.

Ques. What are the conditions on the admission side of the pump?

Ans. Locks are not likely here, but even the pressure of air or gas in the line can cause knocking and vibration.

Ques. What causes air or gas in a gathering line?

Ans. It is usually the result of pumping tank down too low.

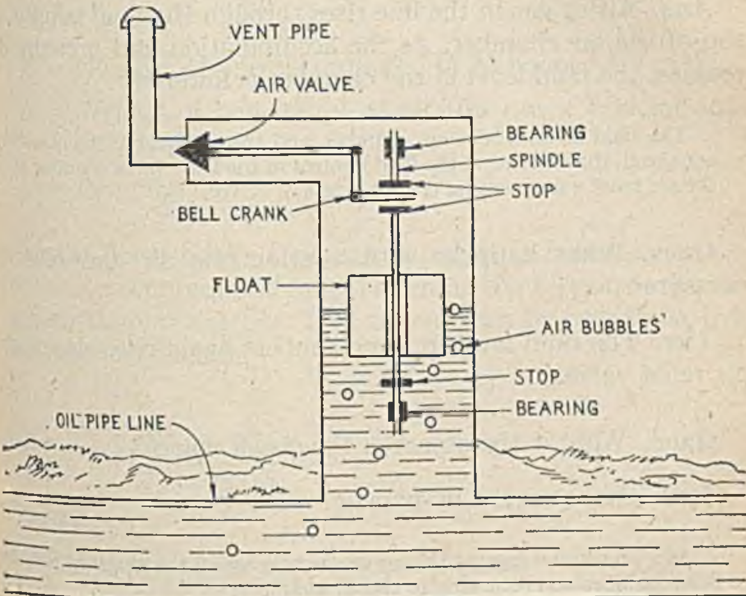


Fig. 104.—Elementary diagram of automatic air relief valve as designed by the author.

Ques. What is provided to eliminate the necessity of operators coming around to start and stop the pumps?

Ans. Automatic controls.

An effective device is an air relief valve and bleeder.

On line installations, the bottom end of an air chamber fits into the top of a short riser welded onto the top of the line.

Ques. Describe its operation.

Ans. Air or gas in the line rises through the fluid column to top of the air chamber. As the accumulation and pressure increases, the fluid level in the chamber is lowered.

The fluid levels in the air chamber and the float valve chamber being equalized, the lowering of the float control in the float valve chamber opens the air relief valve venting the air or gas.

Ques. What happens with venting the air and relief of pressure?

Ans. The fluid level in the chambers again rises closing the air relief valve.

Ques. What is the object of the check valves?

Ans. They prevent by-passing.

As a precaution against the air or gas by-passing the vent line from the float chamber and returning to the air chamber.

Through the bottom fluid equalizing line a back pressure check valve is placed in the latter line. In the same way a back pressure check valve is placed in the air line at the top to prevent by-passing of fluid from the float chamber to the top of the air chamber.

Ques. How is the device located in installing?

Ans. The air chamber is set vertically on top of the line or pump.

A simple type air relief valve is shown in fig. 104.

Approximate Horse Power Estimate.—The following formula is used for making approximate horse power calculations:

$$H.P. = \frac{B \times W \times D}{1440 \times 33{,}000}$$

in which

B = Number of barrels produced in 24 hours.

W = Weight of one barrel of oil (for quick calculation = 330 lbs.).

D = Pumping depth in feet.

To the horse power thus calculated, it is necessary to add the horse power required for overcoming the friction of the sub-surface pumping system. This amount can be calculated from the emperical formula

$$HP_F = \frac{1/8\, W_R + 2L \times N}{33{,}000}$$

in which

HP_F = Horse power to overcome friction.

W_R = Weight of rods.

L = Length of polished rod stroke in feet.

N = Number of strokes per minute.

The sum of these two values represents power requirements at the polished end. The theoretical power input at the prime mover will equal

$$H.P._M = \frac{HP + HP_F}{E}$$

$H.P._M$ = Power input at the prime mover.

$H.P.$ = Power to lift fluid.

$H.P._F$ = Power to overcome friction.

E = Over all efficiency of pumping installation usually .5 to .7.

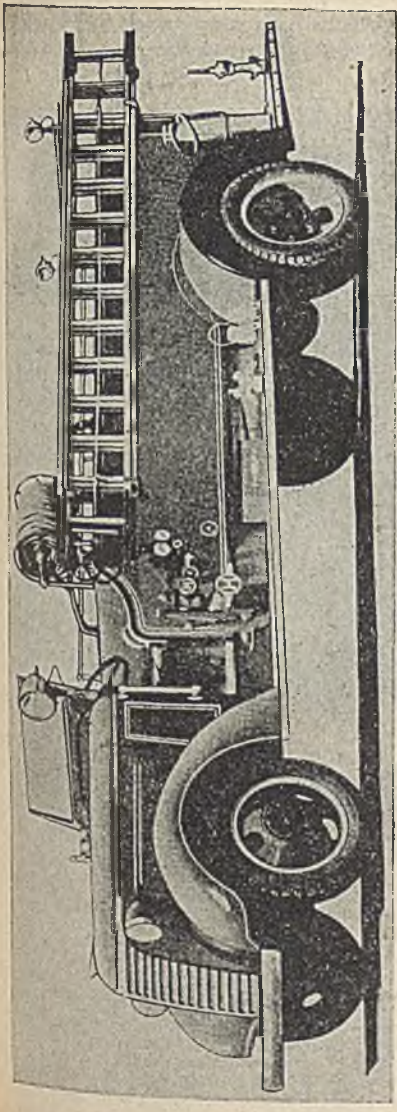


Fig. 1.—500 gallon triple combination automotive fire pumper.

CHAPTER 31

Automotive Fire Pumps

The pumping equipment of a typical fire truck comprises:

1. Two stage volute type centrifugal unit.

2. Rotary gear type primary pump, including various connections and controls.

Fig. 2 shows assembly of gauges and controls on side of the truck by the driver's seat.

The main or pressure volume pump is driven by the engine through a special pump transmission. This transmission also drives the rotary gear priming pump, which exhausts the air from the main pump when necessary.

Except for priming the instructions here given are practically identical whether operating from a hydrant or by draft which means from an open water supply involving lift.

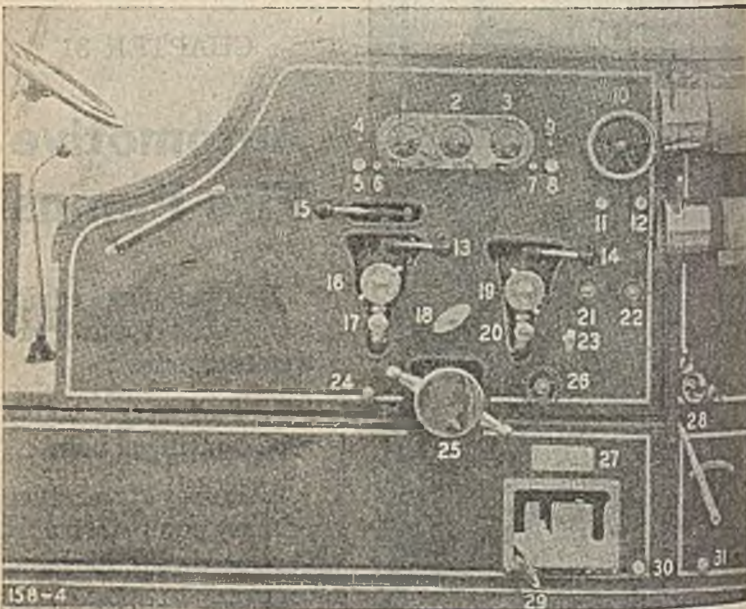


Fig. 2.—Gauges and control board.

THE PARTS ARE

1. Pump Pressure Gage.
2. Engine Tachometer.
3. Pump Compound Gage.
4. Cross-over Valve Grease Fitting.
5. Pump Pressure Gage Shut-Off Valve.
6. Underwriters' Pressure Gage Connection.
7. Underwriters' Vacuum Gage Connection.
8. Pump Compound Gage Shut-Off Valve.
9. Gage Panel Light Switch.
10. Pressure Regulator Hand Wheel.
11. Governor Valve.
12. Pressure Regulator Drain Valve.
13. Pump Discharge Gate Valve #1.
14. Pump Discharge Gate Valve #2.
15. Pump Discharge Gate Valve #3.
16. Pump Discharge Gate #1.
17. Pump Discharge Gate #1 Bleed Valve.
18. Cross-over Valve.
19. Pump Discharge Gate #2.
20. Pump Discharge Gate #2 Bleed Valve.
21. Fill Tank Valve.
22. Discharge to Hose Reel Valve.
23. Priming Valve Control.
24. Cooling Water Valve.
25. Pump Suction Cap.
26. Booster Tank to Pump Valve.
27. Underwriters' Engine Speed Plate.
28. Ground Throttle Control.
29. Pump Gear Shift Lever.
30. Hose Reel Drain Valve.
31. Booster Tank Drain Valve.

The following instructions relate to draft pumping.

Locating Apparatus.—Locate pump on firm ground as close to source of water (river, pond, cistern, etc.) to reduce lift.

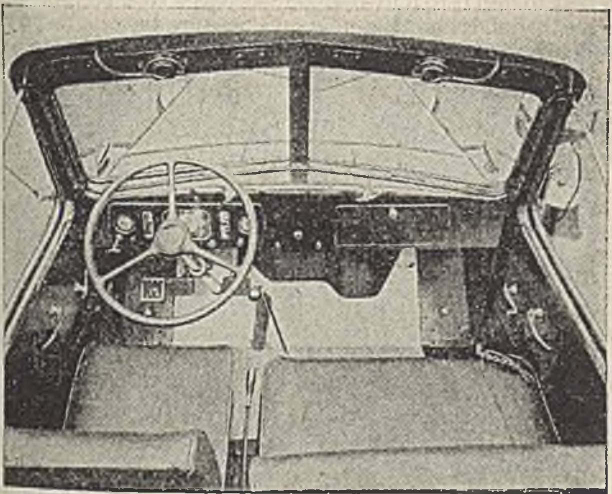


Fig. 3.—Operators seat. View showing dash and operating controls.

Before Leaving Driver's Seat.—With engine idling, shift the transverse lever on seat riser to disengage drive to rear axle.

Then shift road transmission to fourth (direct drive) gear, and latch in position.

NOTE.—The pump transmission is controlled by a horizontal lever extending through the apron at the engineer's station on the left side of the apparatus. This lever has two motions: When raised vertically, it disengages the clutch; when moved horizontally, it shifts the gears. It is important that it be raised as far as possible before any horizontal movement is made, so as to be sure of complete release of the clutch. It is also well to pause briefly after raising the lever, before moving it horizontally, to permit the gears to slow down, before shifting. If this pause be too long, however, the gears may stop, and thus prevent their meshing. In this case, the lever may be quickly lowered and raised again to set them in motion.

Connecting Intake Hose.—Attach strainer to free end of intake hose after it is connected to apparatus. Submerge strainer but keep it off the bottom, so as to be free from sand, leaves and mud..

It is best to lash it to a fixed object, such as a tree, post, or stake. Take care to see that all connections on intake hose are air tight and that rubber gaskets are in good condition. Close all openings preliminary to priming.

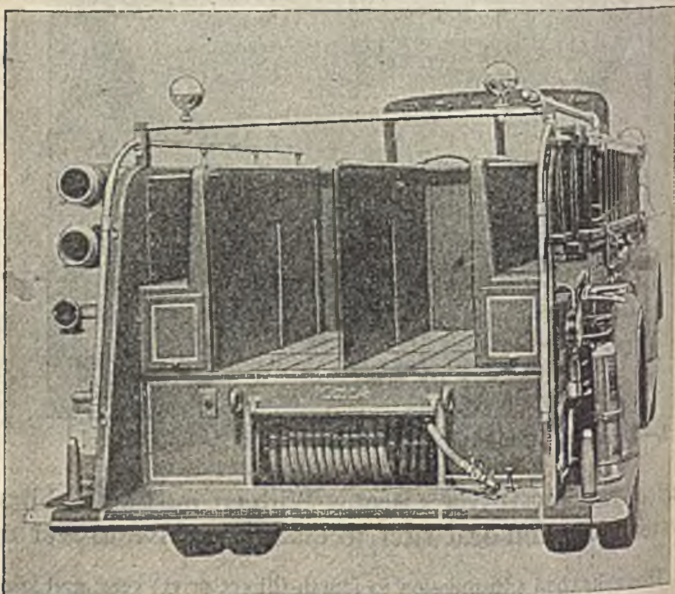


Fig. 4.—Rear view of fire truck showing general arrangement.

Priming the Pump.—With engine idling, shift the horizontal lever controlling the pump transmission, from neutral to prime position. Pull out primer valve handle and engage priming pump.

Open throttle moderately and pump should prime in about 15 seconds. Failure to prime indicates leaks. Don't speed up engine, but look for leaks.

Upon exhausting air from pump, water will be drawn by vacuum thus created into the eye of each impeller, and expelled onto the ground from the priming pump discharge under the apparatus.

When water first reaches the priming pump, it will come out mixed with air. Wait a few seconds until the discharge from the priming pump is uniform, then close priming valve, shift into main pump, open throttle until a pressure of about 20 pounds is built up.

Gauges.—There are two gauges. 1, pressure gauge and 2, combined pressure and vacuum gauge.

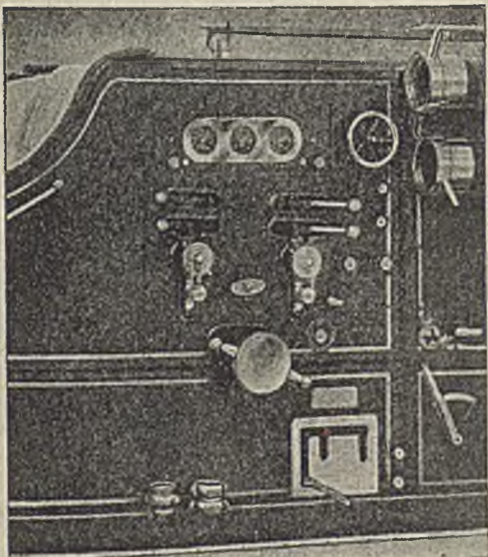


Fig. 5.—Pump controls. Compact and efficient arrangement with all gauges and levers grouped together within easy reach of the operator.

The pressure gauge registers pressure on discharge side of the pump, the other gauge (usually called "compound gauge") registers pressure or vacuum on the intake side of the pump.

Starting the Pump.—With engine slowed down to idling speed, shift the horizontal lever controlling the pump transmission, from prime to pump position.

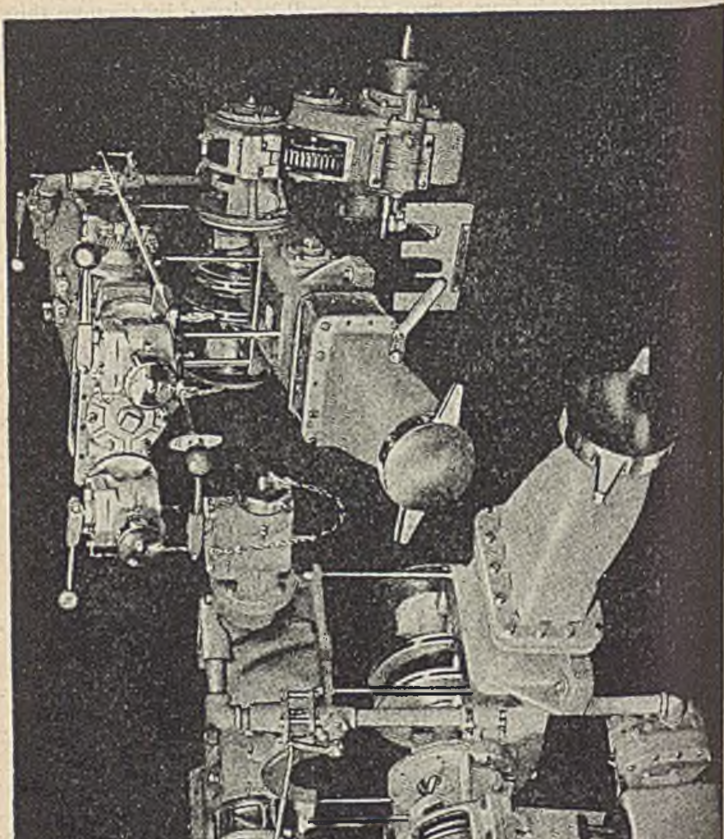


Fig. 6.—Two views of pressure volume centrifugal pumps.

See note on page 844O. Immediately upon starting, the pump should show 20 pounds pressure on the pressure gauge.

Pumping.—Should discharge pressure not rise to the required discharge pressure, while a vacuum of at least 10 inches is showing on the compound gauge, then smaller nozzle tips must be used on the lines, or the cross over valve must be changed from volume to pressure position.

Pressure Operation.—For pressure beyond what can be attained in the volume range of the pump, a change to pressure range or the use of smaller nozzle tips is necessary.

If already in the pressure range, then smaller tips or fewer lines are the only answers, for to demand more of the pump would be to ask for performance beyond its capacity.*

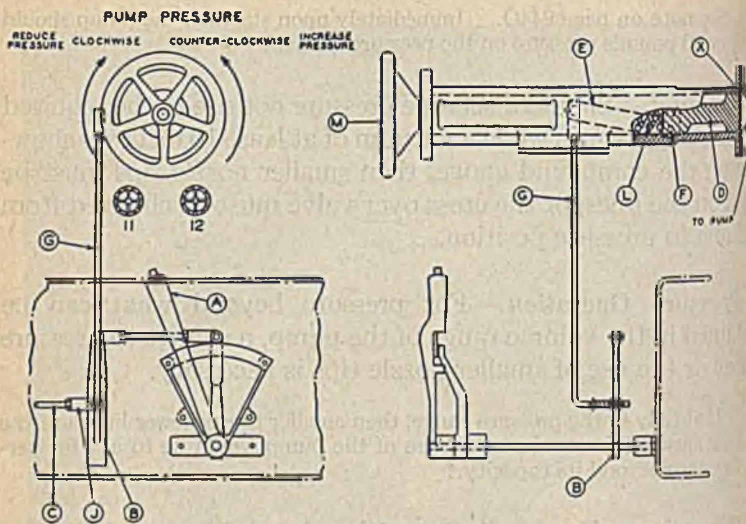
There is no intermediate position for the cross over valve. Always turn it to one extreme or the other.

Pressure Regulator.—To automatically maintain a given pressure, a pressure regulator (or pressure governor) is provided.

This device acts on the engine throttle to speed up, or slow down, the engine automatically to compensate for changes in the elevation of the nozzles or play pipes, and for occasions when one or more of them is opened or shut off. See figs. 2 and 3.

*NOTE.—For operation in the pressure range, the cross over valve is used to change the flow of water through the pump from parallel (each impeller delivering half the total volume of water at the pressure gain of one stage) to series (the discharge of one impeller feeding the intake of the other, thus delivering half the maximum volume at twice the pressure gain of one stage).

NOTE.—Changing from volume to pressure, or back from pressure to volume, is accomplished by throttling the engine until a discharge pressure of 25 pounds shows on the pressure gauge, then turning the cross over valve control to the opposite extreme. The throttle may then be opened immediately until the desired working pressure shows. The change-over should never be made at a pressure greater than 25 pounds, since this will cause the check valves to close too violently (evidenced by a sharp metallic click) and thus subject them to probable damage.



Figs. 7 and 8.—Ground throttle and pressure regulator. **Ground throttle control.** There are two controls, hand throttle lever A, and regulator hand wheel M. Fig. 7 shows both in the off position. When throttle lever A is moved toward the open position it rotates slotted lever B, which pulls back accelerator rod C, and opens engine throttle. **Pressure regulator.** Pump pressure on diaphragm X, causes piston D, to move forward rotating bell crank E, at point F, pushing up rod G, and clevis J. The slot in lever B, is inclined rearward when hand throttle is open; therefore, the downward movement of rod G, causes a forward movement of rod C, closing engine throttle. This movement continues until the piston reaches its maximum travel, ¼ in. at which point clevis J is near the end of the slot and throttle is fully closed. The pump pressure, however, is opposed by the pressure of springs L, on the other side of the piston. This spring pressure is increased by turning hand wheel M, counter clockwise, thus increasing pressure at pump.

NOTE.—*To set the regulator.* 1. With pump primed and operating, turn hand wheel M, counter-clockwise as far as possible. This sets regulator for maximu mpressure. 2. Open throttle lever A, until desired pump pressure is obtained. Watch pressure gauge. 3. Open governor valve 11. 4. Turn hand wheel M, clockwise until pressure is reduced slightly. This indicates that regulator is set for desired working pressure. 5. Next open throttle lever A, all the way, then regulator will hold desired pressure constant. Pressure can be increased or reduced as desired by hand wheel M, without touching throttle until pumper is to be shut down. 6. Once during every pumping operation, open and close momentarily the regulator drain valve 12. This cleans sediment from pressure chamber. 7. It is important to keep the regulator and all controls working freely and well lubricated.

Engine Speed.—Check by watching the tachometer. Do not run engine beyond its rated maximum speeds.

Auxiliary Cooler.—On some models, to assist in cooling the engine on long, hard stretches of pumping, an auxiliary is provided through which a small portion of the pump water is diverted for the purpose of cooling the engine cooling water.

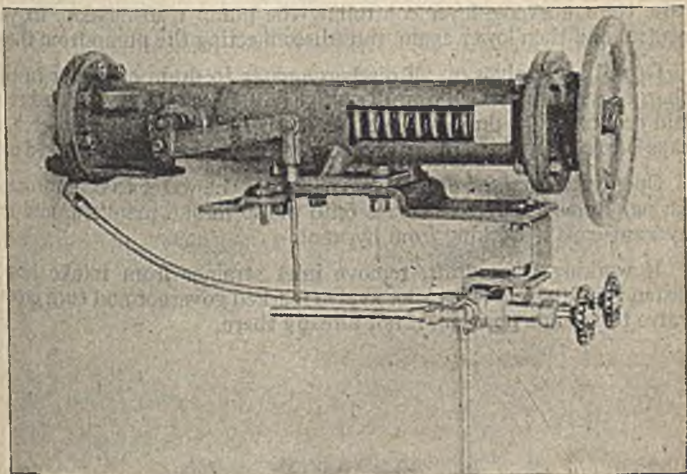


Fig. 9.—Automatic throttle control pressure regulator used in conjunction with centrifugal pump. This device automatically holds pump pressure at desired point regardless of number of hose lines in use or whether they be open or shut.

Temporary Shut Down.—When for the purpose of changing hose, or for any other reason, a shut down is desired when working either on the hydrant or from draft, it is unnecessary to stop the engine or to disconnect the pump, merely close the throttle until the pressure is reduced to about 20 pounds under ordinary conditions, and to possibly 35 pounds on high lifts, when working from draft, and then close the discharge valve.

On resuming, open discharge valve first, then throttle until desired pressure is obtained.

Shutting Down Pump.—When all pumping is completed, shutting the pump down should start with the closing of all discharge gates and closing the hydrant; or in the event of drafting, the raising of intake strainer out of water.

Then slowly close throttle allowing engine to run at idling speed, and raise the transverse lever controlling the pump transmission, move it to neutral and then lower again, thus disconnecting the pump from the drive.

Open bleed valves on all discharge gates to drain off water in the discharge header, by which time discharge hose lines may be disconnected and laid on ground to drain. As soon as discharge hose is disconnected, screw caps in place on discharge gate openings. Next close valve marked cooling.

Open valve marked drain to permit all of the water to drain from pump casing. Immediately screw caps onto pump intake manifold, and replace hydrant caps if working from hydrant.

If working from draft, remove inlet strainer from intake hose and fasten it in its place. Now close valve marked governor and turn cross over valve to volume position, if not already there.

CHAPTER 32

Dredges

In the classification of dredges the designation *hydraulic* dredge for just one type of dredge is rather questionable because every dredge that takes material through an inlet pipe (alleged "suction" pipe) and moves it by a centrifugal pump is a hydraulic dredge.

Manufacturers classify dredges in five general groups:

1. Hydraulic
   - *a*. Plain
   - *b*. Revolving cutter
   - *c*. Traveling inlet screen
2. Dipper
3. Clam shell

To this list may be added—

4. Hopper
5. Seagoing

With respect to hull construction

1. Wood
2. Metal
3. Portable (sectional)

With respect to location of operation

1. Land locked waters
2. Bays and harbors, etc.

The Hydraulic Dredge.—The so-called *hydraulic* dredge is *any dredge which employs a centrifugal dredging pump to pick up and set in motion the material to be handled.*

Ques. What is the distinguishing feature of the hydraulic dredge?

Ans. It is the only type that digs and also transports the material in a single operation to the point of use or disposal.

Ques. What is its adaptation?

Ans. It will excavate silt, sand, clay, gravel, etc., in fact, almost any material short of hard solid rock.

Ques. What is the marine working depth?

Ans. About 100 feet below the surface of the water.

Ques. How far will the dredge deliver the material?

Ans. A mile or more depending upon the size and design of the dredge.

In cases where the distance is more than a mile a booster unit is used which operates another centrifugal pump in series with the main dredging pump.

Ques. How is the material conveyed?

Ans. Through a pipe line installed along the surface of the water, supported on numerous pontoons.

Ques. Mention some advantages of the hydraulic dredge.

Ans. It is the cheapest method of dredging. A further advantage is that both its position and location of its discharge line can be varied readily, permitting very flexible operation in choosing the side of excavation and site for disposal of the excavated material.

Hydraulic Dredges with Plain Inlet.—This type is shown in figs. 1 and 2. It is the accepted standard for sand and gravel production from loose deep deposits which are free from stones and material that would clog up the inlet pipe.

Ques. What protective fitting is placed at the end of the inlet pipe?

Ans. A steel plate head with grating as shown in fig. 3.

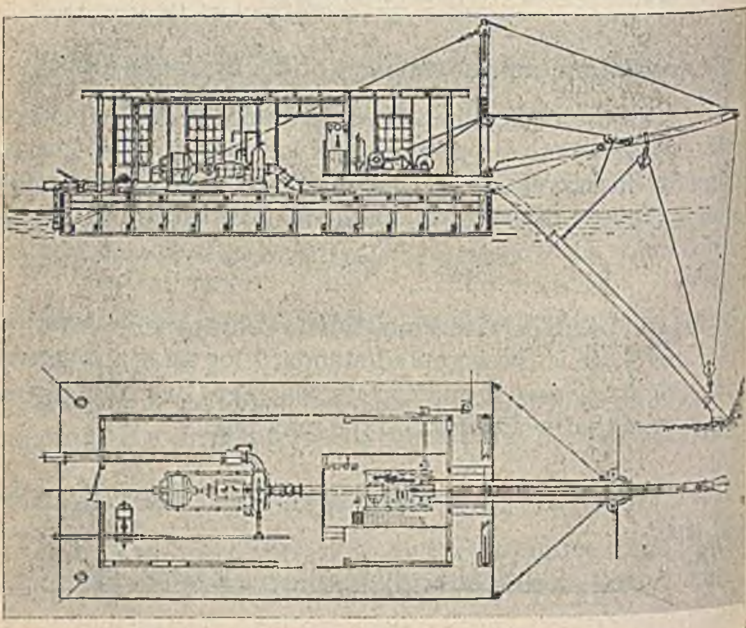
Ques. What equipment is used with the plain type dredge?

Ans. The equipment consists of the dredging pump, a hoist usually with three drums but sometimes with only one, a service pump piping and miscellaneous fittings.

Ques. How is the depth of digging controlled?

Ans. By raising and lowering the inlet pipe by means of a cable.

Ques. How is the position of the dredge in the cut controlled?



Figs. 1 and 2.—Hydraulic dredge with electrically driven dredging pump and plain intake.

Ans. By two forward swinging lines and two anchor lines aft.

Ques. Describe the operation of the dredge.

Ans. In operation, the material is eroded and carried into the inlet with the entering water, which must be in close contact with the bottom at all times to assure picking up the full quantity of solid material.

Ques. For what type of work is this kind of dredge not suited?

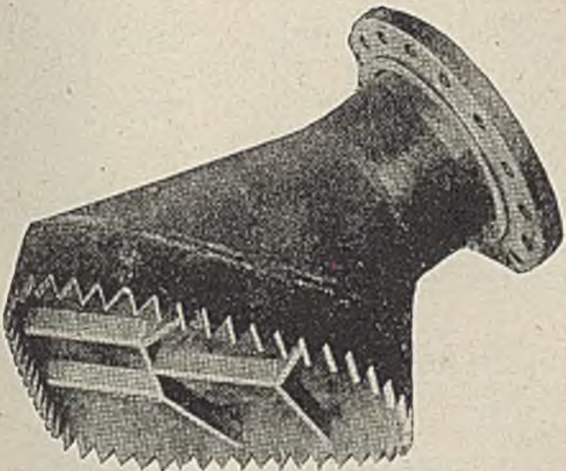


Fig. 3.—Welded steel plate inlet head for plain inlet dredge. The bottom teeth prevent the inlet becoming sealed and the network of bars in the mouth acts as a screen to keep out oversize material that would choke the pump or lines.

Ans. For digging a channel of uniform depth or width.

Ques. How may feeding the material to the dredge be facilitated?

Ans. By surrounding the inlet with high pressure nozzles.

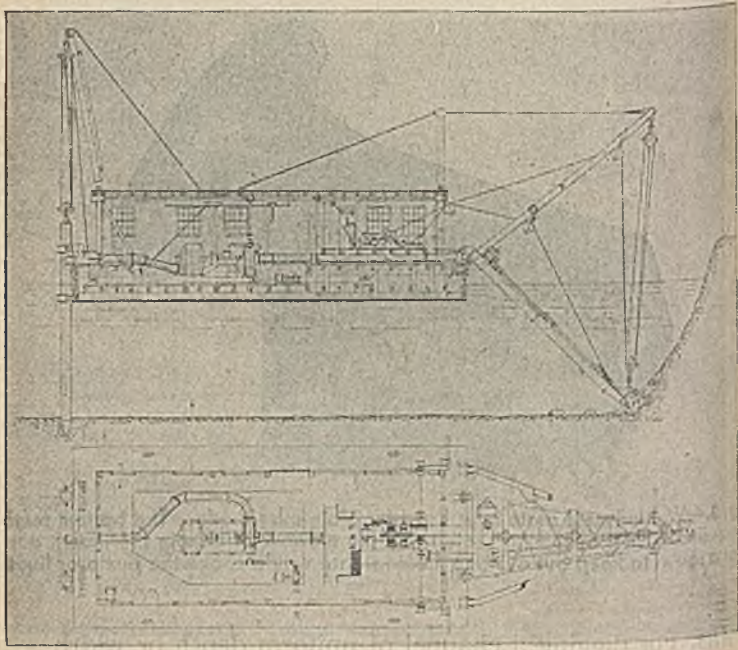
Ques. What is the action of the pressure nozzles?

Ans. They agitate the material immediately in front of the inlet.

Ques. How is high pressure water for the jets obtained?

Ans. From the service pump on board.

This pump should be of ample capacity.



Figs. 4 and 5.—Hydraulic cutter dredge with electrically driven dredging pump. This type dredge is adapted to sand and gravel production as a result of its efficiency in river, channel and harbor work. It has a mechanically driven revolving cutter as shown. The cutter breaks up the material and feeds it to the inlet pipe and acts as a screen to exclude oversized stones. The depth of digging is regulated by lowering and raising the cutter. Advance forward is made by walking the spuds while lateral feeding is controlled by the forward swinging lines.

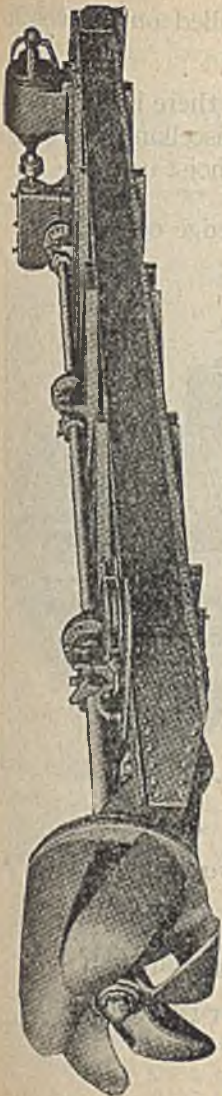


Fig. 6.—12-inch motor driven cutter ladder. A suitable cutter or agitator on the inlet is indispensable to successful and reliable operation of a dredge.

Hydraulic Dredges with Revolving Cutter.—By definition this type dredge is *a hydraulic dredge having in addition to the plain inlet a revolving cutter*, as shown in fig. 6.

Ques. For what use are these dredges adapted and why?

Ans. For general work as it can operate successfully in any kind of material short of solid rock or masses of large boulders.

It is exceptionally efficient in river, harbor and channel work.

Ques. Describe the revolving cutter.

Ans. The cutter with its long shaft is mounted on a "ladder" as in fig. 6, mechanically driven, being located at the inlet.

Figs. 9 to 11 show various cutters.

Ques. What is the action of the cutter?

Ans. It breaks up the material and feeds it uniformly to the intake of the inlet pipe and acts as a screen to exclude oversize stones from the inlet line and pump.

Ques. What apparatus is usually installed on a revolving cutter dredge?

Ans. In addition to the dredging pump there is a five drum hoist, service pump, spuds, piping and miscellaneous fittings. Fig. 7 shows appearance of the five drum hoist used.

Ques. How is the movement of the dredge controlled?

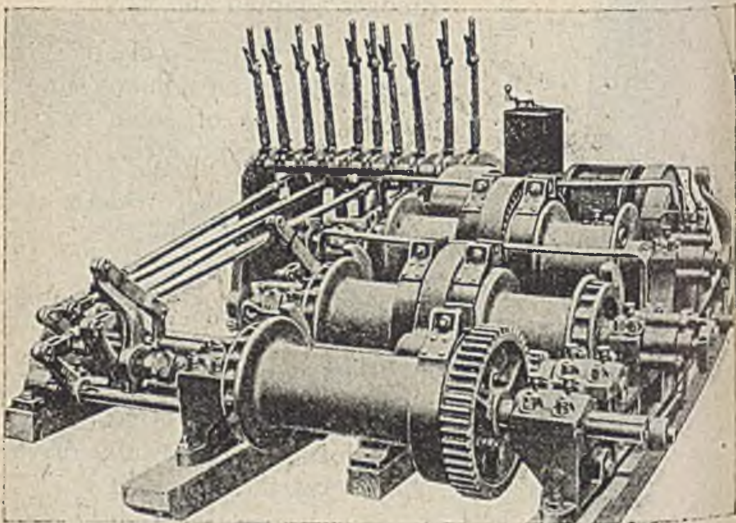


Fig. 7.—Five drum electric driven dredge hoist.

Ans. By two spuds at the stern and two swinging lines forward.

Ques. How is the depth of the dredging regulated?

Ans. By lowering and raising the cutter through.

Ques. How is advance forward made?

Ans. By "walking" the spuds while lateral feeding is controlled by the forward swinging lines.

Ques. What other name is given to the revolving cutter dredge and why?

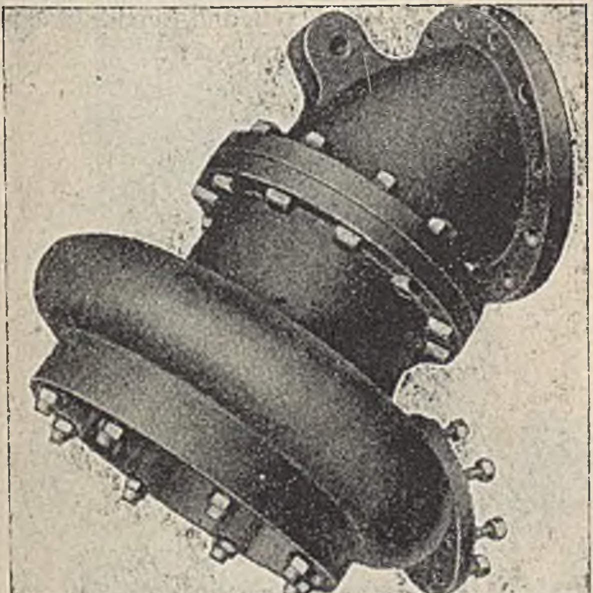


Fig. 8.—Jet agitator with separate 45 degree elbow. A jet agitator or jet head is used on the intake of plain inlet dredge to break up the material ahead of the inlet. Water under pressure must be supplied to the jet.

Ans. The "ladder," because of the so called ladder upon which the cutter shaft is mounted.

Hydraulic Dredges with Traveling Inlet Screen.—By definition this is a hydraulic dredge having the intake of the inlet pipe protected by a traveling screen.

Ques. Describe the screen.

Ans. The device consists of a heavy screen mounted on an endless chain which travels over sprockets and track built up of angles and steel plate. Prongs on some of the links in the chain break up lightly caked masses, carry the oversize material out of the way of the suction and deposit it underneath the dredge.

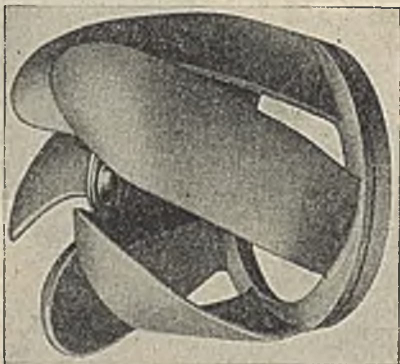


Fig. 9.—Five blade solid cast cutter.

Ques. What is the dual action of the screen?

Ans. It acts both as an agitator and screen to make the feed nearer uniform and to prevent choking of the line and pump.

Ques. Describe the general equipment.

Ans. It consists of the dredging pump, a multi-drum hoist preferably with four drums, a service pump, piping and miscellaneous fittings.

Ques. Describe the dredge movements.

Ans. The depth of digging is regulated by raising and lowering the inlet pipe. Advance forward is made with a line leading forward from the hoist. This line also holds the intake in contact with the material bed, and prevents the dredge being

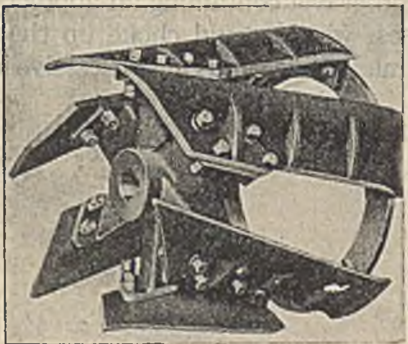


Fig. 10.—Five blade built-up cutter.

Fig. 11.—Solid cast cutter with removable manganese steel cutting edges for excavating clay and similar hard material.

pushed back. The lateral position is controlled by means of two forward swinging lines.

Ques. For what are these dredges chiefly suitable?

Ans. They are primarily adapted to working in loose deposits of sand and gravel containing an appreciable percentage of oversize stones, which would choke up the inlet line and pump of a plain inlet dredge and cause excessive interruptions.

STANDARDS OF HYDRAULIC INSTITUTE

Note.—This Test Code of the Hydraulic Institute is revised from time to time and the latest revision can always be obtained from the Institute.

Test Code

STANDARDS OF HYDRAULIC INSTITUTE
TEST CODE

This code has been adopted by the Hydraulic Institute with Membership as follows:

Aldrich Pump Company, The .... Allentown, Pennsylvania
Allis-Chalmers Manufacturing Co. .... Milwaukee, Wisconsin
American-Marsh Pumps, Inc. .... Battle Creek, Michigan
American Well Works, Inc., The .... Aurora, Illinois
Barrett, Haentjens & Company .... Hazelton, Pennsylvania
Blackmer Pump Company .... Grand Rapids, Michigan
Buffalo Pumps, Inc. .... Buffalo, New York
Byron Jackson Company .... Los Angeles, California
Chicago Pump Company .... Chicago, Illinois
Davidson, M. T., Company .... Brooklyn, New York
Dayton-Dowd Company .... Quincy, Illinois
Dean Brothers Company .... Indianapolis, Indiana
Dean Hill Pump Company .... Anderson, Indiana
DeLaval Steam Turbine Company .... Trenton, New Jersey
Deming Company, The .... Salem, Ohio
Domestic Engine & Pump Company .... Shippensburg, Pennsylvania
Dow Pump & Diesel Engine Company .... Alameda, California
Duriron Company, The .... Dayton, Ohio
Economy Pumps, Inc. .... Chicago, Illinois
Fairbanks, Morse & Company .... Chicago, Illinois
Frederick Iron & Steel Company .... Frederick, Maryland
Gardner-Denver Company .... Quincy, Illinois
Gaso Pump & Burner Mfg. Co. .... Tulsa, Oklahoma
Ingersoll-Rand Company .... New York, New York
Kingsford Foundry & Machine Co. .... Oswego, New York
Kinney Manufacturing Company .... Boston, Massachusetts
Lecourtenay Company .... Newark, New Jersey
Morris Machine Works .... Baldwinsville, New York
National Steam Pump Company .... Upper Sandusky, Ohio
National Transit Pump & Machine Co. .... Oil City, Pennsylvania
Nash Engineering Company .... South Norwalk, Connecticut
Pomona Pump Company .... Pomona, California
Scranton Pump Mfg. Co. .... Scranton, Pennsylvania
Taber Pump Company .... Buffalo, New York
Tuthill Pump Company .... Chicago, Illinois
Warren Steam Pump Company .... Warren, Massachusetts
Weinman Pump Manufacturing Co. .... Columbus, Ohio
Worthington Pump & Machinery Corp. .... Harrison, New Jersey
Yeomans Brothers Company .... Chicago, Illinois

Secretary: C. C. Rohrbach ● 90 West Street, New York

STANDARDS OF HYDRAULIC INSTITUTE
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HYDRAULIC INSTITUTE TEST CODE
CENTRIFUGAL AND ROTARY PUMPS

F-1. OBJECT AND SCOPE

This code contains specific provisions for the testing of centrifugal and rotary pumps by which to determine whether the pump meets the guarantees under which it is sold. The object of the Test Code is to set forth methods under which the performance of centrifugal and rotary pumps can be accurately determined when pumping clear water. The method of determining performance when pumping other liquids shall be subject to agreement. This code is not intended as a treatise or textbook on how to test centrifugal and rotary pumps, but is a statement of the methods governing the test which will give the desired accuracy and a statement of the operating conditions under which the test shall be made. It is intended to be used by experts in this line of work whose general knowledge of hydraulic engineering should be such as to secure reliable test data by following the methods herein set forth.

Centrifugal and rotary pump tests may be classified under two distinct heads:

(1) Acceptance test made in the field on complete installations for the purpose of determining fulfillment or non-fulfillment of contract guarantees between pump builder and purchaser.

(2) Tests made in the shop of the pump builder to determine the fulfillment or non-fulfillment of contract guarantees or agreements between the pump builder and purchaser.

When publishing data resulting from experimental tests, the tests shall be conducted, and the data worked up and presented in accordance with the standards of this code.

This code gives the limiting conditions for all methods of quantitative determination of capacity, head, and power input, whereby the accuracy required for an acceptance test can be obtained.

F-2. MEANING AND INTENT

Except where specifically stated elsewhere herein, this code shall be understood to apply to the tests of centrifugal and rotary pumps proper, and the terms "capacity", "total dynamic head", "efficiency", and "power" are to be taken as referring to pumps. The capacity shall be the absolute quantity and, unless otherwise stated, shall be determined in the manner as set forth under Pars. F-11 to F-27. The total dynamic head shall be measured at the suction and discharge flanges of the pump in the manner as set forth under the heading "measurement of head". In computing the efficiency of centrifugal and rotary pumps, the actual power input to the pump shaft shall be used.

F-3. ALLOWANCES OR MARGINS IN GUARANTEES

No minus tolerances or margins shall be allowed with respect to guarantees for head, capacity or efficiency. A plus tolerance of 10% shall be allowed in capacity at the specified head.

F-4. INSPECTION

A careful inspection shall be made before, during, and after the tests to insure the proper operation of the pump. The water passages of the pump shall be inspected before and after the test to guard against errors during test caused by obstructions clogging the passages. If obstructions are found in the pump, the test shall be re-run. Wearing ring clearances, piezometer openings, discharge and suction gauges, and capacity measuring device, specific gravity of mercury, or other liquids used in manometers, and in general the conditions of all instruments to be used shall be inspected.

All electrical and head measuring instruments shall be calibrated before and after test, and all other instruments that can be conveniently calibrated shall also be calibrated before and after test. Should there be any difference between the two calibrations, the average of the two shall be taken as the correction in computing final results. If the calibrations do not check within one half of one percent, the test shall be re-run.

The dimensions of the suction and discharge openings where pressure readings are to be taken should be checked so that accurate velocity head corrections can be made.

F-5. OPERATING CONDITIONS

The most important factors affecting the operation of a pump are:

1. Dynamic suction conditions
2. Total dynamic head
3. Speed
4. Temperature and specific gravity

The total dynamic suction lift is the vacuum as measured at the suction flange of the pump plus the vertical distance between the gauge connection and the pump centerline. The gauge connecting line must be entirely free from water. The net dynamic suction head is the pressure as measured at the suction flange of the pump corrected for the vertical distance between the centerline of the gauge and the centerline of the pump. The gauge connecting pipe must be entirely filled with water. During field acceptance test, the total dynamic suction lift shall not exceed that specified in the contract, or if the contract specifies a positive head on suction, the net dynamic suction head shall not be less than that specified in the contract. If the conditions as set forth above are not obtainable, the acceptance test shall not be run until such conditions do exist, unless by mutual agreement between both parties, the test will be run and if the performance is what would be expected with the suction conditions as set forth in the contract, the test will be acceptable to both parties, as for capacity, head, and efficiency, but if the performance is not what would be expected with the conditions as set forth in contract, and if there is any question as to noise and vibration, the test shall not be acceptable until the suction conditions are as set forth in the contract.

No valve on the suction that may have been installed as a part of the installation shall be throttled during the duration of the acceptance test.

Acceptance tests shall be made, if possible, at the total dynamic head specified in the contract. If it is impossible to obtain the specified head by throttling or by other means, the test shall be run at the nearest head to the contract conditions that can be obtained, but an agreement between both parties shall be made in writing prior to the test as to what corrections shall be made to determine the fulfillment or non-fulfillment.

When it is impossible to obtain the specified speed on test due to variation of frequency in the electric current or other causes, corrections in the capacity, head, and brake horsepower to correspond to the specified speed, may be made from test data, in the manner set forth under "calculations."

The acceptance guarantees are based upon pumping clear water at 68° F, at which temperature the weight per cubic foot is 62.318 lb. When the temperature or weight per cubic foot is different from that above, the proper correction shall be made. For Rotary Pumps see Par. C-38 Rotary Section.

Before the test begins or before continuing the test after an important change of conditions during operation, the apparatus shall be run under normal conditions for sufficient length of time to bring about equilibrium or steady readings.

F-6. CONDUCT OF TEST

On field acceptance test both parties to the contract shall be represented and shall have equal rights in determining the methods and conduct of the test, unless otherwise provided in the contract. The plan of test and procedure shall be agreed on in advance of the test definitely covering the method to be used in measuring all major quantities, including quantity of water, head, power input, and speed, and it is desirable

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TEST CODE

that the expert personnel who are to conduct each of the various measurements, shall be included in this agreement. All points of disagreement shall be settled in conference leading to agreement, to the satisfaction of both parties, and the results of the test shall be computed and agreed on as acceptable, before the test shall be considered terminated, or the test equipment removed and all results shall be agreed upon by both parties before they are embodied in the final report.

If the points of disagreement cannot be settled by conferences between the parties of the contract, the parties of the contract shall agree on a competent engineer to act as arbitrator, and his decision shall be binding. The expense of the arbitrator shall be borne equally by both parties of the contract.

F-7. PRELIMINARY TESTS

It is advisable to make one or more preliminary tests for the purpose of determining the adequacy of the instruments and apparatus and the training of the personnel.

When conditions do not permit such preparatory runs, operations may be started and the time at which conditions may become satisfactory can be chosen later as the starting time of the test.

F-8. RECORDS

The record sheets hereinafter given for data and results apply to the complete unit, including the drive, but the code itself applies only to the test of the pumps.

The manufacturer's serial number, type, and size, or other means of identification of each unit involved in the test, shall be recorded in order that later there may be no uncertainty as to which unit or units the data obtained refers. The dimensions and physical conditions not only of the machine to be tested, but of all the associated parts of the plant which may have an important bearing on the object of the test should be determined.

F-9. EFFICIENCY

The efficiency of the pump is the ratio of the energy delivered by the pump to the energy supplied to the pump shaft; that is, the ratio of the water horsepower output from the pump to the mechanical horsepower input to the pump shaft.

The efficiency of the pump covered by the contract guarantees, of which this code may form a part, is to be based on the fundamental determination of quantity of water by volume and/or weight, and all other methods of quantity determination shall be for the purpose of arriving as nearly as possible to the exact quantity as might be determined by actual volume determination or weight determination.

The efficiency of the pump under contract of which this code may form a part, may be computed by determining the quantity of discharge by any one of the following methods of measurement:

- (a) Volume or Weight
- (b) Venturi Meter
- (c) Weir
- (d) Nozzle
- (e) Pitot Tube
- (f) Current Meter
- (g) Allen Salt Velocity
- (h) Gibson Method

Other secondary methods for capacity measurements may be employed only by special arrangement or agreement between seller and purchaser, such as salt titration, and Photoflow, method. Whenever such agreements are made, they shall specify in detail the form of measuring device, location and calibration.

F-10. STANDARD UNITS OF VOLUME

The standard units of volume shall be the United States gallon or the cubic foot. The standard U. S. Gallon contains 231 cubic inches or, one cubic foot equals 7.4805 gallons.

The rate of flow shall be expressed in gallons per minute (gpm), cubic feet per second (cfs), or millions of gallons per twenty-four hour day (mgd).

The weight of water at ordinary temperatures 68° shall be taken as 62.318 lb per cubic foot. For density at other temperatures see Paragraph G-47 of Standards of Hydraulic Institute.

F-11. VOLUME OR WEIGHT

The preferred method of determining quantity is by volume or weight and all other methods of quantity determination are considered secondary to these.

Measurement of quantity by weight depends in accuracy upon the accuracy of the scales used and upon the accuracy of the measurement of time. The scales shall be calibrated with standard weights before and after test.

In measurement of capacity by volume sufficient measurements of the reservoir or tank shall be taken to establish its volume within one quarter of one percent. When there is doubt as to the volume of a reservoir it shall be calibrated by weighing the water.

F-12. VENTURI METER

The Venturi Meter is more widely used than any other device for measuring the quantity of water for the determination of efficiency of centrifugal and rotary pumps. The accuracy of the meter depends upon its being installed in such a manner that it will not be adversely affected by flow conditions in the pipe immediately above the meter and immediately below the meter. It has been found that incrustations on the inside of the pipe and bends and elbows in the pipe immediately above the Venturi Meter destroy the accuracy of the meter as a measuring instrument so that great care must be exercised in determining the conditions of installation, and the coefficients applicable under such conditions of installation, in order that accurate quantity determinations may be arrived at by the use of such meter. On acceptance tests the following conditions in regard to the meter shall apply:

Wherever possible the venturi meter should be calibrated in place in the piping system volumetrically before an acceptance test under this code. When this is not possible the specific proportions of the meter shall be the same as that of a meter that has been calibrated by precision methods.

The manufacturer of the meter shall submit a certified curve showing the calibration of the meter. The certification must state the method used in calibration and whether the meter itself was calibrated or whether calibration was obtained with another meter of same specific proportions.

The straight length of pipe preceding the venturi tube must be of the same diameter as the venturi tube inlet and the length must be not less than ten diameters. When it is impossible to provide the length of straight pipe as required above straightening vanes must be used to steady the flow, and means taken to determine that flow is straight and substantially equal in velocity at entrance to venturi.

The size of the venturi tube shall preferably be such that the velocity in the throat is greater than 20 ft. per second for normal capacity of pump tested. This insures not only a differential pressure that can be read with great accuracy, but it also brings the coefficient close to the non-varying part of the curve. In making official tests manometers shall be used and not recording dials.

The equation of the venturi meter is

$$Q = CA_1 \frac{1}{\sqrt{R^4-1}} \sqrt{2gh}$$

Where Q = rate of discharge in cubic feet per second

C = coefficient of discharge from calibration data

A_1 = area of the entrance section at the up-stream pressure connection in square feet

R = ratio of entrance to throat diameter D_1/D_2

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g = acceleration of gravity—32.174 at sea level and about 45 degrees latitude

h = difference of pressure as indicated by a manometer when connected to the inlet and throat pressure connections of the venturi tube, expressed in feet

If the rate of discharge is to be stated in gallons per minute the equation of the venturi meter becomes—

$$gpm = 3.117\ C \cdot A_1 \cdot \frac{1}{\sqrt{R^4-1}} \sqrt{2gh}$$

Where C = coefficient of discharge

A_1 = area of the entrance section at the upstream pressure connection in square inches

R = ratio of entrance to throat diameter D_1/D_2

g = acceleration of gravity—32.174 at sea level and about 45 degrees latitude

h = difference of pressure as indicated by a manometer when connected to the inlet and throat pressure connections of the venturi tube, expressed in feet

F-13. WEIR

The weir is one of the oldest means of measuring large quantities of water and under proper conditions and in the hands of experts gives reliable results provided the velocity of approach to the weir, the smoothness and cleanliness of the crest, the height on the weir with respect to the length, are proper for the weir being used. Proper baffling is very important and small differences from the proper conditions cause considerable variations in the quantity on the weir.

This code provides that an uncalibrated Venturi meter may be used provided that "the specific proportions of the meter shall be the same as that of a meter that has been calibrated by precision methods."

The weir here dimensioned and described and covered by these specifications is of the specific proportions of weirs that have been calibrated by precision methods and coefficient determined. The coefficient here specified is on this specific form of weir only, and this code does not permit any other form of weir or formula for use in acceptance tests.

For acceptance tests where the quantity is to be determined by the weir, the following shall apply:

The weir shall be rectangular in form of opening, sharp crested, with smooth, vertical crest wall, complete crest contraction, free overfall, and have end contraction suppressed. The crest shall be formed of metal about ¼ inch thick, with sharp right angled corner on the upstream edge, an actual crest width of ⅛ inch, smooth and free from rust and grease at the time of test, and beveled at an angle of 45 degrees on the downstream face. The crest edge shall be level. Complete aeration of the nappe shall be secured and observation of the crest conditions and form of nappe shall be made during the test to avoid defective conditions such as adhering nappe, disturbed or turbulent flow, or surging. The side walls of the channel shall be smooth and parallel and shall extend downstream beyond the overfall above the level of the crest. The weir shall be set at right angles with the line of the channel of flow.

The depth "P" of the channel of approach, (Fig. F-1), below the weir crest shall not be less than 3H where H is the maximum head for which weir is to be used.

F-14. Conditions of Installation and Use. The weir or weirs shall be located on the discharge side of the pump, and care shall be taken that smooth flow, free from eddies, surface disturbance, or the presence of considerable quantities of air in suspension, exists in the channel of approach. To insure this condition the weir should not be located too close to the discharge of the pump, and stilling racks and beams should be used when required. The channel of approach should be straight, of uniform cross section and should be unobstructed by racks and booms, for a length of at least 15H from the crest. The racks should be arranged to give approximately uniform velocity across the channel of approach.

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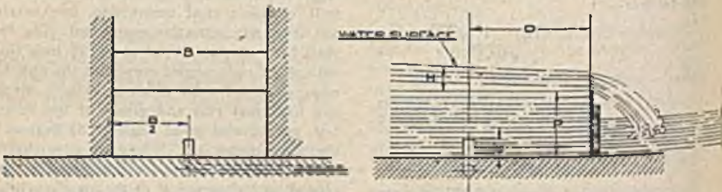
F-15. Measurement of Head. The head (H) on the weir shall be measured by the use of hook gauges placed in stilling boxes located at the side of the channel of approach, upstream from the weir at a distance of not less than five nor more than ten times the maximum head, and communicating through a pipe 2½ inches in diameter, with the opening located in the center of the channel of approach and six inches above the bottom of the channel. (See Fig. F-1).



P TO BE > 3H
D TO BE > 5H MIN. AND < 10H MAX.

PROPORTIONS OF RECTANGULAR WEIR

Fig. No. F-1

F-16. Formula. The discharge is to be computed by the Hamilton Smith formula, as follows:

$$Q = 3.29 \left(b + \frac{D}{7}\right) D^{3/2}$$

Q = quantity in cubic feet per second.
b = length of weir in feet
$D = H + ah$
H = Head on weir in feet
a = 1.33 for a suppressed weir
$h = \frac{V^2}{2g}$ where V = mean velocity of approach in ft/sec

F-17. NOZZLE

A circular nozzle of the converging type used for the determination of quantity is in effect a venturi meter and the coefficient of the nozzle is affected by the same factors which affect the coefficient of the venturi meter. Consequently the same rigid tests or calibrations are to be applied to the nozzle as for the venturi meter.

Wherever possible the nozzle and approach pipe shall be calibrated in place in the piping system volumetrically before an acceptance test. When this is not possible the nozzle, approach pipe and piezometer shall be of homologous design in all particulars to a calibrated nozzle unit providing that the nozzle used is not more than five times the size of the calibrated nozzle.

In making tests with nozzles the following shall apply:

The approach pipe or straight extension ahead of the nozzle shall have a length equal to at least ten diameters of the large end of the nozzle. When this is not possible straightening vanes shall be used to steady the flow and means taken to determine that the flow is straight and substantially equal in velocity at the entrance to the nozzle. The piezometer ring for the attachment of pressure reading instruments shall be a circular ring with four 1/8" or 1/4" diameter holes perpendicular to the axis of the pipe and with the edges of orifices exactly flush with the surface of the pipe and rounded to a radius of 1/16".

The discharge shall be computed from the following formula:

$$Q = CA_1 \frac{1}{\sqrt{R^4 - 1}} \sqrt{2gh}$$

Where Q = rate of discharge in cubic feet per second

C = coefficient of discharge from calibration data

A_1 = Area of the entrance section at the up-stream pressure connection in square feet

R = ratio of entrance to nozzle diameter $\overline{D_1/D_2}$

F-18. PITOT TUBE TRAVERSE

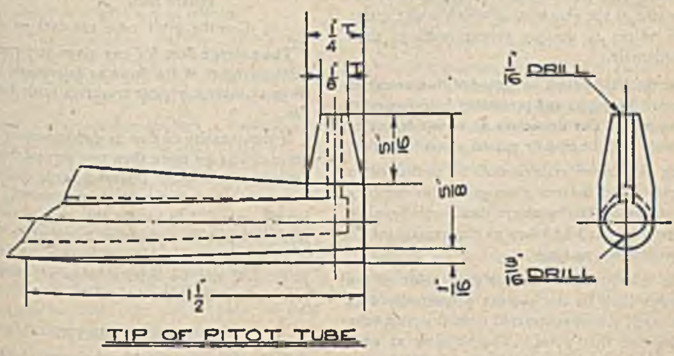
The pitot tube traverse method of water measurement consists of determining the velocity with pitot tube at sufficient points in a measuring section to accurately determine the average velocity for that section.

The pitot tube is a very valuable instrument by which expert hydraulic engineers can determine with a fair degree of accuracy the quantity of water flowing in a conduit where the conditions of flow are steady during the time required to make a traverse and where the conduit is long enough and smooth enough to give the required flow conditions. The coefficient of the pitot tube point itself is affected by cross current flow and the instrument has to be used with care.

The Pitot tube here dimensioned and described has the specific proportion of Pitot tubes that have been calibrated by precision methods and the coefficients determined. The coefficient here specified is applicable to this specific form of Pitot tube only, under the specific conditions herein outlined.

This code does not recognize any other form of Pitot tube for use in acceptance tests.

On acceptance tests where the quantity is to be determined by means of Pitot tube traverse the following shall apply:



TIP OF PITOT TUBE

Fig. No. F-2

The measuring section shall be located in a straight run of pipe on the discharge of the pump at a distance equal to at least ten pipe diameters from any upstream and at least five diameters from any downstream bend, elbow, "Y" branch pipe, valve, or any obstruction to smooth flow.

The area of the cross-section shall be determined with an accuracy within two-tenths of one percent.

The pitot tube shall be of the single opening impact type, made according to the dimensions shown in Fig. F-2, and shall at all times point upstream normal to the plane of the section.

For measuring the pressure head, no less than four piezometers shall be installed in the walls of the pipe in pairs diametrically opposite to each other, and equally spaced around the pipe. The piezometer orifices shall be from ⅛ to ¼ inch in diameter and their edges shall be free from burrs or irregularities and shall be rounded to a radius of 1/16 inch.

A constriction of the diameter not more than one-half of any other passage from the pitot tube point to the measuring gage shall be used and such a constriction shall preferably be in a flat plate with glass walls arranged so that obstructions at the constriction which might act as check valves to destroy proper readings shall be noticeable.

The pitot tube shall be arranged to traverse at least two diameters and preferably four diameters of the pipe. The diameters at which traverses are made shall be equally spaced around the pipe.

The piezometer orifices shall be connected to separate water column glass gauges or mercury manometers. The pressure head at the pitot tube section shall be taken as the average of all the piezometer readings.

The velocity head "h" for each point of the traverses shall be the average pressure head as determined above, subtracted from the pitot tube reading for that point. The velocity at each point of the traverse shall be computed by the formula:

$$V_1 = \sqrt{2gh_1}$$

Where V_1 = the velocity at point No. 1 in ft. per second

g = the acceleration of gravity in ft. per second per second = 32.174

h_1 = the velocity head at point No. 1 as determined above in feet.

The velocities for the traverse at each diameter shall be plotted against equal areas of the measuring section and the area of the curve planimetered to obtain the average velocity for that diameter. The velocities of the traverse of all the diameters shall be averaged to obtain the average velocity for the section.

The flow for each traverse shall be the average velocity for the section times the area of the section, times a pitot tube constant of .994. This constant is applicable only to pipes of not more than 24-inches diameter.

$$Q = CVA$$

Where Q = the flow in cfs.

V = the average velocity in feet per second as determined above.

A = the area of the cross section in square feet.

C = the pitot tube constant = .994.

The average flow for any given test run shall be the average of the flows as determined by the three or more complete traverses made for each run.

If the quantity of flow as determined by a traverse differs by more than two percent from the average of the other traverses made during the same run, the test run shall be repeated. If repeated tests fail to satisfy this requirement, the test shall not be considered to meet the degree of accuracy required by this code and its results shall not be applicable as an acceptance test under this code.

CURRENT METER

F-19. Description. Measurement of quantity of water by this method consists of determining the velocity with current meters at a sufficient number of points in a measuring section to accurately determine the average velocity therein.

The current meter is a useful instrument for determining the velocity of flow in open channels. It is not generally used for precision testing if other methods of quantity determination are applicable.

Considerable work has been done recently with current meters and in order to secure the accuracy required on tests of centrifugal and rotary pumps, the following shall apply.

F-20. Selection of Current Meters. Two types of current meters shall be used, one which will over-register, and one which will under-register the cosine component as little as possible in oblique flow.

Two or more meters of each type shall be used. Half of the meters shall be right hand rotation, and half left hand rotation. The meters shall be so arranged that they alternate with respect to the direction of rotation.

F-21. Calibration of Current Meters. All meters used shall be calibrated before and after the tests to determine their still water rating. The meters shall be calibrated on their supports and mounted as when used in testing.

F-22. The Measuring Section. The bottom and sides of the measuring section shall be regular and smooth and the bottom shall be sufficiently hard for accurate measurement.

Where the measuring section is located in a natural stream, there shall be practically no change in the cross section for 50 feet upstream and 25 feet downstream from the measuring section.

The physical dimensions of the measuring section shall be determined from actual field measurements at sufficient points to keep the error within two-tenths of one per cent. Drawings shall be used only as a check on field measurements.

If, for any given quantity of flow, the discharge as determined by the over-registering type of meter differs by more than one per cent from the discharge as determined by the under-registering type, the measuring section shall be considered unsuitable for current meter measurements for the degree of accuracy required by this code and the results shall not be applicable as an acceptance test.

F-23. Conduct of Test. For all quantities of flow to be measured, a complete traverse of the measuring section shall be made with each type of meter.

If either party to the test questions the proper neutralization of swirl effect, check runs shall be made with the right and left hand rotation meters interchanged, giving a complete traverse of the metering sections with each of the meters used.

During the test the meters shall be supported normal to the measuring section by means of a rigid streamlined rod or frame.

Either the point by point or the integral traverse method of testing may be used. If the point by point method is used, the number of metering points shall not be less than four times the square root of the metering section area in square feet. The meters shall be held at each measuring point for a sufficient length of time so that the velocity can be measured for a period of at least one minute.

The integral traverse method shall be used only when a reliable mechanical means for obtaining uniform traversing speeds is provided. The same traversing speed shall be maintained for all the runs of a test and shall not exceed the rate of four feet per minute.

The meter registration shall be either timed by a stop watch or recorded graphically.

The elevation of the water surface shall be observed continuously during a run of current meter measurements. If during a single traverse, the elevation of the water surface changes by such an amount as to change the area of the cross section by more than one per cent, the run shall be postponed until steady conditions prevail and shall then be repeated.

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TEST CODE

F-24. Computation of Results. The velocity of the water shall be determined from the revolutions of the current meter and its still water rating. For the point by point method, vertical velocity curves shall be plotted and the areas planimetered in order to obtain the mean velocity for each vertical section. A horizontal velocity curve shall then be plotted from the mean vertical values as obtained above or as given directly by the integral traverse method, and the area planimetered to obtain the mean velocity for the entire section.

In drawing the velocity curve at the boundaries the seventh root law shall be used as follows:

$$X^{1/7}V = v$$

Where X = distance from the boundary to the point being plotted, in fractional portion of the total distance from boundary to first metering point (X = 1 at first metering point).

V = velocity at first metering point.

v = velocity at any point X.

The quantity of flow will then be the average velocity as determined above multiplied by the area of the section.

If the variation between the discharges as determined by the over-registering and under-registering meters for any given quantity of flow is less than one per cent, the true quantity of flow shall be taken as the mean of discharges as determined by the two types of meters.

F-25. ALLEN SALT VELOCITY AND GIBSON METHODS

The great development in the hydro-electric field during the past twenty years has occasioned the development of two methods for the determination of a large quantity of flowing water. These two methods are the Gibson Method and the Allen Method invented by Mr. Norman R. Gibson of Niagara Falls, and Professor Charles M. Allen of Worcester, Massachusetts. These two methods provide precision means of measuring large quantities of water. They are to be used only in the hands of experts and preferably by the direct representatives of the inventors to whom it is recommended that application be made when tests are desired to be conducted by either of these methods.

F-26. MEASUREMENT OF QUANTITY BY OTHER METHODS

When the methods of quantity measurement as herein set forth are not available, other methods may be used as the alternate method only after mutual agreement between buyer and seller, as to the coefficients applicable and as to the method to be followed in the conduct of the test. Results obtained by such methods shall not be advertisable.

F-27. ORIFICE PLATES

The test code for the measurement of quantity by means of an orifice plate is just now in the state of flux. Some members of the Institute have used orifice plates with undoubted accuracy for the determination of quantity of liquid, particularly oil, under certain conditions, but these applications are somewhat special. Sufficient tests have not yet been made to write the details of a test code that would cover the degree of accuracy intended to be covered by this code.

The measurement, by means of the orifice plate, when using free discharge, should be subject to the calibrations and requirements as set forth herein for nozzles. The orifice plate in closed pipe would correspond most nearly to the Venturi meter, and the conditions as to calibration and lengths of pipe before the orifice plate would in general apply as herein specified for the Venturi meter. Under this issue of the test code it seemed advisable to leave it optional with the manufacturer as to whether he shall accept the results of tests determined by the use of the orifice plate. In this case it is a matter of special agreement between the buyer and seller as to whether and particularly as to what type and what coefficients shall apply when using orifice plates.

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F-28. MEASUREMENT OF HEAD

The standard unit for measuring head shall be the foot. The relation between the pressure expressed in pounds per square inch and that expressed in feet of head is:

$$\text{Head in ft} = \text{lb/sq in.} \times \frac{144}{\text{Density lb/cu ft}}$$

For water at 68° F, 1 lb/sq in. = 2.310 ft

The total dynamic head on the pump is the difference between the elevation corresponding to the pressure at the discharge flange of the pump and the elevation corresponding to the vacuum or pressure at the suction flange of the pump, corrected for the same datum plane, plus the velocity head at the discharge flange of the pump, minus the velocity head at the suction flange of the pump.

The velocity is to be computed by dividing the measured rate of pumpage in cu ft/sec by the measured area in sq ft at the point of pressure measurement. The velocity head is then to be computed by the following formula:

$$h_v = \frac{V^2}{2g}$$

Where V = velocity in ft/sec

The gauges shall, when practicable, be water columns or manometers, and for high pressures shall be mercury manometers, bourdon gauges, or dead weight gauge testers; when water columns are used, care shall be taken to avoid errors due to the difference between the temperature of the water in the gauge and that of the water in the pump. Gauges and manometers shall be used in accordance with Pars. F-29-F-32 of this code.

The piezometer orifices shall be flush with and normal to the wall of the water passage and the wall shall be smooth and parallel with the flow in the vicinity of the orifices. The piezometer orifices shall be from ⅛ to ¼ inches in diameter and their edges shall be free from burrs or irregularities and shall be rounded to a radius of 1/16 inch. All gauge connections shall be tight against leakage.

In cases where static pressure is unavoidably required in a conduit immediately succeeding an elbow, or other bent conduit such as the scroll case of a centrifugal pump, particular pains must be taken to avoid the error due to the fact that the static pressure on the outside of the curve is higher than the true value. The only way to get a reliable reading is to provide a length of straight pipe at least equal to ten diameters from the pump casing to the pressure hole, and a further length of straight pipe of at least equal to two diameters should be provided between the pressure hole and any succeeding irregularity such as an elbow.

In actual installations it will be often found impossible to provide a straight pipe of the prescribed length between the pump discharge and the pressure hole which will make it impossible to accurately measure the static head. For important installations it is customary to have four static holes over the circumference of the pipe at or near the pump discharge flange with separate pressure measuring instruments for each hole. If the separate manometers show a substantial difference of static pressure of a value that would affect appreciably the total head to be measured then it should be agreed that it is impossible to measure the total head accurately.

F-29. USE OF BOURDON GAGES

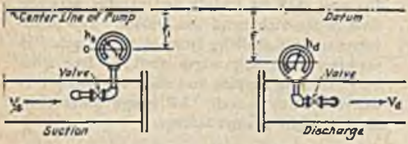

Fig. No. F-3. Absolute Suction Head Above Atmosphere

Total dynamic head in feet

$$H = h_d - h_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction head at datum

$$h'_s = h_s - f_1$$

Caution. Be sure gauge connection pipes are full of water before attaching gauges.

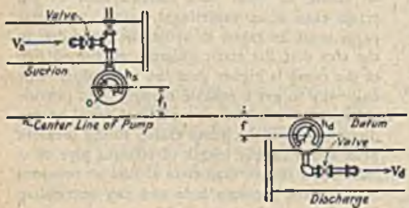

Fig. No. F-4. Absolute Suction Head Above Atmosphere

Total dynamic head in feet

$$H = h_d - h_s - f - f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction head at datum

$$h'_s = h_s + f_1$$

Caution. Be sure gauge connection pipes are full of water before attaching gauges.

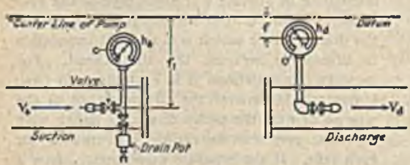

Fig. No. F-5. Absolute Suction Head Below Atmosphere

Total dynamic head in feet

$$H = h_d - h_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction lift at datum

$$h''_s = -h_s + f_1$$

f_1 — distance to pipe tap of connection.

Caution. Be sure discharge gauge connection pipes are full of water, and suction gauge connection pipes filled with air. Occasionally open drain valve on suction connections.

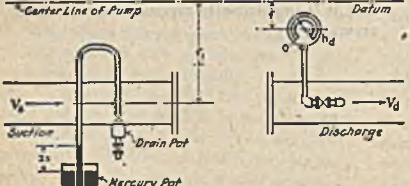

Fig. No. F-6. Alternative to Fig. No. F-5

Total dynamic head in feet

$$H = h_d + h_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

h_s = feet of water

$$H = h_d + Sz_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Where z_s = feet of mercury
S = sp gr of mercury

Suction lift at datum

$$h''_s = Sz_s + f_1.$$

Caution. Be sure discharge gauge connection pipes are full of water.

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Use of Bourdon Gages—Continued

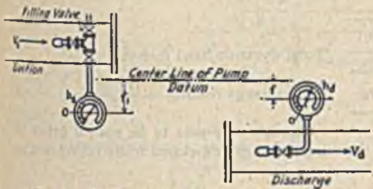

Fig. No. F-7. Absolute Suction Head Below Atmosphere

Total dynamic head in feet

$$H = h_d - h_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction lift at datum

$$h''_s = -h_s + f_1$$

Caution: Be sure both gauge connection pipes are full of water.

F-30. USE OF MERCURY U-TUBES

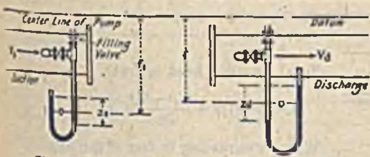

Fig. No. F-8. Absolute Suction Head Above Atmosphere

Total dynamic head in feet

$$H = (S - \tfrac{1}{2})(z_d - z_s) - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction head at datum

$$h'_s = (S - \tfrac{1}{2})(z_s) - f_1$$

Where z_d and z_s = readings in feet of mercury
S = sp gr of mercury

Caution. Connection pipes must be filled with water.

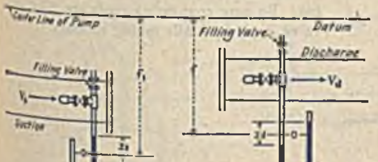

Fig. No. F-9. Absolute Suction Head Less Than Atmosphere

Total dynamic head in feet

$$H = (S - \tfrac{1}{2})(z_d + z_s) - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction lift at datum

$$h''_s = (S - \tfrac{1}{2})(z_s) + f_1$$

Where z_d and z_s = readings in feet of mercury
S = sp of mercury

Caution. Connecting pipes must be filled with water.

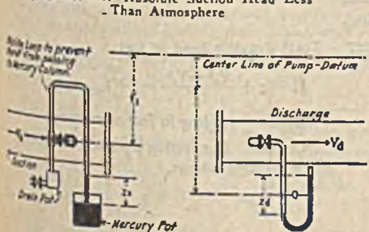

Fig. No. F-10. Alternative to Fig. No. F-9

Total dynamic head in feet

$$H = (S - \tfrac{1}{2}) z_d + S z_s - f + f_1 + \frac{V_d^2 - V_s^2}{2g}$$

Suction lift at datum

$$h''_s = S z_s + f_1$$

Where S = sp gr of mercury.

Caution. Discharge gauge connecting pipe to be filled with water, suction gauge pipe to be air filled.

F-31. USE OF DIFFERENTIAL GAGES

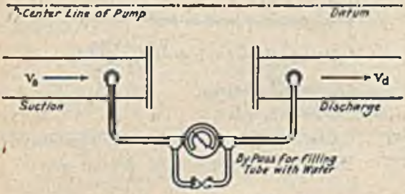

Fig. No. F-11. Bourdon Type

Total dynamic head in feet

$$H = \text{gauge reading} + \frac{V_d^2 - V_s^2}{2g}$$

Caution. By-pass to be opened before taking readings and closed while taking readings.

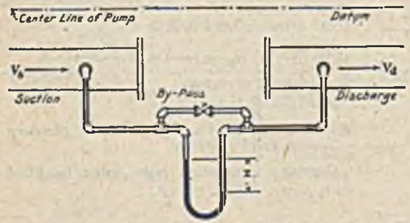

Fig. No. F-12. Mercury U-Tubes
Suction head either above or below atmospheric: discharge head above atmospheric

Total dynamic head in feet

$$H = (S - 1)z + \frac{V_d^2 - V_s^2}{2g}$$

Where z = reading in feet of mercury.
S = sp gr of mercury.

Caution. By-pass to be opened before reading. Be sure connecting pipes are filled, and valve is closed while taking readings.

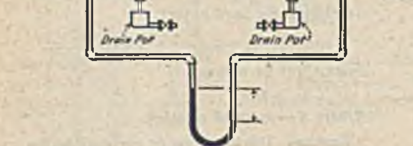

Fig. No. F-13. Mercury U-Tubes
Suction head and discharge head below atmospheric

Total dynamic head in feet

$$H = Sz + \frac{V_d^2 - V_s^2}{2g} - f + f_1$$

Where z = reading in feet of mercury.

Caution. Connecting pipes to be drained before taking readings.

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Use of Differential Gages—Continued

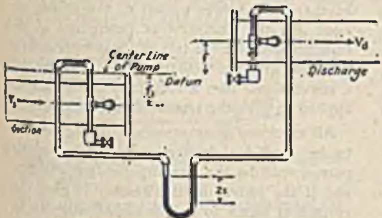


Fig. No. F-13a. Mercury U-Tubes
Discharge and suction at different levels

Total dynamic head in feet

$$H = Sz + \frac{V_d^2 - V_s^2}{2g} + f + f_1$$

Caution. Connecting pipes to be drained before taking readings.

F-32. USE OF WATER GAGES AND U-TUBES.

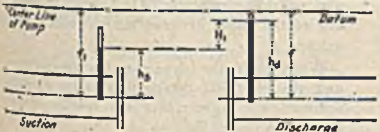


Fig. No. F-14. Absolute Suction Pressure Above Atmospheric
Absolute discharge pressure above atmospheric

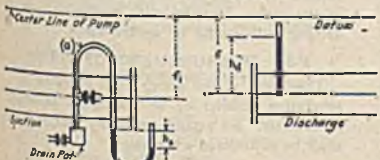
Total dynamic head in feet

$$H = H_1 + \frac{V_d^2 - V_s^2}{2g}$$

$$H = h_d - h_s + f_1 - f + \frac{V_d^2 - V_s^2}{2g}$$

Suction head at datum

$$h'_s = h_s - f_1$$



(a) Rising Loop to prevent Water passing Into U-Tube from Line.

Fig. No. F-15. Absolute Suction Pressure Below Atmospheric
Absolute discharge pressure above atmospheric

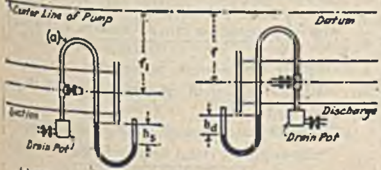
Total dynamic head in feet

$$H = h_d + h_s + f_1 - f + \frac{V_d^2 - V_s^2}{2g}$$

Suction lift at datum

$$h''_s = h_s + f_1$$

Note. Water can not be used in suction gauge if h is greater than height of rising loop



(a) Rising Loop to prevent Water passing into U-Tube from Line.

Fig. No. F-16. Absolute Suction Pressure Below Atmospheric
Absolute discharge pressure below atmospheric

Total dynamic head in feet

$$H = h_s - h_d + f_1 - f + \frac{V_d^2 - V_s^2}{2g}$$

Suction lift at datum

$$h''_s = h_s + f_1$$

Note 1. Water can not be used in suction gauge if h_s is greater than height of rising loop.

Note 2. H, h_d, h_s, f, and f_1 read in feet.

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F-33. MEASUREMENT OF POWER INPUT TO PUMP

The unit of power is the horsepower
1 hp = 550 ft lb/sec

Measurement of power input to the pump, that is, the brake horsepower of the pump falls into two general classes:

(1) Measurements which within themselves determine the actual power or torque delivered to the pump and are, therefore, made entirely during the test by using some form of transmission dynamometer.

(2) Measurements of power input, during the pump test, to the driving element, and the previous or subsequent determination of the relation of the power input to the power output of this driving element, under identical conditions of the pump test.

The use of transmission dynamometers and motors that have been calibrated with a transmission dynamometer shall be taken as giving the absolute power input to the pump.

The majority of centrifugal pumps under test are driven by electric motors and it is the custom to measure the electrical input to the motor, multiply such observations by the efficiency of the motor to thus determine the power input to the pump shaft. The guarantees of performance of pumps covered by this code are based strictly on the actual mechanical power delivered to the pump shaft and not to the computed power based on some arbitrary electrical measurements for determining the efficiency of the motor. Where a question exists as to the efficiency of the motor, its efficiency shall be determined by the measurement of the electrical energy input and the mechanical energy output by means of prony brake or other acceptable method.

Where the efficiency of the motor is based upon the assumption of no load losses, the burden of proof as to the efficiency of the pump motor shall lie with the purchaser and not with the pump builder.

The power delivered to the pump shaft when direct connected to the driving element shall be the actual power output of the driving element. If the efficiency as determined by the code prepared for the test of the driving element does not give the actual or true efficiency, but a conventional one, a correction factor must be applied to give the true efficiency.

All efficiency guarantees on driving elements to be used for driving centrifugal or rotary pumps shall be the true input-output efficiency, and if the conventional efficiency is given, the correction factor to reduce the efficiency to the input-output value must be determined and used.

When measuring power to an electric motor the measurements shall be taken at the motor and not at the switchboard, so as not to include any of the losses between the switchboard and motor.

When both the dynamometer measurement and the electrical determination are carefully followed out, agreements are readily secured that are within ¼%.

When the method of test as outlined above cannot be employed, other methods as outlined in the codes covering each particular type of driving unit should be agreed upon.

F-34. MEASUREMENT OF SPEED

The speed of the pump shall be taken by a revolution counter or an accurately calibrated tachometer. An accurate measurement of speed shall be considered essential.

F-35. CALCULATION OF RESULTS

Output

The water horsepower is found by the following formula:

$$\text{whp} = \frac{\left(\begin{array}{c}\text{Pounds of liquid}\\ \text{pumped per min}\end{array}\right) \times \left(\begin{array}{c}\text{Total dynamic}\\ \text{head in ft of liquid}\end{array}\right)}{33{,}000}$$

Where whp = water horsepower.

When the weight of water is 62.318 lb per cubic foot.

(Density at standard temperature of 68° F)

$$\text{whp} = \frac{\text{gpm} \times (\text{total head in feet})}{3960}$$

Where gpm = gallons per minute.

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If the pump is handling a liquid other than water or water at a temperature resulting in a different weight per cubic foot than 62.318, the above formula must be corrected for the specific gravity of the liquid so that

$$\text{whp} = \frac{\text{gpm} \times (\text{total head in feet})\ S}{3960}$$

Where S = specific gravity of liquid referred to 68° F water.

If the total dynamic head is expressed as pounds per square inch, the formula for theoretical horsepower irrespective of specific gravity of the liquid becomes

$$\text{whp} = \frac{\text{gpm} \times (\text{total head in lb/sq in.})}{1714}$$

F-36. **Input.** The Brake Horsepower input (bhp) when measured by transmission dynamometer is found from the following formula:

$$\text{bhp} = \frac{2\pi LWN}{33{,}000}$$

Where L = Length of lever arm in feet
W = Net weight in pounds
N = Speed in revolutions per minute
π = 3.1416

The Electrical Horsepower input to an electric motor is given by:

$$\text{ehp} = \frac{\text{kw}}{.7457}$$

Where kw = Kilowatt input.

The Brake Horsepower input to a pump driven by an electric motor is:

$$\text{bhp} = \text{ehp} \times \text{Em}$$

Where Em = True efficiency of motor (See par. F-35)

F-37. **Efficiency.** The pump efficiency is found by

$$\text{Ep} = \frac{\text{Output}}{\text{Input}} = \frac{\text{whp}}{\text{bhp}}$$

The combined efficiency of a motor driven unit is found by:

$$\text{Ec} = \frac{\text{whp}}{\text{ehp}} \text{ or } = \text{Ep} \times \text{Em}$$

F-38. **Correction to Constant Speed—Centrifugal Pumps.** For purposes of plotting, the capacity, head and power shall be corrected from the test values at test speed to the rated speed of the pump. Within small limits of speed, say 15%, the corrections are made as follows:

$$Q_2 = \frac{N_2}{N_1} \times Q_1$$

Where Q_1 = Capacity at test speed
Q_2 = Capacity at rated speed
N_1 = Test Speed
N_2 = Rated Speed

$$H_2 = \frac{(N_2)^2}{(N_1)^2}\ H_1$$

Where H_1 = Head at Test Speed
H_2 = Head at Rated Speed

$$HP_2 = \frac{(N_2)^3}{(N_1)^3}\ HP_1$$

Where hp_1 = Horsepower at Test Speed
hp_2 = Horsepower at Rated Speed

F-39. **Plotting of Results.** The Total Head, Efficiency, and Brake Horsepower are usually plotted as ordinates on the same sheet against the Capacity as abscissa as shown on Fig. F-17.

Fig. No. F-17

Sample Plot Showing Characteristic Curves for a Test Where the Data Has Been Corrected to Constant Speed of 1100 rpm

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DATA AND RESULTS OF CENTRIFUGAL OR ROTARY PUMP TEST

F-40. General Information:

1. Date of test
2. Location
3. Owner
4. Manufacturer
5. Object of test
6. Test conducted by
7. Test witnessed by

F-41. Description and Dimensions:

8. Type of pump. (Centrifugal or rotary, horizontal or vertical shaft, split or solid casing, enclosed or open impeller, single or multi-stage, single suction or double suction, volute, or diffusing vane type, etc.)
9. Type of driver (whether motor, turbine, internal combustion engine, belt drive, etc.)
10. Rated capacity..........gpm.........., mgd...........
11. Rated Total Dynamic Head....Feet.... lb/sq in.............
12. Specified Suction Lift (pressure)....... feet
13. Rated Temperature of Liquid .... ° F
14. Rated Speed
15. Liquid pumped
16. Method for measuring capacity
17. Method for measuring head
18. Method for measuring power
19. Size of discharge nozzle.....inches diam
20. Size of suction nozzle......inches diam

AVERAGES OF DATA AS RECORDED

F-42. General:

21. Test number
22. Date
23. Time
24. Conditions peculiar to particular test
25. Readings by
26. Barometer......inches Hg
27. Room temperature......° F
28. Elevation of test installation above sea level......feet

F-43. Pump:

29. Suction lift (pressure)......feet
30. Discharge head......lb/sq in......feet
    (a) Computed velocity in suction pipe at gauge.........ft/sec
    (b) Computed velocity in discharge pipe at gauge........ft/sec
    (c) Head due to difference in velocity heads plus or minus........feet
31. Total dynamic head......lb/sq in...... feet
32. Speed
33. Inlet and throat size of Venturi tube or other meter
34. Meter differential pressure........inches Hg
35. Meter coefficient
36. Brake horsepower (dynamometer readings only)

F-44. DRIVER

(A) Steam Turbine:

37. Pressure of steam to throttle........ lb/sq in. (abs)
38. Quality of steam to throttle.......F or percent moisture.
39. Exhaust pressure........lb/sq in. (abs)
40. Exhaust quality ....F or percent moisture.
41. Steam flow to turbine throttle........ lb/hour
42. Output in horsepower by guarantee or separate test.........
43. For more detailed list see Test Code for Steam Turbines.

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(B) Electric Motor:

44. Potential........volts
45. Current........amperes
46. Frequency........cycles per second
47. Power input........kw, phase 1, 2, 3
48. Speed........rpm
49. Excitation........
50. Output in horsepower by guarantee or separate test.........
51. For more detailed list see appropriate AIEE Codes

(C) Internal Combustion Engine, etc.:

52. Calorific value of fuel........Btu per lb
53. Fuel consumption........lb/hour
54. Exhaust temperature.........°F
55. Output in horsepower by guarantee or separate test.........
56. For more detailed list see appropriate ASME Codes

F-45. PERFORMANCE UNDER ACTUAL TEST CONDITIONS

57. Test number
58. Date
59. Time
60. Conditions peculiar to particular test
61. Volume of water pumped........gpm
62. Total dynamic head........feet
63. Temperature of liquid
64. Specific gravity of liquid
65. Theoretical horsepower
66. Brake horsepower input to pump (by direct measurement, guarantee, or separate test data of driver)
67. Pump efficiency
68. Overall efficiency of pump and driver
69. Duty........million ft lbs per 1000 lb steam.

F-46. PERFORMANCE INDICATED FOR SPECIFIED CONDITIONS

70. Speed........specified rpm
71. Total dynamic head at specified rpm ....feet
72. Capacity at specified rpm........gpm
73. Brake horsepower at specified rpm

F-47. GRAPHICAL PRESENTATION OF RESULTS

74. Characteristic curves

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RECIPROCATING PUMPS

Paragraphs 1 to 47 inclusive of the test code for centrifugal pumps shall apply to the testing of reciprocating pumps with the following additions and exceptions:

Acknowledgment is made to the American Society of Mechanical Engineers for use of certain provisions from the A.S.M.E. test code approved October 25, 1925, but no other provisions of the A.S.M.E. test code are to apply except those herein specifically recited.

F-48. Measurements. The fundamental measurements for a duty trial of a reciprocating steam pumping engine are:

(a) Amount of water pumped, in pounds

(b) Average total head, in feet

(c) Amount of steam supplied, in pounds; or the amount of heat supplied in British thermal units (Btu).

F-49. Instruments and Apparatus. The instruments and apparatus required for a performance test of a steam-driven reciprocating displacement pump include:

(a) Tanks and platform scales for weighing water

(b) Graduated scales attached to the water glasses of the boiler, if the feedwater is measured.

(c) Pressure gauges, vacuum gauges, and thermometers provided with suitable wells

(d) Steam calorimeter

(e) Barometer

(f) Revolution counter or other accurate speed-measuring device

(g) Indicators

(h) A planimeter

(i) A deadweight gauge tester.

The gauge on the discharge main should be attached near the pump discharge nozzle, and that on the suction main near the suction nozzle. These gauges should be attached at an angle of 90° to the direction of flow. The gauge pipes should be provided with valves at the gauges and pet-cock air vents.

When a suction or discharge surface condenser is in use as part of the unit under test, water-head readings should be made outside of and beyond the condenser so that the loss of head through the surface condenser will be charged against the main unit.

F-50. Preparations. Leakage Test of Pump. The tightness of the suction and discharge valves (and the plunger packing, if the pump is of the inside-packed construction) should be ascertained by applying the specified discharge pressure on them and observing the leakage by removing a handhole or manhole cover on the suction side of the part being tested. If essential to the object in view, the valves should be made tight; the plunger packing should leak only enough to insure that the friction of the packing on the plunger is not excessive. In the case of direct-acting pumps with inside-packed plungers the amount of leakage may be estimated by closing the discharge valve and running the pump just fast enough to maintain the normal discharge pressure. If sufficient leakage is found to affect the capacity materially, the conditions should be corrected before proceeding with the test work.

F-51. Steam Rate. The steam rate of the engine should be determined by measuring the condensate from the surface condenser, if that type of condenser is in use. If the condenser leaks, the defects causing such leakage should be remedied before test. The amount of condensation from jackets, receivers, and reheaters, as well as the steam used by the auxiliaries concerned, if not included in the condensate from the condenser, should be added thereto. If a surface condenser is not available, the steam supplied to the engine should be determined by feedwater tests, which require the measurement of the water fed to the boiler. The water from separators and drips on the main steam line, the steam used for other equipment not a part of the engine under test, and the water and steam which escape by leakage from the boiler and piping, must also be measured, and these quantities deducted from the total quantity of feedwater.

F-52. Measurement of Water Pumped. The quantity of water pumped should be determined by computing the actual plunger displacement in U. S. gallons or cubic feet. It is advisable, where apparatus is available, to check the total pumpage by the use of venturi tubes, pitot tube, weir, calibrated orifice, etc. or by means of the use of a reservoir, proper precautions being taken to insure reliable results in accordance with the directions of this code. In direct-acting pumps, the actual stroke of each plunger must be determined by the use of graduated stroke scales.

F-53. Operating Conditions. The operating conditions should conform to the object of the test and these should prevail throughout the trial. If an air pump is used to remove air which enters the suction pipe in the supply system of the pumping engine, such a pump is not to be considered a part of the engine equipment so far as it concerns the test.

F-54. Starting and Stopping. The engine and appurtenances should be first thoroughly heated and run under the prescribed conditions until uniformity is secured. When surface-condenser measurement is used, the duration of the test should be not less than eight (8) hours. Where the steam consumption is determined by measuring the feedwater to the boiler, the duration of the test should be not less than ten (10) hours.

F-55. When a surface condenser is used, the test should start by commencing to weigh or measure the condensate and any other quantities of steam consumption involved, at the same time beginning the observations and other necessary test work. At the end of the allotted time the test is stopped by discontinuing the measurements and observations. When feedwater measurements are employed, the test should be started by carefully observing the steam pressure and water level in the boiler, rate of steaming and the level in the feed tank if measuring tanks are used, at the same time beginning the water measurements and taking up the routine work of the test. Toward the end of the prescribed time, when the water levels, steam pressure, and conditions of fire and draft are as near as practicable to the same points as at the start, the observations should be discontinued. If there are differences in the water levels, corrections must be applied to the water measurements.

F-56. Records. Instruments should be read at intervals not greater than fifteen minutes, and if complete test data are required, indicator cards should be taken from each end of each cylinder once each hour when the conditions are uniform, and oftener when there is much variation. The log should contain the record of the readings of all gauges, thermometers, calorimeters, speed indicator, and all other instruments. If desirable, representative steam-pipe diagrams may be taken with an indicator applied near the throttle valve and operated by connection to the reducing motion of the cylinder indicators.

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F-57. A typical set of indicator cards should be selected from the whole number taken, and these should be embodied in the record. The specimen cards selected should be such as to show the average conditions of pressure and cut-off. If steam-pipe diagrams are obtained, specimens of these should also be placed in the record.

F-58. Throttle Pressure. The throttle pressure, or the average pressure in the steam pipe just before the throttle, is that shown by a calibrated steam gage attached to the steam pipe 1½ to 2 diameters from the throttle.

F-59. Calculation of Results. Steam Rate. State in the report of the test the actual steam rate as measured. State also in the report of the test the total amount of "estimated steam" supplied, as corrected for pressure, quality, or vacuum, in accordance with the correction factor based as far as possible on data obtained from the engine under test.

F-60. Heat Supplied. The heat supplied to a pumping engine and the auxiliaries concerned as part of the engine under test is the total heat content of the steam entering the engine and such auxiliaries, less the total heat returnable from engine and these auxiliaries to boiler.

The heat returnable is the sum of the total heat returned from the various sources as follows:

(a) The heat returnable in the condensate from the air pump, i.e., the product of the heat of the liquid per pound, about 32° F, corresponding to its temperature; and the total weight of such condensate.

(b) The heat returnable in any feedwater heater using exhaust steam or steam from the working charge in the engine.

(c) The heat returnable in high-temperature jacket and other drain water.

F-61. Efficiency, Thermal (Symbol et). Thermal efficiency is defined as the heat equivalent of the work done divided by the heat supplied.

Thermal efficiency of a steam engine or turbine on any steam cycle without reheat or extraction is expressed as follows:

For steam engines:

Indicated thermal efficiency,

$$e_i = \frac{2545}{w_i\,(h-h_1)}$$

For steam engines and turbines:

Brake thermal efficiency,

$$e_b = \frac{2545}{w_b\,(h-h_1)}$$

For engine generators and turbo-generators:

Combined thermal efficiency,

$$e_k = \frac{3413}{w_k\,(h-h_1)}$$

Where w_i — Steam rate referred to indicated horsepower

w_b — Steam rate referred to brake horsepower

w_k — Steam rate referred to net kilowatts

h — Heat content, Btu per lb at the throttle

h_1 — Heat returnable to boiler, Btu per lb

The thermal efficiencies of the internal combustion engine are expressed as follows:

Indicated thermal efficiency, $e_i = \frac{2545}{Q_i}$

Brake thermal efficiency, $e_b = \frac{2545}{Q_b}$

Combined thermal efficiency, $e_k = \frac{3413}{Q_k}$

Where Q_i — Btu per net i. hp-hr

Q_b — Btu per b. hp-hr

Q_k — Btu per net kw-hr

The thermal efficiency for complete plants will be expressed in the same way, using ihp, bhp, net kw, water hp, air hp, etc., as the reference. For example: The overall thermal efficiency of a coal-fired electric plant is

$$\frac{3413}{\text{Calorific value of coal} \times \text{lb coal per net kw-hr}}$$

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It is to be noted that in present usage the terms "thermal efficiency" and "cycle" are not limited to the thermodynamic sense only. They are therefore not to be interpreted in the strictly special manner usual in thermodynamics.

Their use in the general sense is established by custom and is perfectly understandable.

The thermal efficiency of a steam engine, steam turbine or internal combustion engine is an overall efficiency which may be separated into two factors: the cycle efficiency and the engine efficiency.

Thermal efficiency = Cycle efficiency × Engine efficiency.

The engine efficiency may, therefore, be found by dividing the thermal efficiency by the cycle efficiency.

F-62. **Efficiency, Rankine Steam Cycle.** This is the heat cycle efficiency, of the Rankine steam cycle.

$$e_r = \frac{h_1 - h_2}{h_1 - h_3}$$

where h_1 = heat content of steam, at initial condition

h_2 = heat content of steam after adiabatic expansion

h_3 = heat content of the liquid at exhaust pressure

F-63. **Efficiency, Engine (Symbol e_e).** Engine efficiency is the ratio of heat converted to work in the actual engine to the heat available for conversion to work by the cycle employed, or it is equivalent to the actual work done by the engine, divided by the work which could be done by a perfect engine on the same cycle. Engine efficiency is usually determined from a measured thermal efficiency and a calculated efficiency. For steam engines and turbines, all heat cycles are composed of one or more Rankine cycles. Exact knowledge of the steam cycle and a determination of its cycle efficiency is necessary to obtain engine efficiency from a measured thermal efficiency. A single-expansion steam engine or turbine always operates on a simple Rankine cycle. Multiple-cylinder engines or turbines, in which heat is added or abstracted between cylinders, operate on a number of different Rankine cycles, whose cycle efficiency must be determined for several actual heat quantities and pressure differences. The cycle efficiency of this composite cycle is different from the efficiency of a single Rankine cycle covering the entire range.

The term "Engine Efficiency" may thus be amplified if desired, by writing "Engine Efficiency on the Rankine Cycle," or "Engine Efficiency (Rankine)." It must be understood, however, that this is a distinctly different thing from the Rankine Steam Cycle Efficiency as defined in Paragraph F-62. The term "Rankine Efficiency" has been used for both of these things and it must be noted that its use is here abandoned for this reason. The term "Engine Efficiency" may also be amplified by writing "Engine Efficiency of a Turbine-Generator" or "Engine Efficiency of Turbine only." It is not desirable to use such terms as "Turbine-Generator Efficiency" or "Turbine Efficiency." This term may be further amplified when necessary by prefixing the word "Indicated" or "Brake."

F-64. **Head.** Refer to Par. F-28.

F-65. **Discharge Head.** Refer to Par. F-28.

F-66. **Suction Lift.** Refer to Par. F-5.

F-67. **Suction Head.** Refer to Par. F-5.

F-68. **Leakage of Pump.** The percentage of leakage in a reciprocating pump is found by dividing the quantity of leakage during the test, as computed from the rate of leakage determined on the leakage trial, by the plunger displacement during the test, and multiplying the quotient by 100.

F-69. **Capacity.** The capacity by displacement, in cubic feet per 24 hours, is found by multiplying the net or effective area in square feet of all the plungers by the length of the stroke in feet, the number of discharge strokes per minute, and the constant 1440. The equivalent capacity in U. S. gallons per 24 hours is found by multiplying the capacity in cubic feet as determined above by the constant 7.48.

F-70. Water Horsepower. Refer to Par. F-35.

F-71. Engine Indicated Horsepower. The average engine indicated horsepower for the entire test is found by dividing the average water horsepower as computed in Par. F-70 by the average combined hydraulic and mechanical efficiency of the entire unit as computed in Par. F-74.

F-72. Duty per 1000 Lb of Steam. The duty in foot-pounds of work done per 1000 pounds of steam is found by multiplying the number of pounds of water pumped during the test by the average total dynamic head in feet, dividing the product by the number of pounds of steam supplied during the test, and multiplying the quotient by 1000.

F-73. Duty per 1,000,000 Btu The duty in foot-pounds of work done per 1,000,000 Btu is found by multiplying the number of pounds of water pumped during the test by the average total dynamic head in feet, dividing the product by the number of Btu supplied, and multiplying the quotient by 1,000,000.

F-74. Friction and Mechanical Efficiencies. The average combined hydraulic and mechanical efficiency of the entire unit is 100 times the average of the ratios of instantaneous water horsepower to engine indicated horsepower shown by all normal indicator cards taken during the test.

F-75. Data and Results. The data and results of the test should be reported in accordance with the form given herewith. Unless otherwise indicated, the items should be the averages of all observations.

DATA AND RESULTS OF RECIPROCATING STEAM-DRIVEN DISPLACEMENT PUMP TEST

(ASME, Test Codes of 1926.)

General Information

(1) Date of test ....
(2) Location ....
(3) Owner ....
(4) Builder ....
(5) Test conducted by ....
(6) Object of test ....
(7) Duration of test ....

Description and General Dimensions

(8) Type of unit ....
(9) Number of steam cylinders..............................
(10) Diameter of steam cylinders { hp......in; ip......in; lp......in
(11) Diameter of steam piston rods........in
(12) Stroke of steam pistons ....in
(13) Plungers, number ....
(14) Single or double acting ....
(15) Inside or outside packed ....
(16) Diameter of plungers ....in
(17) Stroke of plungers ....in
(18) Diameter of plunger rods (if any) ......in
(19) Net area of each plunger..........sq ft
(20) Type of condenser ....
(21) Location of condenser ....
(22) Cooling surface in condenser........sq ft
(23) Type and size of condenser pumps......
(24) Type and size of any exhaust or receiver, feedwater heaters, steam reheaters and jacket pumps or other auxiliaries a part of the unit ....

Test Data and Results

Pressures

(25) Steam pressure at throttle.......lb/sq in.
(26) First receiver pressure ....lb/sq in.
(27) Second receiver pressure........lb/sq in.

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(28) Hp jacket ....lb/sq in.
(29) Ip jacket ....lb/sq in.
(30) Lp jacket ....lb/sq in.
(31) Barometer......inches of mercury...... .... lb/sq in.
(32) Corresponding absolute pressure........ .... lb/sq in.
(33) Vacuum in exhaust pipe near lp cylinder ......inches of mercury.......lb/sq in.
(34) Corresponding absolute pressure near lp cyl ....lb/sq in.
(35) Vacuum in condenser .... inches of mercury ....lb/sq in.
(36) Corresponding absolute pressure in condenser ....lb/sq in.

Temperatures

(37) Steam at throttle valve ....° F
(38) Exhaust steam ....° F
(39) Water pumped ....° F
(40) Condensate leaving surface condenser... .... ° F
(41) Condensate or feedwater entering feedwater heaters ....° F
(42) Condensate or feedwater leaving feedwater heaters ....° F
(43) Temperature rise in feedwater heaters.. .... ° F
(44) Condensate from jackets ....° F
(45) Condensate leaving 1st receiver........ .... ° F
(46) Condensate leaving 2nd receiver........ .... ° F
(47) Condensate from other drains.......° F
(48) Engine-room air ....° F
(49) External air ....° F

Total Head

(50) Discharge by gauge ....lb/sq in.
(51) Equivalent discharge head...ft of water
(52) Vacuum of pressure shown by gauge on suction main ....ft of water
(53) Discharge head referred to datum line.. .... ft of water
(54) Suction head or suction vacuum referred to datum line ....ft of water

(55) Total dynamic head pumped against.... .... ft of water
(56) Actual suction lift (if any) measured to highest point of discharge valve deck... .... ft of water

Quality of Steam near Throttle

(57) Dryness factor of steam ....per cent
(58) Superheat in steam (if any).........° F

Total Steam Quantities

(59) Total steam as measured supplied to engine and the auxiliaries concerned in its operation as specified ....lb
(60) Correction factor conforming to conditions agreed upon ....per cent
(61) Total "estimated steam" supplied conforming to conditions agreed upon....lb

Total Pump Quantities

(62) Total quantity of water pumped by plunger displacement..U.S. gal or cu ft
(63) Equivalent total weight of water pumped .... lb
(64) Total quantity of water pumped as shown by other means of measurement ....U.S. gal or cu ft

Hourly Steam Quantities

(65) Steam, as measured, supplied per hour lb
(66) "Estimated steam" supplied per hour lb

Hourly Pump Quantities

(67) Quantity of water pumped per hour by plunger displacement ....U.S. gal or cu ft
(68) Equivalent weight of water pumped per hour ....lb
(69) Quantity of water pumped per hour as shown by other means of measurement ......................U.S. gal or cu ft
(70) Slip in percentage of plunger displacement ....per cent

Heat Supplied

(Based on steam supplied, as measured, Item 59)

(71) Total heat per pound of steam at throttle .... Btu

(72) Total heat in steam supplied........Btu

(73) Total heat returnable from engine to boiler ....Btu

(74) Net heat supplied ....Btu

(75) Net total heat supplied per hour....Btu

Heat Supplied

(Based on total "estimated steam", Item 61)

(76) Total heat per pound of steam at throttle .... Btu

(77) Total heat in steam supplied........Btu

(78) Total heat returnable from engine to boiler ....Btu

(79) Net heat supplied ....Btu

(80) Net total heat supplied per hour....Btu

Speed

(81) Total number of revolutions ....

(82) Total number of single strokes ....

(83) Actual length of stroke, if direct-acting.. .... ft

(84) Revolutions per minute ....rpm

(85) Single strokes per min........Str per min

(86) Piston and plunger speed per minute.... ....ft per min

Power

(87) Total work done during test........ft-lb

(88) Water horsepower ....whp

(89) Combined hydraulic and mechanical efficiency ....per cent

(90) Indicated steam horsepower ....ihp

(91) Mean effective pressure (referred to lp cylinder if multiple-expansion engine).. .... lb/sq in.

(92) Indicated water horsepower.......iwhp

Economy Results

(Based on steam supplied as measured, Item 65)

(93) Steam supplied per ihp-hr............lb

(94) Heat supplied per ihp-hr............Btu

(95) Heat supplied per water hp-hr......Btu

Economy Results

(Based on "estimated steam," [1]Item 66)

(96) Steam supplied per ihp-hr............lb

(97) Heat supplied per ihp-hr............Btu

(98) Heat supplied per water hp-hr......Btu

Efficiency Results

(99) Thermal efficiency (referred to ihp) based on steam as measured [2545 ÷ Item 94] ....per cent

(100) Thermal efficiency (referred to ihp) based on "estimated steam"[2545 ÷ Item 97] ....per cent

Duty

(101) Duty per 1000 lb of steam as measured ....ft-lb

(102) [1]Duty per 1000 lb of "estimated steam" .... ft-lb

(103) Duty per 1,000,000 Btu based on steam supplied, as measured................ft-lb

(104) [1]Duty per 1,000,000 Btu based on "estimated steam" ....ft-lb

[1]Note.—The correction factor used shall be stated in percentage, and the basis and method of determining it described.

Duty Trials and Tests of Reciprocating Displacement Pumps Driven by Prime Movers Other Than Reciprocating Steam Engines

F-76. The rules given above apply to all reciprocating water ends of pumps regardless of the type of driving apparatus.

F-77. For rules governing the driving machine, see code which applies to such machine (as an oil engine, gas engine, water wheel, etc.)

F-78. In cases in which the pump is driven by an electric motor, the characteristics and efficiency of the motor, and the power input to the motor should be obtained according to Par. F-11

F-79. The form of the tabulation of the test results for other than steam-driven reciprocating pumps will be dependent on the object of the test and will, in general, be a combination of the parts of the summary which refer to the pump and the form recommended in the code applying to the driving machine:

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