SECTION C

AIR COMPRESSORS AND BLOWERS

CHAPTER 43

Air

Ques. What is air?

Ans. A gas consisting of a mechanical mixture of 23.2% (by weight) of oxygen, 75.5% nitrogen and 1.3% argon.

Carbonic acid (carbon dioxide) is present to the extent of about .03 or .04%. Obscure constituents are .01% Kryplon with small amounts of several other gases.

Ques. What are the percentages by volume?

Ans. Oxygen 21%, nitrogen 78.06% and argon .94%.

Ques. What is "*free air*"?

Ans. Air at atmospheric pressure.

Ques. What mistake is often made with respect to the definition of free air?

Ans. The interpretation of atmospheric pressure.

Because of the altitude, barometer and temperature vary at different localities and at different times, it follows that the term *free air* does not mean air under identical conditions.

Considerable confusion has existed regarding the term *free air* and it has often been interpreted to be air at sea level conditions, namely at an absolute pressure of 14.7 lbs. per sq. in. and at a temperature of 60° Fahr. This is not the correct interpretation of the expression of *free air*.

Ques. What is the correct interpretation of the term "free air"?

Ans. Air at the atmospheric condition at the point where the compressor is installed.

Ques. What is normal air?

Ans. This has been variously defined but may be taken to refer to air with 36% relative humidity at 68° Fahr.

This is the average condition of the atmosphere in a temperate climate.

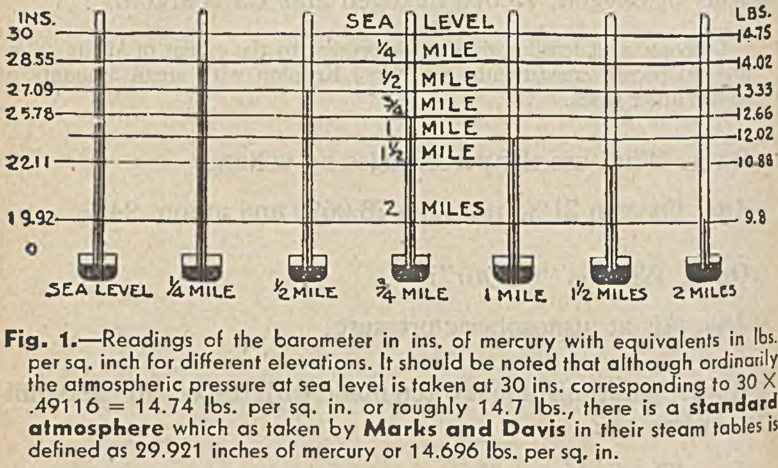


Fig. 1.—Readings of the barometer in ins. of mercury with equivalents in lbs. per sq. inch for different elevations. It should be noted that although ordinarily the atmospheric pressure at sea level is taken at 30 ins. corresponding to 30 × .49116 = 14.74 lbs. per sq. in. or roughly 14.7 lbs., there is a **standard atmosphere** which as taken by **Marks and Davis** in their steam tables is defined as 29.921 inches of mercury or 14.696 lbs. per sq. in.

Ques. What is humidity?

Ans. The watery vapor always present in the atmosphere being greater during warm weather.

Ques. What is absolute humidity?

Ans. The actual quantity of water in the air usually expressed as so many grains of moisture in a cubic foot of air.

A grain is $\frac{1}{7000}$ part of a pound. The amount of water the air is capable of holding is determined by the temperature. *The warmer the air the more moisture it can contain.* For instance, at 80° Fahr. it can hold nearly twice as much moisture as at 60°.

Ques. What is relative humidity?

Ans. The degree of saturation of the air with water vapor as determined by the use of the wet and dry bulb thermometers.

Relative humidity denotes ***the relation*** *between the actual amount of water in the air and the maximum amount it is possible for the air to hold at the same temperature expressed in per cent.*

Example.—Air which is saturated has a relative humidity of 100%, while the air at the same temperature and holding but one half of the saturation amount has a relative humidity of 50%.

Ques. Upon what does the feeling as to whether the air is moist or dry depend?

Ans. Upon the relative and not the absolute humidity.

Ques. What is "humidity" generally understood to mean?

Ans. Relative humidity or percentage of saturation.

NOTE.—***Dew point.*** By definition this is *the temperature at which air becomes saturated with water vapor.*

Ques. How is the relative humidity measured?

Ans. By the wet and dry bulb thermometer. (A form of hydrometer.)

It consists of two thermometers mounted side by side, the bulb of one being kept moist by means of a loose cotton wick tied around it, the lower end of which dips into a vessel of water.

Ques. How does it work?

Ans. On account of evaporation from the bulb, the wet bulb is cooled and indicates a lower temperature than the other

RELATIVE HUMIDITY TABLES

FAHRENHEIT DEGREES. BAROMETER 29 INCHES.

Reading of Dry-Bulb Thermometer	Difference in Degrees Between Wet- and Dry-Bulb Thermometers											
	1	2	3	4	5	6	7	8	9	10	11	12
32	90	79	69	60	50	41	31	22	13	4		
33	90	80	71	61	52	42	33	24	16	7		
34	90	81	72	62	53	44	35	27	18	9	1	
35	91	82	73	64	55	46	37	29	20	12	4	
36	91	82	73	66	56	48	39	31	23	14	6	
37	91	83	74	66	58	49	41	33	25	17	9	1
38	91	83	76	67	59	61	43	35	27	19	12	4
39	92	84	76	68	60	52	44	37	29	21	14	7
40	92	84	76	68	61	53	46	38	31	23	16	9
41	92	84	77	69	62	54	47	40	33	26	18	11
42	92	85	77	70	62	55	48	41	34	28	21	14
43	92	85	78	70	63	56	49	43	36	29	23	16
44	93	85	78	71	64	57	51	44	37	31	24	18
45	93	86	79	71	65	58	52	45	39	33	26	20
46	93	86	79	72	65	59	53	46	40	34	28	22
47	93	86	79	73	66	60	54	47	41	35	29	23
48	93	87	80	73	67	60	54	48	42	36	31	25
49	93	87	80	74	67	61	55	49	43	37	32	26

Reading of Dry-Bulb Thermometer	Difference in Degrees Between Wet- and Dry-Bulb Thermometers																
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	15	17
50	93	87	81	74	68	62	56	50	44	39	33	28	22	17	12	7	2
51	94	87	81	75	69	63	57	51	46	40	35	29	24	19	14	9	4
52	94	88	81	75	69	63	58	52	46	41	36	30	25	20	15	10	6
53	94	88	82	75	70	64	58	53	47	42	37	32	27	22	17	12	7
54	94	88	82	76	70	65	59	54	48	43	38	33	28	23	18	14	9
55	94	88	82	76	71	65	60	55	49	44	39	34	29	25	20	16	11
56	94	88	82	77	71	66	61	56	50	45	40	35	31	26	21	17	12
57	94	88	83	77	72	66	61	56	51	46	41	36	32	27	23	18	14
58	94	89	83	77	72	67	62	57	52	47	42	38	33	28	24	20	15
59	94	89	83	78	73	68	63	58	53	48	43	39	34	30	25	21	17
60	94	89	84	78	73	68	63	58	53	49	44	40	35	31	27	22	18
61	94	89	84	79	74	68	64	59	54	50	45	40	36	32	28	24	20
62	94	89	84	79	74	69	64	60	55	50	46	41	37	33	29	25	21
63	95	90	84	79	74	70	65	60	56	51	47	42	38	34	30	26	22
64	95	90	85	79	75	70	66	61	56	52	48	43	39	35	31	27	23
65	95	90	85	80	75	70	66	62	57	53	48	44	40	36	32	28	25
66	96	90	85	80	76	71	66	62	58	53	49	45	41	37	33	29	26
67	95	90	86	80	76	71	67	62	58	54	50	46	42	38	34	30	27
68	96	90	85	81	76	72	67	63	59	55	51	47	43	39	35	31	28
69	95	90	86	81	77	72	68	64	59	55	51	47	44	40	36	32	29

Reading of Dry-Bulb Thermometer	Difference in Degrees Between Wet- and Dry-Bulb Thermometers																
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
70	96	90	86	81	77	72	68	64	60	56	52	48	44	40	37	33	30
71	95	90	86	82	77	73	69	64	60	56	53	49	45	41	38	34	31
72	96	91	86	82	78	73	69	65	61	57	53	49	45	42	39	35	32
73	95	91	86	82	78	73	69	65	61	58	54	50	46	43	40	36	33
74	96	91	86	82	78	74	70	66	62	58	54	51	47	44	40	37	34
75	96	91	87	82	78	74	70	66	63	59	55	51	48	44	41	38	34
76	96	91	87	83	78	74	70	67	63	59	56	52	48	45	42	38	35
77	96	91	87	83	79	75	71	67	63	60	56	52	49	46	42	39	36
78	96	91	87	83	79	75	71	67	64	60	57	53	50	46	43	40	37
79	96	91	87	83	79	75	71	68	64	60	57	54	50	47	44	41	37
80	96	91	87	83	79	76	72	68	64	61	57	54	51	47	44	41	38
82	96	92	88	84	80	76	72	69	65	62	58	55	52	49	46	43	40
84	96	92	88	84	80	77	73	70	66	63	59	56	53	50	47	44	41
86	96	92	88	85	81	77	74	70	67	63	60	57	54	51	48	45	42
88	96	92	88	85	81	78	74	71	67	64	61	58	55	52	49	46	43
90	96	92	89	85	81	78	75	71	68	65	62	59	56	53	50	47	44
92	96	92	89	85	82	78	75	72	69	65	62	59	57	54	51	48	45
94	96	93	89	86	82	79	75	72	69	66	63	60	57	54	52	49	46
96	96	93	89	86	82	79	76	73	70	67	64	61	58	55	53	50	47
98	96	93	89	86	83	79	76	73	70	67	64	61	59	56	53	51	48
100	96	93	90	86	83	80	77	74	71	68	65	62	59	57	54	52	49

To use tables, find difference in reading between dry- and wet-bulb thermometers. Under that figure, opposite dry-bulb reading, find per cent of Relative Humidity. *Example:* Dry bulb 75°, wet bulb 63°, difference 12. Under 12 and opposite 75 find 51% Relative Humidity.

thermometer, the difference depending upon the rate of evaporation and hence upon the amount of aqueous vapor present in the air.

Ques. How is the relative humidity obtained from the readings?

Ans. By referring them to tables.

Ques. How can accurate readings be obtained?

Ans. Only when the air is caused to pass very rapidly over the moistened wick.

Ques. How is this condition obtained?

Ans. By mounting the thermometer on a handle or chain for whirling. This assembly being known as a **sling psychrometer.**

To observe the wet and dry bulb temperatures of the air, the wet bulb is thoroughly saturated with clean water, preferably distilled. The instrument is then whirled at a rate of 100 or more *r.p.m.*

The whirling should be continued for a half or three-quarters of a minute, then stopped and read quickly, the wet bulb first. Record the wet and dry bulb readings and make, immediately, one or more subsequent observations to check.

The following precautions should be observed:

1. The wet bulb covering should be of clean, closely fitting fabric free from sizing or other foreign matter.

2. Do not touch wet bulb covering with oily fingers.

3. Use clean water, preferably distilled.

4. If air be in motion, face the breeze while making the observation.

5. Step from side to side, while whirling, to prevent body influence.

6. If observations be made out of doors, it is well to seek shade from direct sunlight.

7. Be sure the wet bulb has been cooled to the minimum.

8. The stationary wet and dry bulb hygrometer is frequently subject to an error greater than 20% of the wet bulb depression, and is not a reliable instrument.

Ques. What may be said of the stationary instrument mounted on the wall?

Ans. It's no good unless in a strong air current.

Ques. What is the effect of moisture in air upon an air compressor?

Ans. It decreases its efficiency.

Ques. Why?

Ans. The presence of water vapor in air being compressed materially increases the total heating capacity of the air because of the latent heat of the water vapor. The increased temperature increases the pressure and power required for compression.

Weight of Air.—According to the Smithsonian Institution physical tables, the weight of pure air at 32° Fahr. and a barometric pressure of 14.696 lbs. per sq. in., is .08071 lb. per cubic foot. The weight and volume of air change for variations in temperature and pressure.

Atmospheric Pressure.—By definition, atmospheric pressure is: *The force exerted by the weight of the atmosphere on every point with which it is in contact.*

Ques. How is the pressure of the atmosphere measured?

Ans. In inches of mercury barometer reading or the corresponding pressure in pounds per square inch.

Ques. What is the standard atmosphere?

Ans. 29.921 ins. of mercury which is equal to 14.696 lbs. per sq. in.

That is 1 in. of mercury = ***14.696*** ÷ ***29.921*** = ***.49116*** lb. per sq. in.

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.

Volume and Weight of Air

(at atmospheric pressure for different temperatures)

Temperature, Degrees Fahr.	Volume of 1 Pound of Air in Cubic Feet	Weight per Cubic Foot, Pounds	Temperature, Degrees Fahr.	Volume of 1 Pound of Air in Cubic Feet	Weight per Cubic Foot, Pounds
0	11.57	0.0864	325	19.76	0.0506
12	11.88	0.0842	350	20.41	0.0490
22	12.14	0.0824	375	20.96	0.0477
32	12.39	0.0807	400	21.69	0.0461
42	12.64	0.0791	450	22.94	0.0436
52	12.89	0.0776	500	24.21	0.0413
62	13.14	0.0761	600	26.60	0.0376
72	13.39	0.0747	700	29.59	0.0338
82	13.64	0.0733	800	31.75	0.0315
92	13.89	0.0720	900	34.25	0.0292
102	14.14	0.0707	1000	37.31	0.0268
112	14.41	0.0694	1100	39.37	0.0254
122	14.66	0.0682	1200	41.84	0.0239
132	14.90	0.0671	1300	44.44	0.0225
142	15.17	0.0659	1400	46.95	0.0213
152	15.41	0.0649	1500	49.51	0.0202
162	15.67	0.0638	1600	52.08	0.0192
172	15.92	0.0628	1700	54.64	0.0183
182	16.18	0.0618	1800	57.14	0.0175
192	16.42	0.0609	2000	62.11	0.0161
202	16.67	0.0600	2200	67.11	0.0149
212	16.92	0.0591	2400	72.46	0.0138
230	17.39	0.0575	2600	76.92	0.0130
250	17.89	0.0559	2800	82.64	0.0121
275	18.52	0.0540	3000	87.72	0.0114
300	19.16	0.0522	....	.....	.....

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

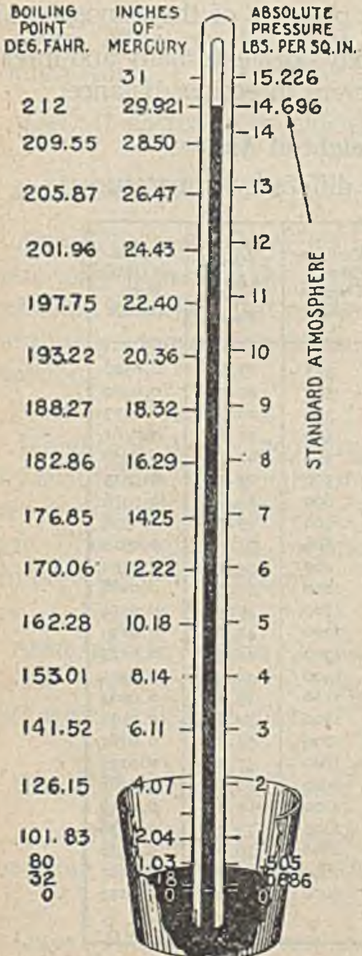


Fig. 2.—Mercurial barometer illustrating the relation between "inches of mercury" and absolute pressure in lbs. per sq. in.

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.

Ques. How?

Ans. It decreases approximately one half pound for every 1000 feet ascent.

The following table of atmospheric pressure at various altitudes, according to the National Advisory Committee for Aeronautics, is here given.

Atmospheric Pressure and Barometer Readings.

(For various altitudes)

Altitude above Sea Level Feet	Atmospheric Pressure Lbs. Per Square Inch	Barometer Reading Inches of Mercury
0	14.69	29.92
500	14.42	29.38
1000	14.16	28.86
1500	13.91	28.33
2000	13.66	27.82
2500	13.41	27.31
3000	13.16	26.81
3500	12.92	26.32
4000	12.68	25.84
4500	12.45	25.36
5000	12.22	24.89
5500	11.99	24.43
6000	11.77	23.98
6500	11.55	23.53
7000	11.33	23.09
7500	11 12	22.65
8000	10.91	22.22
8500	10.70	21.80
9000	10.50	21.38
9500	10.30	20.98
10000	10.10	20.58
10500	9.90	20.18
11000	9.71	19.75
11500	9.52	19.40
12000	9.34	19.03
12500	9.15	18.65
13000	8.97	18.29
13500	8.80	17.93
14000	8.62	17.57
14500	8.45	17.22
15000	8.28	16.88

Source—National Advisory Committee for Aeronautics—report No. 246.

Fig. 2 shows the relation between barometer readings and absolute pressure.

Gauge and Absolute Pressure.—When the hand of a steam or air gauge is at zero, the pressure actually existing is 14.696 lbs. per sq. in. (referred to a 29.921 inch barometer, or that of the atmosphere).

Ques. Why then does the hand of the gauge register zero?

Ans. Because in the steam boiler as well as any other vessel under pressure, the important measurement is the *difference* of pressure between the inside and outside.

Ques. What is this difference of pressure or effective pressure for doing work called and why?

Ans. Gauge pressure because it is measured by the gauge on the boiler or other pressure vessel, as in fig. 3.

Ques. What is absolute pressure?

Ans. Pressure measured from the true zero or point of no pressure.

It is equal to the pressure indicated upon the gauge plus that of the atmosphere which is ordinarily taken as 14.7 lbs. Thus 68 lbs. ***gauge*** pressure is 68 + 14.7 = 82.7 lbs. (per sq. in.) ***absolute.***

Ques. How is absolute pressure expressed as gauge pressure?

Ans. By subtracting 14.7.

Thus 82.7 lbs. ***absolute*** = 82.7 − 14.7 = 68 lbs. ***gauge.***

Pressures Below Atmospheric.—Absolute pressures below atmospheric 14.7 lbs. per sq. in. are expressed either in lbs. per sq. in. (absolute) or the equivalent in inches of mercury.

Ques. What is the meaning of the term referred to a 30 inch barometer?

Ans. It means that the variable pressure of the atmosphere is in value such that it will cause the mercury in the barometer to rise 30 ins.

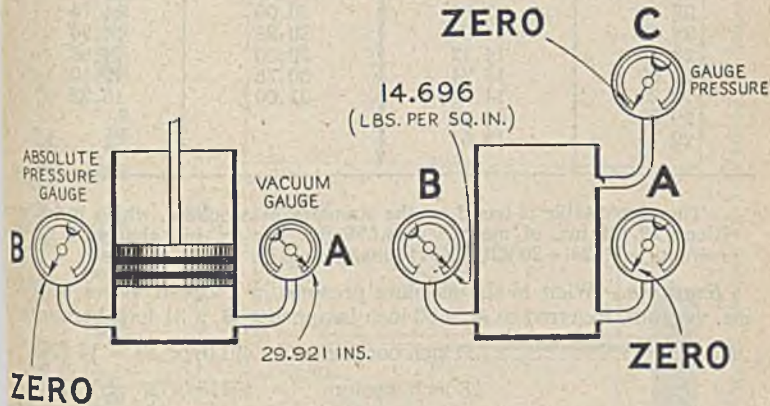


Fig. 3.—Elementary diagrams illustrating the difference between **gauge** and absolute **pressure. A,** vacuum gauge. **B,** absolute pressure gauge.

Ques. How is the pressure in lbs. per sq. in. obtained from the barometer reading?

Ans. **Rule:** Barometer reading in ins. $\times$ .49116 = pressure per sq. in.

Thus, a 30 inch barometer reading signifies a pressure of

$$.49116 \times 30 = 14.75 \text{ lbs. per sq. in.}$$

The following table gives the pressure of the atmosphere in lbs. per sq. in. for various readings of the barometer.

Barometer — Pressure

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.

Example.—What is the absolute pressure in a closed vessel with 28 ins. vacuum. Referred to **A**, a 30 inch barometer; **B**, a 31 inch barometer.

A.

30 inch barometer = .49116 × 30 = 14.735

28 inch vacuum = .49116 × 28 = 13.753

Absolute pressure in vessel referred to 30 inch barometer = .98 lb.

That is when the barometer reads 30 inches and there is a 28 inch vacuum in the closed vessel the difference in pressure between that of the atmosphere and that in the tank is .98 lbs. Another way of calculating it is as follows:

30 in. − 28 in. = 2 in.

Now since 1 inch of mercury = .49116 lb. pressure per square inch

2 × .49116 = .98 lb.

B.

31 inch barometer = .49116 × 31 = 15.226

28 inch vacuum = From **A** = 13.752

Absolute pressure in vessel referred to 31 inch barometer = 1.474 lbs.

The Meaning of

"Referred To A 30 In. Barometer."

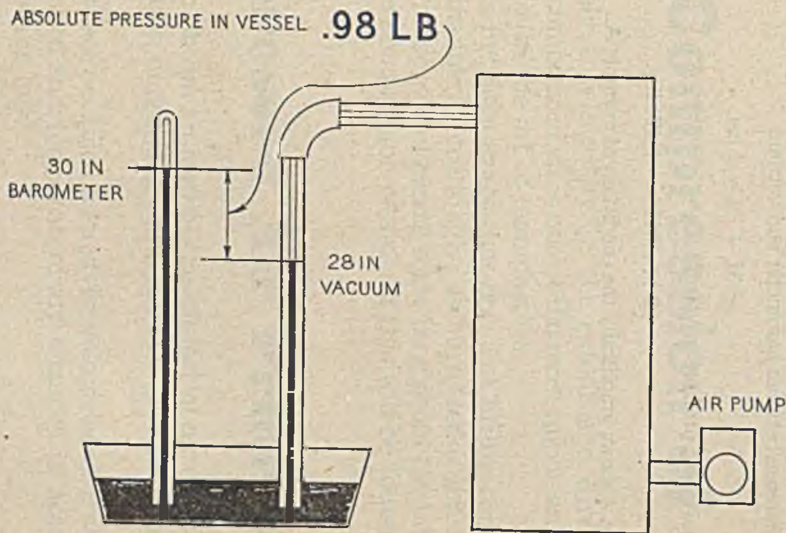


Fig. 4.—Absolute pressure in a closed vessel due to a 28-inch vacuum referred to a 30-inch barometer.

Using the shorter method of calculation—

Pressure in vessel referred to 31 inch barometer

$$= 3 \times .49116 = 1.474 \text{ lbs.}$$

B.

Difference between barometer and vacuum

$$= 31 - 28 = 3 \text{ ins.}$$

Equivalent pressure in vessel

$$= 3 \times .49116 = 1.474 \text{ lbs.}$$

Fig. 4 shows graphically the conditions of example **A.**

CHAPTER 44

The Compression of Air

Second only to electricity in the extent and diversity of application, compressed air is one of the most important factors in every phase in the art of manufacture.

Scarcely an industry exists that does not utilize compressed air in some manner. Accordingly, as an introduction to compressors, the reader should learn something about the basic laws governing the compression of air. All this will be found in this chapter.

Compressed Air Principles

In order to obtain an idea of the subject of air compression, there are certain underlying principles and laws that should be understood.

When the space occupied by a given volume of air is changed, both its pressure and temperature are changed in accordance with the following laws:

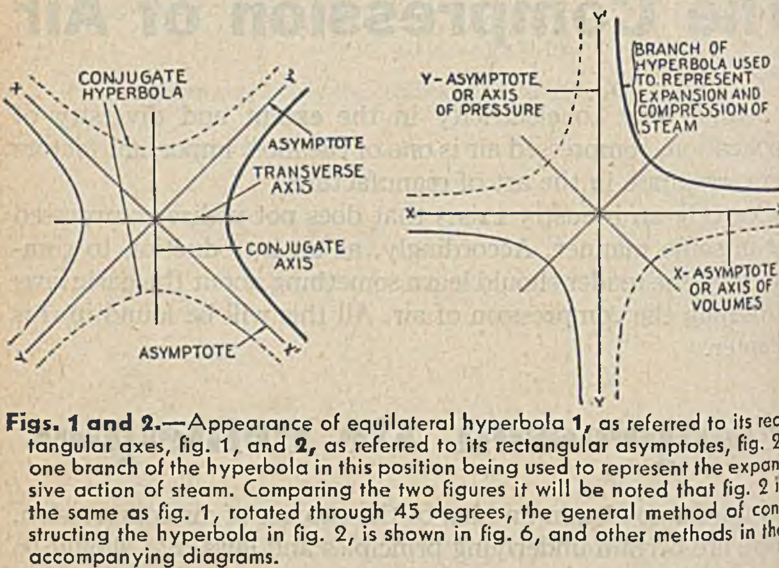
1. Boyle's law
2. Charles' law
3. Joules' law
4. Gay Lussac's law.

Ques. What is compressed air?

Ans. Air forced into a smaller space than it originally occupied.

Ques. What happens when air is compressed?

Ans. Both its pressure and temperature rise.



Figs. 1 and 2.—Appearance of equilateral hyperbola **1**, as referred to its rectangular axes, fig. 1, and **2**, as referred to its rectangular asymptotes, fig. 2, one branch of the hyperbola in this position being used to represent the expansive action of steam. Comparing the two figures it will be noted that fig. 2 is the same as fig. 1, rotated through 45 degrees, the general method of constructing the hyperbola in fig. 2, is shown in fig. 6, and other methods in the accompanying diagrams.

Ques. What is the object of compressing air?

Ans. The power available from compressed air is used in many applications as a substitute for steam, electricity or other force as in operating shop tools, rock drills, etc.

Ques. What happens when the space occupied by a given volume of air is enlarged?

Ans. Both its pressure and temperature are lowered.

Ques. What is the enlargement of volume of compressed air called?

Ans. Expansion.

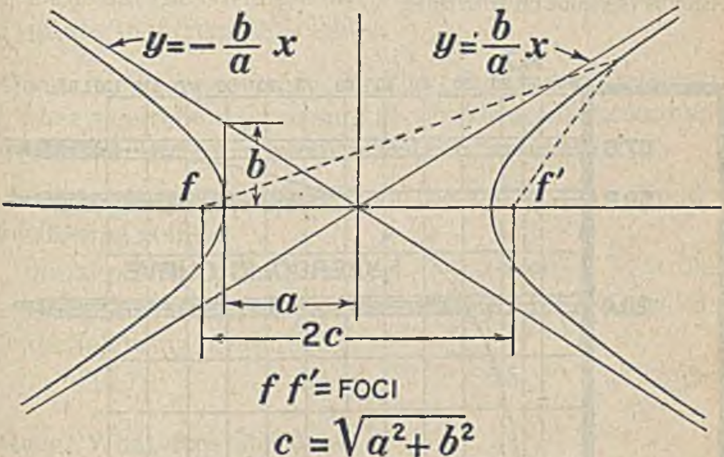


Fig. 3.—The hyperbola. By definition, a curve generated by a point moving so that the difference of its distances from two fixed points is always constant.

Ques. Name an important process whose operation depends upon the cooling of a gas by expansion

Ans. Refrigeration.

BOYLE'S LAW

Boyle's Law.—*At constant temperature, the* **absolute** *pressure of a gas varies* **inversely** *as its volume.*

Boyle's law may be illustrated by the following experiment. In fig. 4 is shown a cylinder having a piston sliding air tight in its length.

If air be compressed in front of the piston as it is forced from one end toward the other, the pressure exerted by the air will increase in ratio as the volume is diminished (assuming constant temperature).

This fact may be shown by inserting in the wall of the cylinder at different points a number of tubes, each provided with an air tight piston upon which bears a helical spring holding it, as at **A**, when the pressure on the piston is the same on both sides.

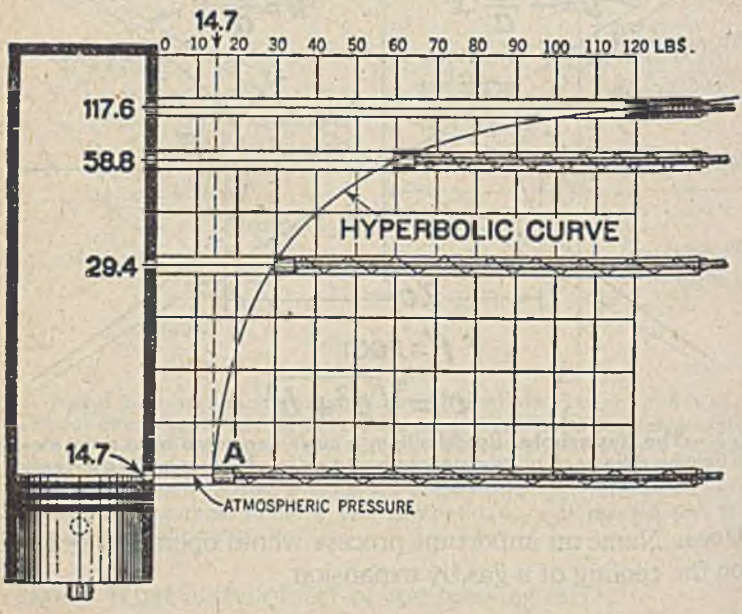


Fig. 4.—Experiment illustrating Boyle's law. This law was discovered by Robert Boyle. Now when the piston moves in the cylinder, the pressure will gradually rise due to the compression of the air and the small pistons will move against the tension of the springs to increasing distances. As the piston moves from the end of the cylinder to the following points: initial position, ½, ¾ and ⅞ stroke the positions of the small pistons as shown in the figure will indicate the following pressures: 14.7 lbs., 29.4 lbs., 58.8 lbs., 117.6 lbs. If a curve be drawn so as to pass through the center of each of the small pistons, it will show the pressure corresponding to every position of the large piston.

The area of each small piston is assumed to be one square inch and the springs of such tension that the pistons will move upward through one of the spaces between the horizontal lines on the diagram with each 10 lbs. of added pressure in the large cylinder.

Ques. What happens when the piston moves in the cylinder?

Ans. The pressure will gradually rise due to the compression of the air and the small pistons will rise against the tension of the springs to increasing heights.

Ques. As the piston moves in the cylinder compressing the air, what would be the pressure at various points according to Boyle's law—that is, assuming constant temperature.

Ans. As the piston moves from the end of the cylinder to the following points:

initial position	½ stroke	¾ stroke	⅞ stroke

the positions of the small pistons (as shown in fig. 1) will indicate the following pressures:

14.7	29.4	58.8	117.6 lbs.

Ques. What does this show?

Ans. It shows under the conditions of Boyle's law that if the volume be diminished by half, the pressure is doubled.

Ques. How can the pressure at all points be shown?

Ans. If a curve be described so as to pass through the center of each of the small pistons, it will show the pressure corresponding to every position of the large piston. See fig. 4.

Ques. What is this curve called and why?

Ans. The hyperbolic curve because it is an equilateral or rectangular hyperbola, referred to its rectangular asymptotes.

Ques. Give two definitions of a hyperbola.

Ans. 1. *A curve formed by a plane cutting a cone at any angle with its base* **greater** *than that of its side.* 2. *A curve generated by*

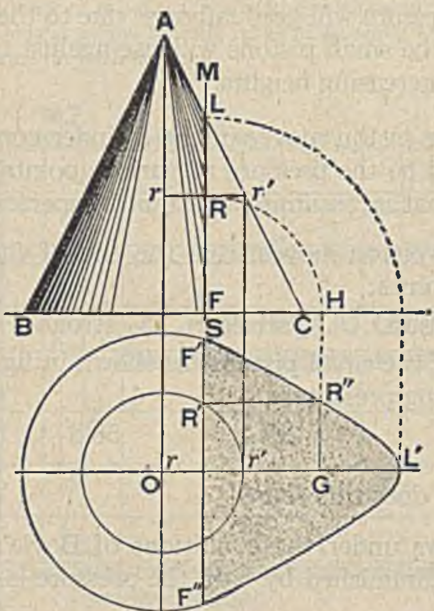


Fig. 5.—Hyperbola or intersection with the surface of a plane parallel with the axis. Construction of the hyperbolic curve for compression at constant temperature, that is isothermal compression. Let plane MS, cut element AC, at L, and base at F. Project F, down to plan cutting base at F′ and F″, which are two points in the curve. With F, as center and radius FL, swing point L, around and project down to axis of plan obtaining point L′, in the curve. Now any other point as R, may be obtained as follows: Swing R, around with F, as center and project down to plan with line HG. Describe a circle in plan with radius rr′ (= radius of cone at elevation of point R) and where this circle cuts the projection of R at R′, project over to line HG, and obtain point R″, which is a point in the curve. Other points may be obtained in a similar manner. The curve is traced through points F′,R″,L′, and similar points on the other side of the axis ending at F″. Such curve is called a hyperbola.

a point moving so that the **difference** *of its distances from two fixed points* (*called the* **foci**) *is always constant.*
See accompanying illustrations of the hyperbola.

The Hyperbolic Curve for Compression.—*This is the curve which gives values for compression according to Boyle's law—that is,* **at constant temperature.**

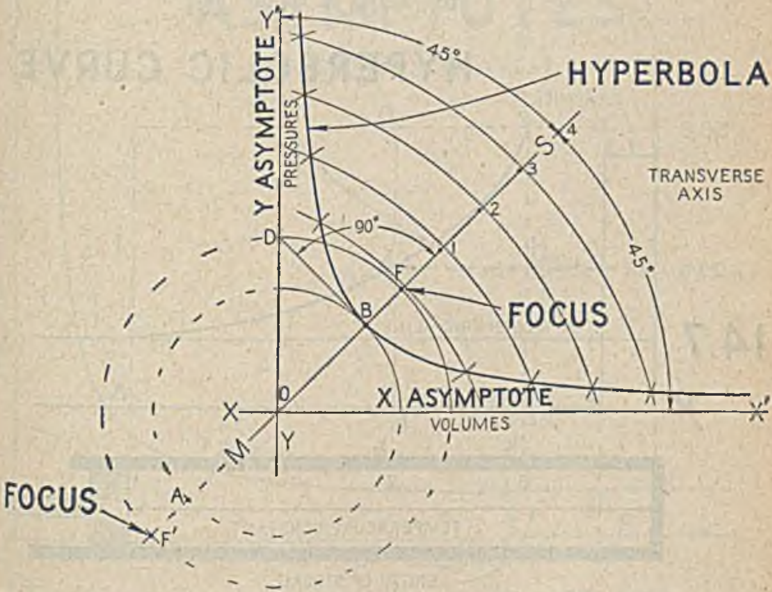


Fig. 6.—To describe an equilateral or rectangular hyperbola referred to its rectangular asymptotes. **General method:** Draw the axis of volumes, or horizontal asymptote X X′, and the axis of pressures, or vertical asymptote Y Y′, cutting X X′, at O, or hyperbolic center. Through O, draw M S, at 45° to X X′. Take any point on M S, as B, and with radius O B, describe a circle, cutting M S, in B and A, giving A B, the transverse axis. At B, erect a perpendicular cutting Y Y′, at D, giving O D, the directrix. With O D, as radius describe a circle cutting M S, at F and F′; these points are the foci of the hyperbola. On M S, take any number of points 1, 2, 3, etc., and from F, and F′ as centers, with A1, A2, etc., as radii describe arcs, cutting the other arcs in points as shown which are points on the curve.

In fig. 7 draw the zero or vacuum line AB (any convenient scale) = length of stroke or volume displaced by the piston and extend it to C, making BC of length = clearance, that is, if the clearance volume be say 8% of piston displacement, then length of BC = 8% of AB or .08 × AB. Now with piston at position 1, and cylinder filled with air at atmospheric pressure (14.7 lbs. per sq. in.) let piston move from position.

1 to 2 to 3

reducing the volume of the air from

1 to ½ to ¼

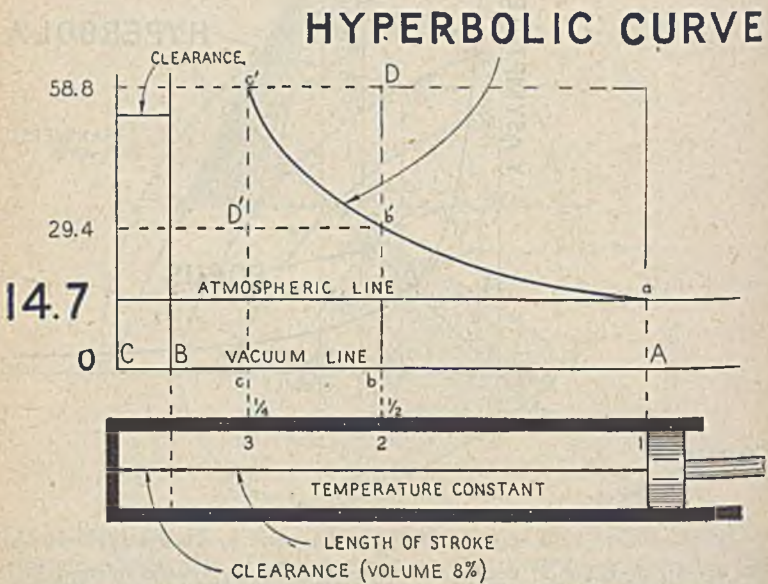


Fig. 7.—The hyperbolic curve for compression constructed from the data: Initial pressure = 14.7 lbs. per sq. in.; compression ratio = 4. Clearance 8%.

According to Boyle's law, *the pressure and volume vary inversely* (*at constant temperature*).

Accordingly erect dotted perpendiculars at piston positions 2 and 3 and lay off $bb' = 2\ Aa$ and $cc' = 4Aa$. Through the points a, b', c', thus found describe a curve which is the hypobolic curve for compression of air according to Boyle's law.

Ques. What kind of a hyperbola is this curve?

Ans. An equilateral hyperbola.

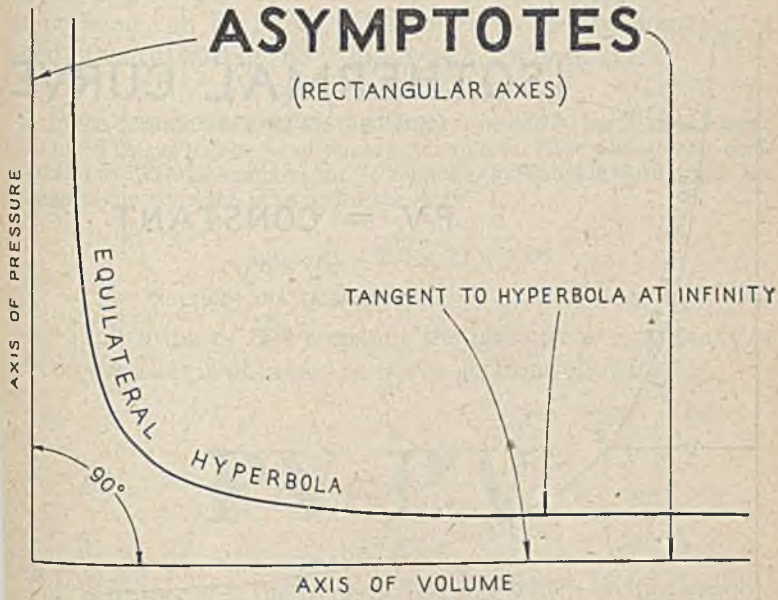


Fig. 8.—The equilateral hyperbola referred to its rectilinear asymptotes.

Ques. What is an equilateral hyperbola?

Ans. A hyperbola whose asymptotes are perpendicular to each other as in fig. 8.

Ques. What is an asymptote?

Ans. A right line which an infinite branch of the hyperbolic curve continually approaches, but never touches.

In other words a tangent to the curve at infinity.

Ques. What name is generally given to the hyperbolic curve for compression and why?

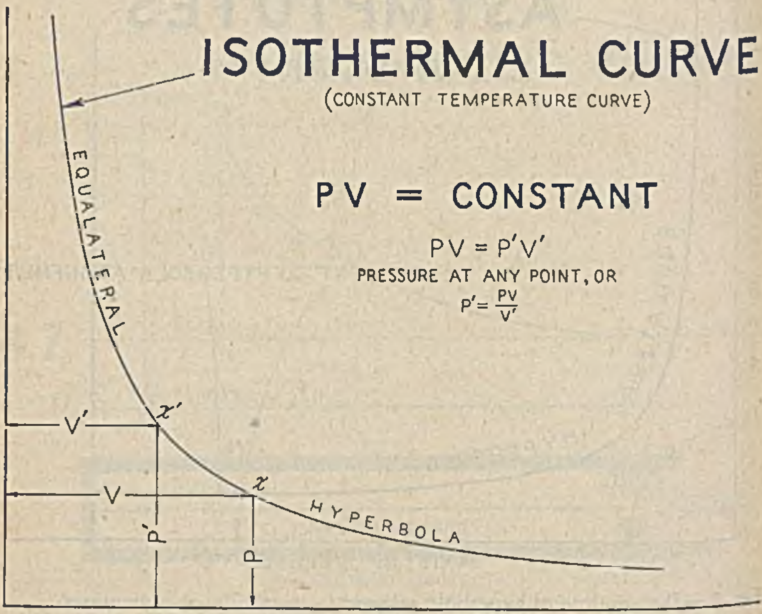


Fig. 9.—The equilateral hyperbola any point on which as x, gives PV = constant P being pressure and V volume. PV = P′V′.

Ans. It is usually called the isothermal curve (fig. 9) because the compression takes place at constant temperature.

This would not be possible in practice unless the heat due to the work done on the air in compressing it were removed as fast as it was generated.

Compression Constant.—According to Boyle's law:

pressure × volume = constant............(1)

If air be compressed to one-third volume and the final pressure of compression be say 100 lbs. absolute, then by application of the compression constant the pressure at any point of the compression can be found. Thus in this case the constant is found by substituting the values which apply in (1).

In the diagram fig. 10 divide the volume scale into 30 parts, then volume 10 (solid black) will be ⅓ of the original volume. Now with the air compressed to 100 lbs. (absolute) and ⅓ volume as indicated in the diagram the compression constant is, substituting in (1)

$$\textit{constant} = 100 \times 10 = 1000$$

By application of this constant the pressure at any point of the compression is obtained as in the section following.

PV=P′V′

This is property of the hyperbolic curve which defines graphically Boyle's law in which *the product of the pressure times volume at any point is equal to the product of the pressure times the volume at any other point. Thus in fig.* 9,

$$PV = P'V' \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

in which

P = pressure at any point *x*

V = volume at any point x

P′ = pressure at any other point x'

V′ = volume " " " " "

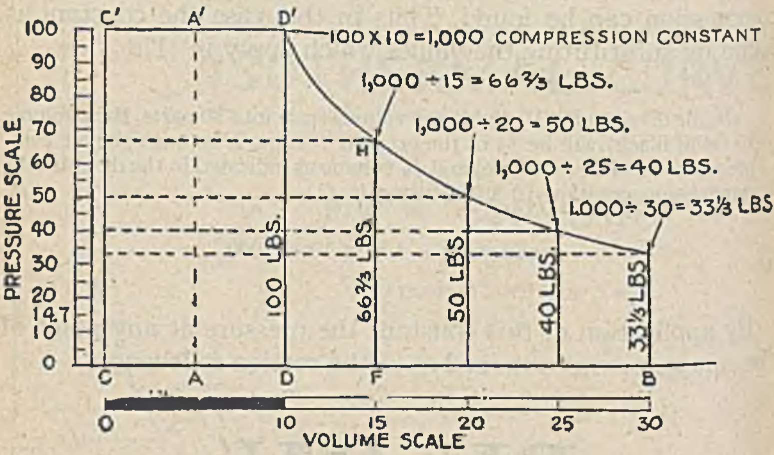


Fig. 10.—Diagram illustrating the **expansion constant** and its use. According to **Boyle's law,** pressure × volume = constant. If, as indicated in the diagram, steam be admitted to a cylinder during 10 inches of the stroke and expanded to 30 inches, the expansion constant = 100 × 10 = 1,000, from which the pressure at any other point = constant ÷ volume, that is, when the piston is at

15 ins.	20 ins.	25 ins.	30 ins.

of the stroke, the expansion constant ÷ volume is

1,000 ÷ 15	1,000 ÷ 20	1,000 ÷ 25	1,000 ÷ 30

which is equal to

66⅔ lbs.	50 lbs.	40 lbs.	33⅓ lbs.

Similarly volume = constant ÷ pressure, that is, when the pressure due to the expansion is

66⅔ lbs.	50 lbs.	40 lbs.	33⅓ lbs.

the expansion constant ÷ pressure is

1,000 ÷ 66⅔	1,000 ÷ 50	1,000 ÷ 40	1,000 ÷ 33⅓

which is equal to

15 ins.	20 ins.	25 ins.	30 ins.

Now to find the pressure at any point solve equation 2 for P′; thus,

$$P' = \frac{PV}{V'} \text{.......................} (3)$$

but from equation (1) PV = constant hence equation 3 may be written

$$P' = \text{constant} \div \text{volume} \text{..............} (4)$$

Referring to fig. 10 and applying (4) when the piston is at

15 in.	20 in.	25 in.	30 in.

from the end of the compression stroke the compression constant or *constant ÷ volume* is

1000 ÷ 15	1000 ÷ 20	1000 ÷ 25	1000 ÷ 30

which is equal to

66⅔ lbs.	50 lbs.	40 lbs.	33⅓ lbs.

For a given pressure of compression to find the volume or degree of compression solve equation 3 for V thus

$$P' = \frac{PV}{V'}$$

from which

$$V = \frac{P'V'}{P} = \frac{\text{constant}}{P} \text{................} (5)$$

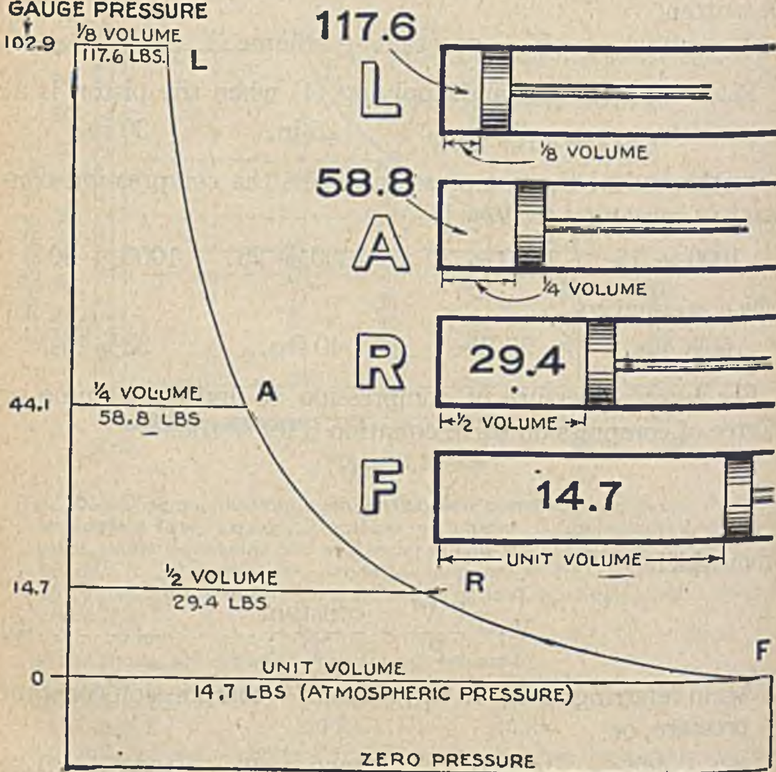
Again referring to fig. 10 the volume = compression constant ÷ pressure, or

1000 ÷ 66⅔	1000 ÷ 50	1000 ÷ 40	1000 ÷ 33⅓

which is equal to

15 ins.	20 ins.	25 ins.	30 ins.

Final Pressure of Compression.—If air be compressed from the beginning of the compression stroke to the end of that stroke, the pressure at the end of that stroke or final pressure of compression as it is called, will depend on the initial pressure absolute and the ratio of compression.



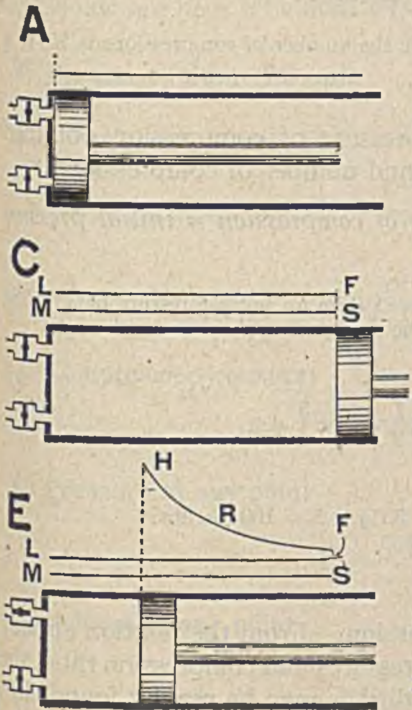
Figs. 11 to 15.—Diagram and compression stroke progressively shown illustrating **Boyle's law.** As the piston travels from position **F,** to **R, A, L,** the pressures are 14.7, 29.4, 58.8, 117.6 lbs. absolute, respectively, being inversely proportional to the volume. The points **F, R, A, L,** on the compression curve correspond to the piston positions **F, R, A, L** as shown.

Ques. Upon what does the ratio of compression depend?

Ans. Upon the final volume of compression.

Ques. What is the ratio of compression?

Ans. The ratio of initial volume to final volume.



Figs. 16 to 21.—Compression cycle illustrated by indicator diagrams and elementary compressor. **A,** beginning of intake stroke; **B,** intermediate position of intake stroke; **C,** end of intake stroke, note atmospheric intake line **LF,** and vacuum line **MS; D,** intermediate position **R,** of compression stroke; **E,** point of maximum compression, note compression curve **FRH; F,** end of cycle, note horizontal discharge line **HD,** indicating discharge into receiver at constant pressure.

Example.—If the final volume be 10% what is the ratio of compression?

$$\text{ratio} = \text{initial volume} \div \text{final volume}$$
$$= 100 \div 10 = 10$$

Ques. What other name is given to the ratio of compression?

Ans. The number of compressions.

Thus in the preceding example the number of compressions is 10, that is 10 compressions.

Ques. How is the final pressure of compression obtained knowing the initial pressure and number of compressions?

Ans. **Rule:** *Final pressure of compression = initial pressure × compression ratio.*

Example.—If air be compressed from an initial pressure of 33⅓ lbs. absolute to ⅓ volume what is the final pressure.

Here the compression ratio

$$= 1 \div \frac{1}{3} = 1 \times \frac{3}{1} = 3$$

Applying rule

$$\text{final pressure} = 33\tfrac{1}{3} \times 3 = 100 \text{ lbs. abs.}$$

Compare this with fig. 10.

Initial Pressure of Compression.—From the relation already given between final, initial pressures and compression ratio, the initial pressure for given conditions may be readily found thus

$$\text{final pressure} = \text{initial pressure} \times \text{ratio}$$

from which

$$\text{initial pressure} = \text{final pressure} \div \text{ratio} \ldots\ldots (6)$$

Example.—If the final pressure of compression be 100 lbs. absolute and number of compressions or ratio be 3, what is the initial pressure?

Substituting in (6)

$$\text{initial pressure} = 100 \div 3 = 33\tfrac{1}{3} \text{ lbs.}$$

Summary: Boyle's Law.—The following relations based on Boyle's law are here tabulated for convenient reference.

1. **Pressure volume formula**

$$PV = P'V' \quad \ldots\ldots (1)$$

$$P = \frac{P'V'}{V} \quad \ldots\ldots (2)$$

$$V = \frac{P'V'}{P} \quad \ldots\ldots (3)$$

2. **Compression constant**

$$PV = \text{constant} \quad \ldots\ldots (4)$$

3. **Pressure at any point**

$$P = \frac{\text{constant}}{V} \quad \ldots\ldots (5)$$

4. **Volume at any point**

$$V = \frac{\text{constant}}{P} \quad \ldots\ldots (6)$$

5. **Ratio of Compression**

$$R = V_i \div V_f \quad \ldots\ldots (7)$$

NOTE.—In all the foregoing the values given as "lbs." are, of course, intended to be "lbs. per sq. in."

in which

R = ratio or number of compression
V_i = initial volume
V_f = final volume
$R = P_i \div P_f$ ....(8)

6. Initial pressure of compression

$P_i = R \div P_f$ ....(9)

in which

P_i = initial pressure absolute
P_f = final " "

7. Final pressure of compression

$P_f = R \div P_i$

CHARLES' LAW

Charles' Law.—*At* **constant pressure,** *the volume of a gas is* proportional to its **absolute** temperature; **at constant** volume, *the pressure is proportional to its absolute temperature.*.

The conditions expressed as a formula are:

1. At constant volume

$$\frac{P}{T} = \frac{P'}{T'} \quad \text{.......................(1)}$$

in which

P = initial pressure absolute
P' = final " "
T = initial temperature absolute
T' = final " "

*NOTE.—Charles law is also called Gay Lussac's law, but is generally known as Charles' law.

2. At constant pressure

$$\frac{V}{T} = \frac{V'}{T'} \quad \text{.............................} (2)$$

in which

V = initial volume (usually in cu. ft.)
V′ = final " " " " "

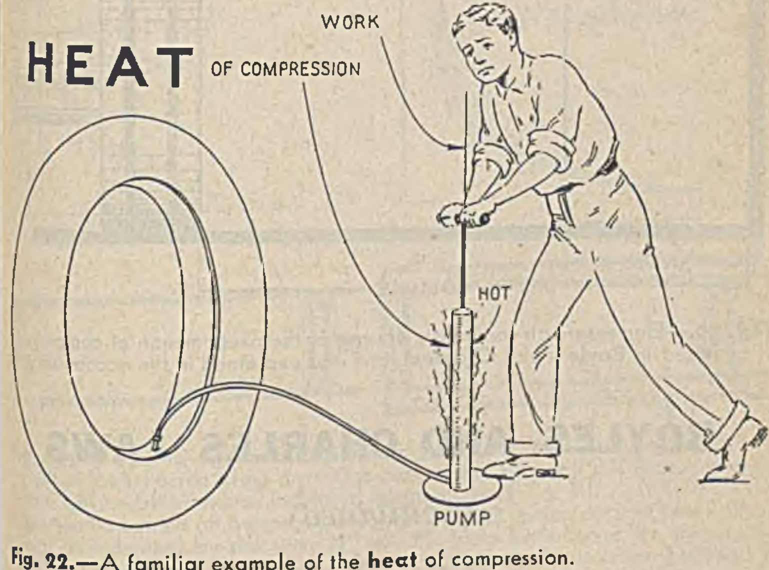


Fig. 22.—A familiar example of the **heat** of compression.

When Air is Compressed Heat is Generated.—This is illustrated by the familiar operation of pumping up a bicycle or automobile tire as an fig. 22—and this is the big problem in the design of compressors to get rid of the heat.

If this heat could be removed as fast as it is generated so as to maintain the air at constant temperature during compression,

isothermal compression would be obtained corresponding with **Boyle's law**, as already explained and represented by the hyperbolic or isothermal curve, fig. 9. *However, no such result is obtained in practice because it's* **impossible** *to remove all the heat as fast as generated.*

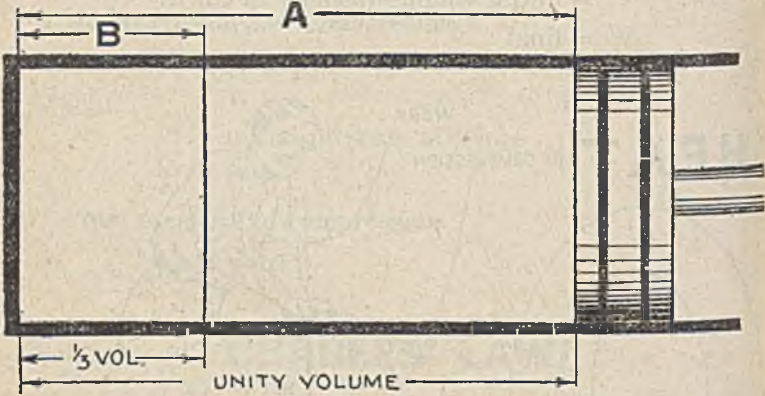


Fig. 23.—Elementary air compressor illustrating the phenomenon of compression as stated in **Boyle's** and **Charles'** laws and explained in the accompanying text.

BOYLES AND CHARLES LAWS

(*Combined*)

In the ordinary process of air compression two elements are at work toward the production of a higher pressure.

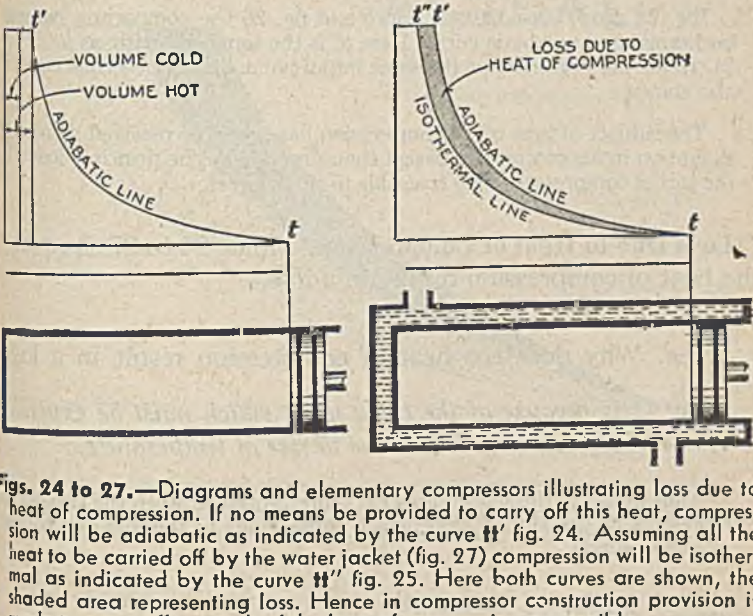
1. The reduction of volume by the advancing piston.

2. The increasing temperature due to the increasing pressure corresponding to the reduced volume.

The application of the two laws is illustrated in fig. 23 which shows a cylinder fitted with an air-tight piston.

If the cylinder be filled with air at atmospheric pressure (14.7 lbs. per sq. in. absolute) represented by volume **A**, and the piston be moved to reduce the volume to say ⅓ **A**, as represented by **B**, then according to ***Boyle's law,*** the pressure will be trebled or = 14.7 × 3 = 44.1 lbs. absolute or 44.1 − 14.7 = 29.4 ***gauge*** pressure.

In reality, here is where **Charles' law** comes in, *a pressure gauge on the cylinder* would at this time show a *higher pressure*



Figs. 24 to 27.—Diagrams and elementary compressors illustrating loss due to heat of compression. If no means be provided to carry off this heat, compression will be adiabatic as indicated by the curve **tt'** fig. 24. Assuming all the heat to be carried off by the water jacket (fig. 27) compression will be isothermal as indicated by the curve **tt"** fig. 25. Here both curves are shown, the shaded area representing loss. Hence in compressor construction provision is made to carry off as much of the heat of compression as possible.

than 14.7 **gauge** pressure *because of the* **increase in temperature** *produced in compressing the air.* This is *called* **adiabatic** *compression.*

Ques. What is adiabatic compression?

Ans. Compression without receiving or giving up heat.

Ques. What is an adiabatic curve?

Ans. A curve similar to the hyperbolic curve but having longer ordinates, that is the curve lies higher than the isothermal curve.

Fig. 24 shows an adiabatic curve and fig. 26 the comparison between isothermal and adiabatic curve. Here *tt'* is the same adiabatic as *tt'* in fig. 24. In fig. 26, beginning at the same initial point *t*, the isothermal curve is also shown.

The subject of heat of air compression has probably received more consideration in air compressor design than any other. The principal losses in the earlier compressors were traceable to this source.

Loss Due to Heat of Compression.—Figs. 24 to 27 show why the heat of compression results in a loss.

Ques. Why does the heat of compression result in a loss?

Ans. It is because of the extra work which must be expended to overcome the excess pressure due to rise of temperature.

Thus in fig. 26 the shaded area *t'l t''* intercepted between the two curves represents an amount of work which corresponds to the loss due to heat of compression.

Ques. Why is this excess work lost?

Ans. Because after the compressed air leaves the cylinder it **cools** *and the pressure* **drops** *to what it would have been if compressed at* **constant** *temperature.*

Ques. Is all this work lost in the operation, and why?

Ans. No, because in compressors where working efficiency is considered, some means of cooling the cylinder is provided such as projecting fins or jackets for the circulation of cooling water.

Ques. How much work is lost?

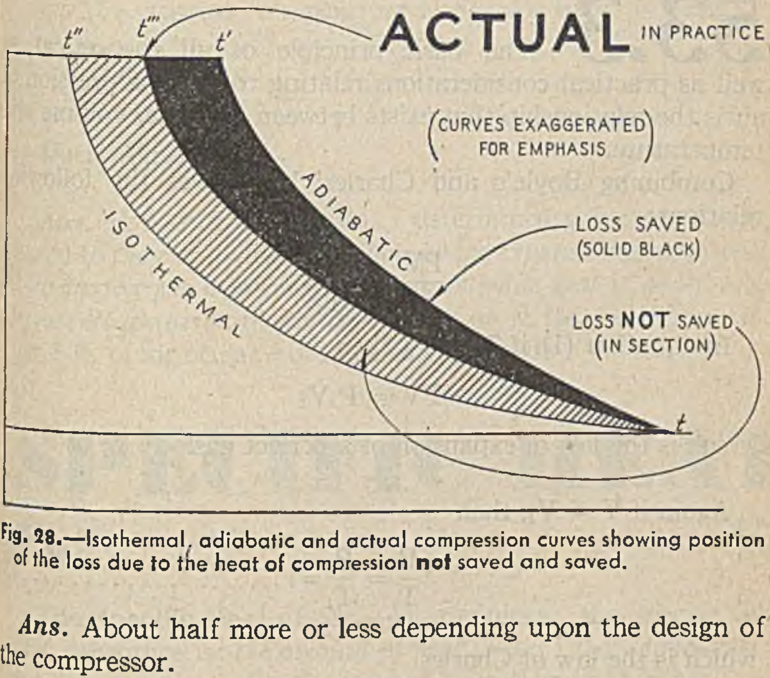


Fig. 28.—Isothermal, adiabatic and actual compression curves showing position of the loss due to the heat of compression **not** saved and saved.

Ans. About half more or less depending upon the design of the compressor.

This is shown in fig. 28. Here to bring out the conditions clearly the curves are greatly exaggerated. Lying somewhere between the isothermal and adiabatic curves is a curve as tt''' which corresponds to the ***actual*** results obtained in practice. Of course this curve tt''' is different with each compressor and will divide the total area $t'tt''$ into two areas $t''tt'''$ (sectioned) and $t'''tt'$ (solid block) in various proportions depending upon the efficiency of the compressor.

Ques. What should be noted about these areas?

Ans. The solid black area $t'''tt'$ represents part of the loss saved due to cooling. The sectional area $t''tt'''$ represents that part of the loss not saved.

53.3—The basic principle of all theoretical as well as practical considerations relating to the compression of air is the relationship that exists between pressure, volume and temperature.

Combining Boyle's and Charles' laws gives the following relations:

$$\frac{PV}{T} = \frac{P_1V_1}{T_1} \quad \text{.....................(1)}$$

In equation (1) if $T = T_1$, then

$$PV = P_1V_1$$

which is the law of expansion of a perfect gas.

Again if $V = V_1$, then

$$\frac{P}{T} = \frac{P_1}{T_1} = \quad \text{.....................(2)}$$

which is the law of Charles.

The relationship between pressure, volume and temperature of a perfect gas is given by the following equation:

$$\frac{PV}{T} = R \quad \text{.....................................(3)}$$

in which

P = absolute pressure in lbs. per sq. ft.

V = volume in cu. ft. of one lb. of air at the given pressure and temperature

T = absolute temperature in degrees Fahr. = 459.6

R = constant

Ques. What is R?

Ans. R is an experimentally determined constant which is equal to *the mechanical work done by the expansion of unit weight of a perfect gas at a constant pressure while heat is added to increase its temperature one degree Fahr. = ft. lbs. per degree Fahr. for* 1 *lb.* of air or gas = **53.3** later explained.

SPECIFIC HEAT

C_p C_v

The Specific Heat of Air.—By definition, the specific heat of a substance is: *the amount of heat (B.t.u.) that is required to raise the temperature of* 1 *pound of the substance* 1° *Fahr.*

There are two kinds of specific heat of air:

1. Specific heat at *constant pressure.* Symbol C_p.

2. Specific heat at *constant volume.* Symbol C_v.

The specific heat at constant pressure is the *total* specific heat and is made up of

1. The internal work of raising the temperature of the air.
2. The external work of pushing away the atmosphere to make room for its expansion.

EXTERNAL WORK

53.3 ft. lb. .0686 *B.t.u.*

External Work at Constant Pressure.—Fig. 30 shows a vertical cylinder with a frictionless air-tight piston whose area = 1 sq. ft. or 144 sq. in. When the cylinder contains 1 lb. of air at armospheric pressure the height of the piston = 12.39 ft., that is 12.39 cu. ft. of air at atmospheric pressure weigh 1 lb.

If the air be heated the volume will increase, moving the piston upward and external work is performed in pushing away the surrounding atmosphere.

Air when heated 1° Fahr. will increase in volume $\frac{1}{491.6}$ of its original volume.

Accordingly, movement of piston due to heating the air 1° Fahr.

$$= 12.39 \times \frac{1}{491.6} = .0252 \text{ ft.}$$

The total pressure of the atmosphere on the upper face of the piston

$$= \begin{cases} \text{Area piston} \times \text{pressure of atmosphere} \quad \text{load} \\ 144 \quad \times \quad 14.7 \quad = 2116.8 \text{ lbs.} \end{cases}$$

External work = $\left\{ \begin{array}{l} \text{Total pressure} \times \text{distance moved} \\ 2116.8 \quad \times \quad 0.252 \quad = \mathbf{53.3} \text{ ft. lbs.} \end{array} \right.$

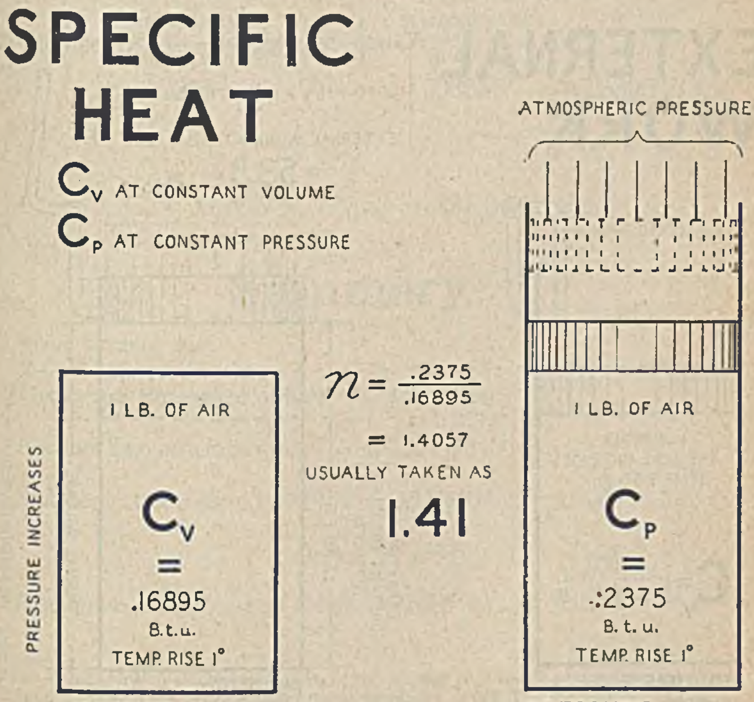
or

$$53.3 \div 777.52 = .06855 \ \textit{B.t.u.}$$

Commonly taken as .0686.

For emphasis:

53.3 foot pounds or **.0686** *B.t.u*



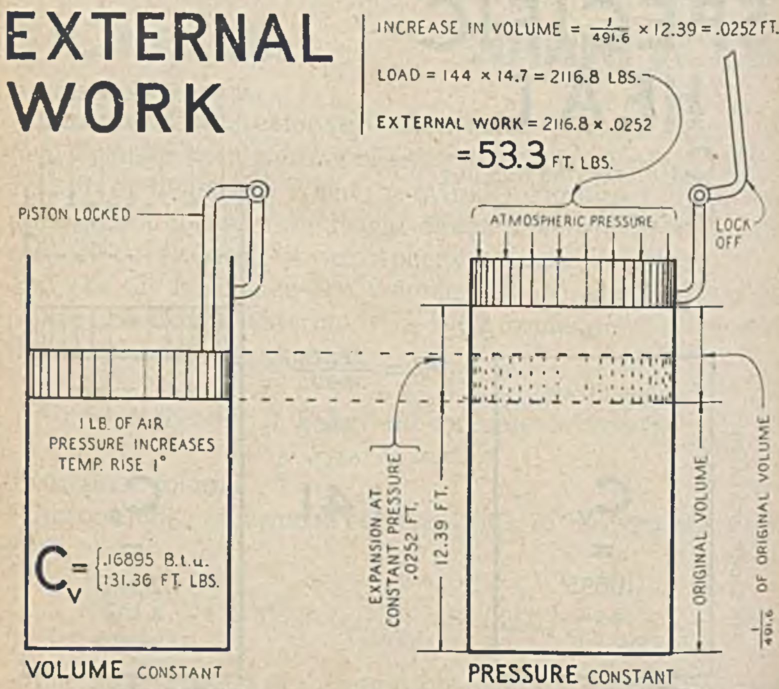
Figs. 29 and 30.—Diagrams illustrating specific heat at constant **volume** C_v and specific heat at constant pressure C_p. This explains how they get **n = 1.41** the exponent for adiabatic compression.

The total specific heat or specific heat at constant pressure which, as stated, includes both the internal and external heat, has been found by Regnault to be

$$.2375 \ B.t.u.$$

or

$$.2375 \times 777.52 = 184.66 \text{ ft. lbs.}$$



Figs. 31 and 32.—The mechanical work done by the expansion of unit weight of a perfect gas at constant pressure while heat is added to increase its temperature one degree Fahr. Fig. 31, piston locked for constant volume; fig. 32, piston free to move for constant pressure.

C_v = .16895

Specific Heat at Constant Volume.—If in raising the temperature of 1 lb. of air 1z Fahr. the volume be kept constant, no *external work* is done, accordingly the specific heat at constant volume, or

$$C_v = \left\{ \begin{matrix} \text{Specific heat at} \\ \text{constant pressure} \\ .2375 \end{matrix} \right\} - \left\{ \begin{matrix} \text{external work} \\ \text{in } B.t.u. \\ .06855 \end{matrix} \right\} = .16895\ B.t.u.$$

or

$$C_v = .16895 \times 777.52 = 131.36 \text{ ft. lbs.}$$

Summary

Value of units used

Mechanical Equivalent of heat = 777.52 ft. lbs.

Absolute zero of temperature = 491.6° Fahr.

Specific heat at constant pressure = .2375 B.t.u. or 184.66 ft. lbs.

" " " " volume = .16895 " " 131.36 " "

External work for constant volume = .0686 " " **53.3** " "

NOTE.—Analogous to the external work of the specific heat of air at constant volume is the external *latent heat* of vaporization in the formation of steam. The author does not agree with the generally accepted calculation for the external latent heat, or external work of vaporization and holds that it is wrong in principle as epxlained at length in Audel's Engineers' and Mechanics' Guide No. 1, page 31. The author holds that the calculation should be based on a stationary water level and not a receding water level as the water already existed at the beginning of vaporization which means that the atmosphere was already displaced to the extent of the volume occupied by the water.

n

1.41—Adiabatic compression is quite different from isothermal compression in that temperature boosts up the pressure so that the adiabatic curve lies outside the isothermal curve. That is, during compression the temperature rises unchecked and reacts on the air to increase its pressure. In order, therefore, to write an expression for adiabatic compression, it is necessary that

$$\frac{V_1}{V}$$

be increased by an amount equivalent to the amount of external work done on the air by heat reaction during compression, as indicated by an exponent n.

According to various authorities this exponent for adiabatic compression of dry air is

$$n = \frac{\text{Specific heat at constant pressure}}{\text{Specific heat at constant volume}}$$

$$= \frac{C_p}{C_v} = \frac{.2375}{.16895} = 1.4057$$

Commonly taken as **1.41**

Thus $PV^{1.41}$ denotes adiabatic compression and PV isothermal. Any intermediate compression between adiabatic and isothermal by PV^n, the value of n will vary considerably.

According to Peele in ordinary single cylinder dry compressors n is roughly 1.3, whereas in the best single stage wet compressor (with spray injector) n becomes 1.2 to 1.25. In large multi-stage units n may be as low as 1.15.

"CARDS"

(So called)

The sloppy expression "*cards*" was probably introduced because the *indicator* pencils a diagram on a "card" or small rectangular piece of paper a little larger but similar to a visiting card, but not so stiff.

The enclosed figure drawn on the card, whether by hand or by the indicator, is properly called a diagram.

It's not the card but what's drawn on the card that is of interest.

In place of a card a piece of tracing cloth could be attached to the indicator drum and a diagram taken and yet most people would still call it a card—which indicates a lack of brain control. They say "take a card" when they mean "take a diagram."

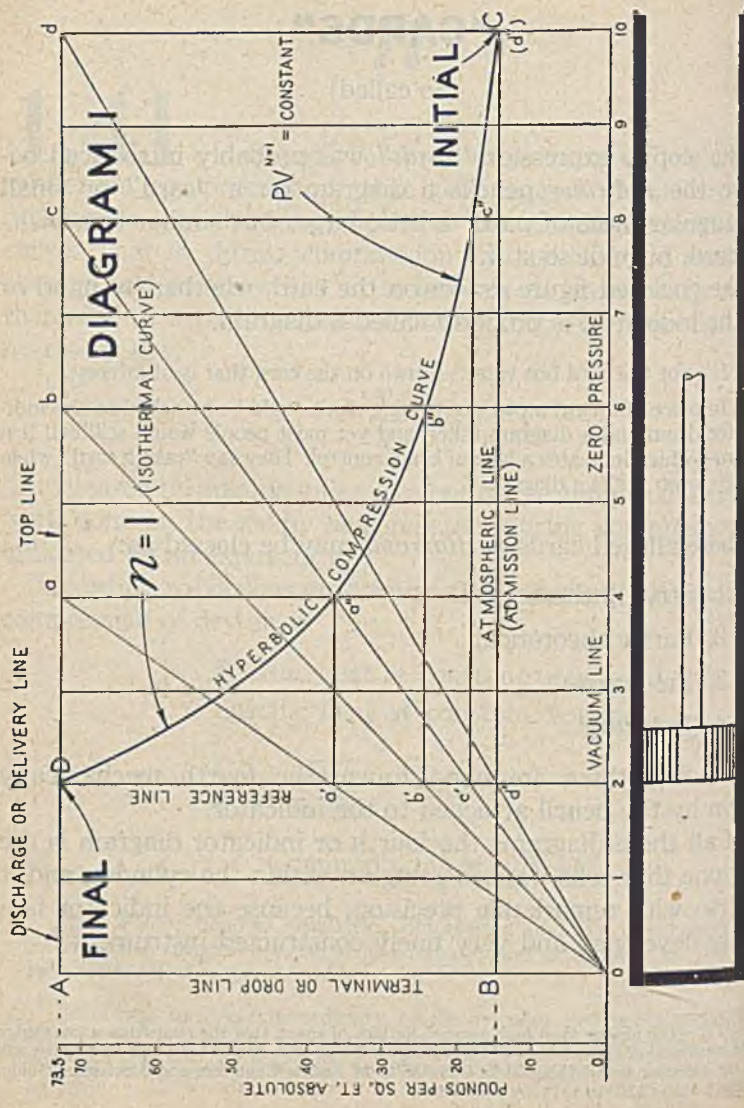
These alleged cards or *diagrams* may be classed as:

1. Strictly theoretical
2. Partly theoretical
3. Predicted
4. Indicator

The first three are hand drawn, the fourth mechanically drawn by the pencil attached to the indicator.

Of all these diagrams the fourth or indicator diagram is the only one that tells what is going on within the cylinder, and it does so with remarkable precision, because the indicator is a highly developed and very finely constructed instrument.*

*NOTE.—The author must here assume, for lack of space, that the reader has a knowledge of the construction and working of the indicator. Those who do not have this knowledge will find the indicator fully explained in Chapter 45 of Audels Engineers and Mechanics Guide, Volume 3, Pages 1269 to 1314, by the author.



Figs. 33 and 34.—Strictly theoretical diagram for the conditions given in Example I.

The various diagrams will now be explained in the sections following.

Strictly Theoretical Diagram.—This diagram represents perfect or ideal performance for any given degree of isothermal compression. Such performance is not possible in practice because of the assumptions, viz.:

1. Admission at atmospheric pressure
2. Isothermal compression
   (That is according to Boyle's law)
3. Zero clearance
4. Instantaneous opening and closing of valves without resistance.

The following example will illustrate how to construct a strictly theoretical diagram for given conditions.

Example.—Construct a strictly theoretical diagram with conditions given as follows:

Admission at atmospheric pressure; that is, initial pressure = 14.7 lbs. per sq. in. absolute. Five compressions.

Since compression is isothermal $n = 1$, that is PV $^{(n=1)}$ constant and this corresponds to the hyperbolic curve.

For five compressions

Final pressure = initial pressure × number of compressions
" " 14.7 × 5
= 73.5 lbs. absolute

Select any convenient scale as 10, 20, 30 lbs. per inch (depending upon the size of the paper) and draw vacuum and atmospheric lines, fig. 33.

At the left end erect two ⊥s close together to represent the pressure scale. Take any convenient length on the vacuum line and divide it into say 10 points (0, 1, 2, 3, etc.). This will be a scale of volumes. At these points (0, 1, 2, 3, etc.) erect ⊥s.

Complete pressure scale, taking an inch or any fraction of an inch to equal 10 lbs. Locate elevation for 73.5 lbs. This will be height of point A.

Through A, draw a horizontal line. Now since there are 5 compressions piston positions at the end of the 5th compression will be at point

$$10 \div 5 = 2$$

giving line 2D which is the line of reference in constructing the hyperbolic curve.

Through O draw diagonals *oa*, *ob*, etc. to the intersection of the "top line" with ⊥s through 4, 6, etc.

Through points a′b′c′, etc., where the diagonals intersect the reference line, draw horizontals to intersect the ⊥s 4*a*, 6*b*, etc., giving points a″b″ which are points on the hyperbolic compression curve.

Note that the diagonal *od* passes through the intersection of the reference line and atmospheric line which indicates that the atmospheric line has been correctly located.

Through points a″b″c,″ etc., describe the hyperbolic curve which gives values for pressure at every point of compression. According to Boyle's law that is when

$$n = 1$$

Thus the height of the curve above the vacuum line at points

8	6	4	2

corresponds to absolute pressure of

18.4 lbs.	24.5 lbs.	36.8 lbs.	73,5 lbs.

The cycle properly begins at A, the end of discharge or delivery of air. When the piston reaches the end of the stroke the discharge valve instantly closes, and since there is no clearance, there is no air, hence the pressure instantly drops, represented by the terminal or drop line AB.

At the same instant the admission (inlet) valve opens and air is admitted to the cylinder, following the receding piston at atmospheric pressure.

This admission continues for the entire (receding) stroke from B to C. Compression now begins at C, the initial point and continues to D, the final point or point of final compression, which in this case is 73.5 lbs. per sq. in. Now if the discharge at this pressure (opening just enough to maintain the pressure constant) the discharge or delivery of air will be represented by the horizontal discharge line DA.

When the piston reaches the end of the stroke, all the air has been discharged and the cycle is completed.

Partly Theoretical Diagram.—Where all the conditions of the strictly theoretical diagram are not adhered to, then the diagram is no longer fully theoretical or ideal. In such cases it might be called *partly* theoretical. In the example to be given the conditions which are different are:

1. Admission at **below** *atmospheric pressure.*
2. Clearance.

Compare these with the four conditions for the strictly theoretical diagram.

In the first place *no compressor can admit free air at atmospheric pressure.* It is impossible because of frictional resistance offered to the free flow of air.

In the second place, no compressor can be built without clearance. These are the two items which represent the chief differences of the partly theoretical and strictly theoretical diagrams.

Example.—Construct a partly theoretical diagram for: Admission at 12 lbs. pressure per sq. in. absolute. Five compressions; 5% clearance.

For 5 compressions

$$\text{Final pressure} = 12 \times 5 = 60 \text{ lbs. abs.}$$

Use same pressure scale as in ***Diagram 1*** (fig. 33). Draw at left pressure scale; next vacuum and atmospheric lines. Reproduce diagram 1 in dotted lines (phantom) for comparison.

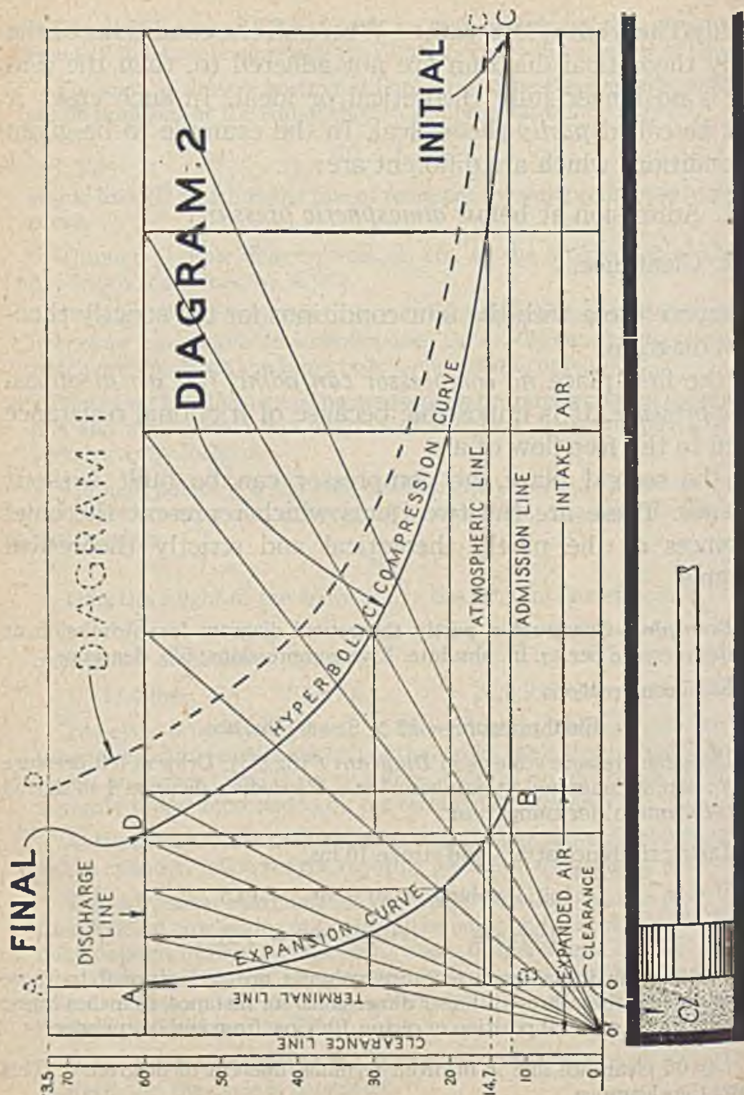
Taking clearance at 5% and stroke 10 ins.

$$\text{Initial volume} = 10 + 5\% = 10.5$$

$$\text{Clearance} = 10 \times .05 = .5$$

That is, the initial and clearance volumes are proportional to these figures. They also represent linear dimensions, for instance, in inches linear clearance .5 in.; initial position of piston 10.5 ins. from end of cylinder.

Lay off clearance line .5 in. from terminal line OA of diagram 1. This gives the clearance.



Figs. 35 and 36.—Partly theoretical diagram for the conditions given in **Example 2.**

From O on vacuum line lay off 10-inch stroke. For 5 compressions the piston displacement for each compression is

$$(10 + .5) \div 5 = 2.1 \text{ ins.}$$

which is distance of piston from end of cylinder at end of stroke. That is, in the diagram the reference line is drawn 2.1 ins. from clearance line.

Since final pressure is 60 lbs. draw top line through the 60-lb. division on the scale.

Draw admission line for 12 lbs.; also erect ⊥s from the vacuum line, dividing the displacement + clearance into 5 equal parts.

Construct the hyperbolic curve by the diagonal method used for diagram 1. Locating center for diagonals at o′ on clearance line instead of terminal line OA as in diagram 1.

When the piston reaches the end of the stroke the clearance space (*Cl*) is filled with air at the discharge pressure and temperature.

As the piston recedes, the clearance air expands, doing work on the piston. This expansion of the clearance air is represented in the diagram by the expansion curve AB, using the terminal line OA as a reference line for expansion.

Evidently (since it was specified that admission was at 12 lbs.) no external air can enter the cylinder until the clearance air has expanded to the point B, or to 12 lbs. absolute.

By comparing the diagram with diagram 1 here shown in phantom, the effect of clearance and a lower admission line is readily seen.

Ques. What is the effect of clearance as indicated by the diagram?

Ans. It puts the compression curve lower than it would be without clearance.

Ques. What are other effects of clearance?

Ans. It reduces the horse power and also the quantity of air delivered from the cylinder in almost exactly the same ratio.

Predicted Diagram.—By definition the predicted diagram is *a diagram drawn by a designer to apply to a given set of conditions,* using his judgment based on experience in locating the various lines with variance from theoretical so that the diagram thus drawn will represent actual performance as near as can be predicted.

By analyzing actual diagrams of compressors similar in type size and running conditions to the one to be designed, a predicted diagram may be drawn which will fairly well represent the results to be obtained.

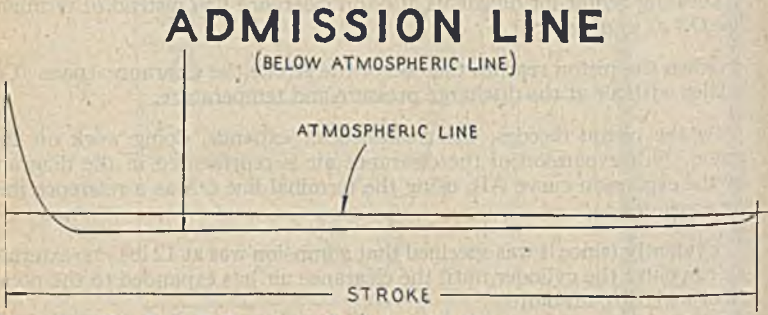


Fig. 37.—Predicted diagram 1. Admission line.

Consider now the various events of the cycle, that is

1. Admission
2. Compression
3. Delivery
4. Expansion

Admission.—Before free air can enter into the cylinder through the admission valves, the air remaining in the clearance space between piston and head, at the end of the stroke must be expanded on the return stroke to a little less than atmospheric pressure. In fact *it is* **impossible** *to admit air at* *atmospheric pressure* because of resistance offered to the incoming air.

Fig. 37 shows lowering of admission line due to resistance events.

Ques. Why then do some indicator diagrams show admission at practically atmospheric pressure?

Ans. Because the pressure of the incoming air has been boosted due to sudden rise in temperature, giving a false pressure.

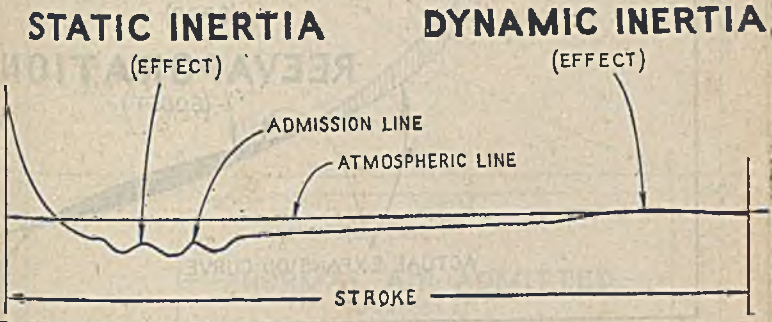


Fig. 38.—Predicted diagram 2. Inertia effect on admission line.

Ques. What effect on admission results in reciprocating compressors.

Ans. Due to the sudden starting and stopping of the air, its static and dynamic inertia tend to lower and boost respectively, the admission pressure, as in fig. 38. A similar distortion occurs on a steam engine expansion curve due to condensation and re-evaporation, as shown in fig. 39. Note how the actual expansion curve falls below and then rises above the hyperbolic curve.

Ques. How is the charge of air at atmospheric pressure determined (neglecting temperature boost and slippage)?

Ans. This is determined as in fig. 40 by measuring the length AB and dividing it by the length DC, on the assumption that the cylinder handles an amount of air at admission pressure and temperature represented by the length of AB.

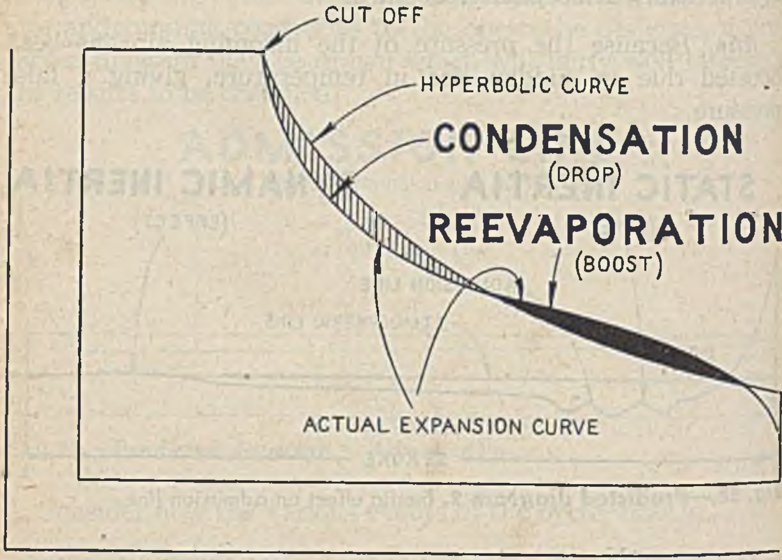


Fig. 39.—Predicted diagram 3. Steam engine diagram analogy of "inertia effect" on compressor curve. The steam expansion curve referred to the hyperbolic curve shows the distortions due to condensation and re-evaporation.

No air charge is admitted until the clearance air has expanded to the admission line. At this point air is admitted to the end of the stroke below atmospheric pressure.

Evidently the air must be compressed from the initial point of compression C to some point as B, where atmospheric pressure is reached.

Evidently since the expanded air DA, was already in the cylinder and BC, was lost, the actual charge or amount of free (normal) air admitted is represented by AB.

Ques. What chiefly causes the actual air admitted to differ from the amount indicated in fig. 40?

Ans. The temperature effect.

Referring to fig. 40 (according to "Compressed Air Data") the temperature at neither B nor A is by any means the same as that of the intake air. The value shown by the indicator diagram is from 4% to 12% greater than the actual delivery measured by more accurate means, depending upon

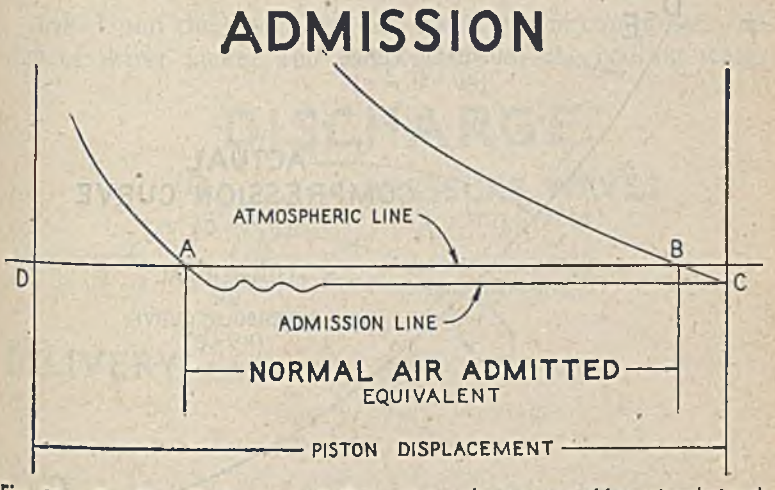


Fig. 40.—Predicted diagram 4. Determining the amount of free air admitted, temperature changes and slippage.

how much heat is absorbed by the incoming air due to the style of air valve gear and the extent of water jacketing.

The actual delivery of a single stage compressor pumping to 100 lbs. gauge from atmosphere will be from 65% to 80% of its piston displacement, depending upon its clearance space and valve gear.

A compound machine compressing to 100 lbs. gauge with an intercooler pressure of about 27 lbs. gauge will deliver from 80% to 90% of its piston displacement, also depending upon both clearance and style of valves.

Compression.—In practice, the actual compression curve will lie somewhere between the isothermal and adiabatic curves, the exact location depending upon the efficiency of the water jacket, temperature of the circulating water, etc.

According to Compressed Air Data, "In present-day reciprocating compressors, the air is compressed very nearly adiabatic. The water jacketing

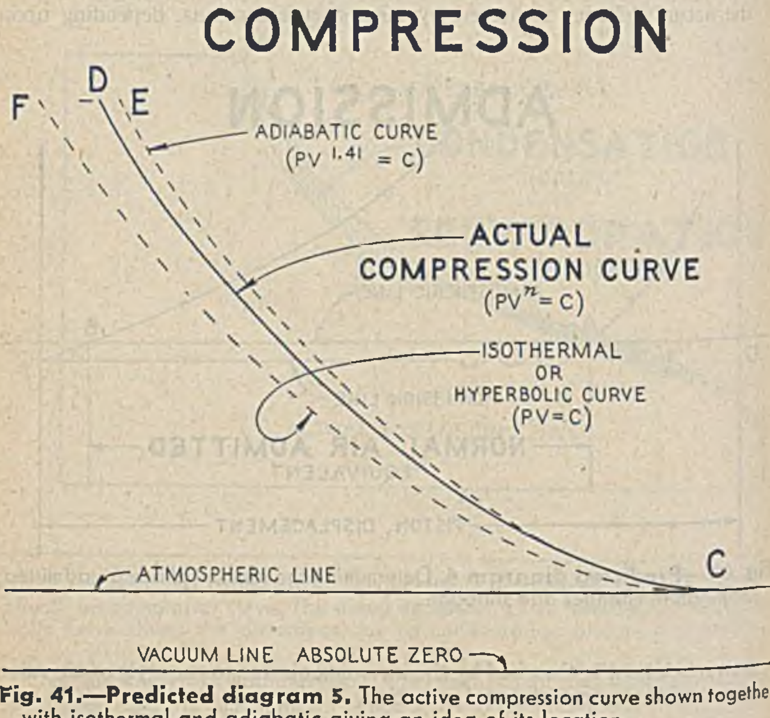


Fig. 41.—Predicted diagram 5. The active compression curve shown together with isothermal and adiabatic giving an idea of its location.

brings the compression curve somewhat below the adiabatic, but the excess pressure required to force the air through valves makes the work done in compressing and discharging the air very close to that which would be calculated on the assumption of adiabatic compression for the entire cylinder of air."

Fig. 41 shows *isothermal* and *adiabatic* compression curves with the *actual curve* ($PV^n = C$) lying in between.

Ques. What is the value of n on the ordinary single stage air compressor?

Ans. As given by Church, 1.33 and by Union as 1.25.

Ques. Upon what does the exact value of n depend?

Ans. Upon the size, type and speed of the compressor, design of water jacket and temperature of the cooling water.

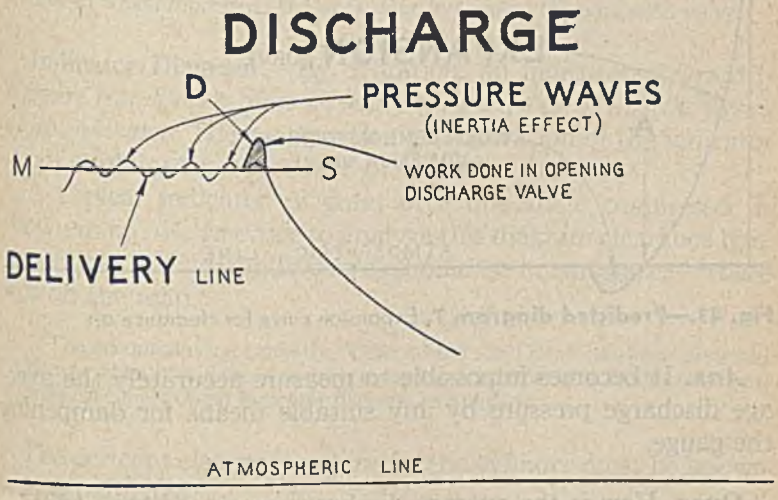


Fig. 42.—Predicted diagram 6. Delivery line.

Delivery.—The delivery line *is not straight but* **wavy** as in fig. 42. This is due to the intermittent and pulsating discharge of the compressor which causes pressure waves in the discharge pipe between compressor and receiver.

This is analogous to the hunting of a synchronous motor or to inertia effect, the variable discharge introducing static and dynamic inertia which reacts upon the normal flow of the air.

According to C.A.D., "If the natural period of the discharge pipe be in resonance with the speed of the compressor, or nearly so, pressure waves of considerable amplitude are induced in the discharge pipe and the discharge receiver."

Ques. What trouble is encountered on account of these pressure waves?

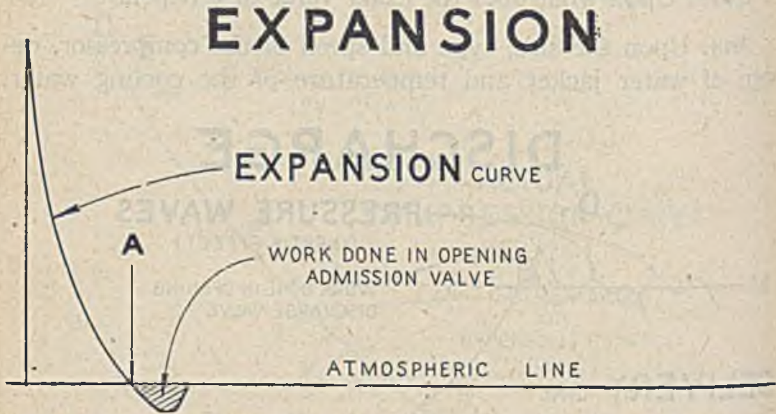


Fig. 43.—Predicted diagram 7. Expansion curve for clearance air.

Ans. It becomes impossible to measure accurately the average discharge pressure by any suitable means for dampening the gauge.

Ques. How is the average discharge pressure obtained?

Ans. By drawing a horizontal line through the wavy delivery line as MS, fig. 42, at an elevation that will strike an average of the pulsating delivery pressure.

Ques. What causes the big shaded wave D?

Ans. The shaded area represents the work done in opening the discharge valve.

Expansion.—This, as has been explained, is the **expansion** *of* **clearance air** *on the admission stroke.*

Most writers refer to this as *re-expansion,* on the assumption that the same clearance air after expansion again filled the clearance space at one end of delivery. However, the probability of this happening seems extremely remote, and accordingly *expansion* seems to be the correct word—not *re-expansion.*

Fig. 43 shows a typical expansion curve. It will be noted that the curve after reaching the atmospheric line, drops below forming a hook, the shaded area of which represents the work done in opening the admission valve.

Indicator Diagram.—By definition, an indicator diagram is *a figure traced on a piece of paper* (*called a "card"*) *by the pencil of an indicator.* The construction and operation of the indicator is not explained here for lack of space.*

A typical indicator diagram of a one stage compressor is shown in fig. 44. In order to analyze this diagram clearance line, isothermal and adiabatic curves should be drawn to see "where it is on the map."

The accompanying table by "Compressed Air Data" has been prepared for use when it is desired to plot theoretical, adiabatic or isothermal compression curves on an indicator diagram as in fig. 44.

The per cent clearance volume of the cylinder must be known or approximated.

Find clearance line in the usual way. Divide total card length from beginning of compression to the clearance line into ten equal parts. Draw vacuum or zero line and atmospheric line at 14.7 lbs. level.

*NOTE.—For full treatment of the indicator see Audels Engineers & Mechanics Guide, Vol. 3, Pages 1269 to 1314, by the author.

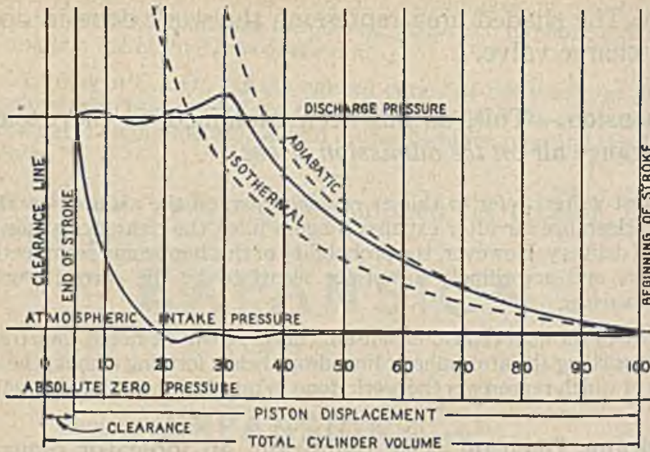


Fig. 44.—Plotting theoretical curves on indicator diagrams Compressed Air Data.

Table of Absolute Pressure in Air Compressor Cylinder

(According to Compressed Air Data)

Final Percent of Total Cylinder Volume (Clearance Included)	Absolute Final Pressure with 14.7 lbs. Absolute Initial Pressure		Factor for Any Initial Pressure	
	Isothermal	Adiabatic*	Isothermal	Adiabatic*
100	14.70	14.70	1.000	1.000
90	16.34	17.02	1.111	1.158
80	18.37	20.07	1.250	1.365
70	20.99	24.17	1.428	1.644
60	24.50	30.05	1.667	2.044
50	29.40	38.65	2.000	2.629
45	32.67	44.79	2.222	3.047
40	36.73	52.74	2.500	3.588
35	42.00	63.55	2.857	4.323
30	49.00	78.68	3.333	5.359
25	58.80	101.6	4.000	6.913
20	73.5	138.7	5.000	9.438
15	98.0	207.3	6.667	14.10
10	147.0	364.7	10.000	24.81
5	294.0	959.0	20.000	65.24

*Based on value for n of 1.3947.

From the table may be read directly the absolute pressure at various points of the stroke. All plotting of pressures must be from absolute zero.

Where the pressure at the start of compresion is not atmospheric, the factors given in the last two columns of the table may be used.

The absolute pressure to be plotted at any point of the stroke is obtained by multiplying the corresponding factor by the absolute admission pressure.

In the table the factors are obtained from the isothermal and adiabatic formulae.

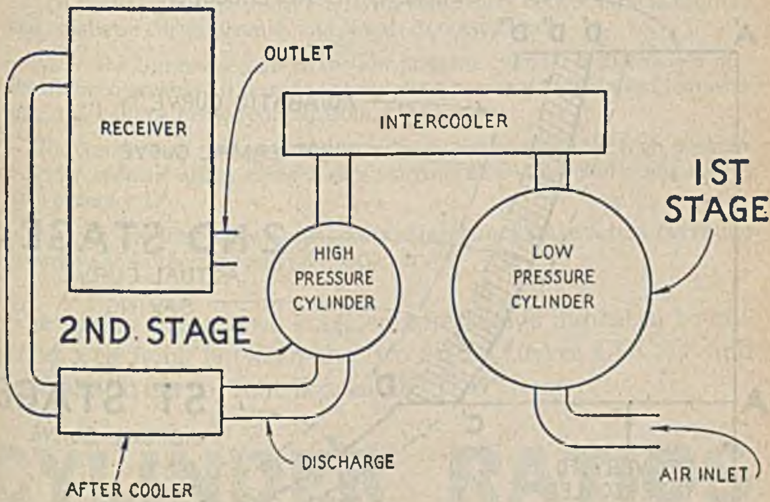


Fig. 45.—Two stage compression system.

TWO STAGE COMPRESSION

It must be evident from the foregoing explanations that isothermal compression is the most economical requiring the expenditure of the least amount of power. However, this method is impossible in practice, but an approach to it is obtained by compression in stages and cooling the air between each stage.

Ques. Describe two stage compression.

Ans. Air is first admitted into a low pressure cylinder and discharged into an intercooler where the air is cooled before entering the next or high pressure cylinder. Here it is compressed to the final pressure.

The diagram, fig. 45, shows the general arrangement of the units of the two-stage compression system.

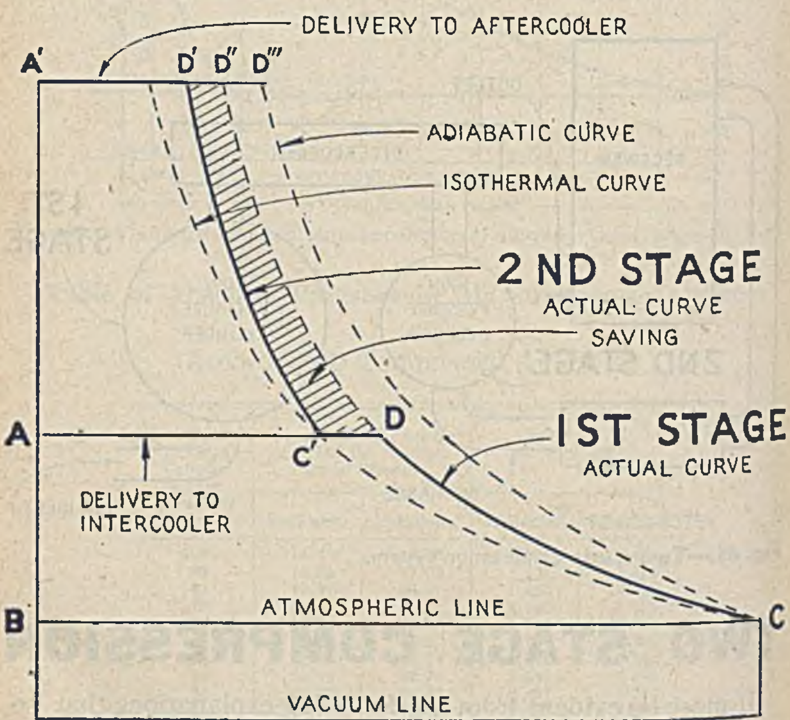


Fig. 46.—Combined diagram illustrating two stage compression. **In practice** single phase compression is employed for pressures up to about 50 or 60 lbs. absolute; two stage for 50 to 60 lbs.; three stage, for from 500 to 1,000 lbs. and four stage for higher pressures. Compressors which work against pressures less than 45 lbs. are usually called blowing engines.

Effect of Two Stage Compression.—The effect of two stage compression as regards power requirements is shown in fig. 46. The total result of dividing the compression into two stages is to obtain a nearer approach to isothermal compression.

The diagram, fig. 46, is a combined diagram for the two stages.

Compression begins in the low pressure cylinder at C, and continues to some intermediate point as D.

The *first stage actual curve* CD, lying somewhere between the isothermal and adiabatic curves from C and shown in dotted lines.

At D, the compressed air in the low pressure cylinder is discharged into the intercooler where it is cooled at constant pressure to its initial temperature, the volume being reduced from AD to AC′.

The compressed air now flows from the intercooler into the high pressure cylinder ***second stage*** compression starting at C′ and continuing to the final pressure D′.

As with the first stage actual curve, the second stage actual curve lies between the isothermal and adiabatic curves.

The saving due to two stage compression is indicated by the shaded area lying between the two actual curves CDC′D′ and the adiabatic curve, that is area DC′D′D″.

CYLINDER RATIOS
FOR
TWO STAGE COMPRESSION

According to Union, the correct ratio of cylinders for 2 stage compression is obtained by the following formula:

$$r = \sqrt{\frac{P_3}{P}}$$

In which r = ratio of cylinders.

P_3 = Absolute terminal pressure in lbs. per sq. in.

P = Atmospheric pressure in lbs. per sq. in.

Thus in two stage compression, extract the square root of the number of atmospheres to be compressed. This proportion of cylinder volumes divides the work equally between the different stages.

The intercooler pressure (P_1) in a two stage compressor is obtained by the following formula:

$$P_1 = P \times \sqrt{\frac{P_3}{P}} - P$$

In which P_1 = Intercooler pressure between first and second stages.

The following table gives the correct cylinder ratio and intercooler pressure in two stage compression for gauge pressures from 50 to 500 pounds per square inch.

Gauge Pressure Pounds	Absolute Pressure, Pounds	Number of Atmospheres	Correct Ratio of Cylinder Volumes	Intercooler Gauge Pressure	Gauge Pressure Pounds	Absolute Pressure, Pounds	Number of Atmospheres	Correct Ratio of Cylinder Volumes	Intercooler Gauge Pressure
50	64.7	4.40	2.10	16.2	200	214.7	14.60	3.82	41.4
60	74.7	5.08	2.25	18.4	210	224.7	15.28	3.91	42.8
70	84.7	5.76	2.40	20.6	220	234.7	15.96	3.99	44.0
80	94.7	6.44	2.54	22.7	230	244.7	16.64	4.08	45.3
90	104.7	7.12	2.67	24.5	240	254.7	17.32	4.17	46.6
100	114.7	7.80	2.79	26.3	250	264.7	18.00	4.24	47.6
110	124.7	8.48	2.91	28.1	260	274.7	18.68	4.32	48.8
120	134.7	9.16	3.03	29.8	270	284.7	19.36	4.40	50.0
130	144.7	9.84	3.14	31.5	280	294.7	20.04	4.48	51.1
140	154.7	10.52	3.24	32.9	290	304.7	20.72	4.55	52.2
150	164.7	11.20	3.35	34.5	300	314.7	21.40	4.63	53 4
160	174.7	11.88	3.45	36.1	350	364.7	24.80	4.98	58.5
170	184.7	12.56	3.54	37.3	400	414.7	28.20	5.31	63.3
180	194 7	13 24	3.64	38.8	450	464.7	31.60	5.61	67.8
190	204.7	13.92	3.73	40.1	500	514.7	35.01	5.91	72.1

CHAPTER 45

The Work of Compression

Work.—The word *work* is a widely used word, and is in the vocabulary of a great many people who have a very vague idea as to its meaning. Included among them are many who do not know the difference between *work* and *power*.

Considering the subject of this chapter, it seems fitting that the reader should at least first check up on the term work.

Ques. What is work?

Ans. *The overcoming of resistance through a certain distance by the expenditure of* **energy.**

Here two words are encountered—*resistance* and *energy*. Unless the exact meaning of each be known the definition is not clear.

Ques. What is resistance?

Ans. *The quality of not yielding to* **force** *or* **external pressure**; that quality of a body which acts in opposition to the pressure of another.

A typical illustration of *force* and *resistance* is shown in fig. 1. The flat tire and sandy road should help to clarify these terms.

Ques. What is energy?

Ans. That will be explained later.

Ques. How is work measured?

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

Ques. What is a foot pound?

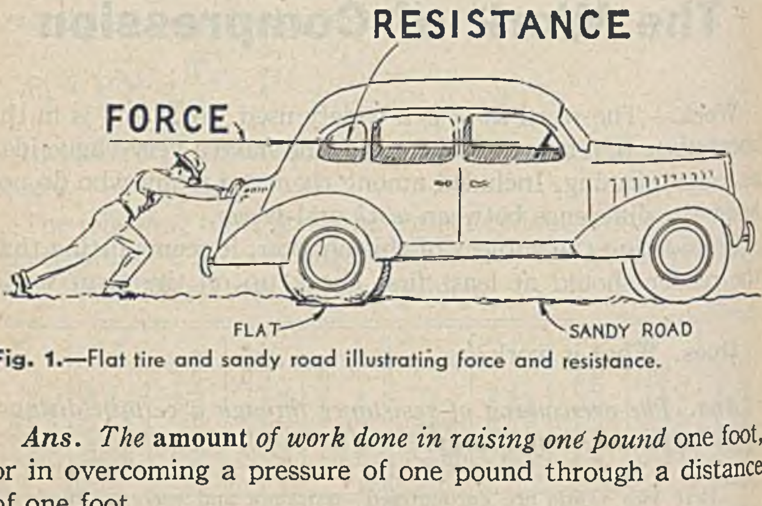


Fig. 1.—Flat tire and sandy road illustrating force and resistance.

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.

Note especially that it makes no difference how long it takes to raise the pound one foot (all day if necessary, or only one second if in a hurry)—time has nothing to do with it. Fig. 3 illustrates the foot-pound.

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

*NOTE.—Although this has been treated in the Chapter on Physics, additional explanations are here given for the convenience of the reader and to possibly further explain terms.

These effects are indicated through the touch and feeling as well as by the expansion, fusion, combustion or evaporation of the matter upon which it acts*.

Ques. How is heat measured?

Ans. By a standard unit called the British thermal unit (*B.t.u.*)

Ques. What is the British thermal unit?

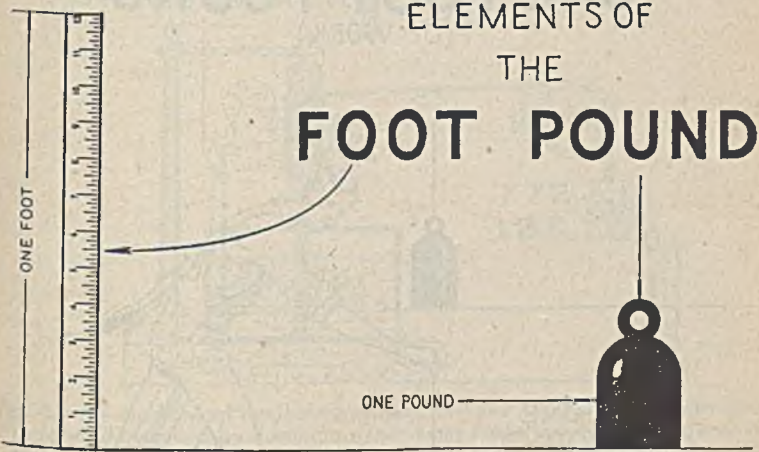


Fig. 2.—Elements of a foot pound—**one foot** and **one pound.**

Ans. $\frac{1}{180}$ *part of the heat required to raise the temperature* of one pound of water from 32° to 212° Fahr.

Ques. Mention one undesirable effect of heat.

*NOTE.—See Chapter 1, Page A49 on Physics for further explanation on heat.

Ans. In the compression of air the work of compression is converted into heat, heating the air and increasing the pressure above that corresponding to Boyle's law requiring more power.

Heat and Work.—*Heat develops mechanical force and motion,* hence *it is convertible into work.*

Fig. 3.—Man raising one pound one foot illustrating the **foot pound.**

Ques. What is the relation between the unit of heat and the unit of work?

Ans. It was shown by experiments made by Joule (1843-50) that 1 *unit of heat* = 772 *units of work.*

Ques. What is this called?

Ans. The "mechanical equivalent of heat" or Joule's experiment.

Fig. 5 shows work being transformed into heat.

Ques. Is the equivalent as obtained by Joule correct?

Ans. No.

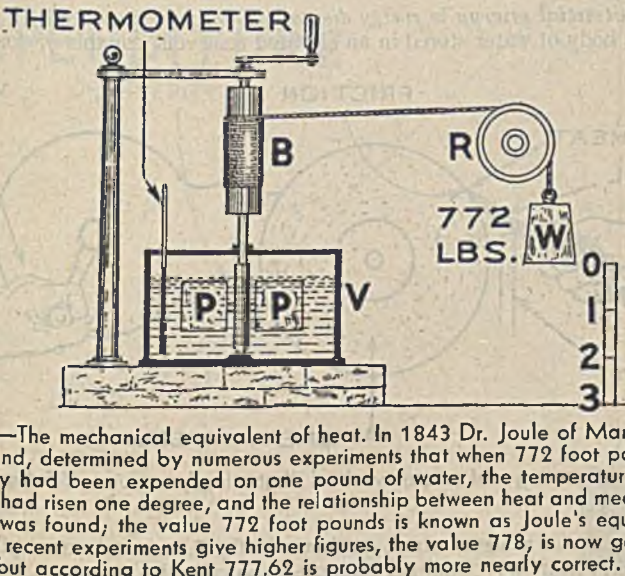


Fig. 4.—The mechanical equivalent of heat. In 1843 Dr. Joule of Manchester, England, determined by numerous experiments that when 772 foot pounds of energy had been expended on one pound of water, the temperature of the latter had risen one degree, and the relationship between heat and mechanical work was found; the value 772 foot pounds is known as Joule's equivalent. More recent experiments give higher figures, the value 778, is now generally used but according to Kent 777.62 is probably more nearly correct. **Marks and Davis** in their steam tables have used the figure 777.52.

Later experiments gave various results. 778 is sufficiently accurate for ordinary calculations. However, 777.5 is probably more nearly correct. The value 777.52 was used by Marks and Davis in their Steam Tables.

Energy.—By definition, *energy is stored work*, that is, *the ability to do work*, in other words, the ability to move against resistance.

Ques. Must a body do work to possess energy?

Ans. No.

Work is done by an ***expenditure of energy*** as in fig. 6.

Ques. Name two kinds of energy.

Ans. Potential energy and kinetic energy.

Potential energy is *energy due to position* as represented for instance by a body of water stored in an elevated reservoir, capable of doing work

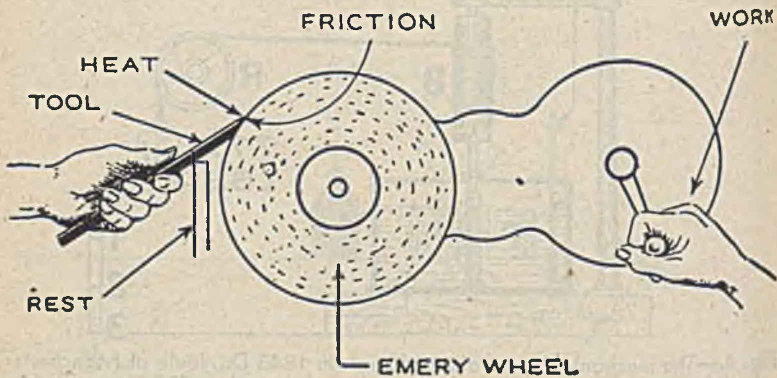


Fig. 5.—Grinding tool on emery wheel illustrating heat due to **friction.**

by means of a water wheel, as shown in fig. 7, or an elevated pile driver monkey as in fig. 8. Another example is shown in figs. 9 to 11.

Kinetic energy is *energy due to momentum*, that is, *the energy of a moving body*, which is equivalent to saying, *dynamic inertia.*

For instance, a heavy rapidly moving freight train possesses considerable kinetic energy and considerable work in the form of braking (friction converted into heat) has to be done to bring it to rest. This is dynamic inertia as shown in fig. 12. Conversion of potential energy into dynamic energy is shown in fig. 13. Another example of kinetic energy is shown in fig. 14.

Conservation of Energy.—If the layman knew anything about the conservation of energy, there would be less of these perpetual motion inventors, trying to accomplish the impossible.

By definition, *energy can be transmitted from one body to another or transformed in its manifestations, but can neither be created nor destroyed.*

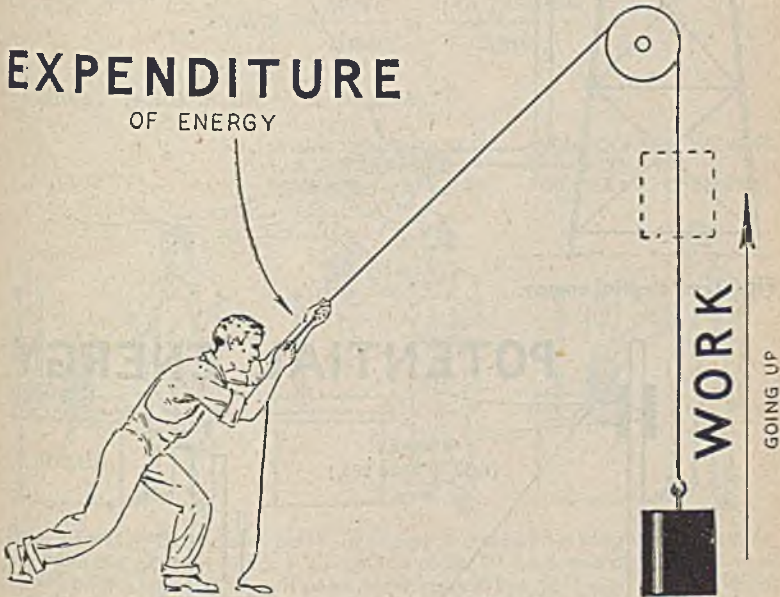


Fig. 6.—Work being done by the expenditure of energy.

Ques. Can energy be dissipated, that is converted into a form from which it cannot be recovered?

Ans. Yes.

Thus in the case of that very wasteful contraption the alleged house heating boiler, a great percentage of the heat escapes up the chimney because

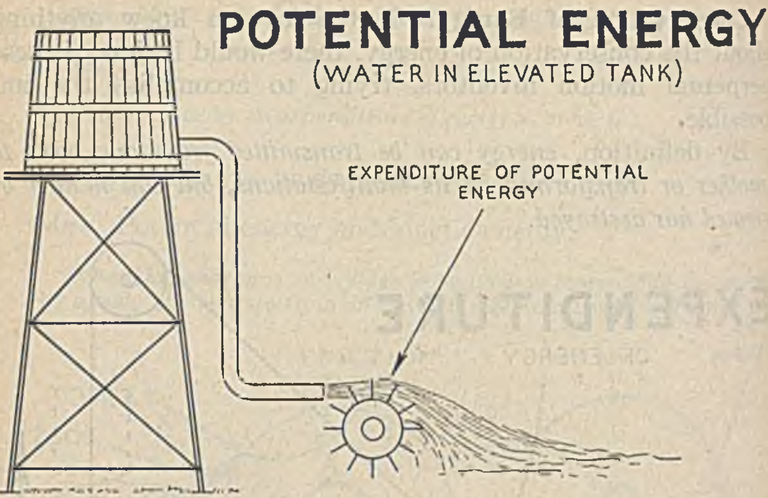


Fig. 7.—Potential energy.

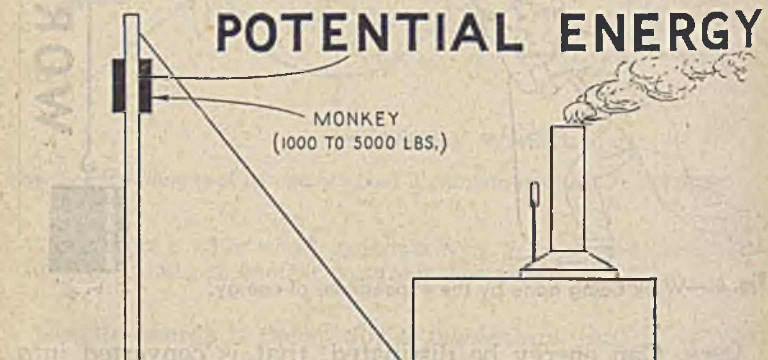


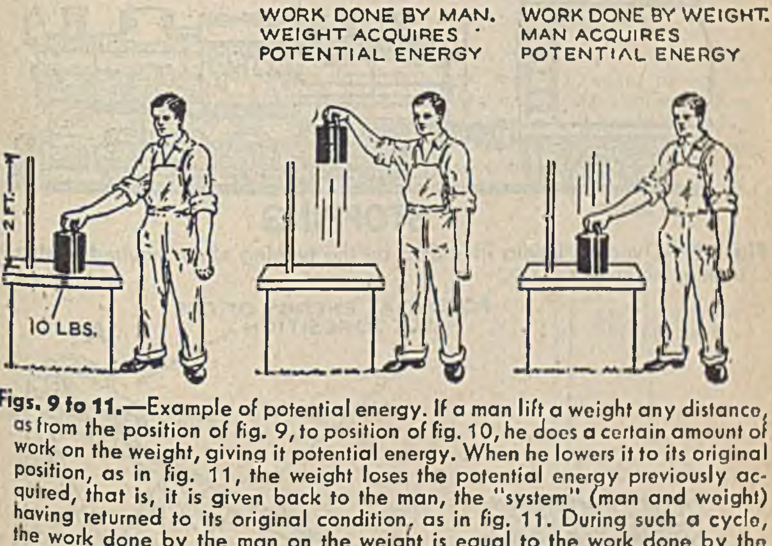
Fig. 8.—Potential energy; second illustration.

the amount of heating surface provided is so small as to be ridiculous. Some designs are first rate cellar heaters and with a red hot stove pipe present a distinct fire hazard. This goes for all sectional cast iron alleged house heating boilers—thermally no good.

Power.—By definition, power is *the rate at which work is done*, that is, work divided by the time in which it is done. Thus,

$$\text{Power} = \frac{\text{work}}{\text{time}} = \frac{\text{foot pounds}}{\text{time}}$$

Ques. What is the unit of power?



Figs. 9 to 11.—Example of potential energy. If a man lift a weight any distance, as from the position of fig. 9, to position of fig. 10, he does a certain amount of work on the weight, giving it potential energy. When he lowers it to its original position, as in fig. 11, the weight loses the potential energy previously acquired, that is, it is given back to the man, the "system" (man and weight) having returned to its original condition, as in fig. 11. During such a cycle, the work done by the man on the weight is equal to the work done by the weight on the man.

Ans. The horse power.*

*NOTE.—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 is one horse power?

Ans. 33,000 foot pounds per minute.

An operation illustrating one horse power is shown in fig. 15.

Ques. What is the object of the gearing in fig. 15 between the engine and boiler?

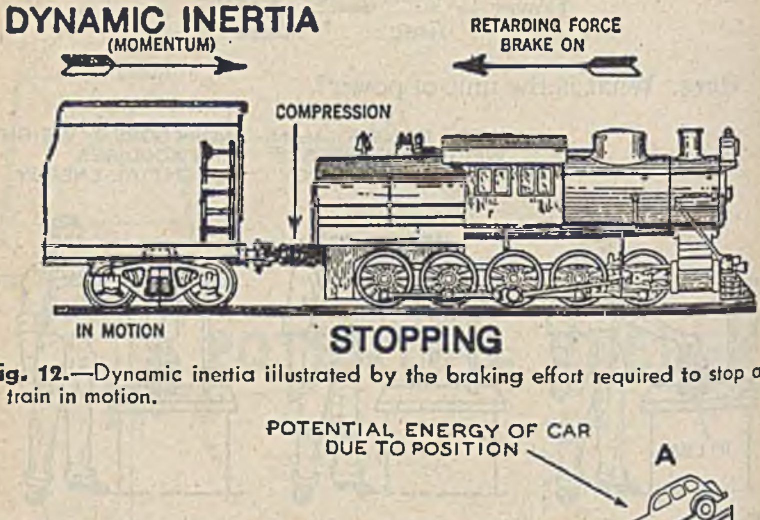


Fig. 12.—Dynamic inertia illustrated by the braking effort required to stop a train in motion.

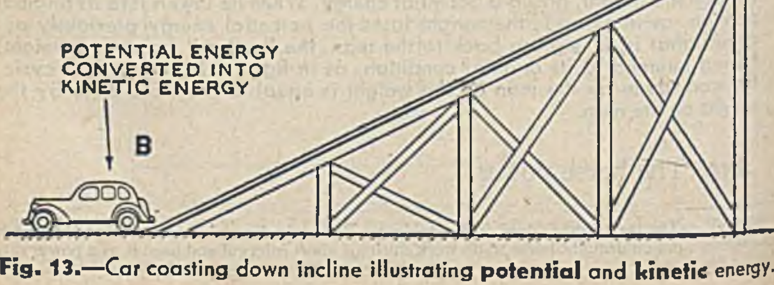


Fig. 13.—Car coasting down incline illustrating **potential** and **kinetic** energy.

Ans. It is to reduce the load (pounds) on the piston.

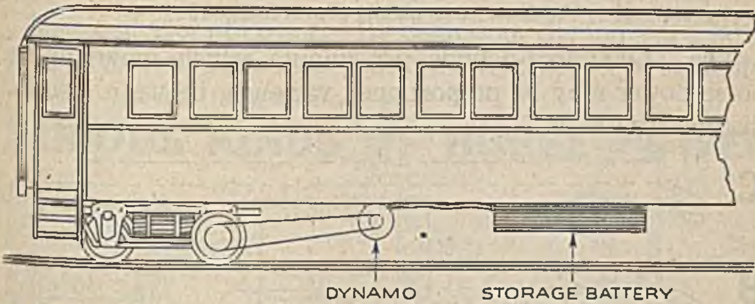


Fig. 14.—Kinetic energy. The illustration represents a railway car with axle lighting system. If the car be set in motion and then no further power be applied its momentum or kinetic energy will drive the dynamo which in turn will charge the storage battery, and acting like a brake will gradually bring the car to rest. During this operation, the kinetic energy, originally possessed by the moving car, is absorbed by the dyanmo (neglecting friction) and delivered to the battery as electrical energy which may be used in lighting the car.

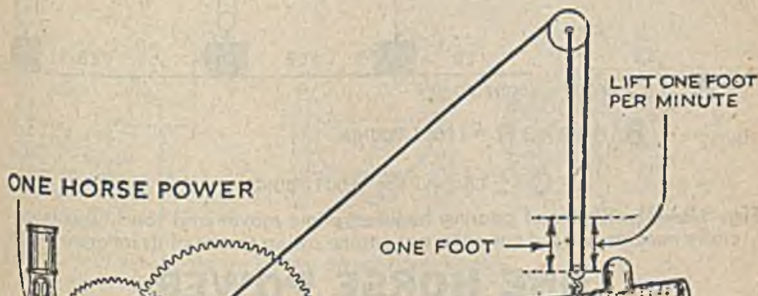


Fig. 15.—Engine developing one horse power. Neglecting power lost by friction of the reduction gears the engine delivers one horse power—this is **brake** horse power. Define brake horse power.

Thus, in fig. 16, a one-pound weight is shown with three block and fall hook-ups. In these, a movement of one, two and four feet is required to raise the weight one foot. The pull (or load on piston in fig. 15) is inversely proportional to the distance as plainly shown in fig. 16.

The 33,000 foot-pounds per minute which make up the horse power may be proportioned variously between load and distance raised as

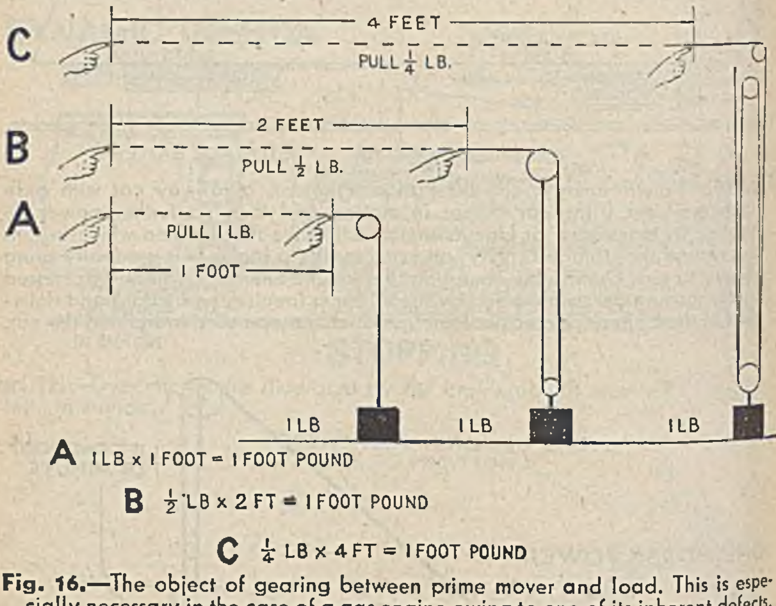


Fig. 16.—The object of gearing between prime mover and load. This is especially necessary in the case of a gas engine owing to one of its inherent defects.

ONE HORSE POWER

33,000 pounds		1 foot in one minute				
3,300	"	10 feet	"	"		"
330	"	100	"	"	"	"
33	"	1000	"	"	"	"
3.3	"	10000	"	"	"	"
1	"	33000	"	"	"	"

There are various kinds of horse power such as: Nominal, indicated, brake, effective, hydraulic, boiler, electrical, etc. The horse power of interest here is the indicated horse power.

Indicated Horse Power.—By definition, the indicated horse power is *the actual power developed within a cylinder as calculated from the indicator diagram.*

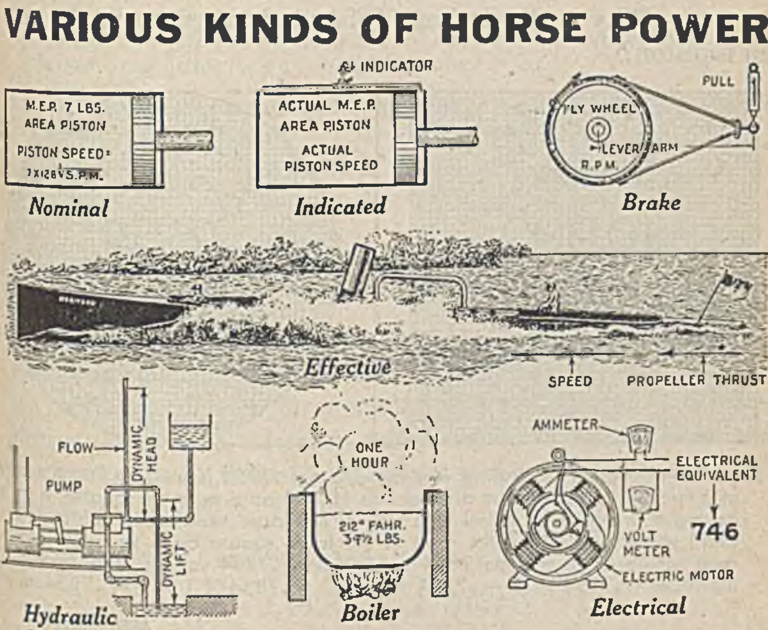


Fig. 17.—What should be known about horse power.

In the operation of any engine or compressor, it represents the power at the instant the diagram was taken. In the case of a steam driven compressor the indicated horse power at the steam end does not represent the power delivered to the compressor piston, being in excess by an amount equal to the power lost by friction.

The indicated horse power at the compressor end is the actual power delivered to compress the air.

How to Calculate Horse Power.—There are various horse power formulae and before considering them a few preliminary explanations are necessary.

Ques. Why is the decimal .7854 used to ascertain the area of a piston?

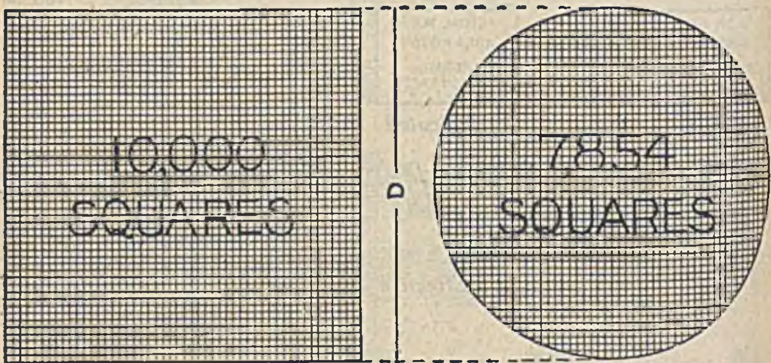
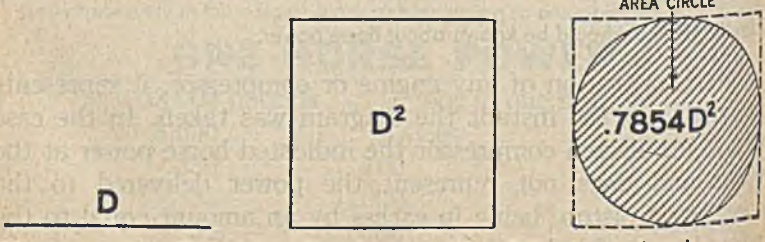


Fig. 18.—Diagram illustrating why the decimal .7854 is used to find the area of a circle. If the square be divided into 10,000 parts or small squares, a circle having a diameter D, equal to a side of the large square will contain 7854 small squares, hence, if the area of the large square be 1 sq. in., then the area of the circle will be 7854 ÷ 10,000 or .7854 sq. ins., that is, area of the circle = $.7854 \times D^2 = .7854 \times D \times D = .7854 \times 1 \times 1 = .7854$ sq. in.



Figs. 19 to 21.—Pictorial explanation that $.7854\ D^2$ = area of circle.

Ans. Because it represents the relation between a circle and circumscribed square as shown in fig. 18.

Ques. Define piston speed?

Ans. The total distance traveled by the piston in one minute —not the actual velocity at any given instant.

Rule.*—Multiply twice the number of revolutions per minute by the stroke in inches and divide the product by* 12 *to reduce to feet.*

Example.—A compressor has a stroke of six inches and runs at 500 revolutions per minute. What is the piston speed?

$$\text{piston speed} = \frac{2 \times 6 \times 500}{12} = 500 \text{ ft. per minute}$$

Ques. What is the usual but objectionable method of calculating horse power?

Ans. RULE: *Multiply the mean effective pressure in lbs. per square inch by the area of the piston in square inches and multiply the product by the length of stroke* **in feet** *and by the number of strokes per minute (twice the number of revolutions); divide this last product by* 33,000 *and the answer will be the horse power of a double acting engine.*

Expressed as a formula

$$HP = \frac{2\ PLAN}{33{,}000} = \frac{2\ (.7854D^2)\ PLN}{33{,}000} \quad \ldots\ldots\ldots (1)$$

in which

P = mean effective pressure in lbs. per sq. in.
L = length of stroke **in feet**
A = area of piston in sq. ins. = .7854 × diameter^2
N = number of **revolutions** per minute
D = diameter of piston in inches

In the formula the numerator is made up of the total number of foot pounds done in one minute; the denominator or 33,000 is the number of foot pounds per minute per horse power.

Ques. What is the matter with this formula?

Ans. It requires a ridiculous waste of time in calculating.

Ques. Why?

Ans. In the first place the stroke which is usually given in inches has to be reduced to feet; in the second place the constants 2, .7854, 33,000 should be eliminated by reducing the whole expression to its lowest terms, thus:

$$*HP = \frac{2P\frac{L}{12} \times .7854\, D^2N}{33,000} = .000003697\, PLD^2N$$

Calling the constant .000004, which is near enough for ordinary calculations, and changing the order of the factors, the formula becomes:

$$HP = .000004(D^2L)\, NP \ldots\ldots\ldots\ldots (2)$$

Note in using this formula, first put down the cylinder dimensions in inches, squaring diameter, thus:

$$D^2L$$

prefix the constant 4, omitting the ciphers

$$4D^2L$$

insert the number of revolutions and mean effective pressure

$$4D^2LNP$$

*NOTE.—In using this formula L, is the length of stroke in *inches*.

Substitute a given set of values and point off by "sense of proportion." The reader should take an example, and calculate horse power first by formula (1) then by formula (2) noting the difference in time and mental effort in using formula 2, not forgetting to consider the extra chances in formula 1 of making mistakes in figuring.

Effect of the Piston Rod on Power.—It must be evident, since the piston rod passes through the stuffing box in the cylinder head, that the area of the piston subjected to pressure is reduced on one side by an amount equal to the area of the rod, whereas the full area of the other side of the piston is subjected to pressure. Accordingly, the power developed at the crank end will be less than at the head end. To get the average effect *½ area of piston rod is deducted from area of piston*, but this refinement is usually neglected in ordinary calculations as a waste of time.

ISOTHERMAL COMPRESSION

In treating of the work of compression first, will be considered compression according to Boyle's law, that is, **isothermal** *compression*, or compression at **constant** *temperature*, in formula form

$$PV^{n=1} = \text{constant}$$

Ques. What is that $n = 1$ placed after V for?

Ans. The exponent n provides for the conditions under which compression takes place. Taking $n = 1$, indicates that there is no heat of compression (*isothermal compression*), that is, substituting 1 for *n*, gives PV^1 which is the same as PV, hence no extra work.

Where the heat of compression is considered (always present in actual compression) the value of *n* is always greater than one.

Ques. How is isothermal compression, that is compression in which PV = constant indicated?

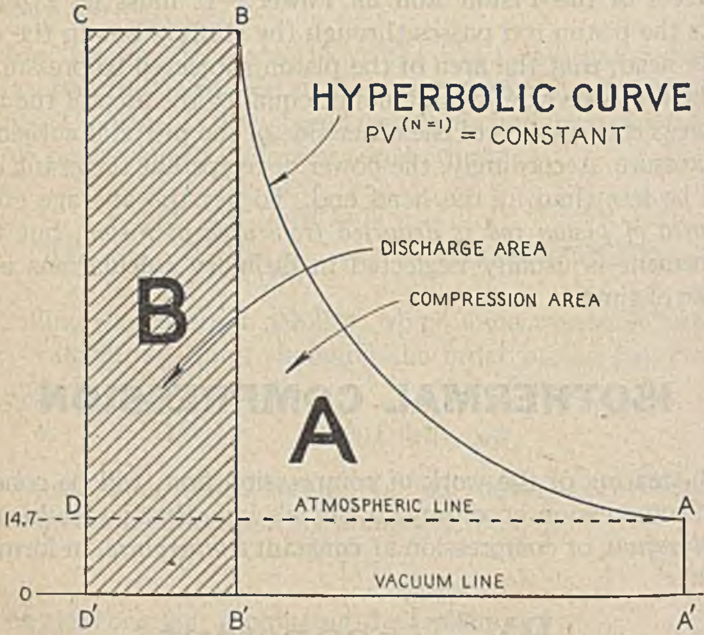


Fig. 22.—Theoretical diagram for isothermal compression to illustrate work as indicated by diagrams.

Ans. It is represented graphically by the equilateral hyperbola referred to its rectilinear asymptotes.*

*NOTE.—See pages 15 to 22, for explanation relating to the hyperbola.

The work of compression and during discharge at constant pressure during discharge is equal to the area of the indicator diagram, fig. 22.

Work as Indicated by Diagram.—Take for example a theoretical diagram as in fig. 22. To understand the diagram, consider first

1. The forward or compression stroke, and then
2. The return or admission stroke

From the explanations which follow (assuming no clearance) it will be seen that the net work done in compressing and discharging the air is equal to *the work of compression plus the work of expulsion of the air from the cylinder minus the work done on the piston by the pressure of the atmosphere during admission.*

1. ***The forward or compression stroke.***

Fig. 22 is theoretical diagram which shows what happens on the forward stroke considering only the advancing side of the piston. Let D′A′ (same as DA) = length of stroke.

At beginning of stroke with cylinder full of air at atmospheric pressure A, let the air be compressed to some pressure as B. as indicated by the hyperbolic curve AB, and then be discharged from the cylinder at constant pressure as indicated by the line BC.

Draw B′B dividing the diagram ABCD′A′ into two areas A′ABB′ or **A**, and B′BCD′ or **B**.

The work done during compression and discharge *for each square inch of the piston*

= area **A** + area **B**.

Ques. Why is the work done equal to area **A** + area **B**?

Ans. To illustrate, consider the work done during discharge represented by the shaded area **B**, or B′BCD′, fig. 22, the shaded area **B** represents the work done during discharge because:

Table of
Hyperbolic Logarithms

No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.
1.1	0.0953	4.5	1.5041	7.9	2.0669	19.0	2.9444
1.2	0.1823	4.6	1.5261	8.0	2.0794	20.0	2.9957
1.3	0.2624	4.7	1.5476	8.1	2.0919	21.0	3.0445
1.4	0.3365	4.8	1.5686	8.2	2.1041	22.0	3.0910
1.5	0.4055	4.9	1.5892	8.3	2.1163	23.0	3.1355
1.6	0.4700	5.0	1.6094	8.4	2.1282	24.0	3.1781
1.7	0.5306	5.1	1.6292	8.5	2.1401	25.0	3.2189
1.8	0.5878	5.2	1.6487	8.6	2.1518	26.0	3.2581
1.9	0.6419	5.3	1.6677	8.7	2.1633	27.0	3.2958
2.0	0.6931	5.4	1.6864	8.8	2.1748	28.0	3.3322
2.1	0.7419	5.5	1.7047	8.9	2.1861	29.0	3.3673
2.2	0.7885	5.6	1.7228	9.0	2.1972	30.0	3.4012
2.3	0.8329	5.7	1.7405	9.1	2.2083	31.0	3.4340
2.4	0.8755	5.8	1.7579	9.2	2.2192	32.0	3.4657
2.5	0.9163	5.9	1.7750	9.3	2.2300	33.0	3.4965
2.6	0.9555	6.0	1.7918	9.4	2.2407	34.0	3.5263
2.7	0.9933	6.1	1.8083	9.5	2.2513	35.0	3.5553
2.8	1.0296	6.2	1.8245	9.6	2.2618	36.0	3.5835
2.9	1.0647	6.3	1.8405	9.7	2.2721	37.0	3.6109
3.0	1.0986	6.4	1.8563	9.8	2.2824	38.0	3.6376
3.1	1.1312	6.5	1.8718	9.9	2.2925	39.0	3.6636
3.2	1.1632	6.6	1.8871	10.0	2.3026	40.0	3.6889
3.3	1.1939	6.7	1.9021	10.5	2.3513	41.0	3.7136
3.4	1.2238	6.8	1.9169	11.0	2.3979	42.0	3.7377
3.5	1.2528	6.9	1.9315	11.5	2.4430	43.0	3.7612
3.6	1.2809	7.0	1.9459	12.0	2.4849	44.0	3.7842
3.7	1.3083	7.1	1.9601	12.5	2.5262	45.0	3.8067
3.8	1.3350	7.2	1.9741	13.0	2.5649	46.0	3.8286
3.9	1.3610	7.3	1.9879	13.5	2.6027	47.0	3.8501
4.0	1.3863	7.4	2.0015	14.0	2.6391	48.0	3.8712
4.1	1.4110	7.5	2.0149	15.0	2.7081	49.0	3.8918
4.2	1.4351	7.6	2.0281	16.0	2.7726	50.0	3.9120
4.3	1.4586	7.7	2.0412	17.0	2.8332		
4.4	1.4816	7.8	2.0541	18.0	2.8904		

NOTE.—Hyperbolic or Naperian logarithms are common logarithms multiplied by 2.3025851.

$$\underset{\text{(load)}}{\textit{work done} = B'B} \times \underset{\text{(distance moved)}}{B'D'} = \underset{\text{(area)}}{B'BCD'}$$

From this it must be evident that the work done during compression = area **A**, or A′ABB′.

Ques. How is this area **A** determined, other than by planimeter measurement?

Ans. The exact value of the area **A** may be readily obtained by referring to the table of hyperbolic logarithms, page 84.

Ques. Why?

Ans. Because since the compression curve is a hyperbola, the hyperbolic logarithm of the compression ratio (number of compressions) expresses the relation between the area **A**, during compression, and the area **B**, during discharge.

That is to say, taking the shaded discharge area

$$\mathbf{B} = \text{unity}$$

$$\textit{Compression area } \mathbf{A} = \textit{hyperbolic logarithm } r$$

and

$$\text{total area } \mathbf{A} + \mathbf{B} = 1 + \text{hyperbolic logarithm } r$$

Ques. What is r?

Ans. The number of compressions; or compression ratio

Example.—If air be compressed to ¼ its volume, as in fig. 23, and the ¼ volume discharged at the final pressure, then calling the area representing the ¼ volume during discharge 1, the area during compression = hyperbolic logarithm of 4, which is from the table on page 84. = 1.3863.

This means that if 1 represent the work (or area **B**) during discharge, 1.3863 represents the work (or area **A**) during compression. Accordingly the total work done during the stroke is equal to

$$\mathbf{A} + \mathbf{B} = 1 + hyp.\ log.\ 4 = 1 + 1.3863 = 2.3863$$

To further explain the ratio of compression r compare this with the expansion of steam as in a steam engine which is the same as the compression of air only "*in reverse.*"

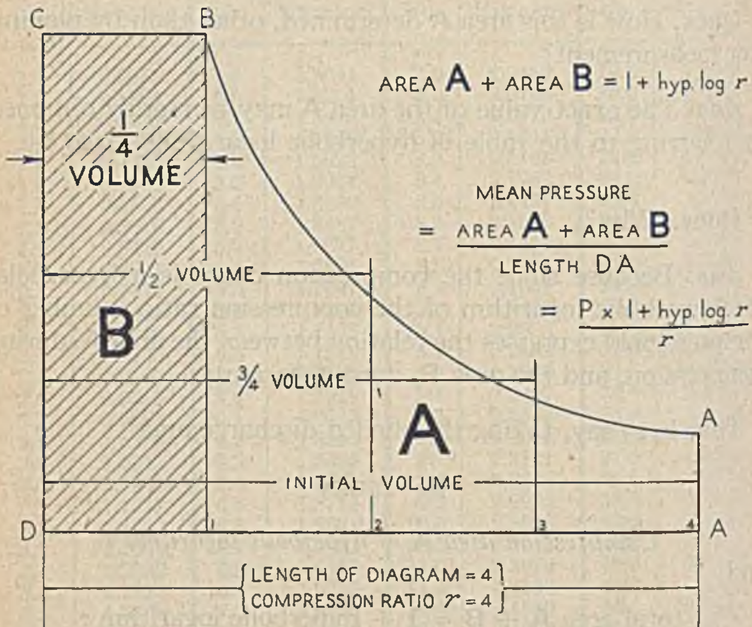


Fig. 23.—Theoretical diagram for isothermal compression (without clearance) for conditions given in the accompanying example.

Example.—A steam engine operates with a terminal pressure of 14.7 lbs. absolute, ¼ cut off (4 expansions), no clearance, find work done during the stroke.

In fig. 24 draw the vacuum and atmospheric lines D′A′ and DA. Divide D′A′ into 4 equal parts. Erect perpendiculars at A′ and D′, and construct the hyperbola extending from A to B.

If A′A = 14.7 lbs. absolute, B′B will equal 14.7 × 4 = 58.8 lbs. abs., using the same scale. That is, if steam at 58.8 lbs. abs. be cut off at ¼ stroke and expanded to the end of the stroke or expanded 4 times the initial volume, the terminal pressure (pressure at the end of the stroke) =

$$58.8 \div 4 = 14.7 \text{ lbs. abs.*}$$

Since the pressure is constant during admission, this will be represented by the horizontal line CB., the point C, being at the intersection of this horizontal, with the perpendicular from D′, the beginning of the stroke.

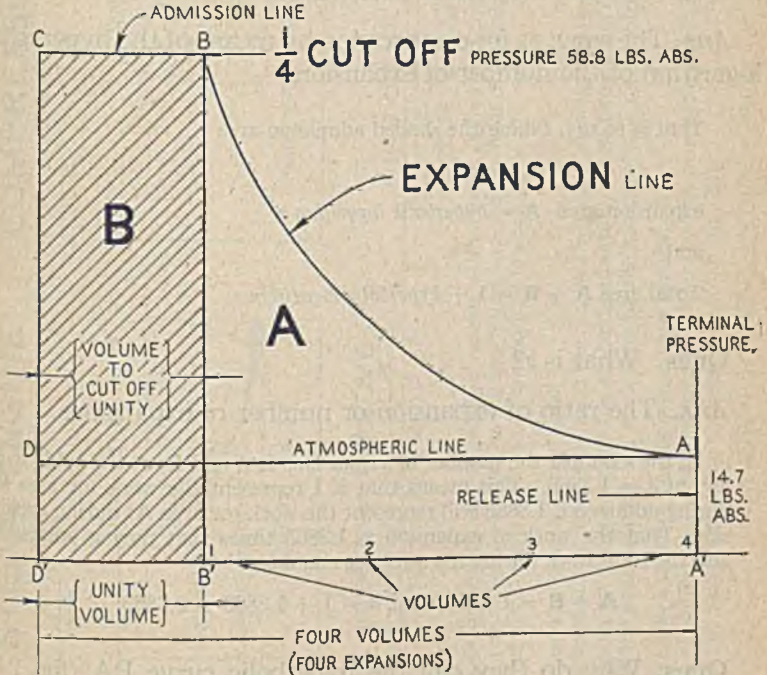


Fig. 24.—Theoretical diagram to illustrate steam engine operation at ¼ cut off (4 expansions) atmospheric terminal pressure.

*NOTE.—Of course in this illustration steam is assumed to act according to ***Boyle's law***, but it should be *distinctly understood* that in practice *it does not* because of *condensation* during the early part of expansion and *re-evaporation*, occurring later. This results in the actual expansion line first falling below, then rising above, the hyperbolic curve.

Now similarly as in the preceding explanation, for the shaded area **B**, or CBB′D′

	(load)		(distance moved)	(area)
work done =	B′B	×	B′D′	= CBB′D′

From this it must be evident that the work done during *admission* = Area **B** or CBB′D′.

Ques. How is the area **A** determined?

Ans. The same as for compression by means of the hyperbolic logarithm of the number of expansions.

That is to say, taking the shaded admission area

B = unity

expansion area **A** = *hyperbolic logarithm r*

and

Total area **A** + **B** = 1 + *hyperbolic logarithm r*

Ques. What is r?

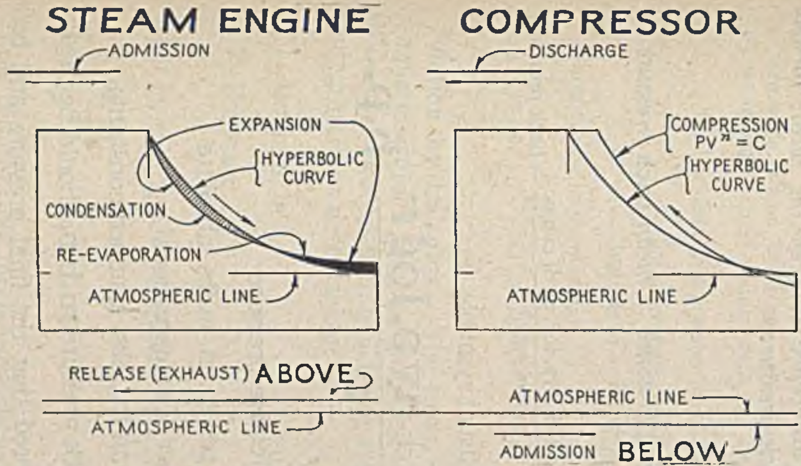
Ans. The ratio of expansion or number of expansions.

In the example the number of expansions is 4 and from the table hyp. log. of 4 = 1.3863. This means that if 1 represent the work (or area **B**) during admission, 1.3863 will represent the work (or area **A**) during expansion. That the work of expansion is 1.3863 times that during admission and the total work during the stroke is equal to

A + **B** = 1 + *hyp. log.* 4 = 1 + 1.3863 = 2.3863

Ques. Why do they call the hyperbolic curve BA (fig. 24) a line?

Ans. Although a line is generally considered as being straight, a straight line may be considered as being a curve with infinity radius.



Figs. 25 to 31.—Comparison of the principal events of the steam engine and compressor cycle. Note effect of condensation and re-evaporation on the expansion line and the heat of compression on the compression curve: the hyperbolic curve is the basis of comparison in each case.

2. The return or admission stroke.

During the forward stroke work is done on the air, by the piston; however during the return stroke, work is done on the piston by the pressure of the air being admitted. These two events occur at the same time on opposite sides of the piston. In fig. 22, work done by the atmosphere is represented by the area A'ADD'.

3. The net work during the compression stroke.

Since, as just explained, work is done on the piston by the atmosphere during admission, it must be evident that the net work during the compression stroke (fig. 22)

$$= A'ABB' + B'BCD' - A'ADD', = ABCD$$

The Mean Effective Pressure.—This is the average resultant pressure acting on the piston during the stroke, that is, the effective pressure which compresses and discharges the air. It must be evident from the foregoing that the mean effective pressure (*m.e.p.*) is *the* **difference** *between the mean* **forward** *pressure and the mean* **back** *pressure.*

That is

m.e.p. = *mean* **forward** *pressure* — *mean* **back** *pressure*

Accordingly,

$$m.e.p. = \frac{\text{final pressure abs.} \times 1 + \text{hyp. log, no of comp.}}{\text{number of compressions}} - \text{back pres. abs.}$$

or as expressed with the usual symbols

$$m.e.p. = \frac{\mathrm{P} \times 1 + \mathrm{hyp.log}\, r}{r} - \mathrm{b.p.} \quad \ldots (1)$$

in which

m.e.p. = mean effective pressure

P = final pressure per sq. in. in lbs. absolute

r = ratio or number of compressions

b.p. = back pressure in lbs. abs. assumed constant;

(if not, the mean back pressure **m.b.p.** should be used)

It should be remembered that the final pressure and back pressure are taken in *lbs. per sq. in.* **absolute.**

Ques. In equation (1) why does dividing P × 1 + hyp. log *r* by *r* give the mean forward pressure?

Ans. In the formula, r, being the ratio of compression represents also the length of the diagram and when the area of the diagram is divided by its length the result is the mean height of the diagram which measures the mean pressure.

Example.—What is the mean effective pressure (*m.e.p.*) when compressing isothermally to ¼ volume with 14.7 lbs. initial pressure?

$$\text{final or discharge pressure} = 14.7 \times 4 = 58.8 \text{ lbs. abs.}$$

$$\text{back pressure} = 14.7 \text{ lbs. abs.}$$

$$\text{hyp. log of } 4 = 1.3863$$

Substituting in formula (1)

$$m.e.p. = \frac{58.8(1 + 1.3863)}{4} - 14.7 = 20.38 \text{ lbs. per sq. in.}$$

Solution by Calculus.—The mean effective pressure in the example just given may be found by the calculus.

There are two cases in which the area "*under the curve*" may be integrated, as with respect to—

1. Volume, or
2. Pressure

That is, taking for the differential

1. dv, or
2. dp

These differentials (areas) are shown graphically in the diagrams, figs. 32 and 33.

In either case the area "*under curve*" means the area enclosed between the curve, one of its axes, and two vertical or horizontal ordinates.

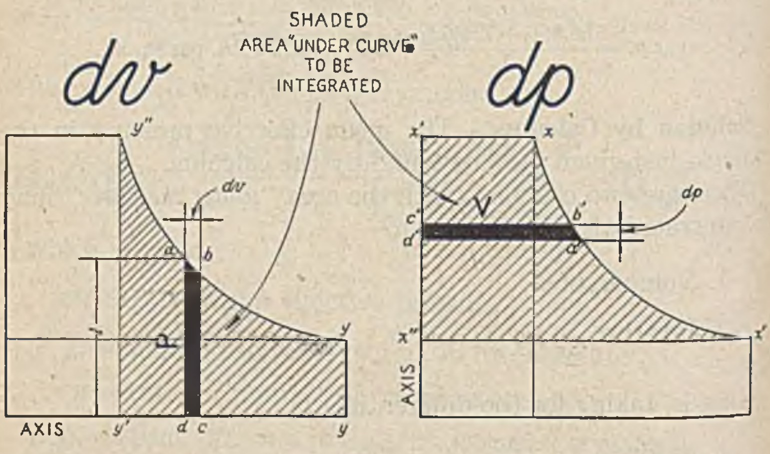
In fig. 32, the area to be integrated is $yyy'y''$, which is made up of an infinite number of narrow strips, as *abcd*. The area of each of these strips is equal to

$$\text{width} \times \text{height}$$

or

$$dc \times l$$

dc being an infinitesimal part of the *volume* is called *dv*, and since the height *l* corresponds to the pressure at that point it is called P.



Figs. 32 and 33.—Diagram indicating two methods of integrating in finding **m.e.p.** by the calculus.

Similarly in fig. 33, the infinitesimally small strip $a'b'c'd'$ is horizontal and its width represents an increment of pressure called *dp*. Evidently, since its length corresponds to volume at that point, it is called V.

CASE I

Integrating with Respect to dv.—The diagram, fig. 34, is a duplicate of fig. 32, since it represents the same conditions of compression as in the example. Since the compression is isothermal

$$n = 1$$

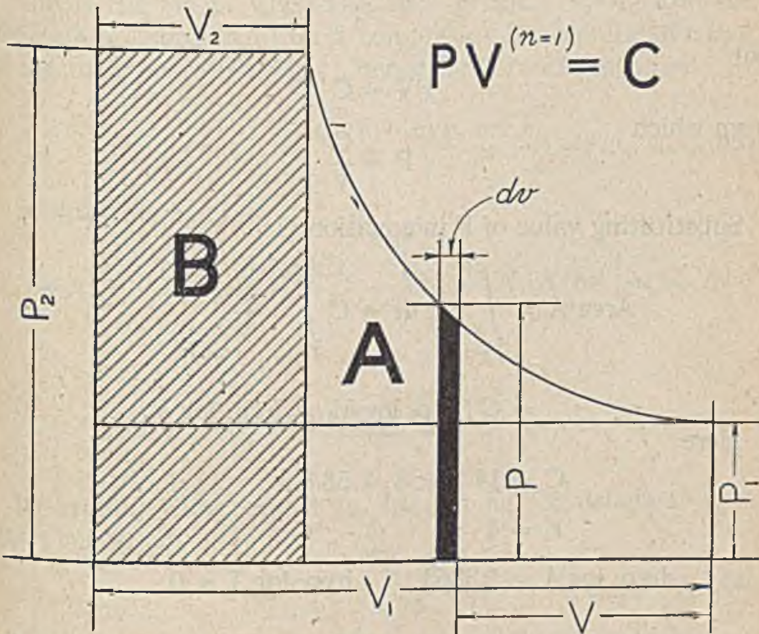


Fig. 34.—Diagram illustrating integration with respect to **dv.**

that is, compression takes place according to **Boyle's law,** from which: *The product of pressure and volume at any point is equal to the product of pressure and volume at any other point,* that is in fig. 34.

$$PV = P_1V_1 = P_2V_2 = C$$

here C is a constant.

Integrating the area A, "under the curve"

$$\text{Area A} = \int_{V_2}^{V_1} P\,dv \qquad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

but

$$PV = C$$

from which

$$P = \frac{C}{V}$$

Substituting value of P in equation (1)

$$\text{Area A} = \int_{V_2}^{V_1} \frac{C}{V}\,dv = C \int_{V_1}^{V_2} \frac{dv}{V}$$

$$= C\,(\text{hyp. log } V_1 - \text{hyp. log } V_2) \ldots\ldots (2)$$

Here

$$C = 14.7 \times 4 = 58.8$$

$$v_1 = 4$$

$$\text{hyp. log } 4 = 1.3863 \qquad \text{hyp. log } 1 = 0$$

Substituting log values in 2

$$\text{Area A} = C\,(1.3863 - 0) = C\,(\text{hyp. log } r)$$

in which r = ratio of compression = 4.

Now in fig. 34, $\log_r$ or 1.3863 = area of A, when area B = 1.

Accordingly, area of entire diagram, that is

$$\text{area } \mathbf{A} + \text{area } \mathbf{B} = C\,(1 + \text{hyp log } r).$$

and

$$m.e.p. = C\,\frac{(1 + \text{hyp. log } r)}{r}$$

This is the *m.e.p.* when the back pressure = 0. However, the back pressure (*b.p.*) on a compressor is theoretically that of the atmosphere (14.7 lbs.), hence the formula becomes

$$m.e.p. = \frac{C\,(1 + \text{hyp. log } r)}{r} - b.p. \ldots\ldots\ldots(3)$$

Substituting in (3)

$$m.e.p. = \frac{58.8\,(1 + 1.3863)}{4} - 14.7 = 20.38 \text{ lbs. per sq. in.}$$

CASE 2

Integrating with respect to dp.—In fig. 35 (shaded) area under curve, or

$$\text{area B} = \int_{P_1}^{P_2} V dp \ldots\ldots\ldots\ldots\ldots\ldots(4)$$

but

$$PV = C$$

from which

$$V = \frac{C}{P}$$

Substituting this value for V in equation (4)

$$\text{Area B} = \int_{p_1}^{p_2} \frac{C}{P}\,dp = C\int_{p_1}^{p_2} \frac{dp}{P}$$

$$= C\,(\text{hyp. log } p_2 - \text{hyp. log } p_1)$$

$$= C \text{ hyp. log } r$$

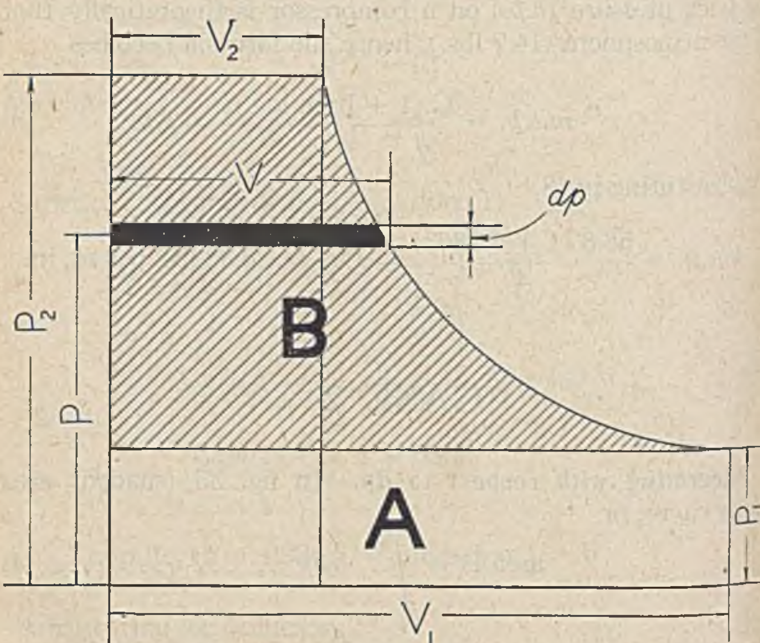


Fig. 35.—Diagram illustrating integration with respect to **dp.**

Here as before

$P_1 = 14.7 \qquad P_2 = 58.8 \qquad r = 58.8 \div 14.7 = 4$

$$\text{hyp. log } r = \text{hyp. log } 4 = 1.3863$$

$$b.p. = 14.7$$

$$m.e.p. = \frac{C(1 + \text{hyp. log } r)}{r} - b.p.$$

Substituting

$$m.e.p. = \frac{58.8\ (1 + 1.3863)}{r} - 14.7 = 20.38$$

ADIABATIC COMPRESSION

By definition, adiabatic compression of air is: *Compression without loss of heat.*

Consider for example a perfectly insulated air cylinder and piston having a full charge of air between the piston and cylinder.

As the piston moves the volume of air becomes smaller and the temperature rises.

On reduction of volume this rise in temperature causes the pressure to rise more than it would if compression took place at constant temperature.

If the temperature were constant, then according to Boyle's law: *At constant temperature the pressure of a gas varies inversely as its volume.* For instance, half the volume, double the pressure.

In the actual compressor air cannot be compressed at constant temperature, consequently there is a loss due to excess pressure caused by the rise of temperature.

Accordingly, if this heat could be removed as fast as generated, so as to maintain a constant temperature during compression, the least amount of work would be required.

If it were possible to compress air without removing any of the heat generated, the maximum loss of power would result in the operation of compressors.

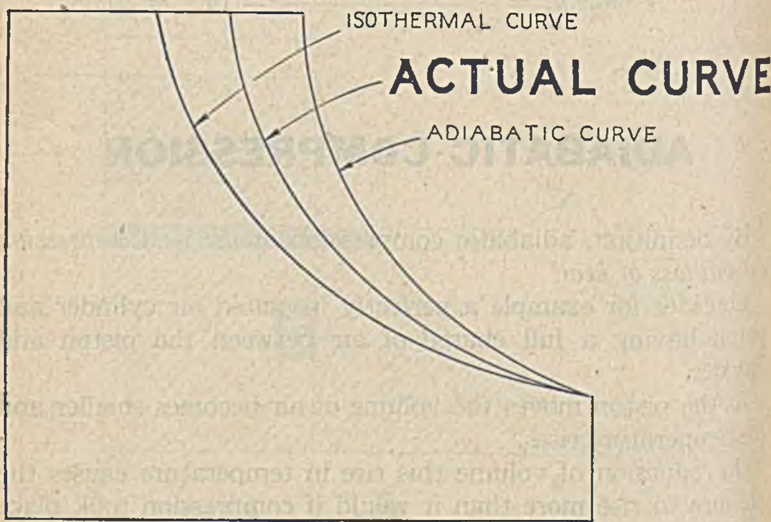


Fig. 36.—Diagram showing location of **actual curve** of compression as lying somewhere between the **isothermal** and **adiabatic** curves—in fact very near the **adiabatic curve.**

In the actual compression of air under practical working conditions, neither of these results are obtained, that is, the curve of compression will lie somewhere between the isothermal and adiabatic curves as shown in fig. 36.

Ques. What is the objection to adiabatic compression?

Ans. A considerable loss takes place because of the effect of rising temperature boosting the pressure above normal. That is, more power is required for adiabatic compression than for isothermal compression.

Moreover, the compressor must work at high temperature which is attended by lubrication difficulties.

Ques. What is the present method of removing heat of compression?

Ans. Water jacketing the cylinder.

Various methods have been tried such as spraying water into the cylinder, circulating cooling water through the piston, etc. The use of cooling spray or so called wet compression has long since been discontinued, as has also the method of circulating water through the piston, because of mechanical difficulties.

Ques. In addition to water cooling, what is the most effective method for reducing the heat of compression?

Ans. By dividing the compression into two or more stages with intercooling between the stages.

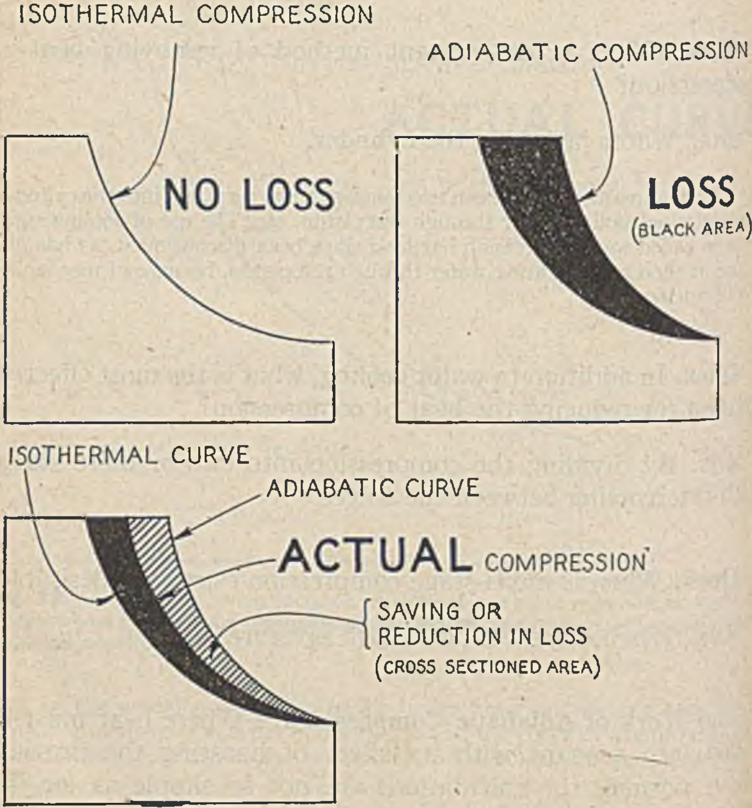
Ques. When is multi-stage compression especially desirable?

Ans. When compressing to high pressures.

The Work of Adiabatic Compression.—Where heat must be taken into account with its effect of boosting the pressure above normal, the calculations are not so simple as for isothermal compression as the compression takes place in accordance of both **Boyle's and Charles' laws.**

Combining both laws—

$$\frac{PV}{T} = \frac{P_1V_1}{T_1}$$



Figs. 37 to 39.—Diagrams showing the effect of heat in causing loss during compression.

here T and T_1 are the initial and final absolute temperatures in degrees Fahrenheit. In the general equation for compression

$$PV^n = C$$

The value of the exponent n depends upon the conditions into which compression takes place, thus if it be isothermal $n = 1$ that is $PV^{n=1} = C$

If all the heat of compression be retained (adiabatic compression) n becomes the ratio of the specific heat at constant pressure to the specific heat at constant volume.

For dry air $n = \frac{C_p}{C_v} = \frac{.2375}{.16895} = 1.4057.$

commonly taken as 1.41, as has previously been explained.

Ques. What is the value of n in practice?

Ans. The value of n on the ordinary single stage air compressor as given by Church is 1.33 and Union as 1.25.

Ques. Upon what does the exact value depend?

Ans. It varies with the compressor and depends upon the size of the cylinder, speed of the machine, design of the water jacket, and temperature of the cooling water.

In present day reciprocating compressors, the air is compressed very nearly adiabatically. The water jacket brings the compression curve somewhat below the adiabatic but the excess pressure required to force the air through valves makes the work done in compressing and discharging the air very close to that which would be calculated on the assumption of adiabatic compression.

Note the following values of n as given by O'Neil.

Values of N

$n = 1$ for isothermal compression.

$n = 1.406$ for adiabatic compression for dry air.

n = 1.3947 for adiabatic compression of air having 36% relative humidity at 68° F. This may be considered an average condition in a temperate climate.

Net Work of Compression by Integration.—The net work of compression consists of three items:

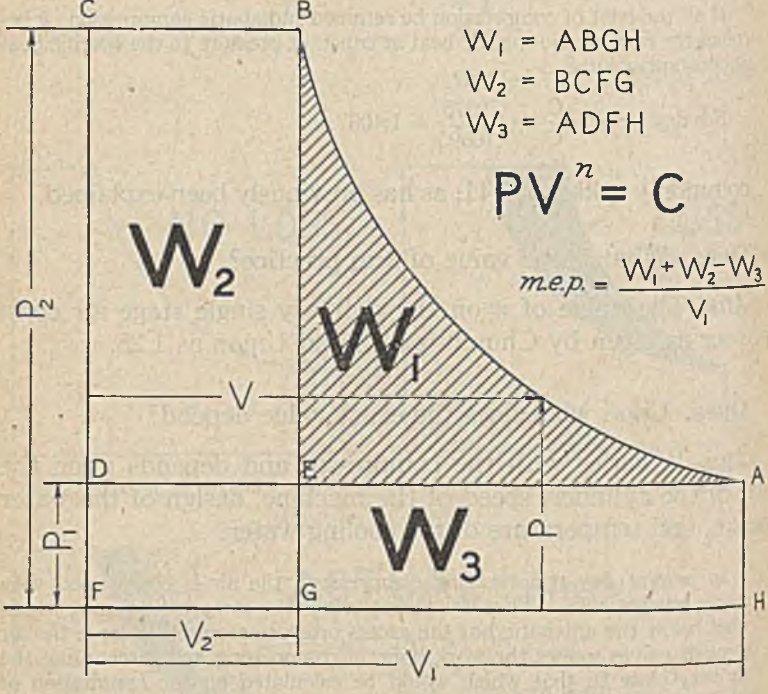


Fig. 40.—Diagram illustrating calculus method of calculating the work and **m.e.p.** of adiabatic compression.

1. Work of compressing
2. Work of discharging
3. Work done by atmosphere during admission.

Expressed as a formula

$$W_n = W_1 + W_2 - W_3$$

in which

W_n = net work of compression
W_1 = work of compressing
W_2 = work of discharging
W_3 = negative work or work done on the piston by the atmosphere during admission.

In the actual compressor the air as before stated is compressed very nearly adiabatically.

Although the effect of water jacketing is to bring the actual compression curve somewhat below the adiabatic curve, opposing this is the excess pressure required to force the air through the valves which makes the net result almost the same as adiabatic compression.

Accordingly in the theoretical diagram the adiabatic compression curve is used as shown in fig. 40.

In the clearance diagram for single stage compression

Area ABGH is the work of compressing
" BCFG " " " " expulsion
" ADFH " " " done on the piston by the incoming charge.

These areas as tabulated are denoted by W_1, W_2 and W_3 respectively.

Ques. How is the work of W_1 of compression obtained?

Ans. By integrating under the curve $PV^n = C$, that is between the limits V_1 and V_2.

Ques. How is the work (W_2) of expulsion obtained?

Ans. It is the product of P_2 multiplied by V_2.

Ques. How is the negative work (W_3) done by the atmosphere obtained?

Ans. It is the product of P_1 multiplied by V_1.

Ques. How is the mean effective pressure obtained?

Ans. By adding the first two areas, deducting the third area and dividing by the length of the diagram, thus:

$$m.e.p. = \frac{(W_1 + W_2) - W_3}{V_1}$$

Thus

$$m.e.p. = \frac{n}{n-1} P_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

For normal air it becomes

$$m.e.p. = 3{,}534\ P_1 \left[\left(\frac{P_2}{P_1} \right)^{.283} - 1 \right]$$

Expressing the ratio of compression $\frac{P_2}{P_1}$ by r, it becomes

$$m.e.p. = 3.534\ P_1 \left(r^{.283} - 1 \right)$$

Volumes mean Pressures, Temperatures, etc.

for

Air Compression from one Atmosphere and 60° Fahr.

Gauge Pressure	Absolute Pressure	Pressure in Atmospheres	Volume with Air at Constant Temperature	Volume with Air not Cooled	Mean Pressure per Stroke. Air at Constant Temperature	Mean Pressure per Stroke. Air not cooled	Final Temperatures. Air not cooled	Gauge Pressure
0	14.7	1.	1.	1.	0.	0.	60	0
5	19.7	1.34	.7462	.81	4.3	4.5	106	5
10	24.7	1.68	.5952	.69	7.62	8.27	145	10
15	29.7	2.02	.495	.606	10.33	11.51	178	15
20	34.7	2.36	.4237	.543	12.62	14.4	207	20
25	39.8	2.7	.3703	.494	14.59	17.01	234	25
30	44.7	3.04	.3289	.4638	16.34	19.4	255	30
35	49.7	3.381	.2957	.42	16.92	21.6	281	35
40	54.7	3.721	.2687	.393	19.32	23.66	302	40
45	59.7	4.061	.2462	.37	20.52	25.59	321	45
50	64.7	4.401	.2272	.35	21.79	27.39	339	50
55	69.7	4.741	.2109	.331	22.77	29.11	357	55
60	74.7	5.081	.1968	.3144	23.84	30.75	375	60
65	79.7	5.423	.1844	.301	24.77	31.69	389	65
70	84.7	5.762	.1735	.288	26.	33.73	405	70
75	89.7	6.102	.1639	.276	26.65	35.23	420	75
80	94.7	6.442	.1552	.267	27.33	36.6	432	80
85	99.7	6.782	.1474	.2566	28.05	37.94	447	85
90	104.7	7.122	.1404	.248	28.78	39.18	459	90
95	109.7	7.462	.134	.24	29.53	40.4	472	95
100	114.7	7.802	.1281	.232	30.07	41.6	485	100
105	119.7	8.142	.1228	.2254	30.81	42.78	496	105
110	124.7	8.483	.1178	.2189	31.39	43.91	507	110
115	129.7	8.823	.1133	.2129	31.98	44.98	508	115
120	134.7	9.163	.1091	.2073	32.54	46.04	529	120
125	139.7	9.503	.1052	.202	33.07	47.06	540	125
130	144.7	9.843	.1015	.1969	33.57	48.1	550	130
135	149.7	10.183	.0981	.1922	34.05	49.1	560	135
140	154.7	10.523	.098	.1878	34.57	50.02	570	140
145	159.7	10.846	.0921	.1837	35.09	51.	580	145
150	164.7	11.204	.0892	.1796	35.48	51.89	589	150

Indicated Horse Power "Per 100 Cu. Ft."—The expression "per 100 cu. ft." means 100 cu. ft. displacement per minute and is a measure of capacity frequently used in horse power calculations. On this basis

$$i.h.p. = \frac{\text{PD} \times (m.e.p. \times 144)}{33{,}000}$$

in which

$i.h.p.$ = indicated horse power, that is of the air cylinder.

$m.e.p.$ = mean effective pressure in lbs. per sq. in.

PD = piston displacement in cu. ft. per minute.

Now if the piston displacement be made 100 cu. ft. per minute, then the $i.h.p.$ per 100 cu. ft. piston displacement per minute

$$= \frac{100 \times (m.e.p. \times 144)}{33{,}000} = \frac{m.e.p.}{2.292}$$

Hence to find $i.h.p.$, to compress 100 cu. ft. of air per minute:

Rule.—Divide the $m.e.p.$ by 2.292.

Example.—What horse power is required to compress 100 cu. ft. of air per minute to 100 lbs. gauge pressure. Initial pressure 14.7 lbs.

Look in table on page 105 for "mean pressure per stroke air not cooled" and find 41.6 lbs. Substituting this in rule

$$i.h.p. = 4.16 \div 2.292 = 18.1$$

Compare this with the table on page 107 which is based on a value of n of 1.3947. Values for corresponding brake horse power given in the table on page 110. Here brake horse power should be interpreted as the brake horse power in the case of a power-driven machine or steam indicated horse power in the case of a direct connected steam-driven compressor.

Mean Effective Pressure

To compress air from atmosphere (14.7 lbs. per sq in.) to various pressures.

and Theoretical Horsepower

Required to compress air at the rate of 100 cu. ft. per min. from atmospheric pressure to various pressures.

Discharge pressure lbs.—gauge	Discharge pressure lbs.—absolute	Discharge pressure atmospheres—absolute	Single- or multi-stage compression isothermal		Single-stage compression adiabatic*		Two-stage compression adiabatic*		Theoretical intercooler gauge pressure	% of power saved by two-stage over single stage adiabatic compression
			M. E. P. per sq. inch	Theoretical Hp. per 100 cu. ft.	M. E. P. per sq. inch	Theoretical Hp. per 100 cu. ft.	M.E.P. per sq. in. referred to low-pressure air cylinder	Theoretical Hp. per 100 cu. ft.		
5	19.7	1.34	4.13	1.8	4.48	1.96	....	....	....	....
10	24.7	1.68	7.57	3.3	8.21	3.58	....	....	....	....
15	29.7	2.02	10.31	4.5	11.4	5.0	....	....	....	....
20	34.7	2.36	12.62	5.5	14.3	6.2	....	....	....	....
25	39.7	2.70	14.68	6.4	16.9	7.4	....	....	....	....
30	44.7	3.04	16.30	7.1	19.2	8.4	....	....	....	....
35	49.7	3.38	17.90	7.8	21.4	9.3	....	....	....	....
40	54.7	3.72	19.28	8.4	23.4	10.2	....	....	....	....
45	59.7	4.06	20.65	9.0	25.2	11.0	....	....	....	....
50	64.7	4.40	21.80	9.5	27.0	11.8	....	....	....	....
55	69.7	4.74	22.95	10.0	28.7	12.6	....	....	....	....
60	74.7	5.08	23.90	10.4	30.3	13.3	....	....	....	....
65	79.7	5.42	24.80	10.8	31.9	13.9	....	....	....	....
70	84.7	5.76	25.70	11.2	33.3	14.6	29.2	12.8	20.6	12.3
75	89.7	6.10	26.62	11.6	34.7	15.2	30.3	13.3	21.6	12.5
80	94.7	6.44	27.52	12.0	36.0	15.7	31.3	13.7	22.7	12.7
85	99.7	6.78	28.21	12.3	37.3	16.3	32.3	14.1	23.6	13.5
90	104.7	7.12	28.93	12.6	38.6	16.9	33.2	14.5	24.5	14.2
95	109.7	7.46	29.60	12.9	39.8	17.4	34.2	14.9	25.5	14.4
100	114.7	7.80	30.30	13.2	40.9	17.9	35.0	15.3	26.3	14.5
110	124.7	8.48	31.42	13.7	43.2	18.9	36.7	16.1	28.1	14.8
120	134.7	9.16	32.60	14.2	45.2	19.8	38.3	16.8	29.8	15.1
130	144.7	9.84	33.75	14.7	47.2	20.7	39.6	17.3	31.5	16.4
140	154.7	10.52	34.67	15.1	49.2	21.5	40.8	17.9	32.9	16.7
150	164.7	11.20	35.59	15.5	51.0	22.3	42.3	18.5	34.5	17.1
160	174.7	11.88	36.30	15.8	....	....	43.6	19.0	36.1	....
170	184.7	12.56	37.20	16.2	....	....	44.7	19.5	37.3	....
180	194.7	13.24	38.10	16.6	....	....	45.8	20.0	38.8	....
190	204.7	13.92	38.80	16.9	....	....	46.8	20.4	40.1	....
200	214.7	14.60	39.50	17.2	....	....	47.8	20.9	41.4	....
250	264.7	18.00	42.70	18.6	....	....	52.5	22.7	47.6	....
300	314.7	21.40	45.30	19.7	....	....	56.5	24.5	53.4	....
350	364.7	24.81	47.30	20.6	....	....	59.6	26.1	58.5	....
400	414.7	28.21	49.20	21.4	....	....	62.7	27.4	63.3	....
450	464.7	31.61	51.20	22.3	....	....	65.3	28.6	67.8	....
500	514.7	35.01	52.70	22.9	....	....	67.8	29.6	72.1	....
550	564.7	38.41	53.75	23.4	....	....	70.0	30.6	76.3	....
600	614.7	41.81	54.85	23.9	....	....	72.3	31.3	80.5	....

*Based on a value for *n* of 1.3947.

Approximate Displacement in Cubic Feet per Minute

Double-Acting Pistons Working in Single Cylinder at Various Piston Speeds. Piston Rods Not Deducted

Diam. of Cyl. Ins.	Piston Speed, Feet per Minute = 2 × stroke (ft.) × r.p.m. (for double acting cylinder)																		
	200	225	250	275	300	325	350	375	400	425	450	475	500	550	600	650	700	750	800
	Displacement in Cubic Feet per Minute																		
3	9.8	11.1	12.3	13.5	14.7	16	17.2	18.5	19.6	21	22.2	23.4	24.6	27	29.4	31.8	34.4	36.8	39.2
4	17.4	19.7	21.8	24	26.2	28.5	30.7	32.9	34.8	37.2	39.4	41.5	43.6	48	52.4	56.7	61.4	65.2	69.5
5	27.2	30.8	34.2	37.5	41	44.5	48	51.4	54.4	58	61.6	65	68.4	75	82	88.7	96	102	109
6	39.2	44.4	49	54	59	64	69	74	78.4	84	88.8	93	98	108	118	128	138	148	157
7	53.4	60.5	67	74	80	87	94	100.0	106.8	114	121	127	134	148	160	174	188	201	214
8	70	79	87.5	96	105	114	123	132	140	149	158	166	175	192	210	227	246	262	280
9	88	100	110	122	133	144	155	166	176	188	200	210	220	244	266	287	310	332	352
10	109	123	136	150	164	178	192	206	218	233	246	260	272	300	328	354	384	409	436
11	132	149	165	182	198	215	232	250	264	282	298	315	330	364	396	430	464	496	528
12	156	177	196	216	236	256	276	296	312	335	354	374	392	432	472	514	552	585	625
13	184	208	230	255	276	300	325	345	368	392	416	440	460	510	552	601	650	690	736
14	212	240	266	295	320	348	375	400	424	455	480	510	532	590	640	695	750	795	850
15	244	276	305	337	368	400	432	460	488	521	552	580	610	674	736	798	864	915	976
16	278	315	350	385	420	455	490	525	556	595	630	660	700	790	840	910	980	1040	1110
17	314	355	395	435	472	512	550	591	628	672	710	750	790	870	944	1020	1100	1180	1260
18	352	400	440	485	530	575	620	660	704	755	800	840	880	970	1060	1150	1240	1320	1410
19	392	445	490	540	590	640	690	740	784	840	890	930	980	1080	1180	1280	1380	1470	1570
20	436	490	545	600	656	710	765	820	872	930	980	1040	1090	1200	1312	1420	1530	1640	1750
22	530	595	660	725	790	860	930	990	1060	1120	1190	1260	1320	1450	1580	1720	1860	1990	2040
24	628	710	790	865	940	1020	1100	1180	1256	1340	1420	1500	1580	1730	1880	2040	2200	2360	2520
26	738	830	920	1020	1100	1200	1300	1390	1476	1570	1660	1750	1840	2040	2200	2400	2600	2770	2970
28	854	960	1070	1180	1280	1390	1500	1600	1780	1820	1920	2030	2140	2360	2560	2780	3000	3200	3420
30	980	1110	1230	1350	1480	1600	1720	1840	1960	2100	2220	2340	2460	2700	2960	3200	3440	3690	3920
32	1120	1260	1400	1540	1670	1820	1960	2100	2240	2380	2520	2660	2800	3080	3340	3630	3920	4200	4480
34	1260	1420	1570	1740	1800	2050	2220	2360	2520	2690	2840	3000	3140	3480	3780	4000	4440	4720	5040
36	1420	1600	1770	1950	2120	2300	2500	2650	2840	3000	3200	3360	3540	3900	4240	4620	5000	5320	5680
38	1560	1780	1960	2170	2360	2560	2750	2950	3120	3350	3540	3750	3920	4340	4720	5110	5500	5850	6250
40	1740	1970	2180	2400	2620	2850	3060	3280	3480	3720	3940	4150	4360	4800	5240	5680	6120	6520	6960

A convenient formula for calculating the displacement of one double-acting cylinder is:— The square of the diameter (inches) × stroke (inches) × rpm × .0009 = piston displacement (c.f.m.). This makes a reasonable allowance for the piston rod on all piston sizes from about 5″ diameter to about 30″ diameter. If there were no allowance for the piston rod the multiplier would be .0009090.

Displacement of Air Compressors.—The capacity of an air compressor is usually expressed in cubic feet per minute. Strictly speaking this is the volume displaced by the net area of the compressor piston.

Ques. What is the net area of the piston

Ans. The area of the piston—½ area of the piston rod.

In ordinary calculations this refinement is not considered.

Example.—Find the displacement of a 12 x 11 single stage double acting compressor, running 300 *r.p.m.*; area rod 2 ins.

Area 12 in. piston	$= .7854 \times 12^2 =$	....	113.10 sq. in.
Half Area 2 in. rod	$= \frac{1}{4}\pi \times 2^2 =$	....	1.57 sq. in.
		Net area	111.53 sq. in.

or $111.53 \div 144 = .77$ sq. ft.

$$\text{displacement} = \underset{\text{(in. sq. ft.)}}{\text{net piston area}} \times \text{stroke} \times \text{strokes}$$

$$= .77 \times \frac{11}{12} \times 2 \times 300$$

$$= 423.5 \text{ cu. ft. per minute.}$$

A table of approximate displacements is given on page 108 for convenient reference.

Calculation of Horse Power.—The following illustrates how to calculate the horse power required for compressors.

1. For isothermal compression

2. For adiabatic compression

Example.—What horse power is required to compress isothermally 400 cu. ft. of free air per minute into a receiver against a gauge pressure of 100 lbs. per sq. in. Assume theoretically that admission takes place at atmospheric pressure.

One method of solving such problems is as follows:

Compression ratio = (100 + 14.7) ÷ 14.7 = 7.8 = r.

Hypobolic logarithm of 7.8 (from table page 84) = 2.0541

1 + hyp. log. of 7.8 = 1 + 2.0541 = 3.0541.

$$m.e.p. = \frac{P \text{ abs.} \times 1 + \text{hyp. log } r}{r} - \text{b.p.} \quad \ldots\ldots (1)$$

in which

$m.e.p.$ = mean effective pressure in lbs. per sq. in. in air cylinder.

P = final or discharge pressure in lbs. per sq. in. abs.

r = ratio of compression

b.p. = back pressure in lbs. abs. per sq. in.

Approximate brake horse power required by air Compressors*

Figures given are b.h.p. per 100 cu. ft. of free air per min. actually delivered.

Altitude Feet	SINGLE-STAGE lbs. per sq. in. gauge			TWO-STAGE lbs. per sq. in gauge			
	60	80	100	60	80	100	125
0	16.3	19.5	22.1	14.7	17.1	19.1	21.3
1000	16.1	19.2	21.7	14.5	16.8	18.7	20.9
2000	15.9	18.9	21.3	14.3	16.5	18.4	20.5
3000	15.7	18.6	20.9	14.0	16.1	18.0	20.0
4000	15.4	18.2	20.6	13.8	15.8	17.7	19.6
5000	15.2	17.9	20.3	13.5	15.5	17.3	19.2
6000	15.0	17.6	20.0	13.3	15.2	17.0	18.8
7000	14.7	17.3	19.6	13.0	14.9	16.6	18.4
8000	14.5	17.1	19.3	12.7	14.6	16.2	18.0
9000	14.3	16.8	18.9	12.5	14.3	15.9	17.6
10000	14.1	16.5	18.6	12.3	14.1	15.6	17.2
12000	13.6	15.9	17.9	11.8	13.5	15.0	16.5
14000	13.1	15.2	17.2	11.3	12.9	14.3	15.7

*Note: Brake horsepower will vary considerably with the size and type of compressor.

Substituting in formula (1)

$$m.e.p. = \frac{114.7 \times 3.0541}{7.8} - 14.7 = 30.2 \text{ lbs.}$$

The size of the compressor piston depends upon conditions, but for calculation of horse power, any size piston may be assumed. Accordingly, for simplicity assume piston to have an area of 1 sq. ft. that is, 144 sq. ins. so that 400 ft. per min. will correspond to the piston speed.

$$\text{Now } i.h.p. = \frac{\text{piston area} \times m.e.p. \times \text{piston speed}}{33{,}000} = (2)$$

Substituting in formula (2)

$$i.h.p. = \frac{144 \times 30.2 \times 400}{33{,}000} = 52.7$$

In a previous section (see page 106) it was shown that for 100 cu. ft. piston displacement

$$i.h.p. = \frac{m.e.p.}{2.292} \dots\dots\dots\dots (3)$$

Substituting in this formula (3) and multiplying by 4 for 400 cu. ft.

$$i.h.p. = \frac{30.2}{2.292} \times 4 = 52.7$$

It must be understood that this is the theoretical power required for compressing the air and to which must be added extra power to overcome friction of compressor and driver.

For illustration assume efficiency 80%.

$$\text{loss} = 100 - 80 = 20\%$$

$$\text{Power required} = 52.7 \times (1 + 20\%) = 63.2$$

Example.—What power is required for compression as in the previous example but *adiabatically* instead of *isothermally*?

This is easily obtained by aid of the mean effective pressure table (prepared by O'Neil) on page 107.

For isothermal compression the *m.e.p.* as calculated in the previous example = 30.2 lbs. per sq. in. Referring to the table the *m.e.p.* for adiabatic compression (100 lbs. gauge pressure) = 40.9 lbs. per sq. in.

The horse power required is obtained by multiplying the horse power found for isothermal compression by the ratio of the two *m.e.p.* that is for adiabatic compression

$$i.h.p. = \begin{cases} \text{theoretical} = 52.7 \times (40.9 \div 30.2) = 71.4 \\ \text{delivered} = 63.2 \times (40.9 \div 30.2) = 85.6 \end{cases}$$

Efficiency.—In general the term efficiency may be defined as: *The ratio of the useful work performed by a prime mover to the energy expended, that is, the* **output** *divided by the* **input.**

It is important to distinguish between several kinds of efficiency and with respect to compressors the following should be considered:

1. Volumetric efficiency.
2. Compression efficiency.
3. Mechanical efficiency.
4. Slippage efficiency.
5. Overall efficiency.

Volumetric Efficiency.—By definition volumetric efficiency is: *The ratio of the actual number of cubic feet of free air (at* 14.7 *lbs. per sq. in. abs. and* 60° *Fahr.) compressed per unit of time to the number of cubic feet of piston displacement during that time.*

Expressed as a formula

$$E_v = \frac{V}{D}$$

in which

E_v = volumetric efficiency.
V = actual cubic feet of free air at 14.7 lbs. per sq. in. abs. and 60° Fahr. compressed per minute.
D = displacement in cubic feet per minute.

Volumetric efficiency depends upon:

1. Pressure at end of admission.
2. Temperature at end of admission.
3. Quality of the air at end of admission.
4. Clearance volume.

Ques. What effect has pressure on the *charge**?

Ans. The lower the absolute pressure the less the volumetric efficiency.

Ques. Can *free air* be admitted at atmospheric pressure?

Ans. No

Ques. Why?

Ans. Frictional resistance encountered by the air on entering reduces its pressure.

Thus in fig. 41 the charge is admitted (assuming no clearance) from A to B. The terminal admission pressure indicated at B, is *below* atmospheric pressure. On the compression stroke the piston moves to some point C,

*Note—The term charge means the amount of air taken in at the end of the admission stroke.

while the pressure of the charge is building up to atmospheric pressure. Accordingly the volume of normal air admitted per stroke is less than the cylinder displacement per stroke LL and under such conditions the volumetric efficiency would be *ll* ÷ LL less than 100%.

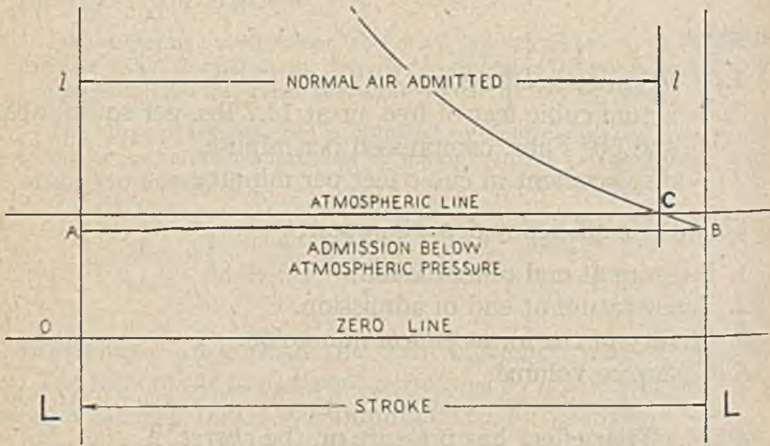


Fig. 41.—Diagram illustrating frictional resistance during admission in reducing the charge.

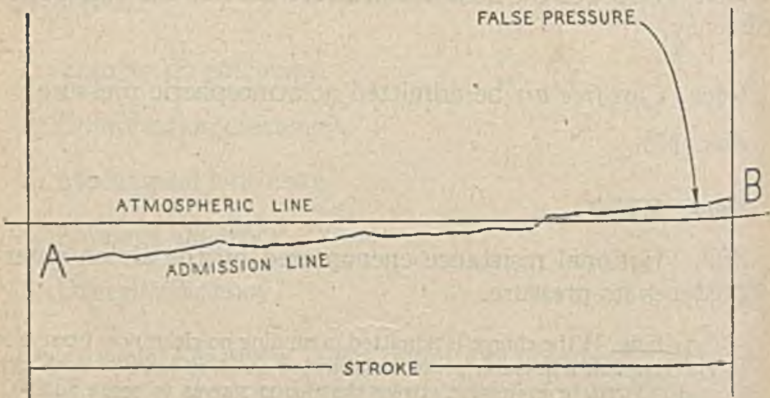


Fig. 42.—Admission line illustrating false or boosted pressure at the end of admission.

Ques. If the admission line on a diagram gradually approached the atmospheric line, what are the reasons?

Ans. 1. False or boosted pressure due to the charge receiving heat from the cylinder walls; 2, gradual reduction in piston speed as the piston nears the end of the admission stroke; 3. inertia effect.

Thus, in fig. 31, at the beginning of admission the column of air in the intake pipe being at rest its static inertia offers resistance which coupled with the frictional resistance in entering reduces its pressure (below atmospheric).

As admission progresses the incoming charge receives heat from the cylinder walls which boosts the pressure; also the slowing down of the piston velocity as it nears the end of the stroke causes the dynamic inertia of the incoming charge to further boost the pressure.

Under exceptional conditions, inertia of the charge coupled with heating could possibly build up a false pressure above atmospheric as indicated at B, fig. 31.

Ques. What is the amount of error with respect to admission shown on indicator diagrams and why?

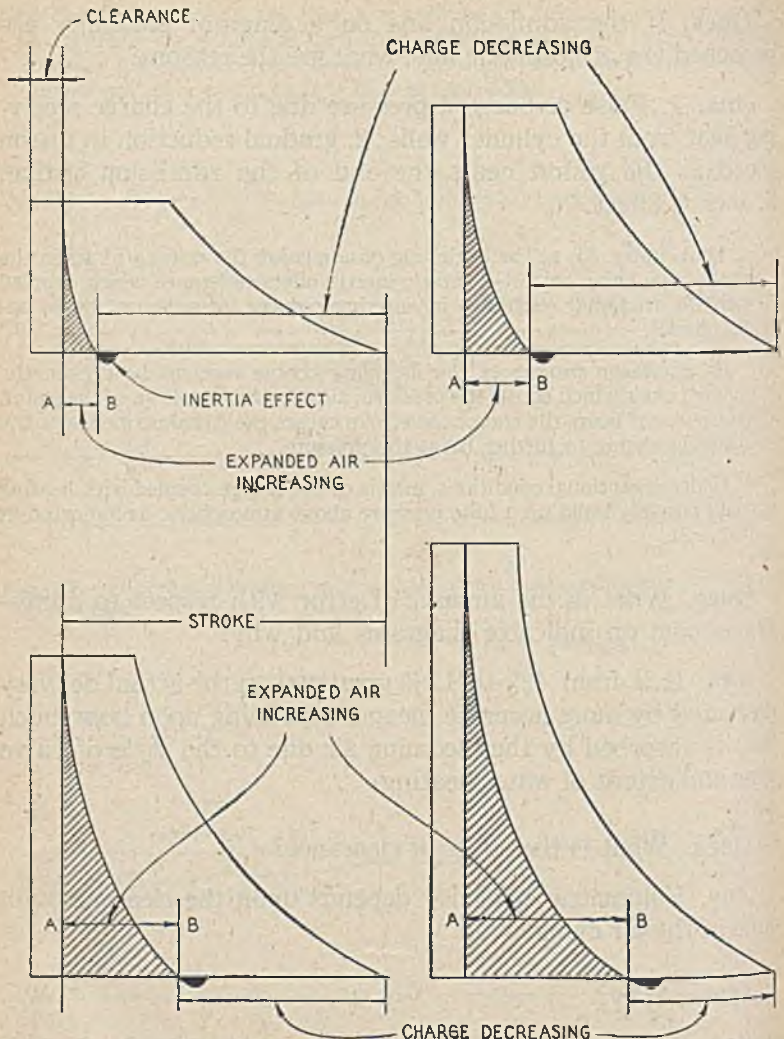
Ans. It is from 4% to 12% greater than the actual delivery measured by more accurate means, depending upon how much heat is absorbed by the incoming air due to the style of valve gear and extent of water heating

Ques. What is the effect of clearance?

Ans. Volumetric efficiency depends upon the clearance volume in the air cylinder.

Ques. Why?

Ans. The greater the clearance volume the greater the volume of the cylinder occupied by the clearance air which expands



Figs. 43 to 46.—Diagrams showing effect of expanding clearance air as the compression ratio increases.

and prevents the entrance of free air during the early part of the admission stroke.

In figs. 43 to 46, the volume of the expanded air in the cylinder is indicated by AB, which as seen increases with the ratio of compression.

This decreases the amount of charge that can be admitted. In the diagrams the little loop (solid block) is the sudden drop due to the static inertia of the column of air in the admission pipe at that moment.

Ques. What should be considered with respect to the admission pipe and why?

Ans. It should be as short as possible, free of elbows, and if long, the size should be increased to avoid frictional losses.

Compression Efficiency.—By definition compression efficiency is: *The ratio of the theoretical power required to compress the amount of air actually delivered to the actual power developed in the air cylinder as shown by indicator diagrams.*

Compression efficiency, according as the theoretical power is based, may be classed as:

1. Isothermal compression efficiency.

2. Adiabatic compression efficiency.

For convenience the factors may be expressed per 100 cu. ft. of free air delivered per minute.*

Compression efficiency expressed as a formula is:

$$E_c = \frac{\text{T.H.P.}}{\text{D.H.P.}}$$

*NOTE.—For more information on compression efficiency see Compressed Air Data by F. W. O'Neil.

in which

E_c = compression efficiency.

T.H.P. = theoretical horse power.

D.H.P. = delivered horse power, or 1, indicated horse power at steam cylinder of a steam driven unit, or 2 brake horse power at the shaft of a power driven unit.

Ques. Upon what does compression efficiency depend?

Ans. Upon the water jacket and cooling devices.

Ques. What type of compressor is sometimes used principally to increase compression efficiency?

Ans. Multi-stage compressors.

Ques. How is compression efficiency determined with respect to a compression diagram?

Ans. In fig. 47, ABCDE is a diagram as traced by the indicator. Starting at A, draw the isothermal diagram AGCDF then as indicated by the diagram

$$\text{Compressor efficiency} = \frac{\text{area AGCDF}}{\text{area ABCDE}}$$

NOTE.—A high volumetric efficiency, for instance, does now necessarily imply a high efficiency compressor. As a matter of fact, a lower volumetric efficiency may indicate liberal ports and valves and consequently very small pressure losses in the compressor, resulting in a high over-all efficiency.

NOTE.—When compressors were bought and sold on the "swept" volume of the air cylinders, the volumetric efficiency was a matter of great importance to the purchaser. Nowadays compressor contracts are usually based on compressor output, and provided that the compressor yields its full specified output of air, the volumetric efficiency is of little interest to the user. As long as the purchaser obtains the specified output of air compressed to the required pressure by the consumption of not more than the specified amount of steam or electrical energy, he need not trouble himself about the volumetric efficiency or any other of the various "efficiencies" which are so often quoted.

According to one manufacturer "actual compression curves will follow the adiabatic curve quite closely as the water jacket has little effect other than to facilitate lubrication.

Mechanical Efficiency.—By definition mechanical efficiency is: *The ratio of the air indicated horse power divided by the steam horse power*, or the brake horse power in the case of a power driven machine. That is

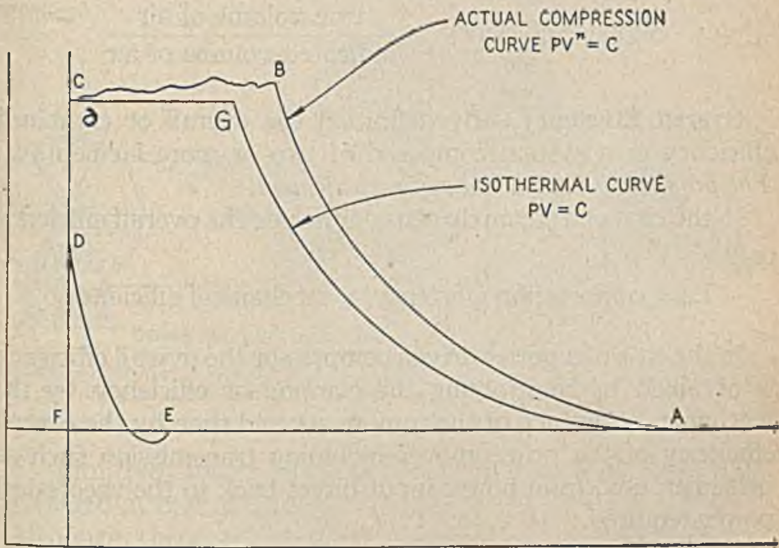


Fig. 47.—Diagram for determining compression efficiency.

For steam driven compressor

$$\text{mechanical efficiency} = \frac{\text{air indicated horse power}}{\text{steam indicated horse power}}$$

and the mechanical efficiency of a power driven compressor is

$$\text{mechanical efficiency} = \frac{\text{air indicated horse power}}{\text{brake h.p. delivered to compressor shaft}}$$

Slippage Efficiency.—By definition, slippage efficiency is: *The ratio of the air actually measured to the apparent volume accounted for by the indicator diagram.*

Expressed as a formula,

$$\text{Slippage efficiency} = \frac{\text{true volume of air}}{\text{indicated volume of air}}$$

Overall Efficiency.—By definition the overall or combined efficiency of a system composed of two or more elements is: *The product of the efficiency of each element.*

In the case of a steam driven compressor the overall efficiency or

$$E_o = \text{compression efficiency} \times \text{mechanical efficiency}$$

In the case of a power driven compressor the overall efficiency is obtained by multiplying the compressor efficiency by the mechanical efficiency of the compressor and then by the overall efficiency of the prime mover including transmission such as belt idler, etc., from power input direct back to the theoretical power required.

Example.—In a certain pumping plant as shown in fig. 48, there is a power pump connected by belt drive to an engine. If the efficiency of various elements be: Boiler 73%; engine thermal 15%; engine mechanical 95%; belt drive 98%; pump 95%, what is the combined or overall efficiency?

The product of the various efficiencies listed in the example is the combined efficiency, that is

$$\text{combined efficiency} = \underset{\text{boiler}}{.73} \times \underset{\text{engine}}{.15 \times .95} \times \underset{\text{belt}}{.98} \times \underset{\text{pump}}{.95} = .097 \text{ or } 9.7\%$$

Multi-Stage Compression.—The object of dividing the compression of air into two or more stages is to obtain the saving due to a nearer approach to isothermal compression. As previously explained for two stage compression the method is as follows:

1. Air is compressed to a certain pressure in a low pressure cylinder.

2. Discharged into an inter-cooler which reduces the temperature.

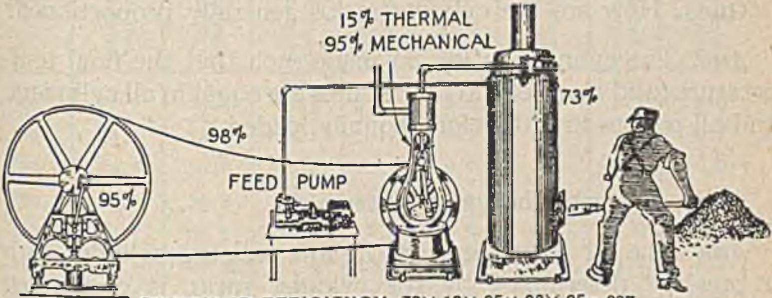


Fig. 48.—Diagram of steam plant illustrating method of determining the overall or combined efficiency.

3. Admitted into a high pressure cylinder compressed and discharged at high pressure.

Although theoretically there is a gain in multi-stage compression even for low final pressure, the saving in such cases is so small as to be offset by the additional expense involved.

Ques. What is the final pressure range for single stage compressors?

Ans. From 80 to 100 lbs. gauge.

Multi-stage compression should be used for higher pressures.

Ques. State the advantages of multi-stage compression.

Ans. 1. Reduces final temperature; 2, reduces horse power required; 3, partially removes entrained moisture; 4, increases volumetric efficiency by reducing clearance expansion loss; 5, generally reduces maximum piston loads below those which would be found in a single stage compressor of equal displacement.

Ques. How are the cylinder ratios generally proportioned?

Ans. The cylinder ratios are made such that the final temperatures and mean effective pressures are equal in all cylinders, and all pistons are, therefore, equally loaded.

Ques. Describe the compression cycle.

Ans. The air compressed in the low pressure cylinder with a pressure determined by the cylinder ratio, is discharged through the discharge valves to an inter-cooler where it is split up into thin streams passing over cold surfaces, whence it is admitted to the high pressure cylinder. It is there compressed to a higher pressure, and again reaches a temperature about the same as that attained in the first cylinder. In two stage machines, this air will be discharged directly to the receiver without further cooling unless conditions are such as to render necessary the use of an after-cooler.

Ques. What is the construction of the inter-cooler?

Ans. Modern practice involves a nest of tubes through which cold water circulates and over and between which the stream of air passes, complete breaking up and subdivision of the stream being secured by baffle plates, and the tubes themselves.

Loss of work due to heat in compressing air from atmospheric pressure to various gauge pressures by simple and compound compression. Air in each cylinder initial temperature 60° F.

Gauge Pressure	One Stage		Two Stage		Three Stage		Four Stage	
	Percentage of Work Lost in Terms of							
	Isothermal Compression	Adiabatic Compression	Isothermal Compression	Adiabatic Compression	Isothermal Compression	Adiabatic Compression	Isothermal Compression	Adiabatic Compression
60	29.9	23.0	13.4	11.8	8.6	7.9	4.7	4.5
70	30.6	23.4	14.1	12.4	8.7	8.0	6.1	5.7
80	32.7	24.6	14.7	12.8	9.7	8.9	6.4	6.0
90	34.7	25.8	16.1	13.8	10.5	9.5	7.3	6.8
100	36.7	26.8	16.9	14.5	10.9	9.8	7.8	7.3
125	41.1	29 2	18.5	15.6	11.6	10.4	8.8	8.1
150	44.8	30.9	20.1	16.7	12.3	10.9	9.1	8.4
200	51.2	33.9	22.2	18.1	14.0	12.3	10.5	9.5
300	61.2	37.9	25.7	20.5	16.6	14.2	12.0	10.7
400	68.7	40.7	28.9	22.4	18.2	15.4	13.1	11.5
500	70.6	41.4	31.2	23.8	19.3	16.2	14.1	12.3
600	80.4	44.5	32.8	24.7	20.4	16.9	14.9	13.0
700	85.0	46.0	34.6	25.7	21.3	17.6	16.1	13.8
800	89.5	47.2	35.7	26.3	22.0	18.1	16.2	13.9
900	93.0	48.2	37.1	27.0	22.6	18.5	16.6	14.4
1000	96.1	49.0	37.9	27.5	23.2	18.8	16.9	14.5
1200	102.8	50.7	40.3	28.8	24.8	19.9	17.7	15.0
1400	108.6	52.0	41.5	29.3	25.9	20.5	18.6	15.7
1600	113.4	53.1	43.5	30.3	26.5	20.9	19.2	16.1
1800	117.5	54.0	44.8	31.0	27.3	21.2	19.6	16.4
2000	122.0	55.0	45.8	31.4	27.5	21.5	19.9	16.5

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Saving Due to Multi-Stage Compression.—The table on page 123 (According to Union) gives the percentage of work lost in the heat of compression in one, two or three stages at various pressures. In these figures no account is taken of jacket cooling, nor is any allowance made for certain inevitable mechanical losses.

Example.—Assume a volume of compresssed air equivalent to 100 effective horse power is to be delivered at a pressure of 100 lbs.

Referring to the table in column 2, the theoretical percentage of lost work in one stage compression is given at 36.7; but because there is bound to be some radiation of heat, the value of 36.7% will not be found in practice, and 30% may be assumed as a practical figure under average conditions. On this basis, it is found that to deliver 100 horse power in compressed air at 100 lbs. pressure, by one stage compression, there will be required 130 indicated horse power.

Looking now at column 4 of the table, the percentage of loss in two-stage compression at this pressure is found to be 16.9%, which is very close to the figure found in practice. Applying this value, it is seen that to deliver the equivalent of 100 effective horse power in air at 100 lbs. pressure, by two-stage compression, about 117 indicated horse power will be required. In this case as between single and two-stage compression, there is a direct saving of 13 indicated horse power or 10%.

Temperature and Lubrication.—If air be compressed in a single cylinder, from atmosphere, and a temperature of 60° Fahr. to a final pressure of 100 lbs. the maximum temperature will be 484° Fahr. This temperature is manifestly destructive to common lubricants, and ordinary oils are burned into a solid gritty coke-like substance, which gives the very reverse to proper lubrication, unless proper cooling devices are employed to keep the parts cold. This carbon deposit collecting in ports and valves may so obstruct and clog them as to cause leakage and throw an additional load on the compressor.

If, however, the same volume of air be compressed in the low pressure cylinder to a pressure of 25 lbs. the highest temperature which can be reached is only 233°, a heat which will not leave a deposit or destroy the lubricating qualities of good oils such as should be used in air compressor work.

This air passing through the inter-cooler will be brought back to approximately the original temperature of 60°, and compressed in the second stage, or high pressure cylinder, from 25 lbs. to 100 lbs. Here the maximum temperature will be little, if any, in excess of that in the first stage cylinder, since the heat of compression is a function of the number of compressions and is almost wholly independent of initial pressure.

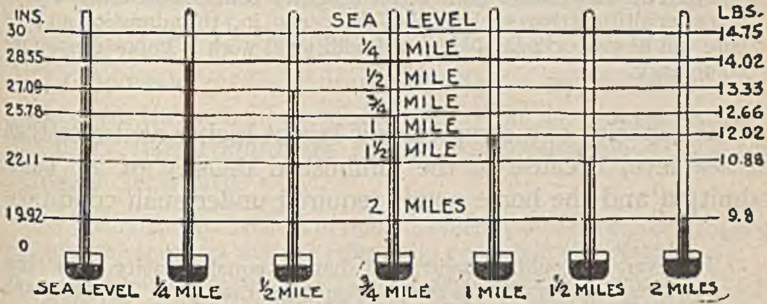


Fig. 49.—Readings of the barometer in ins. of mercury with equivalents in lbs. per sq. inch for different elevations. It should be noted that although ordinarily the atmospheric pressure at sea level is taken at 30 in. corresponding to $30 \times .49116 = 14.74$ lbs. per sq. in. or roughly 14.7 lbs. there is a standard atmosphere which as taken by Marks and Davis in their steam tables is defined as 29.921 inches of mercury or 14.696 lbs. per sq. in.

In multi-stage compressors, therefore, the conditions of temperatures are seen to be most conducive to thorough lubrication of the pistons and valves tending toward durability and tightness of the working parts with sustained efficiency of the machine.

Pressures for Stage Compression.—According to one authority single stage compression is used for pressures up to 100 lbs. Two stage compression for pressures from 80 lbs. to 500 lbs. three stage compression for pressures 500 lbs. to 1500 lbs. and four stage compression for pressures above 1500 lbs.

Effect of Altitude on Compression.—The pressure of the atmosphere is taken ordinarily at 14.7 lbs. per sq. in. at sea level. For a rough appreciation it may be assumed that the pressure decreases ½ lb. per sq. in. for every 1000 feet of ascent.

Barometer readings and corresponding pressures (lbs. per sq. in.) for various elevations are shown in fig. 49, repeated here for convenient reference.

From this illustration it will be seen that if a compressor be operated at a greater altitude than sea level (14.7 lbs. per sq. in.) the admission air pressure will be proportionately less and additional work is imposed upon the compressor.

The capacity of a compressor is less at higher altitudes than at sea level because of the diminished density of air being admitted and the horse power required under such conditions less.

However, it should be understood that for equal capacity more horse ower is required for altitude compression than at sea level. The little table which follows shows effect of altitude upon capacity and power.

For instance, referring to the table for say, 4000 ft. elevation, the decrease in power is given at 6.9% while the loss of capacity is 13%. Thus for equal performance more power is required for elevation compression than at sea level.

Effect of Altitude upon Capacity and Power

Altitude, in Feet	Loss of Capacity, Per Cent	Decrease in Power Required, Per Cent
0	0	0.0
1,000	3	1.8
2,000	7	3.5
4,000	13	6.9
6,000	19	10.1
8,000	24	13.1
10,000	30	16.1

CHAPTER 46

Classification of Compressors

Compressed air is an important factor in practically every phase of the art of manufacture. That is practically every industry uses compressed air. In its application it is second only to electricity.

The rapid development of compressed air appliances has brought about economical results that are reflected in every field of industry.

As the economical application of compressed air is wholly dependent upon its economical production, it is apparent that the modern air compressor must embody every refinement in design and construction.

Evidently to meet the multiplicity of applications in the use of compressed air, many types of compressors have been developed to adapt them to these varied conditions.

A classification of compressors to be comprehensive should be made from numerous points of view. Accordingly air compressors may be classed:

1. With respect to the frequency of the cycle, as

   a. Single acting

   b. Double acting

2. With respect to the nature of the cycle, as

   a. Single stage

   b. Double stage

3. With respect to the moving parts, as

   a. Reciprocating
   b. Centrifugal
   c. Rotary

4. With respect to duty, as

   a. Low pressure; blowers (centrifugal displacement)
   b. Medium pressure (single stage)
   c. High pressure (multi-stage)

5. With respect to the drive

   a. Direct connected
   b. So called "power" driven

6. With respect to the prime mover, as

   a. Steam
   b. Gas engine
   c. Diesel
   d. Electric

7. With respect to the placement of the air cylinder or cylinders, as

   a. Vertical
   b. Horizontal (straight line)
   c. Radial or angle

8. With respect to the number of power cylinders, as

   a. Single cylinder
   b. Multi-cylinder

9. With respect to the degree of expansion of the steam on steam driven compressors, as

  a. Simple
  b. Compound

10. With respect to the method of cooling, as

  a. Water
  b. Air

11. With respect to power transmission, as

  a. Direct drive
  b. Belt drive (Flat V)
  c. Chain drive

12. With respect to number of air cylinders

  a. Simplex
  b. Duplex
  c. Triplex

13. With respect to nature of installation

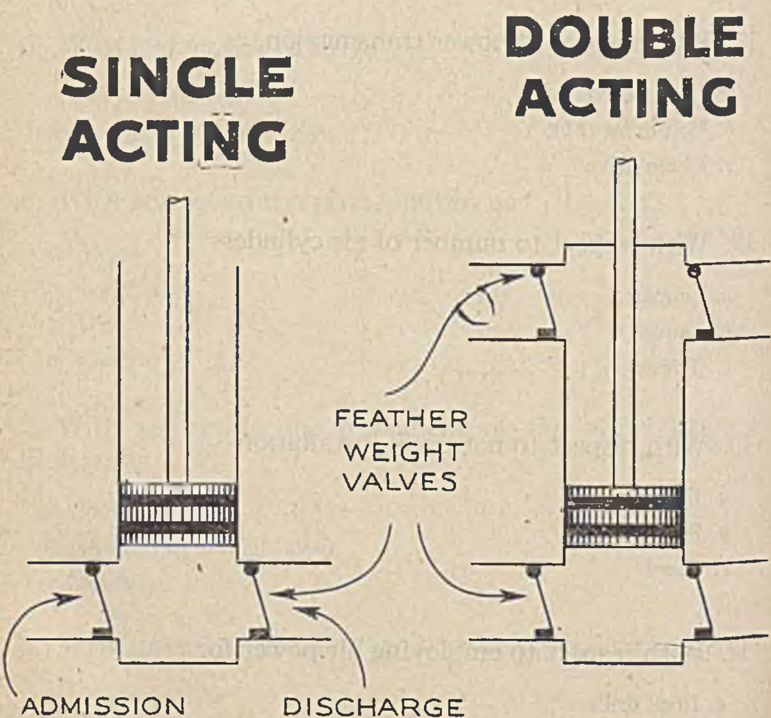
  a. Portable
  b. Semi-fixed
  c. Fixed

14. With respect to employing air power for

  a. Rock drills
  b. Pneumatic hand tools

15. With respect to applications

 a. Rock drills
 b. Pneumatic hand tools
 c. Road building
 d. Trench digging
 e. Quarrying
 f. Mine Prospecting
 g. Tearing up paving
 h. Sand blasting
 i. Spray painting, etc.



Figs. 1 and 2.—Compressors 1. Fig. 1, single acting; fig. 2, double acting.

Ques. What is a single acting compressor?

Ans. One in which compression takes place every other stroke.

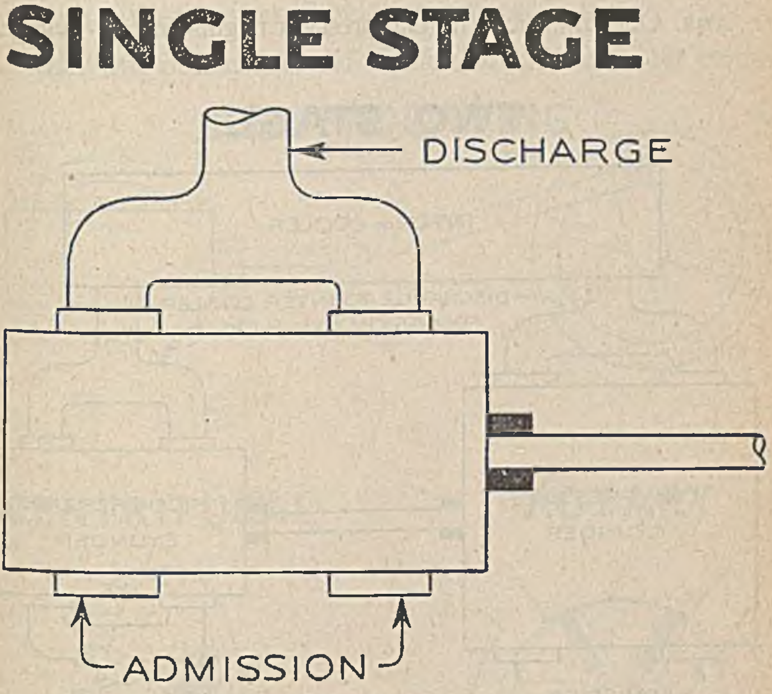


Fig. 3.—Compressors 2. Fig. 3, single stage.

Ques. What is a double acting compressor?

Ans. One in which compression occurs every stroke.

Ques. What is a single stage compressor?

Ans. One in which the compression cycle takes place in a single cylinder.

Ques. What is a two stage compressor?

Ans. One in which the compression begins in one cylinder

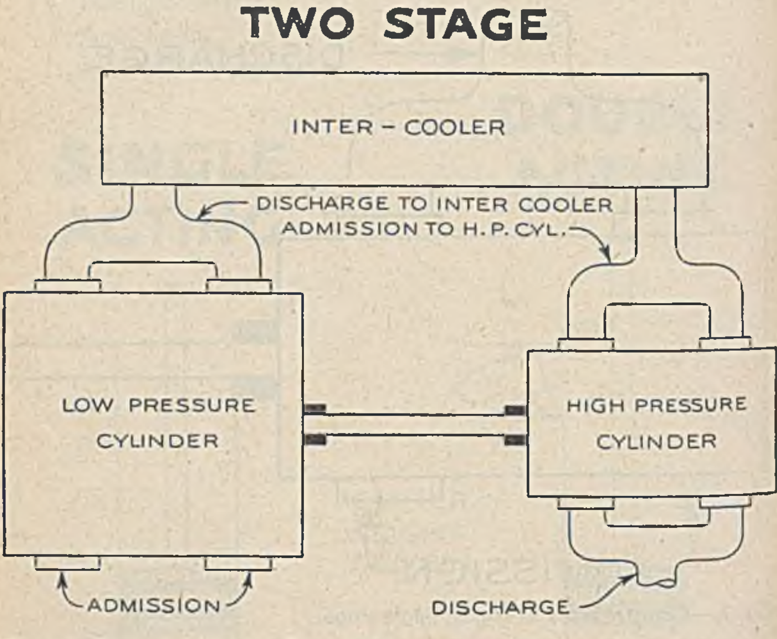


Fig. 4.—Compressors 3.—Two-stage.

and is completed in a second cylinder, thus dividing the temperature range between the two cylinders and permitting cooling between the cylinders.

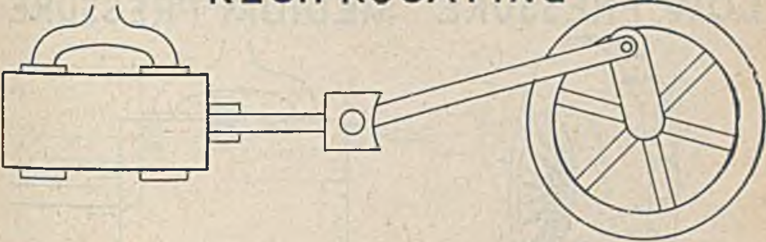
Ques. What is a reciprocating compressor?

Ans. One having a piston arranged to move to and fro—forward and backward.

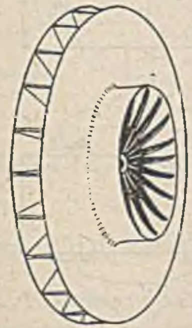
Ques. What is a centrifugal compressor?

Ans. One designed to deliver large quantities of air or gas

RECIPROCATING



CENTRIFUGAL



ROTARY

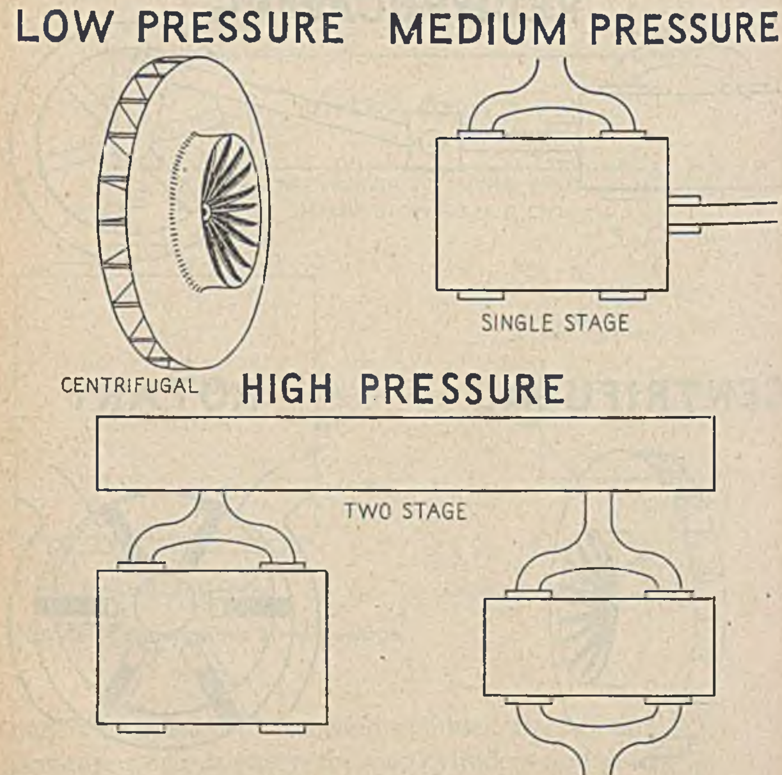
Figs. 5 to 7.—Compressors 4. Fig. 5, reciprocating; fig. 6, centrifugal; fig. 7, rotary.

at low pressure moved by centrifugal force generated by a fast revolving rotor.

The proper name for centrifugal compressor is *blower*.

Ques. What is a rotary compressor?

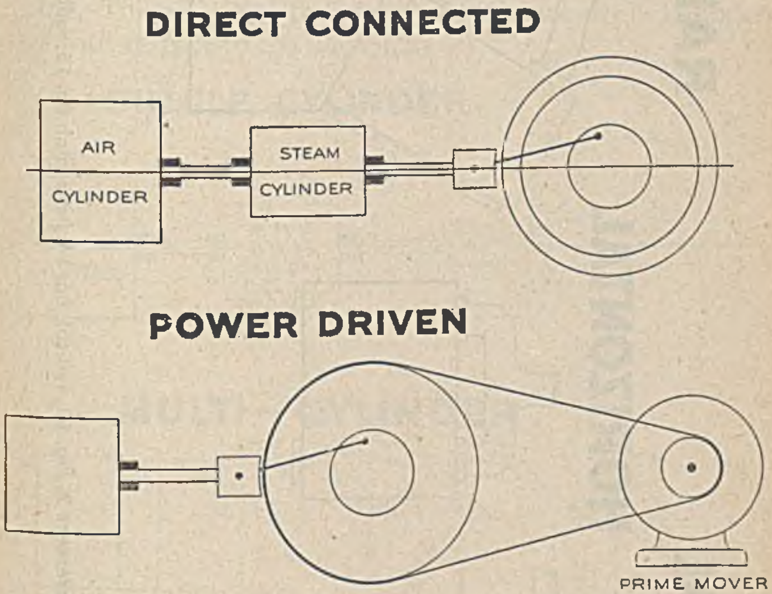
Ans. One having a vane rotor or its equivalent mounted eccentrically in a stationary casing.



Figs. 8 to 10.—Compressors 5. Fig. 8, low pressure; fig. 9, medium pressure; fig. 10, high pressure.

Ques. What is the distinction between low, medium and high pressures?

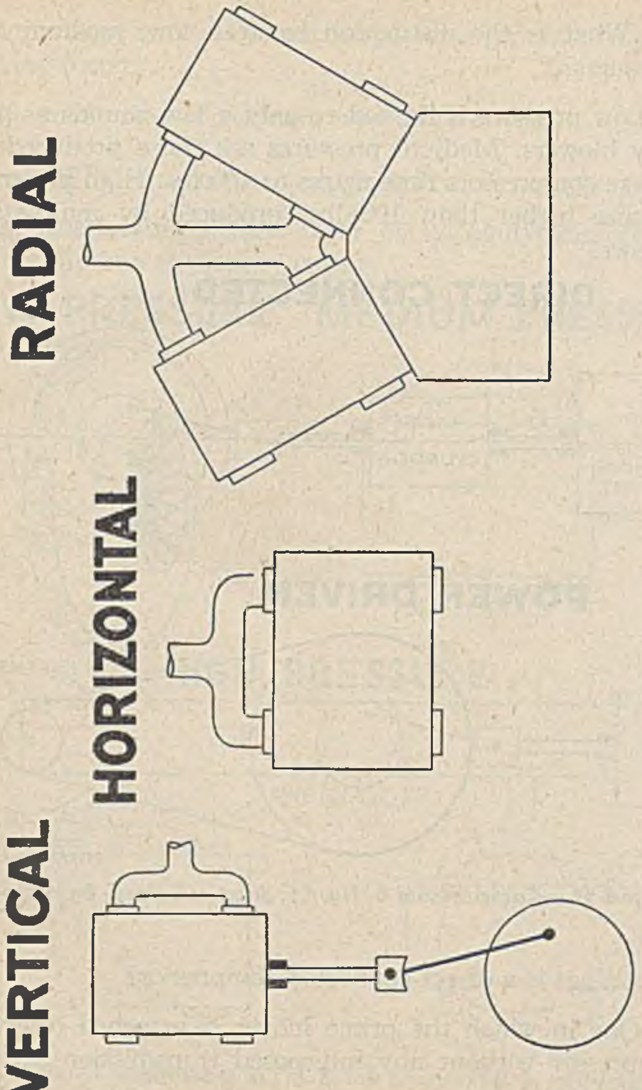
Ans. Low pressure is limited to only a few pounds as produced by blowers. Medium pressures are those produced by single stage compressors ranging up to 100 lbs. High pressures are pressure higher than 100 lbs. produced by multi-stage compressors.



Figs. 11 and 12.—Compressors 6. Fig. 11, direct connected; fig. 12, power driven.

Ques. What is a direct connected compressor?

Ans. One in which the prime mover is attached direct to the compressor without any interposed transmission such as belt, chain, etc.



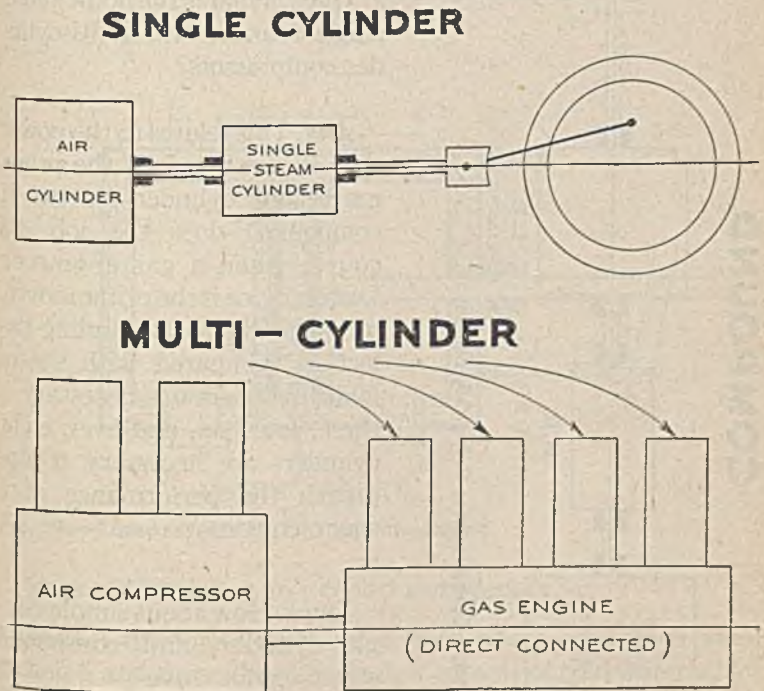
Figs. 13 to 15.—Compressors 7. Fig. 13, vertical; fig. 14, horizontal; fig. 15, radial.

Ques. What is a power driven compressor?

Ans. The term power driven compressor is a very objectionable term for a compressor having a separate prime mover and connected by suitable transmission such as a belt.

Ques. Distinguish between vertical, horizontal and V cylinder placements.

Ans. This relates to the position of the cylinder or cylinders axes with respect to the horizontal.



Figs. 16 and 17.—Compressors 8. Fig. 16, single cylinder; fig. 17, multi-cylinder.

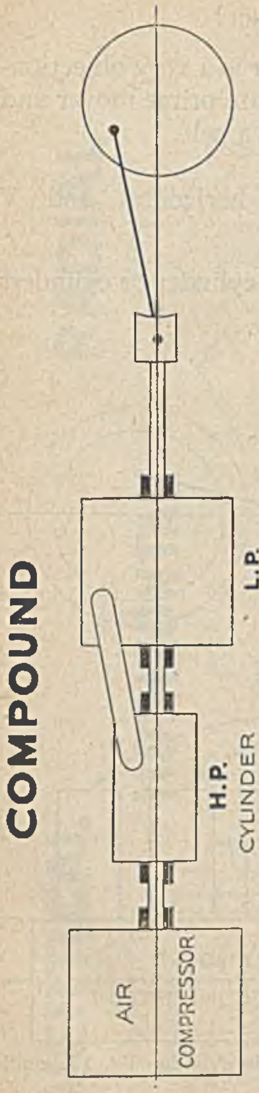


Fig. 18.—Compressors 9. Fig. 18, compound.

A vertical compressor is one whose piston reciprocates in a vertical axis. A horizontal compressor is one whose piston reciprocates in a horizontal axis a V or radial compressor is of the multi-cylinder type whose pistons reciprocate at angles to each other, usually 90°.

See figs. 13 to 15.

Ques. What is the point about single cylinder and multi-cylinder compressors?

Ans. This relates to the power end. Where steam is the prime mover one cylinder (or two if compound) does the job. Of course when a gas engine or Diesel engine is the prime mover, owing to their outstanding defect as compared with steam multi-cylinders are necessary—three, four, six, and even eight cylinders are necessary to approach the performance of a steam engine.

Ques. How about simple (single cylinder) and compound steam engine drive?

Ans. Its not a case of getting a smoother turning effect, but to get better economy that compound or two stage expansion is sometimes used.

The steam engine never has to compete with an internal combustion engine with respect to smooth turning operation—the internal combustion engine with its outstanding inherent defect can never equal a steam engine with respect to the approach to uniform rotary motion, no matter how many cylinders or how heavy a fly wheel.

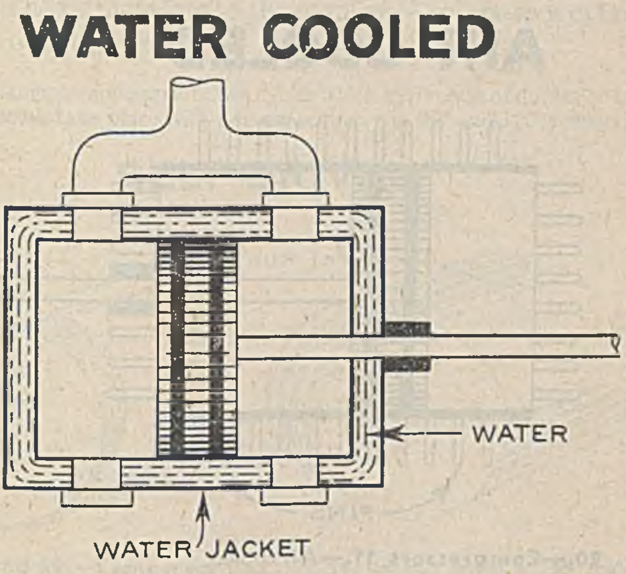


Fig. 19.—Compressors 10. Fig. 19, water cooled.

Ques. What is a water cooled compresssor?

Ans. A misnomer for a compressor whose cylinder is water jacketed and through which flows a current of cold water which functions as a *transmission medium* to carry off *some* of the heat of compression.

Ques. What is an air cooled compressor?

Ans. One whose cylinder has cast integral numerous thin fins to form excess cooling surface exposed to a draught of cool air which forms the medium to carry off some of the heat of compression.

Ques. What is a direct connected compressor?

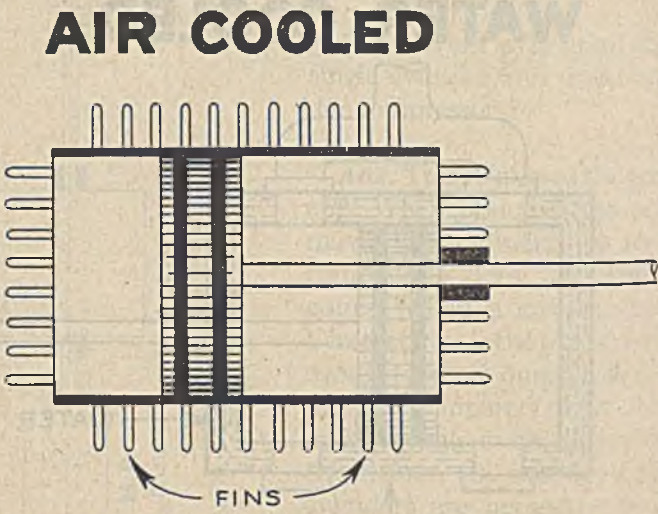


Fig. 20.—Compressors 11.—Air cooled.

Ans. One in which the prime mover is attached rigidly to the compressor moving element and moves in unison with it without the interposition of any power transmitting gear such as belt, etc.

Ques. What is a power driven compressor?

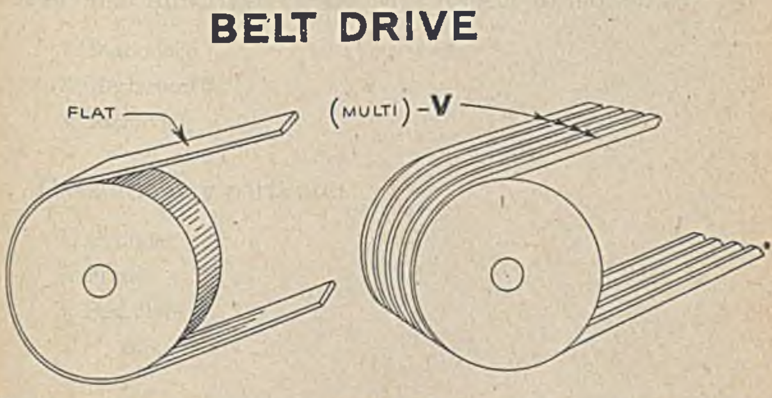
Ans. A ridiculous term (they are all power driven) for a compressor whose prime mover is a separate unit and connected by a *transmission* such as belt, chain, etc.

It's known as such and nothing can be done about it.

Ques. Distinguish between simplex, duplex and triplex compressors.

Ans. The terms relate to the number of compressor cylinders whether one, two or three.

Not stage compression, but to cycles which in the case of duplex or triplex compression take place with a phase sequence of 90° and 120° respectively.



Figs. 21 and 22.—Compressors 12. Fig. 21, flat belt drive; fig. 22, multi V. belt drive.

Ques. What is a portable compressor?

Ans. A small unit such as for contractors which is easily moved from place to place.

Ques. What is a semi-fixed compressor?

Ans. A unit larger than the portable type where skids are used in place of being mounted on a truck, the adaptation being for service where frequent moving is not necessary.

Ques. What is a fixed compressor?

Ans. A compressor mounted upon a permanent base as concrete for service not requiring removal from place to place.

CHAPTER 47

Compressor Parts

The numerous parts of which a compressor is composed may be divided into three classes with respect to motion as:

1. Stationary
2. Reciprocating
3. Revolving

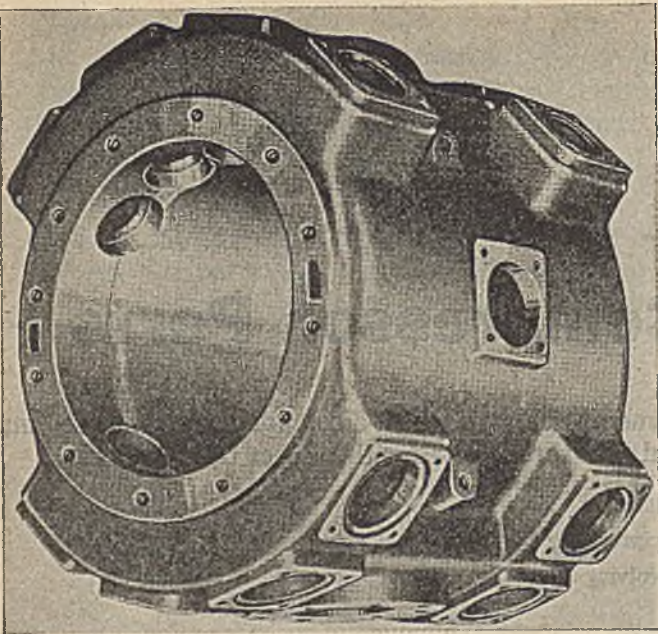
The stationary parts are:

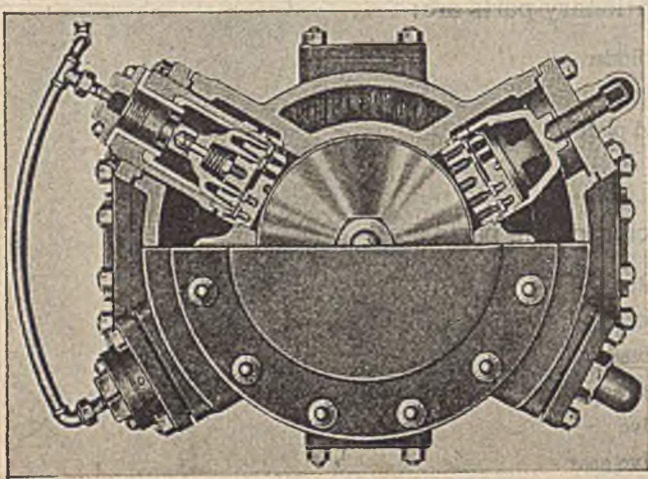
1. Cylinder
2. Frame
3. Bed plate

    etc.

The reciprocating parts are:

1. Piston
2. Piston rod
3. Cross-head
4. Connecting rod (wrist pin end)
5. Valve
6. Valve gear





Figs. 1 and 2.—Views of compressor cylinder showing typical construction.

The rotating parts are:

1. Shaft
2. Connecting rod (crank pin end)
3. Crank pin

The Compressor Cylinder.—To meet the varied conditions under which compressors work in the numerous applications of compressed air, several types of cylinder have been developed. They may be classed:

1. With respect to the method of casting as:

   a. Separately cast

   b. En bloc

2. With respect to the method of cooling as:

   a. Water { running / hopper }

   b. Air { natural / induced }

3. With respect to the cycle of operation, as:

   a. Single stage

   b. Two stage

A typical, separately cast water cooled cylinder is shown in figs. 1 and 2.

In practice and construction the cylinder has walls sufficiently thick for two reborings, and is counter bored at the ends to prevent the rings wearing a shoulder.

In the design shown the valves are located radially in the cylinder and the passages are short and direct to reduce frictional resistance and clearance to a minimum.

Ques. Describe the provision for water cooling.

Ans. Both cylinder walls and heads are water jacketed and as can be seen in the illustrations, bolted hand plates are provided on both sides of the jacket for cleaning.

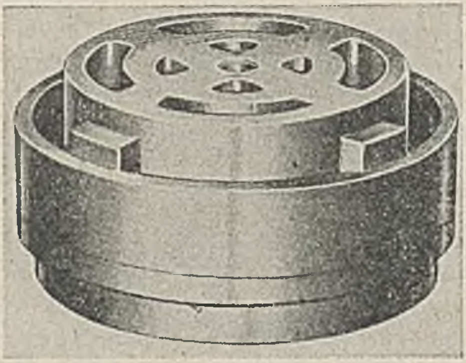
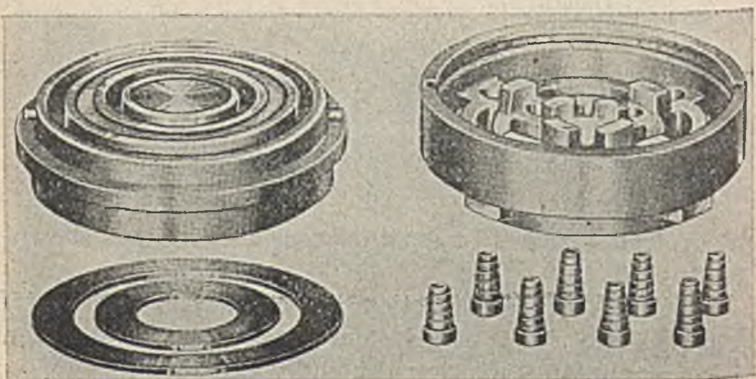


Fig. 3.—Admission valve assembly.



Figs. 4 to 7.—Discharge valve parts. **In this construction** both admission and discharge valves as well as the seats and cages are inter-changeable and are held in place in the cylinder with a yoke which bears on the outer section of the valve assembly with a holding down center bolt through the cover.

Ques. What are "en bloc" or uni-bloc cylinders?

Ans. Two or more cylinders cast integral, that is, all in one casting.

An example of this practice is shown in fig. 8. Here as will be noticed the compressor design follows approved automotive practice.

Ques. Describe the hopper cooled system.

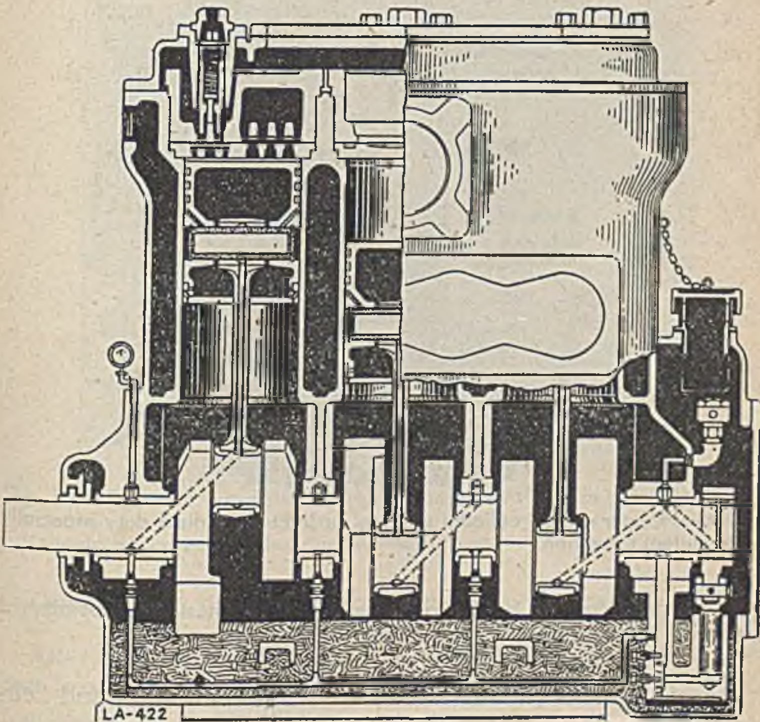


Fig. 8.—Sectional view of a triplex vertical compressor illustrating en block construction.

Ans. This is a non (external) circulating system. The cylinder has an open water jacket of considerable volume as shown in fig. 9.

As stated by one manufacturer the hopper cooled system is suitable for constant operation up to 100 lbs. and intermittent operation up to 150 lbs. The closed cooling system (that is, the "running" or "circulating" water system) must be used for constant service of from 100 lbs. to 200 lbs. and

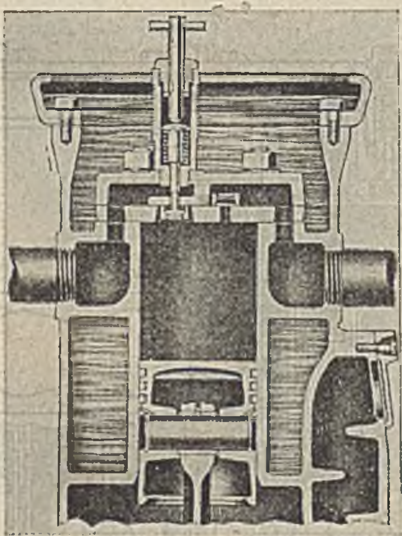


Fig. 9.—Hopper cooled cylinder suitable for light or medium duty especially intermittent operation.

intermittent service of 150 lbs. to 250 lbs. There are some designs not adapted for such severe duty.

Ques. What is the distinction between "running" and "circulating" water?

Ans. Running water in jackets is water not used over again;

circulating water is water re-circulated over and over again as on an automobile system with radiator.

Ques. What is intermittent service at 250 lbs.?

Ans. Service such as pumping up tanks to 250 lbs. and then closing down so that the plant can cool off.

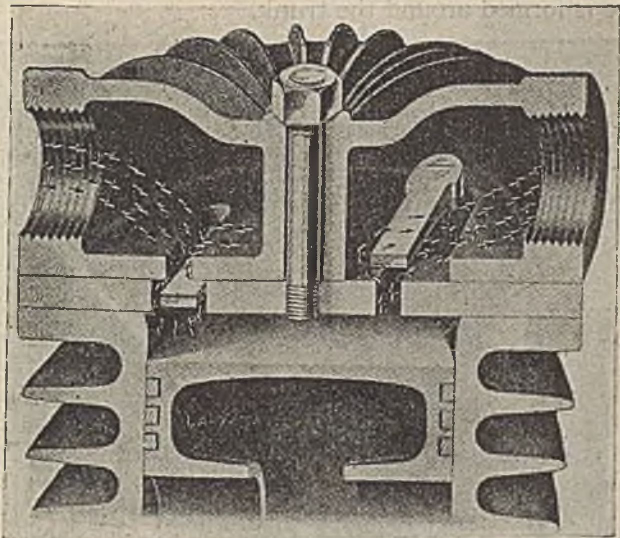


Fig. 10.—Detail of air cooled cylinder head and valve plate. To illustrate the passage of air through the valves both are here shown open, but in operation it should be understood that one is always closed.

Ques. What is the construction of an air-cooled cylinder?

Ans. Numerous fins are cast integral with the cylinder walls and heads to furnish additional radiating surface to carry off the heat.

Fig. 10 shows detail of an air-cooled cylinder. The radiating fins on both cylinder and head are here plainly shown. This particular cylinder is of the single acting type with "natural" air cooling. That is to say, no forced draught is provided as with the induced air cooling system. This latter system is shown in fig. 11.

Ques. What is the essential features of a two stage cylinder?

Ans. This is a cylinder of special construction equipped with a step piston, the low pressure being at the top while the high pressure is formed around the trunk.

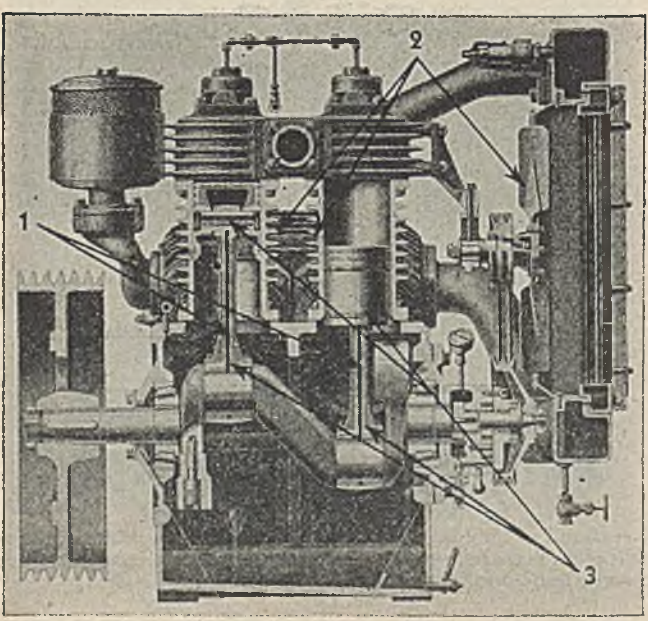
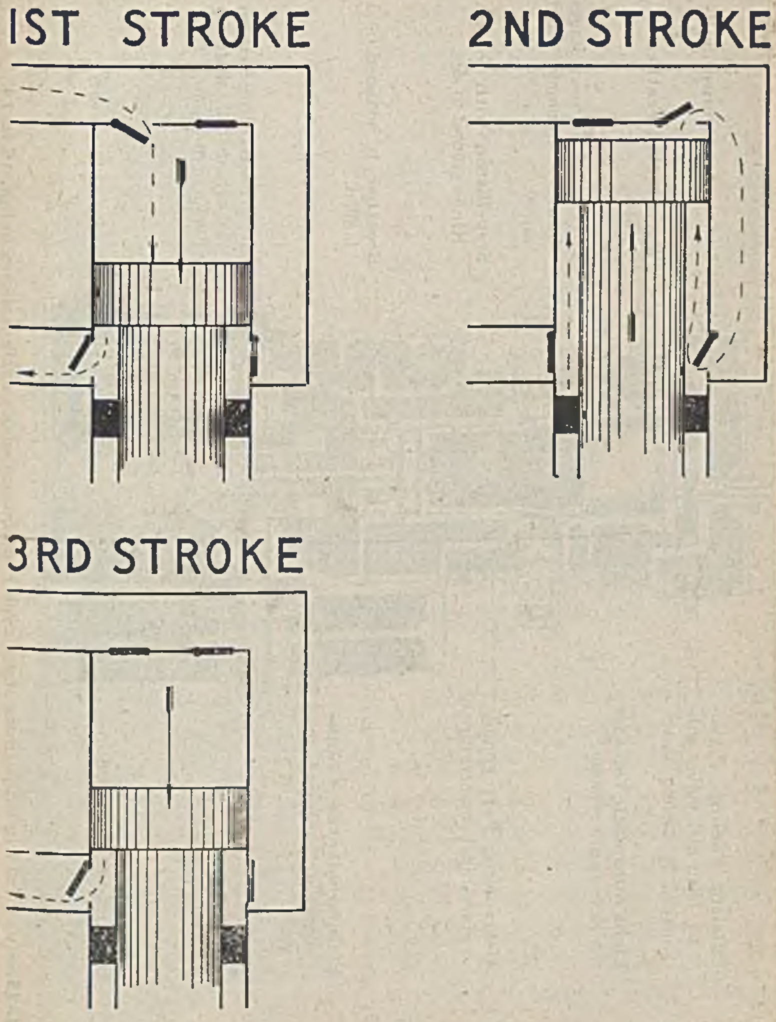


Fig. 11.—Multi cylinder compressor illustrating induced (forced draught) air cooling. **The parts are:** 1, connecting rods; 2, fan; 3, force feed lubrication.

Ques. How does it work?

Ans. The cycle requires three strokes and is shown progressively in figs. 12 to 14. Stroke 1. Admission to first stage; stroke 2, first stage compression and transfer to high pressure



Figs. 12 to 14.—The two stage three stroke cycle, showing events of each stroke.

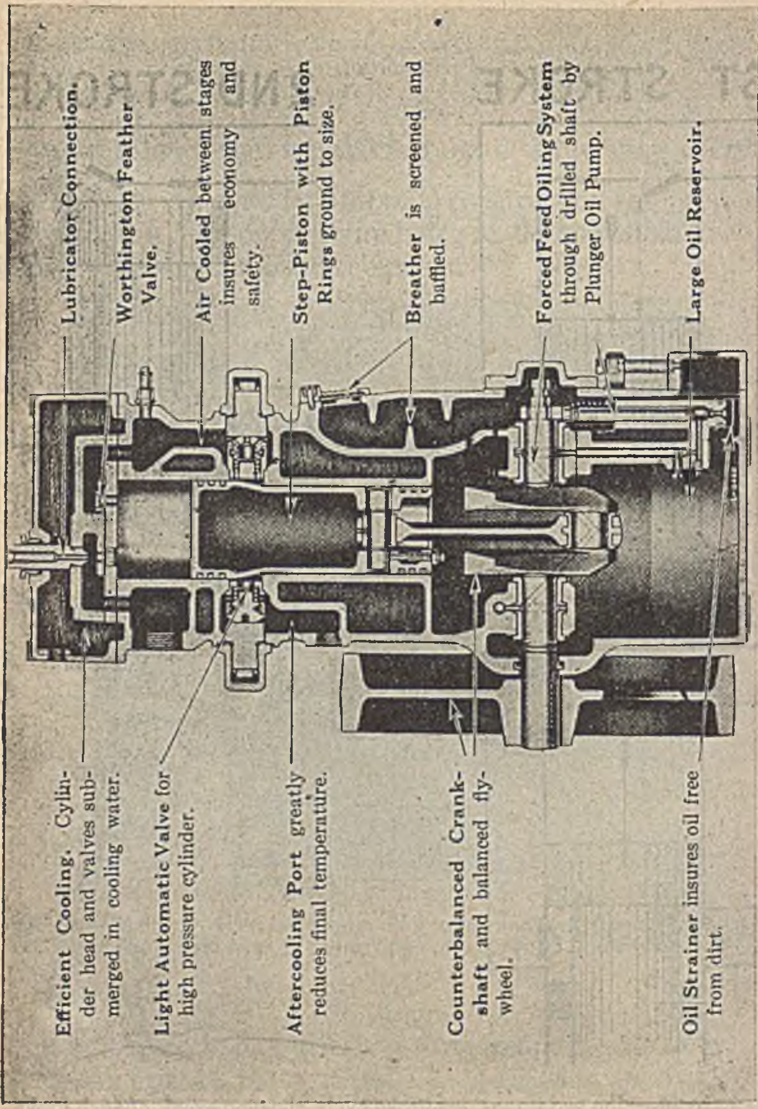


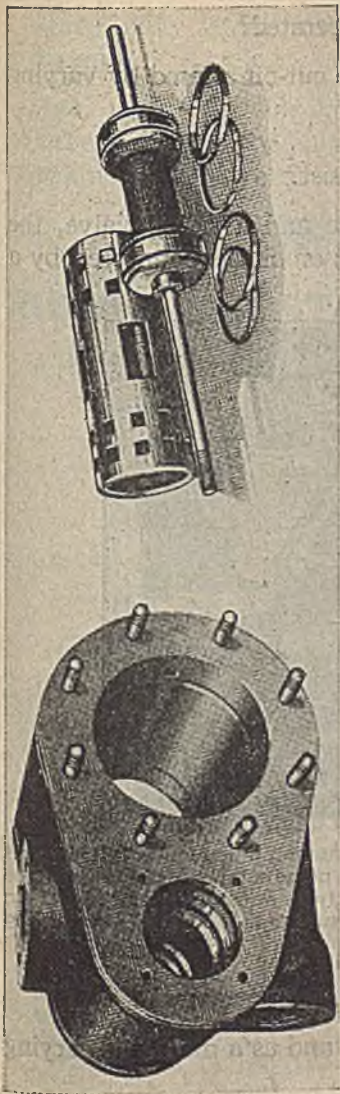
Fig. 15.—Vertical two stage (three stage cycle) compressor whose cycle is shown in figs. 12 to 14.

step piston; stroke 3, second stage compression and discharge at high pressure.

The Steam Cylinder.—The part follows steam engine design, the valve gear provided will depend upon the economy desired. With respect to the feature the valve may be either a single valve usually of the piston type or one with a riding cut-off which permits a degree of expansion beyond the range of the simple valve gear.

Fig. 16 shows a typical steam cylinder designed for piston valve and fig. 17 the valve and its seat which in this case is a bushing. The assembly is shown in fig. 18.

As stated where economical working is required a riding cut-off should be used. There are several riding cut-off gears working on different principles, but the one employed on compressors is *the riding cut-off with* **variable** *lap*. This method of riding cut-off is known as the Meyer gear and is also largely used on marine engines.



Figs. 16 and 17.—Steam cylinder with piston valve and valve details. The valve is of the inside admission type.

Ques. How is the riding valve operated?

Ans. By a fixed eccentric and the cut-off altered by varying the lap of the riding valve.

Ques. Of what does the gear consist?

Ans. It consists of a main valve and a riding valve, the latter being divided into two plates or blocks connected by a right and left handed screw.

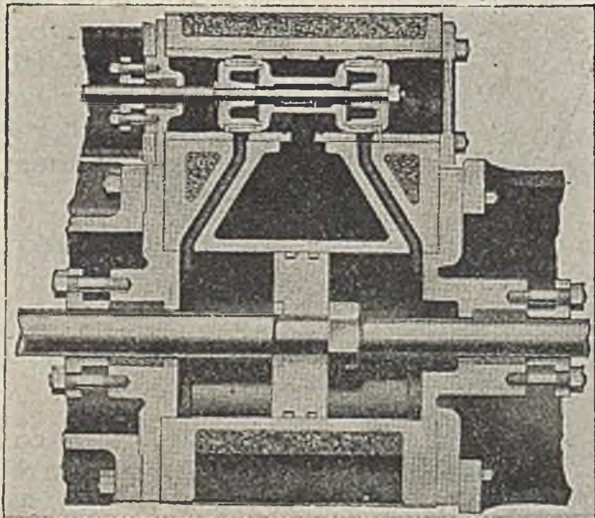


Fig. 18.—Sectional view of steam cylinder showing piston valve. Piston valves are balanced and permit the use of high pressure super-heated steam. The valve here shown is of the inside admission type. Although harder to set than an outside admission valve, it has the advantage that the valve rod stuffing box is subjected only to low exhaust pressure.

Ques. What duties are performed by the screw?

Ans. It serves as a valve spindle and as a means of varying the lap of the riding valve.

Ques. Why is the riding valve necessary for satisfactory economy?

Ans. The simple slide valve cannot be satisfactorily designed to cut off less than one-half and this is not nearly enough for the economical use of steam.

MEYER
RIDING ADJUSTABLE CUT OFF VALVE
(BY THE METHOD OF VARIABLE LAP)

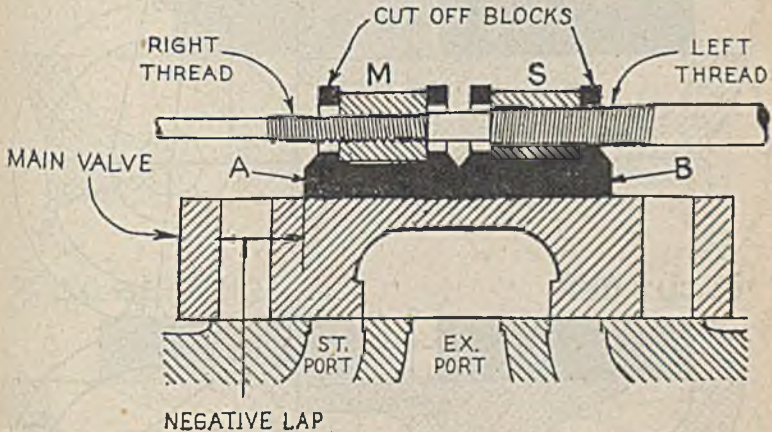
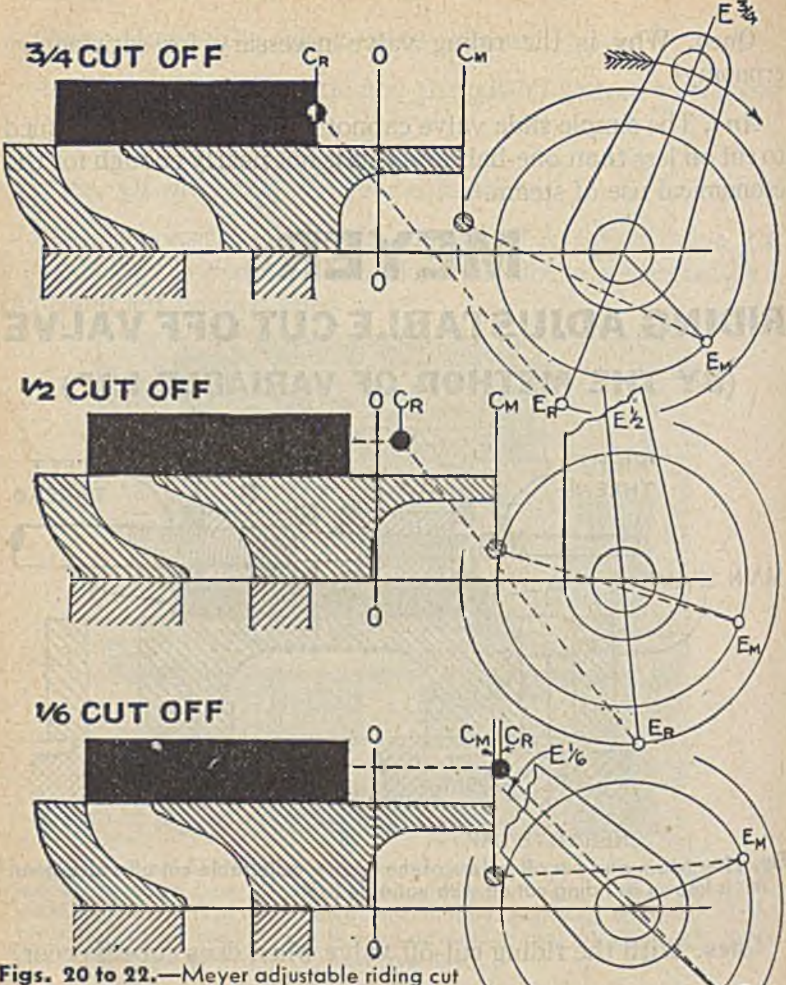


Fig. 19.—Main and cut off valves of the Meyer adjustable cut off valve gear. This is known as riding cut off with variable lap.

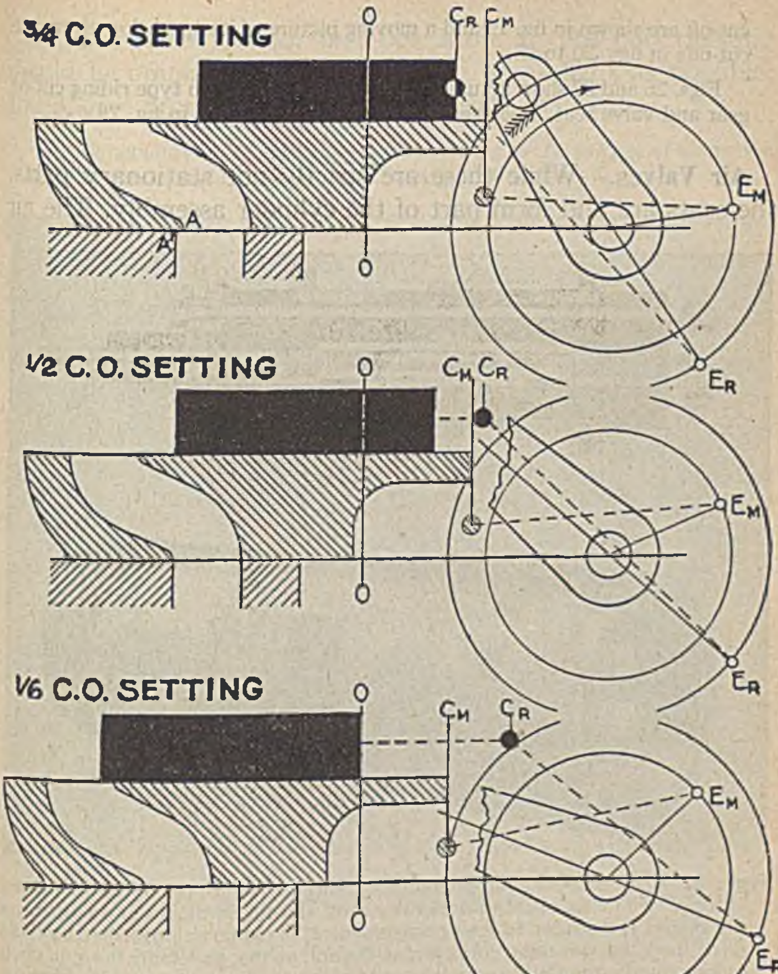
Ques. With the riding cut-off valve when does cut-off occur?

Ans. *When the riding valve is at a distance from the center of the main valve equal to its lap.*

That is, when the steam edge of the riding valve is "line and line" with the steam edge of the bridge of the main valve. The essentials of the Meyer



Figs. 20 to 22.—Meyer adjustable riding cut off gear showing positions of valves eccentrics and cranks for ¾, ½ and ⅙ **cut off** setting. By noting the positions of the eccentrics it is evident that as the cut off is **shortened,** it becomes **sharper.** Figs. 20 and 21 show valves moving in same direction and fig. 22 valves moving in opposite directions.

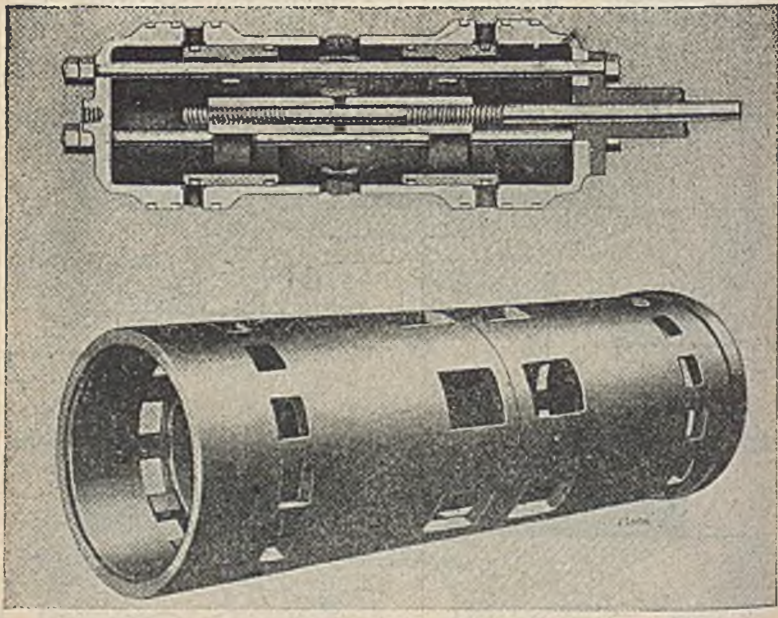


Figs. 23 to 25.—Meyer adjustable siding cut off gear showing positions of valves eccentrics and crank for ***"mid admission"*** corresponding to ¾, ½ and ⅙ cut off setting. It will be noted that for the ⅙ cut off the port opening is reduced.

cut-off are shown in fig. 19 and a moving picture of its operation at various cut-offs in figs. 20 to 25.

Figs. 26 and 27 show actual construction of the piston type riding cut-off gear and valve seat. The assembly in cylinder is shown in fig. 28.

Air Valves.—While these are strictly not stationary parts, the seats are and form part of the cylinder assembly. The air



Figs. 26 and 27.—Meyer adjustable riding cut off piston type valve and "blocks" with view of cylindrical valve seat. The cut off blocks or riding valves are R and I, threaded to their common valve, the latter being rotated by chain drive from the governor. Thus under control of the governor, the cut off is automatically adjusted to vary the speed of the compressor to meet the changes in load demand.

valves are perhaps the most vital part of a compressor with respect to efficient working and in their design there are several conditions to be met for best operation. 1. The port opening should be ample. 2. The valve should be as light as possible, and 3, give proper port opening with very little lift.

Designers have given much thought and time to the subject of air valves and the results obtained have been worthy of the effort.

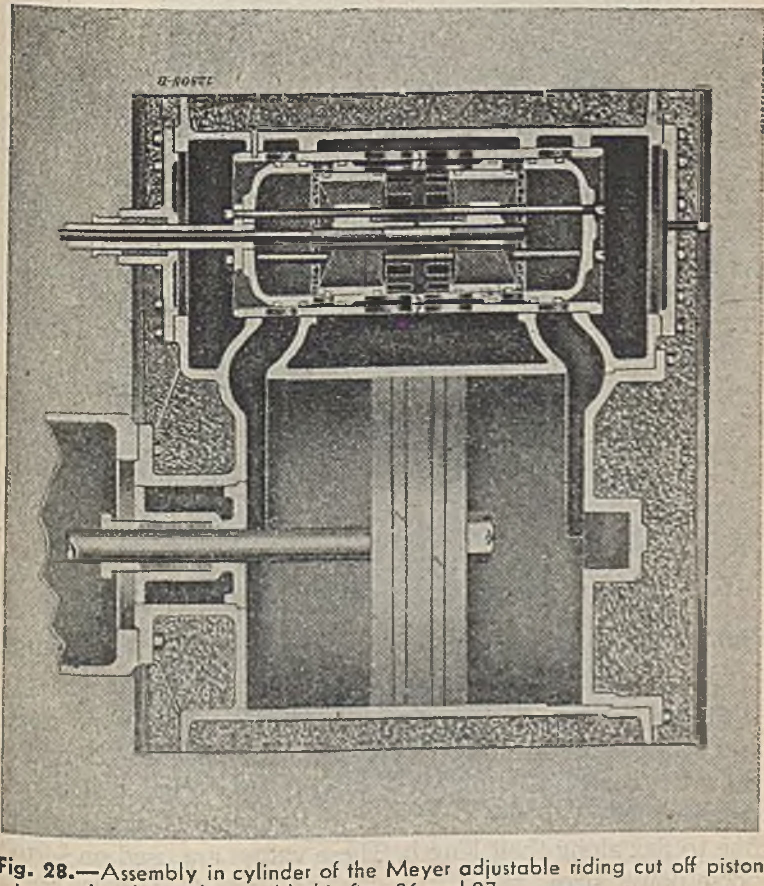
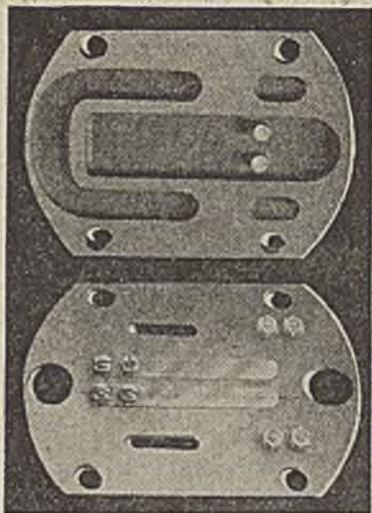


Fig. 28.—Assembly in cylinder of the Meyer adjustable riding cut off piston type valve shown disassembled in figs. 26 and 27.

The various types of air valve in general use may be classed as:

1. Finger
2. Feather
3. Plate
4. Ring { plain / dual cushion }
5. Channel



Figs. 29 and 30—Stainless steel finger valves showing cover valves and seat.

Finger Valves.—For light service as in small units at garage and service stations finger valves are suitable. Figs. 29 and 30 show the construction of these valves. They consist of narrow strips of stainless steel, fastened to the seat at one end and free to flex along their length. These valves are used on ¾ to 5 horse power sizes.

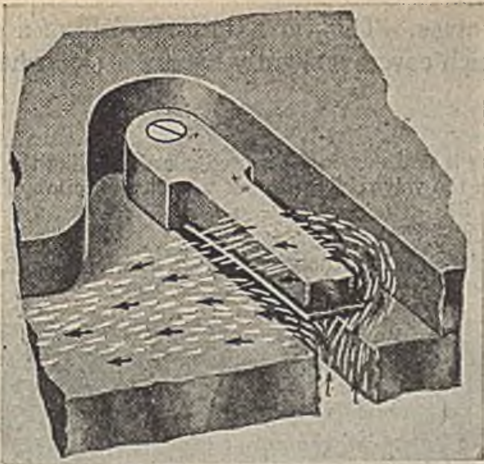


Fig. 31.—Detail of feather valve in cylinder head, showing operation. It consists of a strip of ribbon steel which covers a slightly narrower slot when the valve is closed. In opening it flexes against a curved guard as shown.

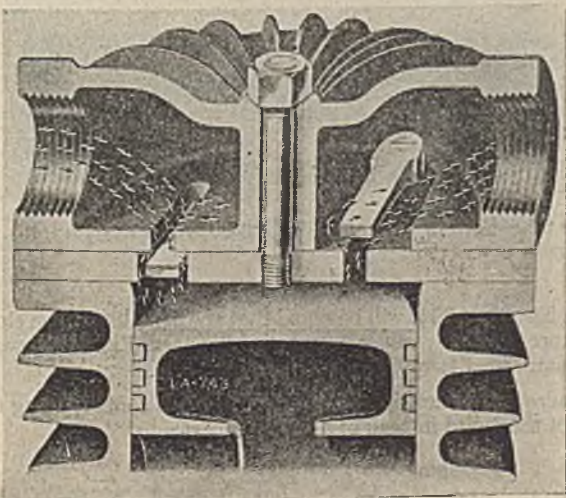
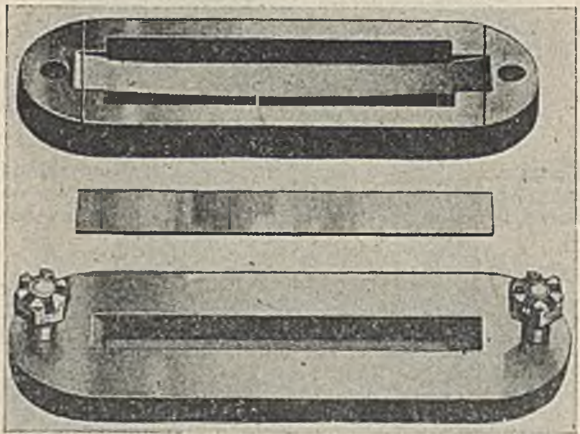


Fig. 32.—Detail of air cooled cylinder head showing feather valves, both in open position to illustrate the passage of air through them.

Feather Valves.—This type valve consists of a strip of ribbon steel which covers a slightly narrower slot when the valve is closed.

In opening it flexes against a curved guard allowing the air to pass on either side of the valve strip with but very little friction. The assembly is shown in fig. 31.

Both admission and discharge valves are shown in fig. 32.



Figs. 33 to 35.—Parts of feather valve. Fig. 33, valve in guard; fig. 34, valve; fig. 35 cover.

From figs. 33 to 35 it will be noted that the complete valve element consists of three parts: 1, valve; 2, seat, and 3, guard. Fig. 33 shows the valve in the guard, fig. 34, valve and fig. 35, seat.

In operation, when opening, the valve makes contact along the curved surface of the guard instead of a direct impact.

Plate Valves.—These valves also known as Rogler valves have a stop plate and are fully described on pages 340 to 344.

Ring Disc Valves.—The valve proper consists of annular rings stamped out of thin alloy steel as shown in fig. 37. One type of valve of this class is cushioned by special springs. Its general appearance is shown in fig. 36 and its operation in the diagrams, figs. 37 to 39.

Although the valve parts are circular the diagrams have been made in plane or developed for better illustration. The annular valve spring has a number of equally spaced flexures or waves as has also the reaction plate. The reaction plate has two waves for each wave of the valve spring as shown in fig. 37.

Here the valve is shown closed and the valve spring resting against the bottom of alternate reaction plate waves.

Unsupported length or effective lever arm of the valve spring is indicated by X.

In operation as the valve begins to open, fig. 38, because of the pressure difference on opposite sides, the valve spring begins to flatten and its reaction points on the reaction plate change from the bottom of reaction plate waves to points along the upward slopes.

On further opening of the valve as in fig. 39 the spring continues to roll up the slopes of the reaction plate waves. The unsupported length of the valve spring or its effective lever arm has now shortened from the original distance X to a distance $\frac{X}{2}$ or half as much.

The result is that the stiffness of the valve spring becomes approximately twice as great after the valve has opened part way as it would have been were the wave type reaction plate absent.

Note.—Valve wear may be said to be in proportion to the pressures under which the valve operates. It is not a question here of allowable wear; a replacement is made as soon as conditions indicate the valve is not functioning properly. The practice is sometimes followed of periodically changing all the discharge valves to prevent an excessive number of shut-downs. This is done in severe service where deposits and the service permit the proper functioning of the valves for a comparatively short period of time only.

The object of this design is to offer slight resistance to valve at beginning of its opening movement, but to provide great cushioning effect as the valve approaches the end of its lift.

Channel Valves.—This type of valve has been designed for heavy duty service. The parts are shown in figs. 40 to 43. They are fig. 40, cover; fig. 41, valve spring; fig. 42, channel valves, and fig. 44, valve seat.

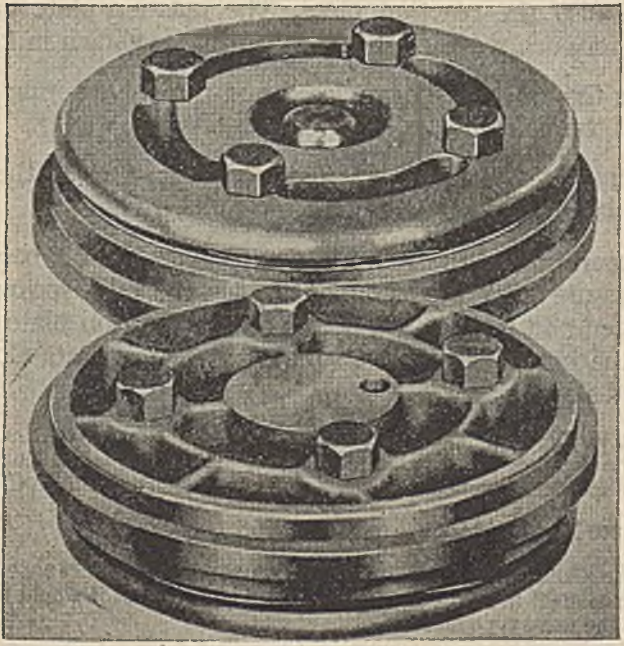
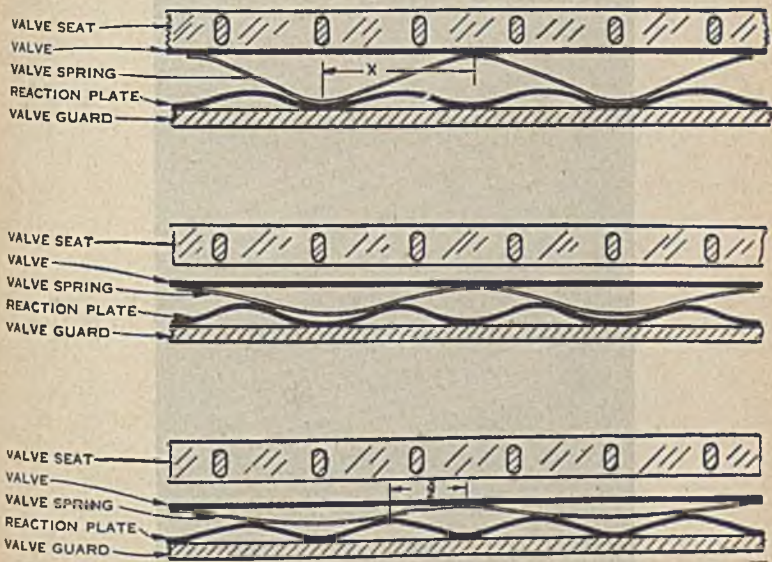


Fig. 36.—Ring disc valve with dual cushion springs.

Operation of the valves is shown progressively in figs. 44 to 46, and is explained with the illustrations. The valves and springs are made of stainless steel.

The Frame.—This part of the compressor should be of ample proportions with metal properly distributed to secure the maximum strength and rigidity. There are numerous types of frame depending upon the type of compressor. One pattern is shown in fig. 47. Evidently this is for a steam driven compressor with overhanging steam and air cylinders.

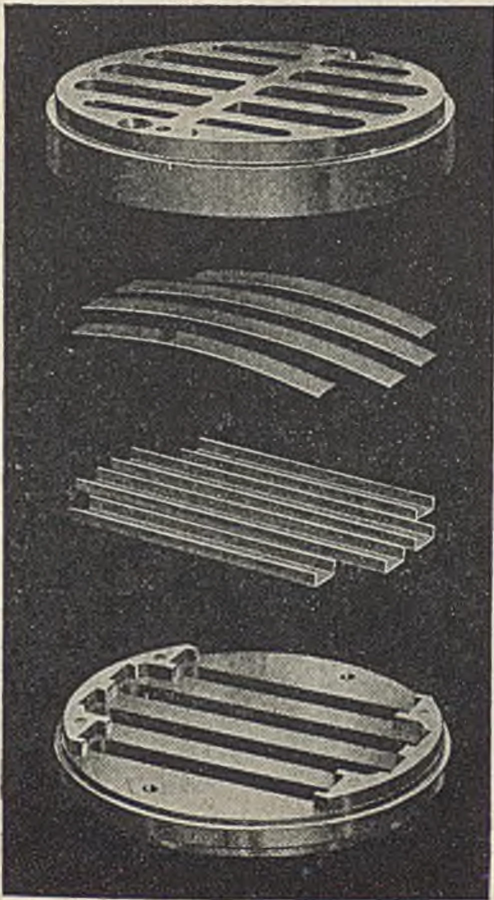


Figs. 37 to 39.—Diagrams showing operation of the ring disc valve with dual cushion springs.

It is extended to the foundation in the form of a broad liberal support reaching its entire length. The cross head guides are bored at the same setting as the fitting for the cylinder.

The frame with its covers form a closed chamber with a quantity of oil

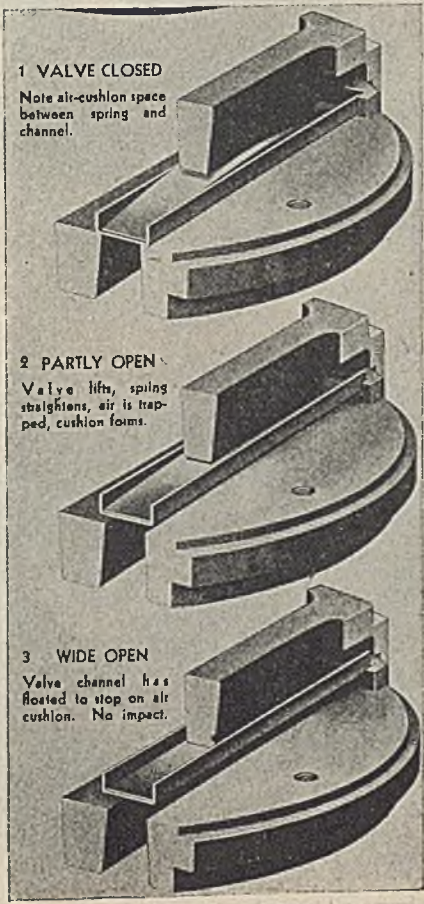
CHANNEL VALVES AND PARTS



Figs. 40 to 43.—Channel valves and parts. Fig. 40 cover; fig. 41, springs; fig. 42, channel valves; fig. 43, valve seat.

in the basin below the crank, into which the crank and connecting rod dip at each revolution.

Another frame of the oil tight type is shown in fig. 48. These frames virtually combine frame and bed plate. Fig. 49 shows a frame for a large vertical compressor with bed plate separate.



Figs. 44 to 46.—Operation of channel valves.

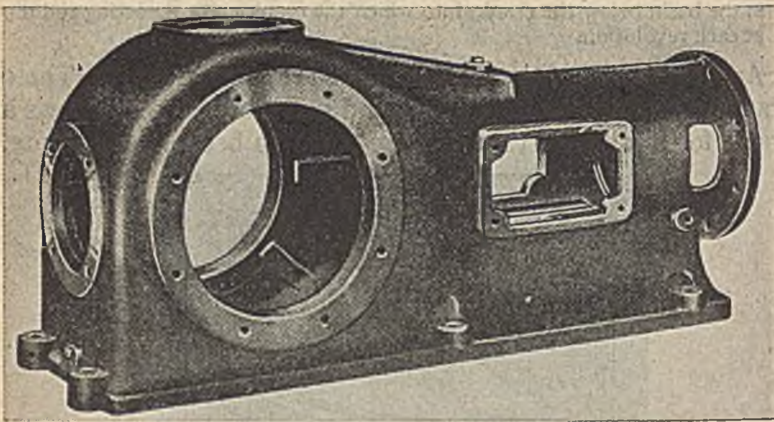


Fig. 47.—Typical frame of the box pattern.

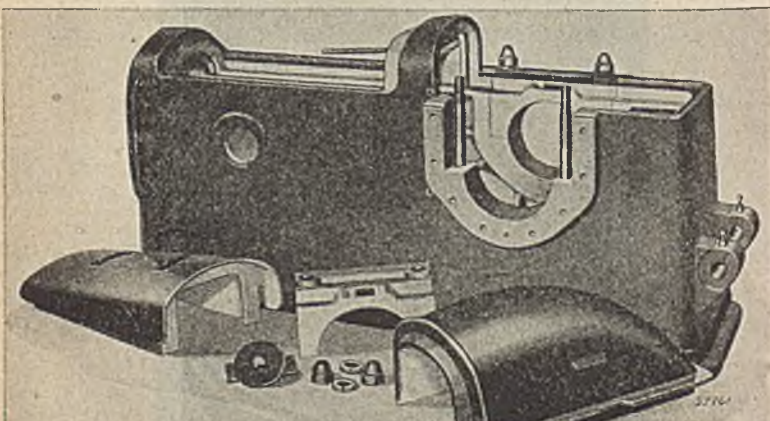


Fig. 48.—Another frame oil type showing main bearings and covers.

Bed Plate.—As just stated on large compressors the bed plate is a separate casting. Fig. 50 shows a typical design which is in fact the bed plate for the frame shown in fig. 49.

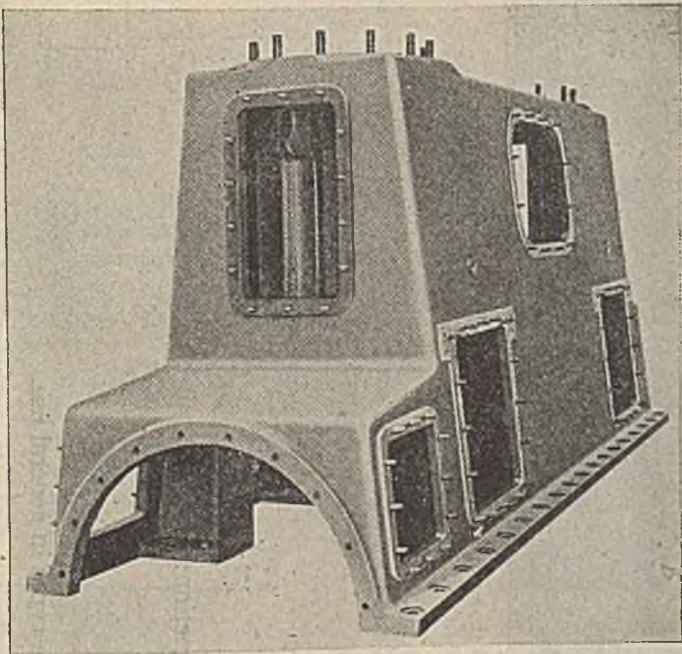


Fig. 49.—Main frame of a large compressor in which the base (bed plate) is a separate casting.

In construction, heavily ribbed cross members are provided to support each of the main bearings. There is a deep sump under the strainer plates to act as a lubricating oil reservoir.

The Piston.—With all the different types of compressors there are numerous forms of pistons. They may be classed:

1. With respect to the cycle frequency as:

   a. Single acting

   b. Double acting

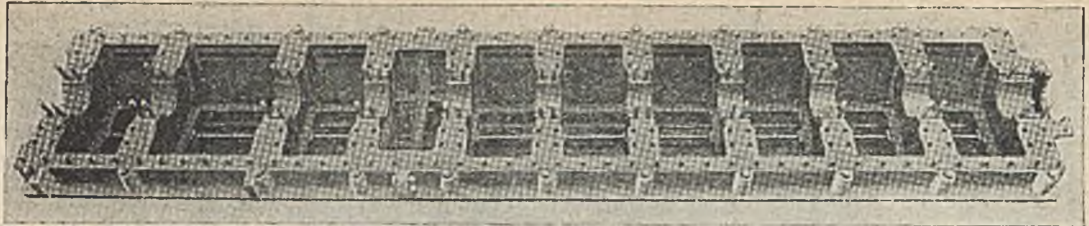


Fig. 50.—Bed plate for the frame shown in fig. 49. Note provision for the multi-throw crank.

2. With respect to form, as:

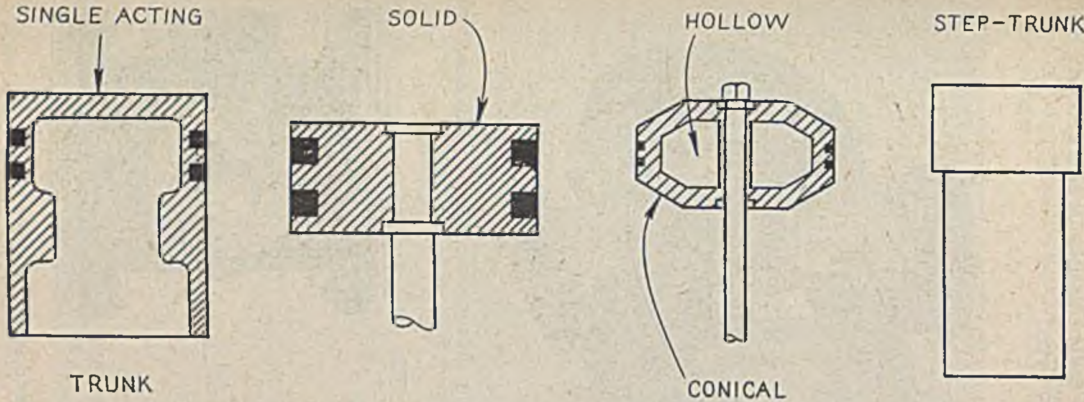
   a. Cylindrical { solid / hollow }
   b. Conical
   c. Trunk
   d. Step trunk

3. With respect to its function:

   a. Driving (steam)
   b. Compressing (air)

Figs. 51 to 54 show some types in general use.

Qualities which a piston should possess are adequate bearing surface to prevent undue wear and sufficient number of rings to prevent leakage.



Figs. 51 to 54.—Various pistons. Fig. 51, single acting trunk; fig. 52, solid, fig. 53, hollow conical; fig. 54, step trunk.

The appearance of single acting pistons is shown in fig. 55, which is an assembly of two-stage pistons for a V two-stage compressor.

Fig. 18 shows a steam piston of the solid cylindrical type and how attached to the thorough piston rod.

Cross Head.—This follows steam engine practice. Fig. 57 shows the box type cross head which is a steel casting. It is provided with shim adjustable parallel fitted bronze shoes top and bottom. At the left, fig. 56, is the wrist pin. This is usually made of carbon steel. Note the shape of the ends. It is fitted into the cross head on a taper, and is secured by a nut and lock nut. It is further secured from turning by the little dowel pin seen on the tapercd end; this fits in the groove. This cross head is of the double gib type.

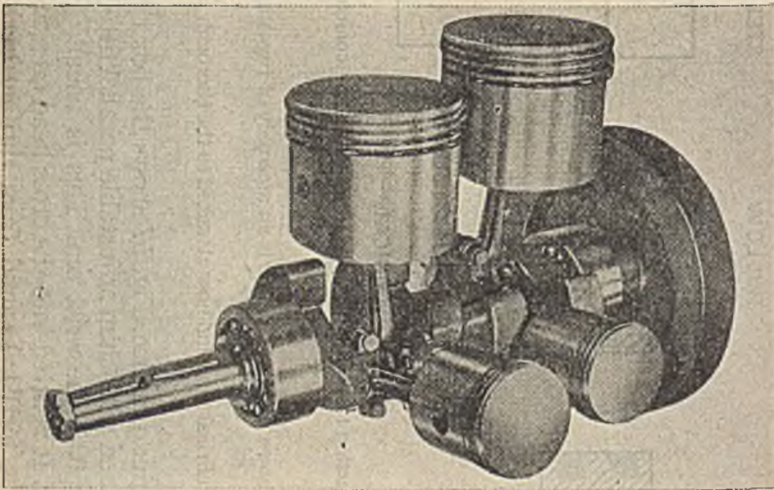
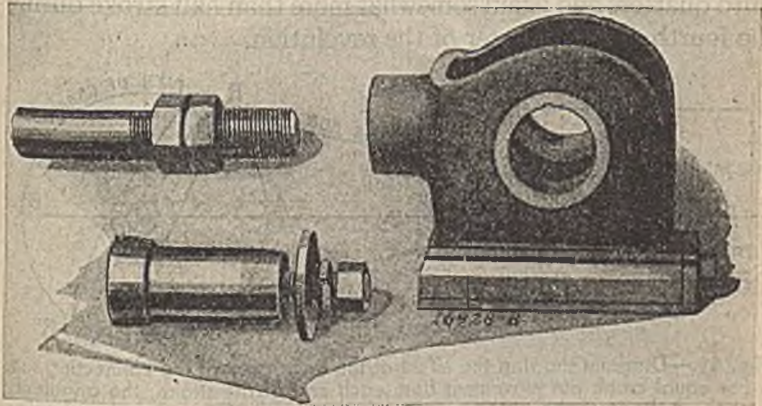


Fig. 55.—Crank shaft with single acting pistons and connecting rods assembled, illustrating ball bearings for main bearings.

Figs. 56 and 57.—Box type cross head with its wrist pin.

The piston rod is attached to the cross head by threaded joint and two piston rod nuts permitting close adjustment of the piston and clearances in the cylinder.

Connecting Rod.—The to and fro or reciprocating motion of the piston is converted into a rotary motion by the connecting rod which joins the cross head to the crank. The connection is made by the wrist and crank pins for which there are suitable bearings at the ends of the rod.



Figs. 58 to 60.—Single slipper type cross head showing wrist pin and piston rod joint construction.

Ques. How long are connecting rods?

Ans. The length measured between the wrist and crank pin centers is usually made from 2 to 2½ times the length of the stroke.

Thurston says the longer proportion gives a long and easy working rod and the former a rather short but yet a manageable one.

Ques. Mention an inherent defect of the connecting rod.

Ans. Its "*angularity*" distorts the motion of the piston.

Ques. What is the nature of this distortion?

Ans. Starting at the beginning of the forward stroke, the inclination or angularity of the rod with respect to the cylinder axis causes the piston to move somewhat more than half its stroke while the crank 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.

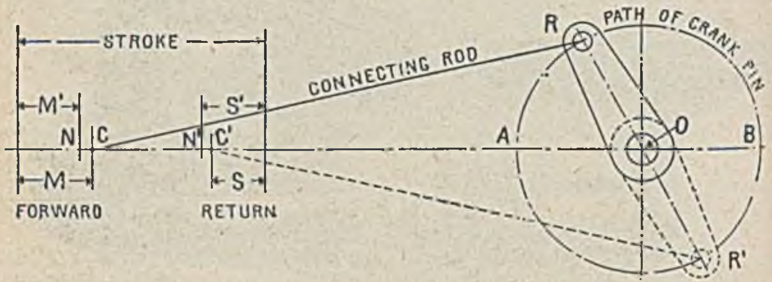


Fig. 61.—Diagram showing the effect of the angularity of the connecting rod. For equal crank pin movement from each end of the stroke, the angularity of the rod causes the piston to travel further on the forward stroke, than on the return stroke.

The effect of the angularity of the connecting rod is shown in fig. 61.

Ques. What other name is given to the angularity of the rod?

Ans. It is sometimes called obliquity of the rod.

Angularity is the preferred word.

Ques. Upon what does the amount of distortion of piston movement depend?

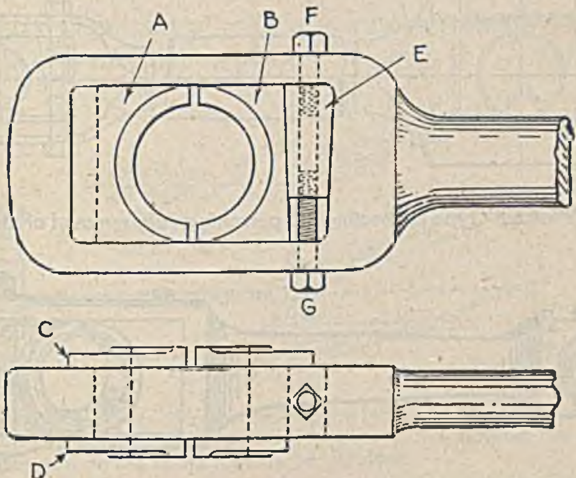
Ans. Upon the length of the connecting rod.

The shorter the rod the greater the distortion.

Ques. How are the rod ends made?

Ans. They may be solid or built up.

Ques. What are the various arrangements for adjusting the brasses to take up wear?



Figs. 62 and 63.—Solid end with block adjustment. **The parts are:** A, B, brasses, A being provided with flanges C, D; E, adjusting block or wedge; F, G, adjustment bolts which retain the block in the desired position.

Ans. Adjustment may be provided for by means of 1, blocks; 2, bolts, or 3, gibs and cotters.

Figs. 62 and 63 show a solid end with block adjustment. A rectangular slot is cut in the enlarged section in which is inserted the brasses A and B, having suitable flanges, C and D, to retain them in the slot. A wedged shape block E is fitted to slide upon B, thus bringing A and B closer together and reducing the size of the bearing. Two bolts, F and G, are threaded in the ends of the block to secure it in position.

To adjust: The bolt F is first loosened and then G tightened to the desired amount. The block is then locked in place by tightening F.

Ques. What is a "gudeon"?

Ans. An obsolete name for a wrist pin.

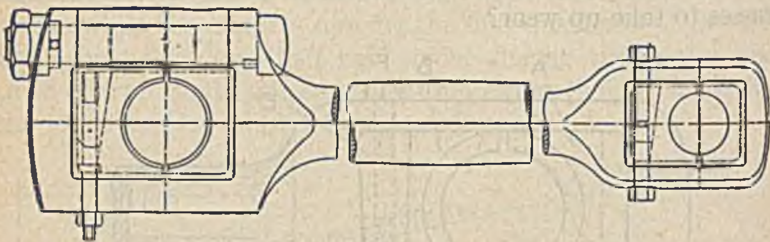


Fig. 64.—"Hatchet" type connecting rod permitting easy removal of the brasses.

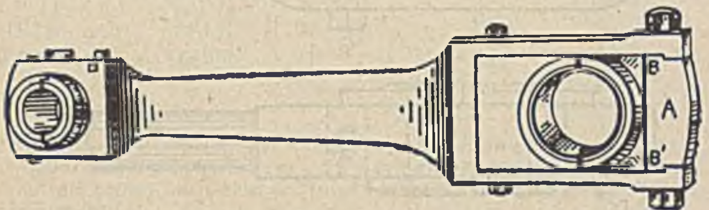


Fig. 65.—Connecting rod type with removable end. The extreme end is a separate piece removable by taking out the end bolts. The brasses are then easily accessible. The rod is provided with block adjustment.

The Valve Gear.—By definition the valve gear is: *The mechanism or combination of parts by which a reciprocating or to and fro motion is imparted to the valve from the rotary* motion of the shaft. The simplest form of valve gear corresponding to the type employed on direct connected steam units consists of:

1. Yoke
2. Stem

3. Guide
4. Rocker lever (on some gears)
5. Eccentric rod
6. Eccentric strap
7. Eccentric

These parts are shown assembled in fig. 68.

Ques. What is the valve yoke?

Ans. By definition a yoke is a frame or clamp for holding the parts of a machine in place.

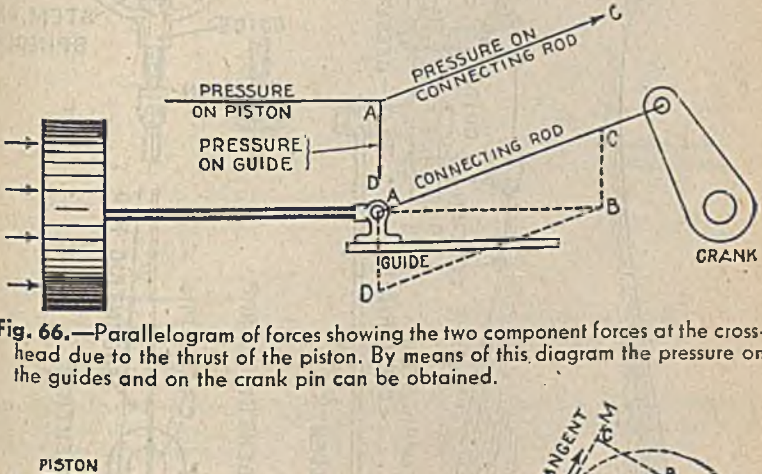


Fig. 66.—Parallelogram of forces showing the two component forces at the cross-head due to the thrust of the piston. By means of this diagram the pressure on the guides and on the crank pin can be obtained.

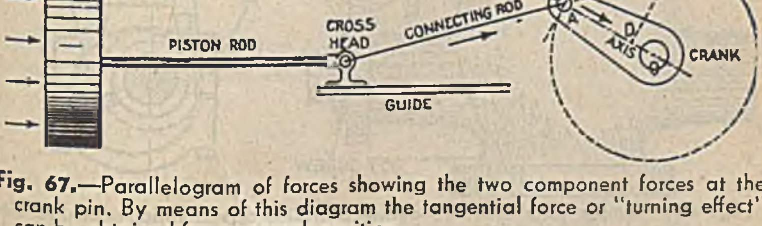


Fig. 67.—Parallelogram of forces showing the two component forces at the crank pin. By means of this diagram the tangential force or "turning effect" can be obtained for any crank position.

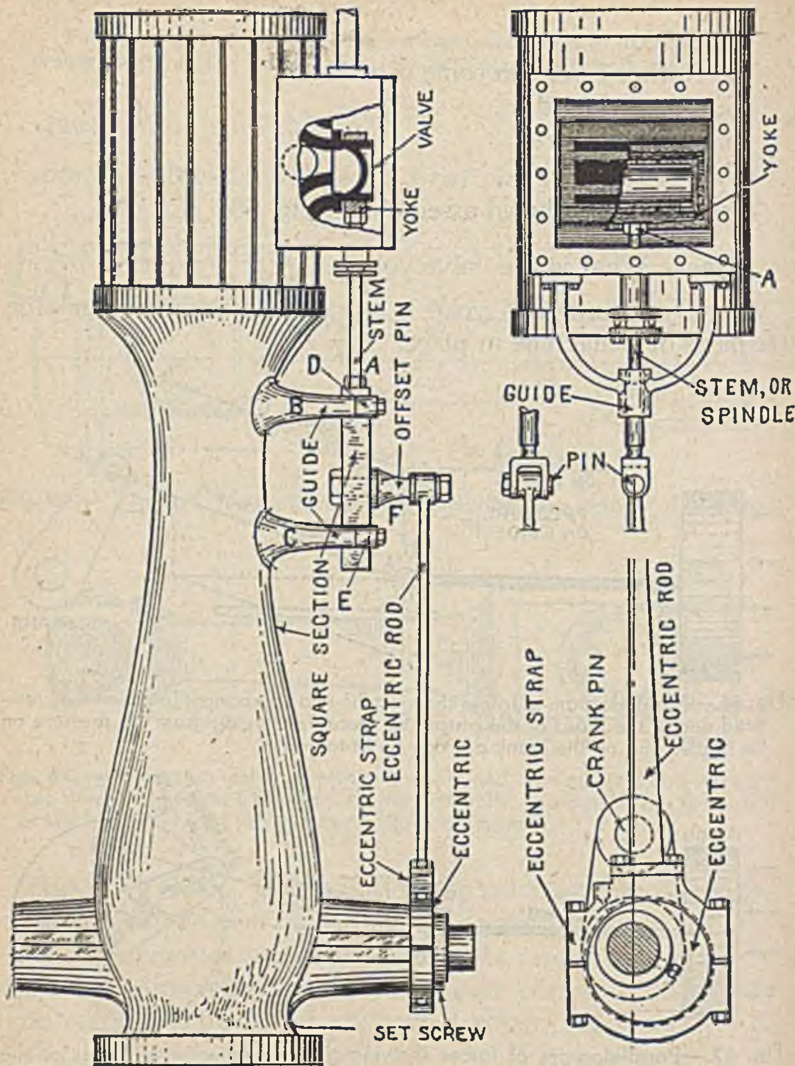


Fig. 68.—Simple form of valve gear, consisting of eccentric, eccentric strap, eccentric rod, pin, guide, stem or spindle with adjustable end, and yoke.

Specifically, the rectangular frame which fits over the box like portion of the slide valve and to which the spindle is attached as shown in fig. 69.

Here the fit between the valve and yoke is such that the valve is free to move in the yoke to adjust itself to the seat. At A, is a slight enlargement to receive the valve stem.

Ques. Describe the valve stem.

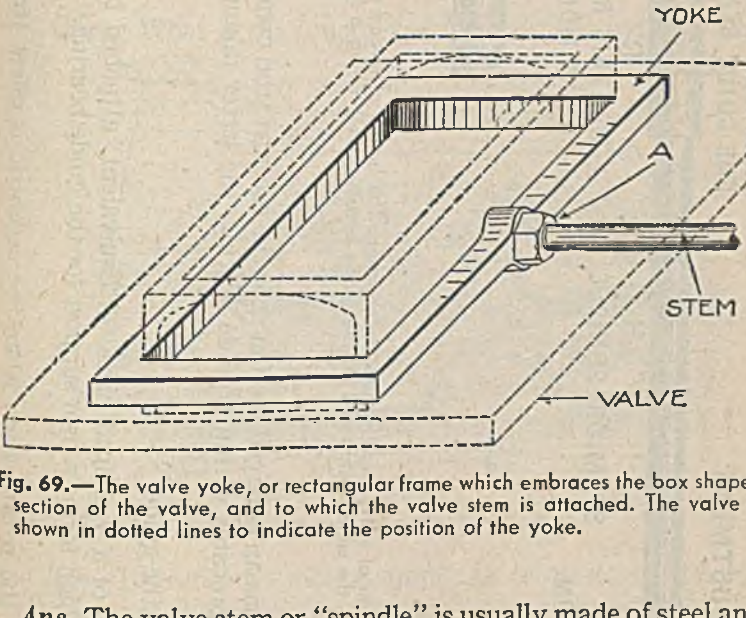
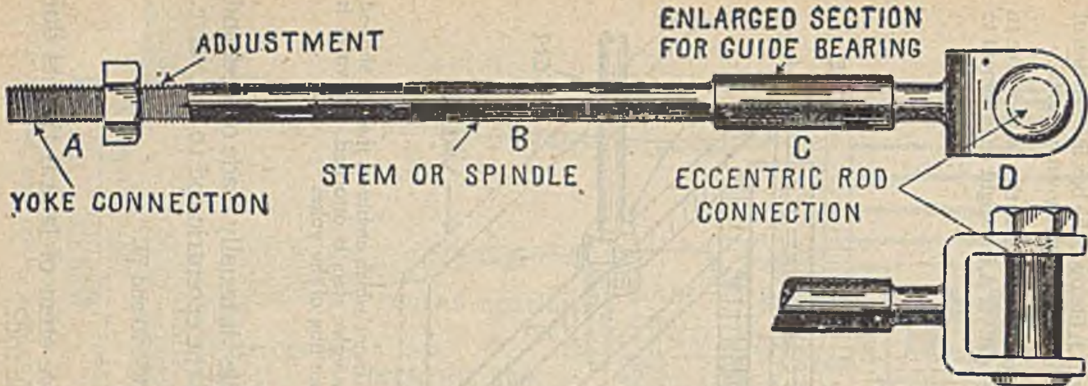


Fig. 69.—The valve yoke, or rectangular frame which embraces the box shaped section of the valve, and to which the valve stem is attached. The valve is shown in dotted lines to indicate the position of the yoke.

Ans. The valve stem or "spindle" is usually made of steel and serves to transmit the motion of the eccentric rod to the valve.

A typical valve stem is shown in figs. 70 and 71.

Ques. What relieves the valve stem of lateral thrusts due to the angularity of the eccentric rod?



Figs. 70 and 71.—Form of valve stem for yoke connection. The length of thread at A, is made sufficient for adjustment. At C, the stem is enlarged for the guide bearing, there being a forked end which carries the eccentric rod pin.

Ans. The stem has an enlarged section near the eccentric rod connection as shown in fig. 70, which reciprocates within a guide bearing, the latter taking the lateral thrust.

Ques. Describe the guide.

Ans. It consists of a V-shaped piece (or equivalent) attached to the crank end of the steam chest which forms a firm support for the guide bearing.

Ques. What is the construction when the eccentric is offset from the valve stem?

Ans. An offset piece which connects the stem and eccentric and having suitable bearing, is provided such as shown in fig. 72.

Ques. What is an indirect rocker lever?

Ans. One interposed between valve stem and eccentric rod in the case of indirect valve gears where the valve stem and eccentric rod move in opposite directions, as in fig. 73.

Ques. What is the function of the eccentric rod?

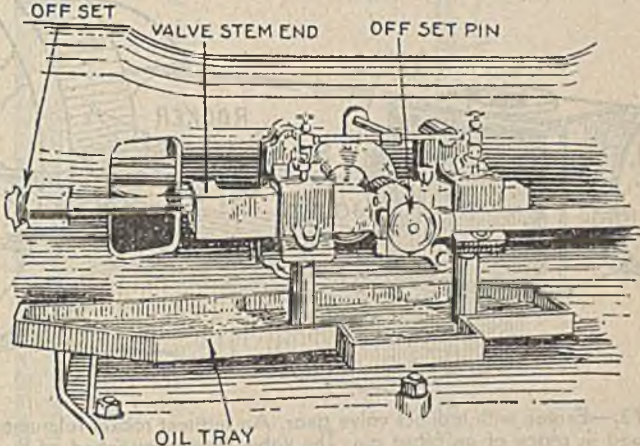


Fig. 72.—Type of valve stem guide used for offset pin.

Ans. It transmits motion from the eccentric to the valve stem and at the same time changes rotary motion into reciprocating motion.

Ques. How is it constructed?

Ans. It maybe either round or of rectangular section. A simple form of rod if shown in fig. 74 with the end connection indicated by dotted lines.

Ques. What modification is made when the rod is placed outside the fly wheel?

Ans. A pin is used in place of an eccentric, if the rod be constructed as in fig. 75.

The Eccentric and Strap.—By definition, an eccentric is: *A disc having its axis of revolution out of its center of figure.*

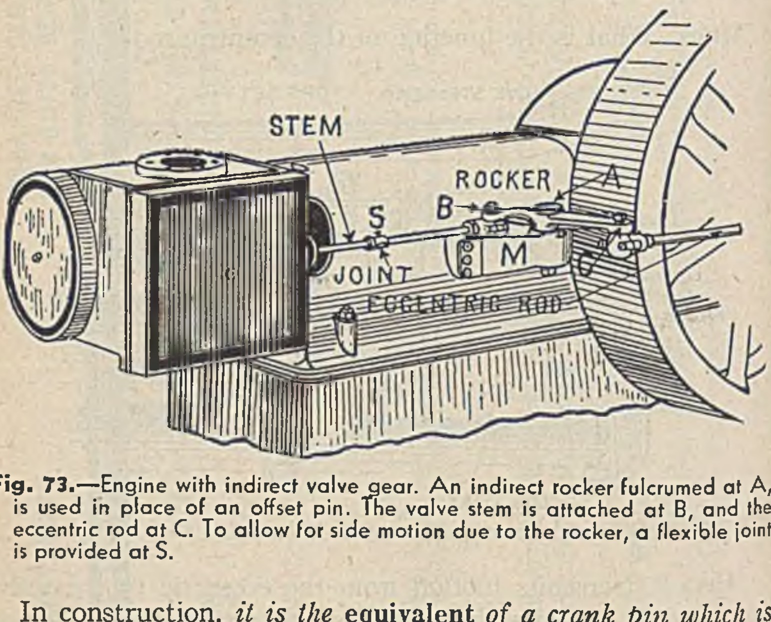


Fig. 73.—Engine with indirect valve gear. An indirect rocker fulcrumed at A, is used in place of an offset pin. The valve stem is attached at B, and the eccentric rod at C. To allow for side motion due to the rocker, a flexible joint is provided at S.

In construction, *it is the* **equivalent** *of a crank pin which is so large in diameter that it embraces the shaft to which it is attached and dispenses with arms.*

Ques. What duty is performed by the eccentric?

Ans. Its object is to change rotary motion of the shaft into reciprocating motion, transmitting to the valve stem.

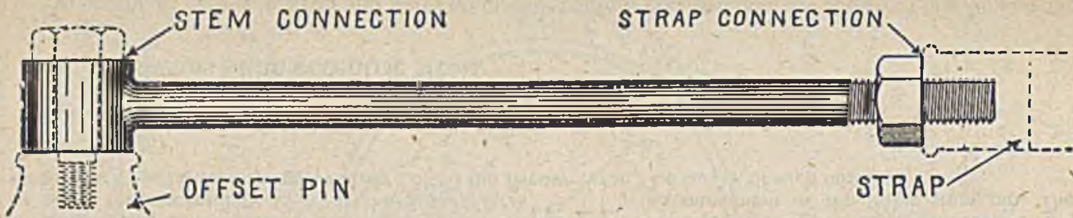


Fig. 74.—Plain eccentric rod having threaded connection for eccentric strap, and an eye for the valve stem pin.

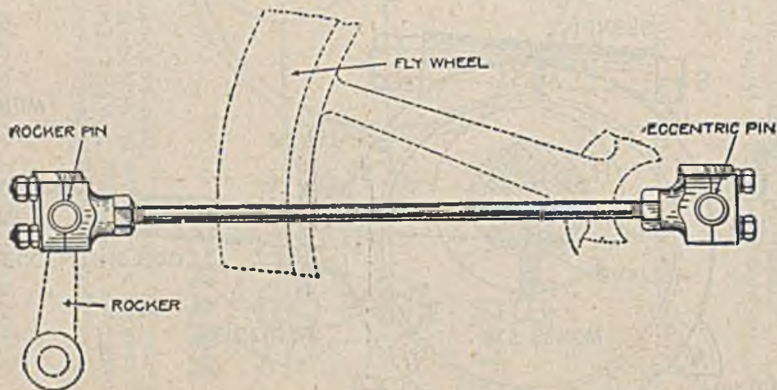
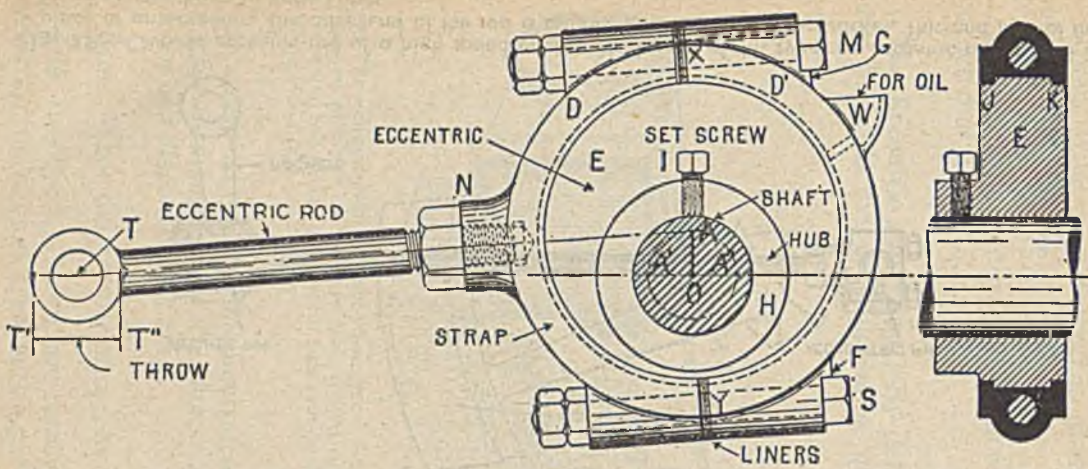


Fig. 75.—Outside eccentric rod of a high speed engine. With rods of this type, an eccentric pin is used in place of an eccentric. The other end of the rod is usually attached to a direct rocker. This and part of the fly wheel are shown in dotted lines.



Figs. 76 and 77.—Eccentric strap, and eccentric rod. The eccentric E, is usually secured on the shaft O, by a set screw I. A groove is turned in the strap to register with a projection on the eccentric so as to prevent side motion. The strap is in two halves, D, D′ and held together by the bolts M, S, liners being inserted between for adjustment. O, is the center of the shaft, and A the center of the eccentric .OA, is the eccentricity; A′A″, or twice the eccentricity OA, is the throw. A′A″ = T′T″, the movement of the valve stem pin. The **eccentricity** is frequently **ignorantly** called the **throw**—don't be guilty of such mistake.

Ques. How is the motion transmitted from the eccentric?

Ans. By means of an eccentric strap.

An eccentric, eccentric strap and valve stem, all of simple construction, are shown in figs. 76 and 77.

Ques. What is the throw of the eccentric?

Ans. Twice the eccentricity or the amount of the reciprocating motion produced.

The throw is equal to the diameter of the circle described by the center of the eccentric as it revolves around the shaft—*not half this distance as it is sometimes* ***ignorantly*** *called.*

Ques. What is the angular advance of an eccentric?

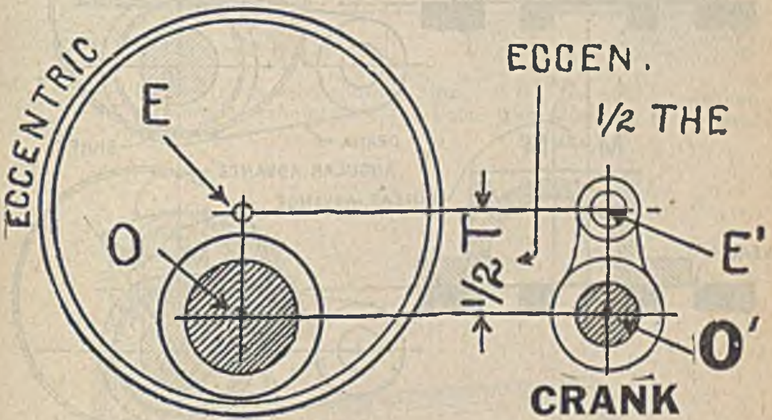


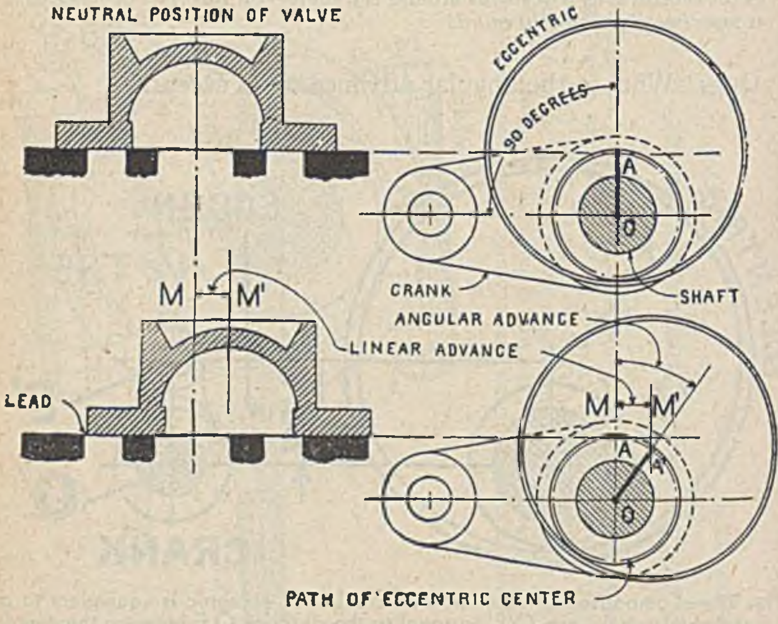
Fig. 78.—Comparison of eccentric and crank. An eccentric is equivalent to a small crank whose arm O′E′ is equal to the distance O E between the center of the shaft, and the center of the eccentric. This distance is the **eccentricity,** or **one-half** the throw. Sometimes **ignorantly** called the throw.

Ans. The number of degrees the eccentric must be turned forward from a position at right angles to the crank to give the valve its *linear advance,* that is an amount equal to outside lap + lead, when the crank is at the beginning of the stroke.

Angular and linear advance are illustrated in figs. 79 and 80.

Ques. What is lap and what is its object?

Ans. It is that portion of the valve face which overlaps the steam port when the valve is in its central or neutral position.



Figs. 79 and 80.—Illustrating linear and angular advance. When the crank is on the dead center, and the eccentric set 90° ahead, the valve should be in its neutral position as shown in fig. 79. The valve, however, when the engine is on the dead center, must be at a distance (M M′, fig. 80) from its neutral position equal to the **lap + lead** or in its position of linear advance. The eccentric then must be turned ahead through an angle A O A′, its angular advance, sufficient to move the valve to its linear advance position M′.

Its object is (outside lap) to shorten the cut off and (inside lap) to increase the compression. See fig. 81.

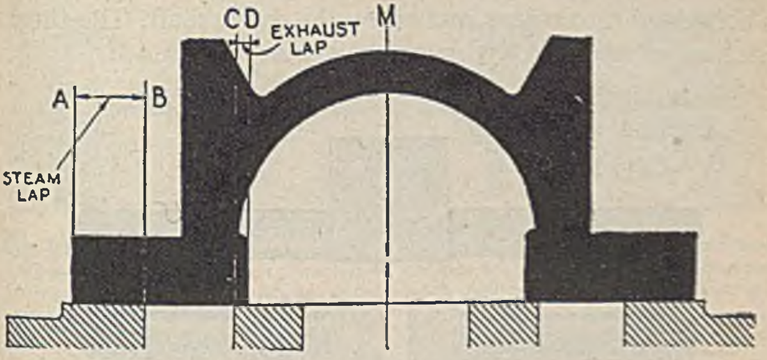


Fig. 81.—The plain D slide valve showing "lap." A B is the outside or steam lap; C D, the inside or exhaust lap. The figure also illustrates the neutral position of the valve.

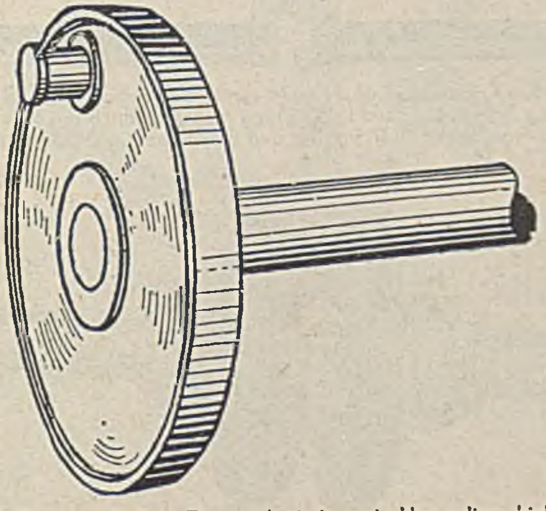
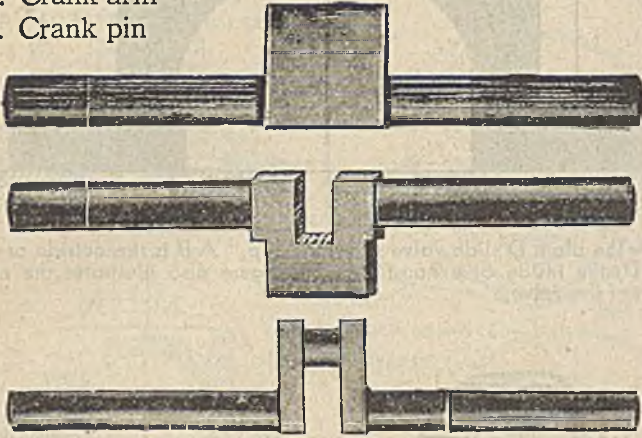


Fig. 82.—Built up crank shaft. The crank pin is carried by a disc which is counterbalanced. Hydraulic pressure is used to force the pin and shaft in place.

The Crank Shaft.—The reciprocating motion of the piston is converted into rotary motion by the crank shaft. The three essential elements of any crank shaft are:

1. Shaft
2. Crank arm
3. Crank pin



Figs. 83 to 85.—Construction of a forged center crank shaft. Fig. 83 shows the rough forging. It is first placed in a slotting machine which removes the metal between the crank arms, as in fig. 84, and then turned in a lathe and finished as shown in fig. 85.

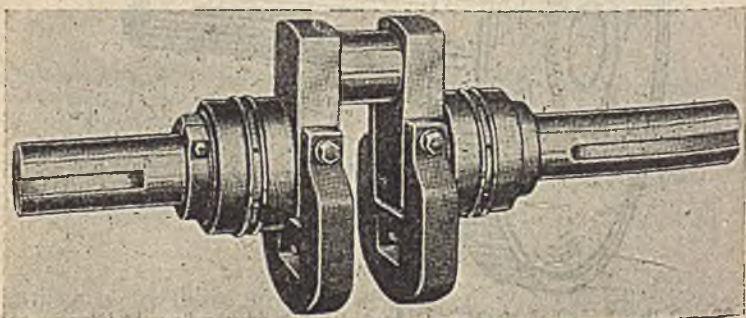


Fig. 86.—Center crank shaft with bolted counter weights.

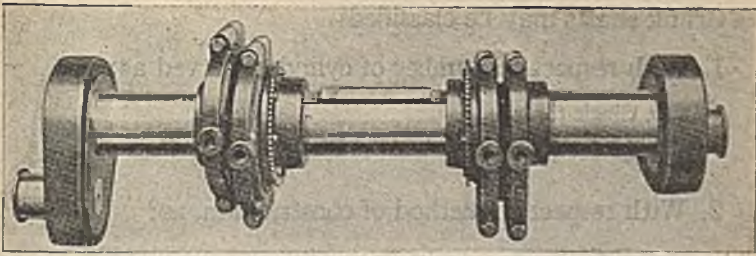


Fig. 87.—Built up crank shaft with arm type cranks for a cross connected two stage compressor direct connected to compound steam end. Note the four eccentrics, two each for Meyer gear of each cylinder. The chain sprockets for governor are also seen.

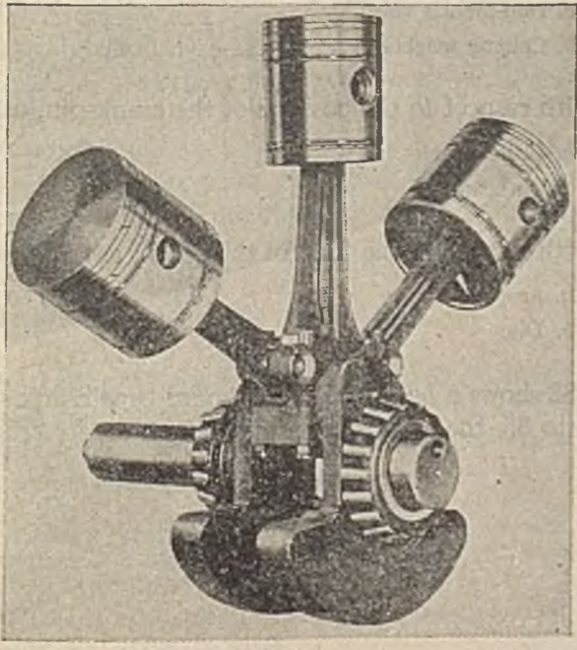


Fig. 88.—Articulated connecting rods of three cylinder compressor assembled to one throw crank shaft. Note the roller main bearing.

Crank shafts may be classified:

1. With respect to number of cylinders served as:

   a. Single throw
   b. Multi-throw

2. With respect to method of construction, as:

   a. Built up
   b. Solid

3. With respect to balancing as:

   a. Non-counter weights
   b. Counter weights

4. With respect to the position of the crank pin, as:

   a. Center crank
   b. Outboard

5. With respect to the form of the crank, as:

   a. Arm
   b. Disc

Fig. 82 shows a built-up disc outboard type crank shaft and figs. 83 to 85, the construction of a forged center crank shaft.

CHAPTER 48

Compressor Types

In the preceding chapter each part that goes into the make up of a compressor is explained and illustrated at length. It remains to show all these parts assembled, that is, the complete compressor as variously constructed to meet different service conditions.

The fundamental types are here shown "in the metal" that is by external and sectional views from which the general appearance of these machines and their construction can be seen.

In this chapter there will be no duplication of the elementary or skeleton diagrams of Chapter 46, but frequent reference will by made to these elementary drawings.

Single Acting Compressors.—For small units where compactness is not of prime importance, single acting compressors are largely used. They are made in many types such as air or water cooled, single or multi-cylinder, one or two stage, etc.

Fig. 1 shows the inside of an air cooled single cylinder one stage compressor direct connected to electric motor drive. The notation gives considerable information about the construction.

Fig. 2 shows a two cylinder air cooled single stage compressor with induced draught—note the fan in the fly wheel. The "radial type" three cylinder air cooled compressor is shown in fig. 3. A two stage single cylinder water cooled compressor is shown in fig. 4.

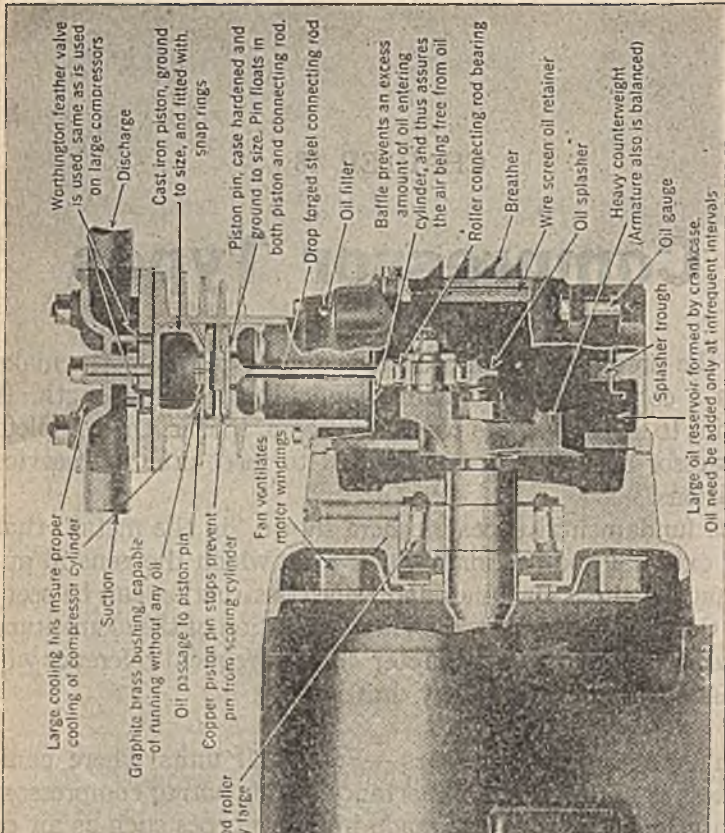


Fig. 1.—Monobloc pattern vertical one cylinder air cooled single stage compressor direct connected to electric motor drive.

Double Acting Compressors.—Of course a double acting cylinder is more compact than single acting, and accordingly for larger capacities double acting cylinders are used as for a given capacity they take up less room than the single acting type. See Chapter 46, fig. 2.

Fig. 5 shows a horizontal double acting water cooled single stage belt driven compressor. The sectional view plainly shows

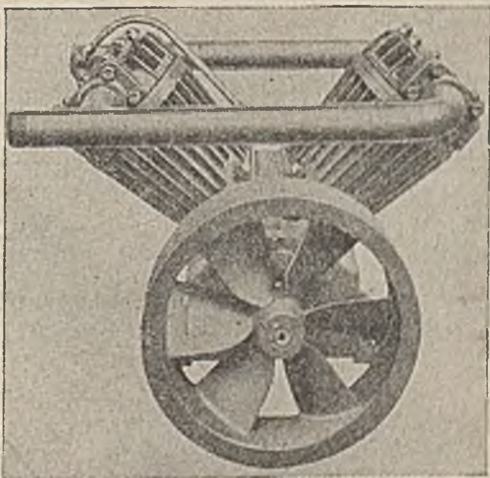


Fig. 2.—Two cylinder single stage induced draught air cooled compressor. Note cylinders placed at 90° which gives better load distribution.

construction of cylinder details such as water jackets, placement of valves, etc.

Compressors of this class are ordinarily made in sizes with compressor cylinders ranging from 6½ × 7 to 20 × 13 ins. Evidently with belt drive any kind of prime mover may be used as may be best suited to the conditions. See Chapter 46, fig 12.

Of course belt drive takes up more room than a direct connected unit and where space is limited a direct connected unit is installed. Here choice is not limited to any particular prime mover as compressors are direct connected to steam engines, internal combustion engines, electric motors, etc.

Fig. 1 shows the direct connected electric motor drive. The same compressor end as was shown in fig. 5 is here shown, fig. 6, direct connected to a steam end.

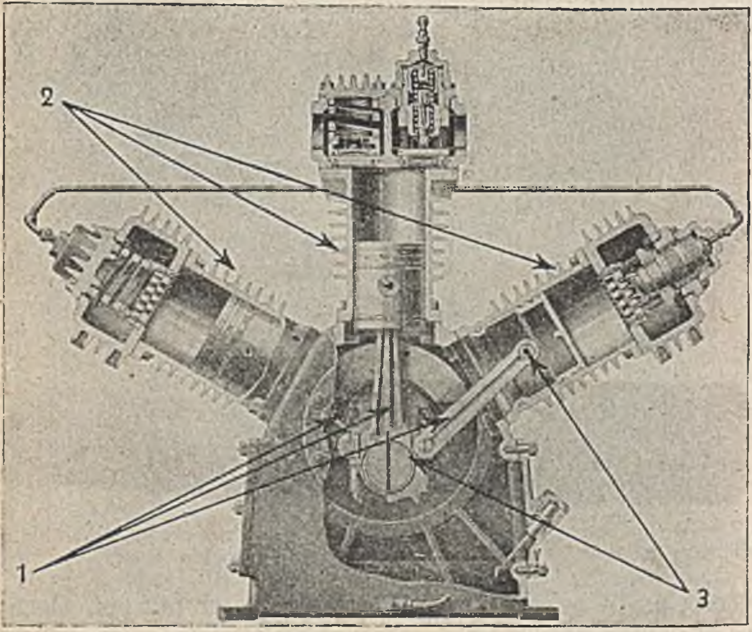


Fig. 3.—Three cylinder "radial type" single stage air cooled compressor. **1**, connecting rods; **2**, cooling fins; **3**, wrist and crank pins.

There is a common piston rod for both steam and air cylinders; this is known as a "tandem" hook up in distinction from the cross connected arrangement later shown. See Chapter 46, fig. 11.

Another tandem unit which looks like fig. 6, but is quite different is shown in fig. 7.

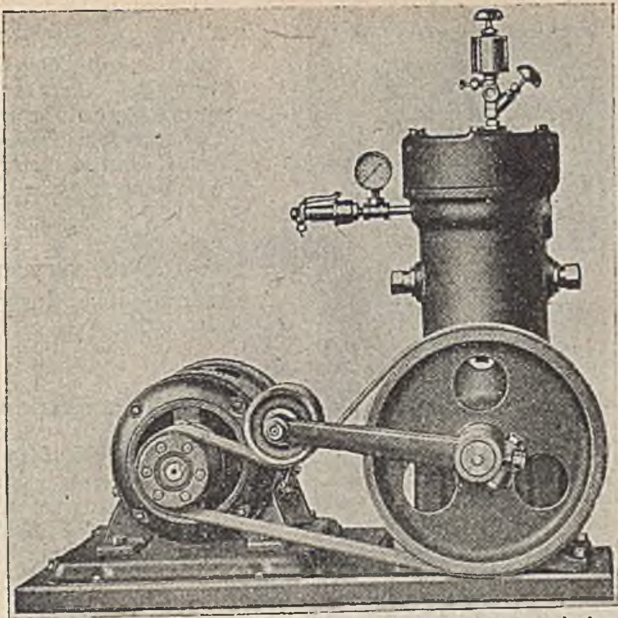


Fig. 4.—Vertical single cylinder step piston two stage water cooled compressor close coupled belt connection to electric motor. Note the idler giving belt tension and increasing the arcs of contact.

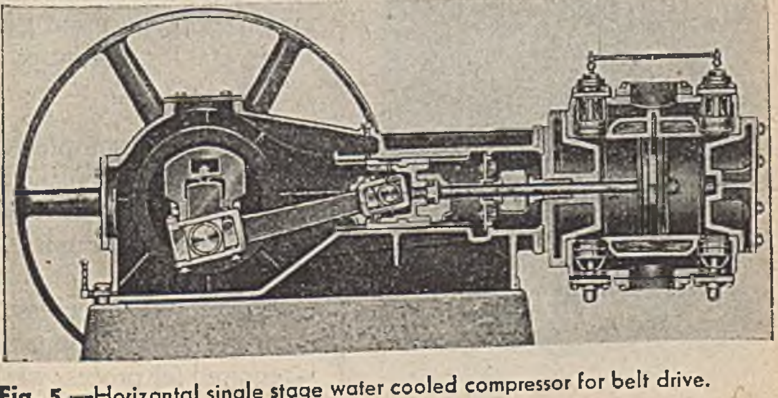


Fig. 5.—Horizontal single stage water cooled compressor for belt drive.

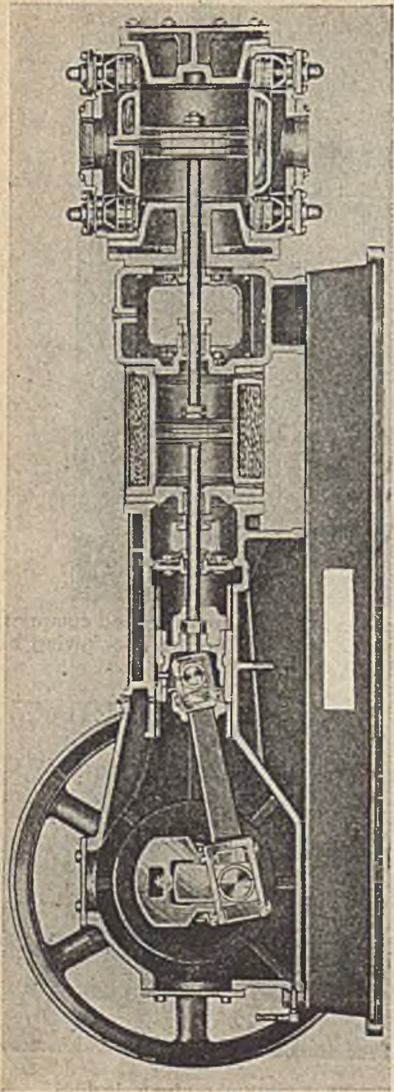


Fig. 6.—Horizontal single stage water cooled compressor direct connected steam drive.

This is a belt drive two stage machine with inter-cooler (all two-stage compressors have inter-coolers). The actual appearance of this compressor with belted electric motor drive is seen in fig. 8.

A machine very similar to the one just described is shown in fig. 9.

This compressor is of the three-stage two-cylinder type. The working of this interesting cycle is shown on page 151. Note the external appearance as seen in the accompanying illustration, fig. 10.

In general, two stage compressors are designed for continuous heavy duty service in supplying air at from 150 to 500 lbs. pressure.

Three stage compressors are designed for continuous extra heavy duty service in supplying air at from 500 to 2500 lbs. These units are used wherever high pressure air or gas in small quantities is required for process work.

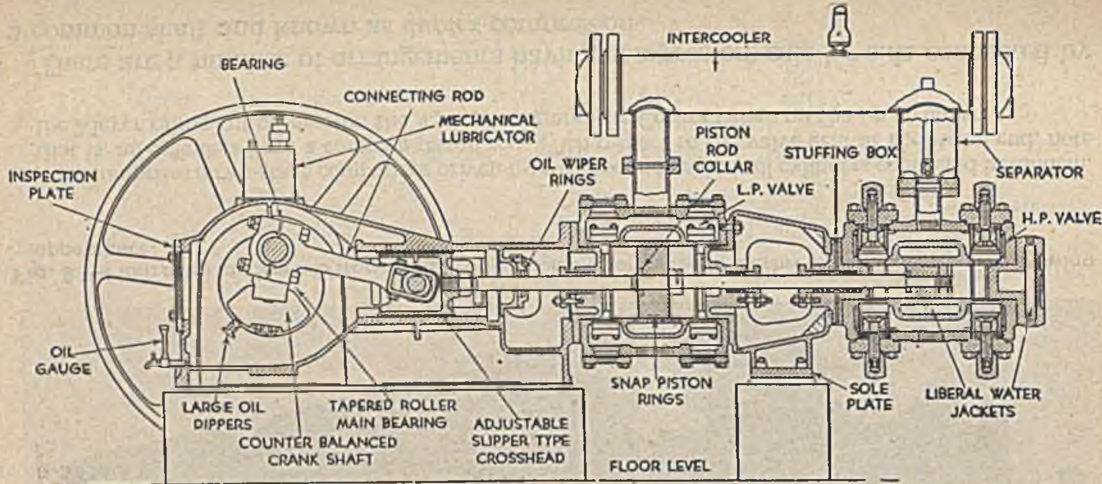


Fig. 7.—Horizontal tandem two stage water cooled belt driven compressor. Sectional view with names of parts. The illustration shows the inter-cooler connecting the two cylinders.

The tandem arrangement lends itself to many combinations of power and compression.

For instance an air cylinder may be driven by a compound steam engine, all cylinders having a common piston rod as shown in the elementary drawing, page 138. While providing a very economical drive the hook up has an extended length and this would have to be considered with respect to space available.

An extension of this hook up is shown in sectional view drawing, fig. 11, which shows construction details.

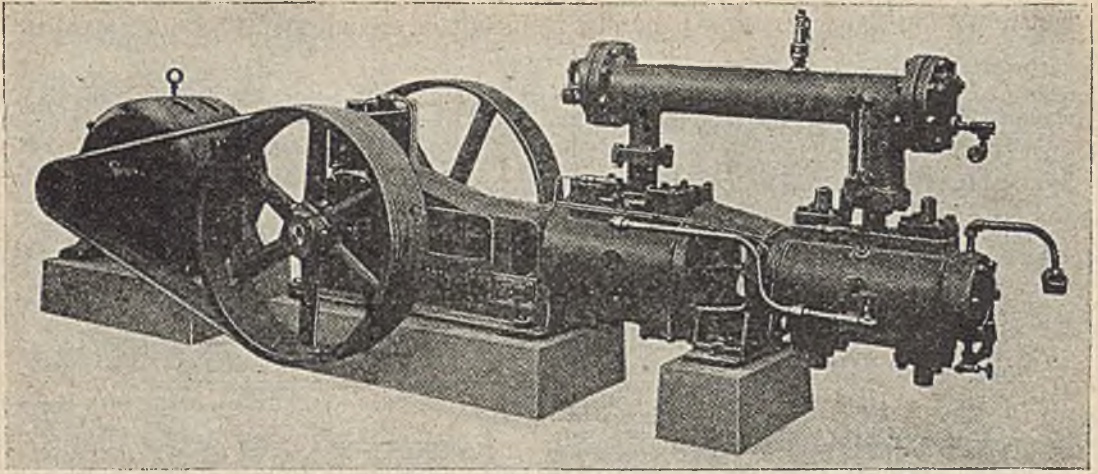


Fig. 8.—Horizontal tandem two stage water-cooled compressor belted to electric motor. External view showing appearance.

It comprises a two-stage compressor driven by a compound engine, all cylinders connected in tandem, that is, all cylinders have a common piston rod. With respect to the valve gear at the steam end, note the Meyer riding cut off gear on the *h.p.* cylinder and the Corliss valves on the *l.p.* cylinder.

There are a number of arrangements having compressors side by side connected by a common shaft and known as *duplex* compressors.

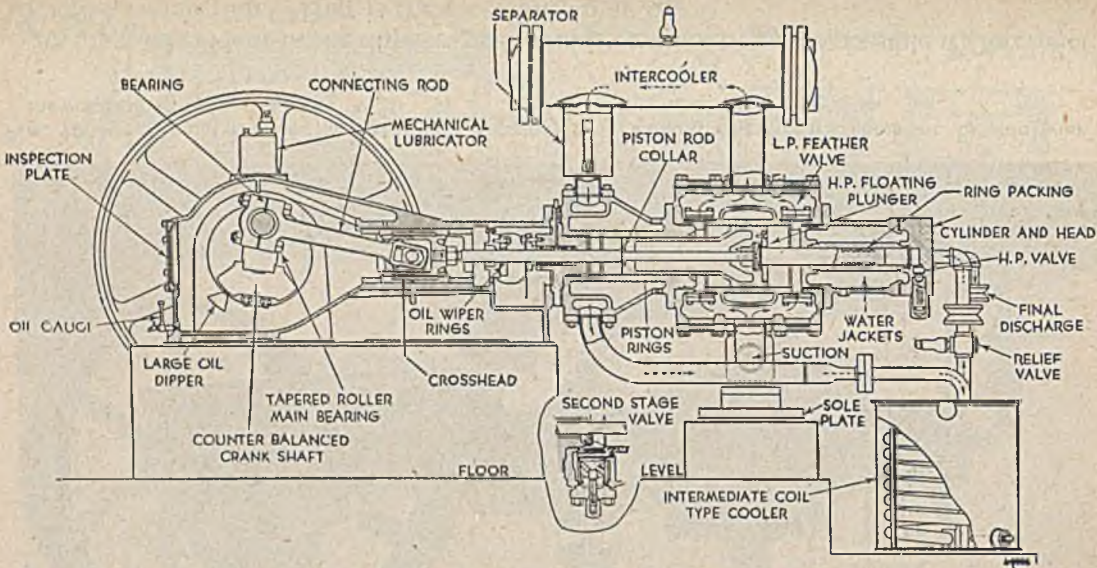


Fig. 9.—Horizontal tandem three stage (step piston) water cooled belt driven compressor. Sectional view with names of parts.

There may be two independent compressors as in figs. 12 and 13 or "cross connected" units as in figs. 14 and 15. Such machines are for the large size units where considerable air capacity is required. Actual construction of a duplex type compressor is shown in figs. 16 and 17.

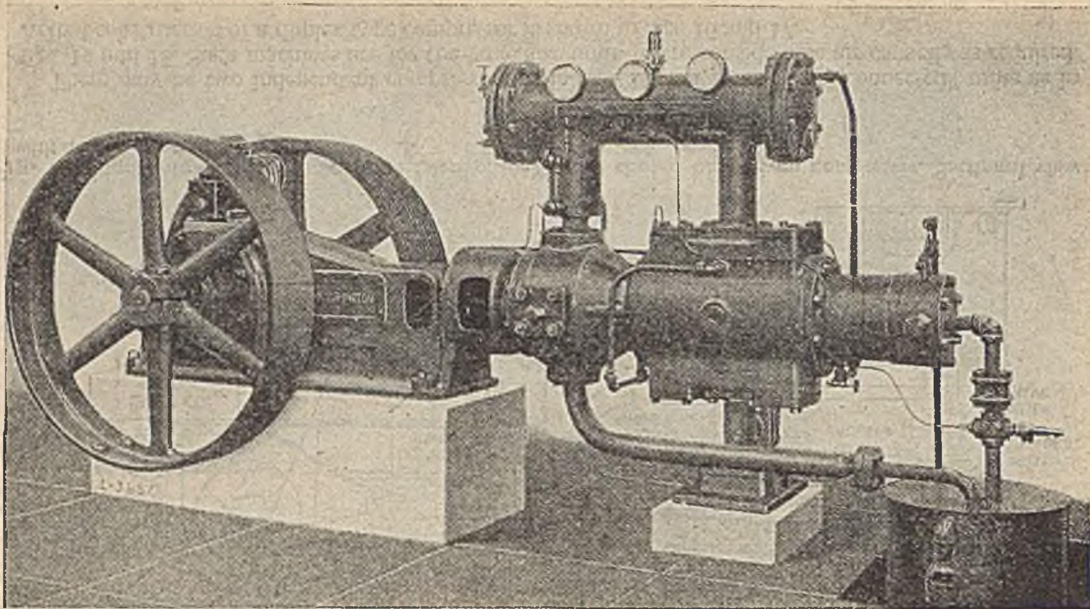


Fig. 10.—Horizontal tandem three stage (step piston) water cooled belt driven compressor. External view showing appearance.

As must be evident in the duplex type various combinations are possible with respect to "cross connection." That is the assemblies may be

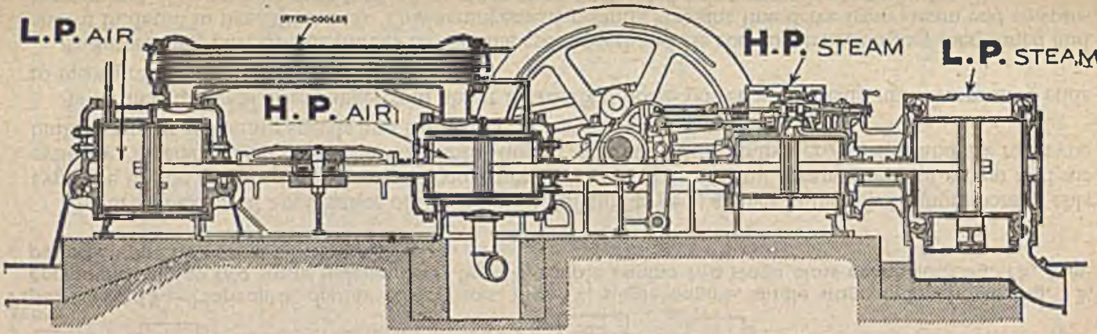
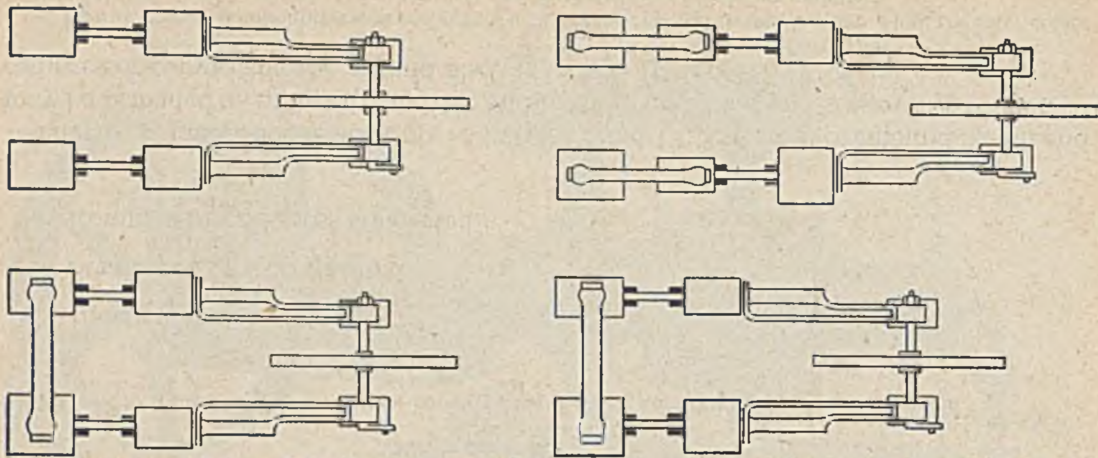


Fig. 11.—Horizontal tandem two stage compressor with direct connected compound steam end.

1. Independent
2. Steam end cross connected
3. Compressor end cross connected

That is; 1, independent hook up, as in fig. 12 and 13; 2, a cross compound steam end may be provided as in figs. 16 and 17; and 3, there may be a two-stage cross connected compressor comprising the air and also, as in figs. 16 and 17.

That is, a cross compound steam end drives a two-stage cross-connected air end. Here the inter cooler forms the cross connection between the first and second stage compressor cylinders.



Figs. 12 to 15.—Horizontal duplex compressors. Fig. 12, single engine, single stage independent; fig. 13, compound engine two stage independent; fig. 14, simple engine two stage cross connected; fig. 15, compound engine two stage cross connected.

The actual external appearance of a compressor of this type is shown in fig. 18. Compressors of this type are of the so-called "four corner construction" which permit any arrangement of steam and air cylinders. Specifically it is of the horizontal duplex double acting double cross head and tie rod type built to run at moderate speeds and for heavy continuous service.

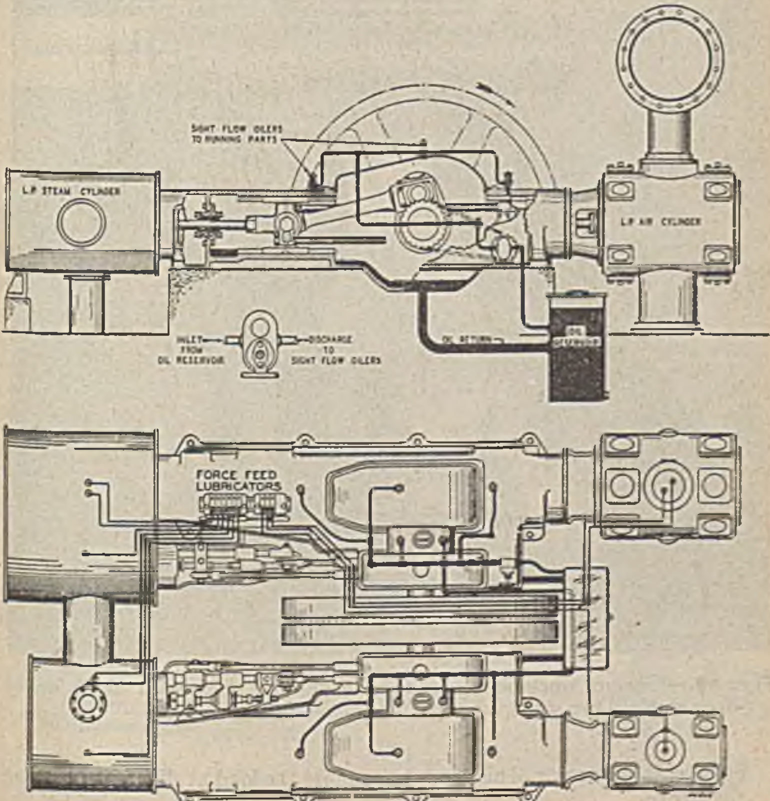
As ordinarily made sizes range from about 50 to 1500 horse power with steam and compressor ends to meet practically any pressure conditions.

An interesting part of this unit is the double cross head and tie rod construction just mentioned and shown in detail in figs. 19 and 20. This arrangement permits straight line drive from steam end to compressor end. The various details of a two stage cross connected compressor is shown in fig. 21.

Inter and after coolers as well as other auxiliary apparatus are explained in later chapters.

Ques. What is the principal advantage of a duplex machine?

Ans. By properly timing the events of the two pairs of cylinders the load is equalized to a considerable extent.



Figs. 16 and 17.—Elevation and plan of direct connected compound steam driven ("four corner") two stage water cooled compressor showing general arrangement and oiling system.

Ques. How are they timed?

Ans. By setting the two crank pins at 90°.

The result is shown by the indicator diagram, fig. 22. The diagram shows that as the load on one air cylinder is increasing, the load on the other

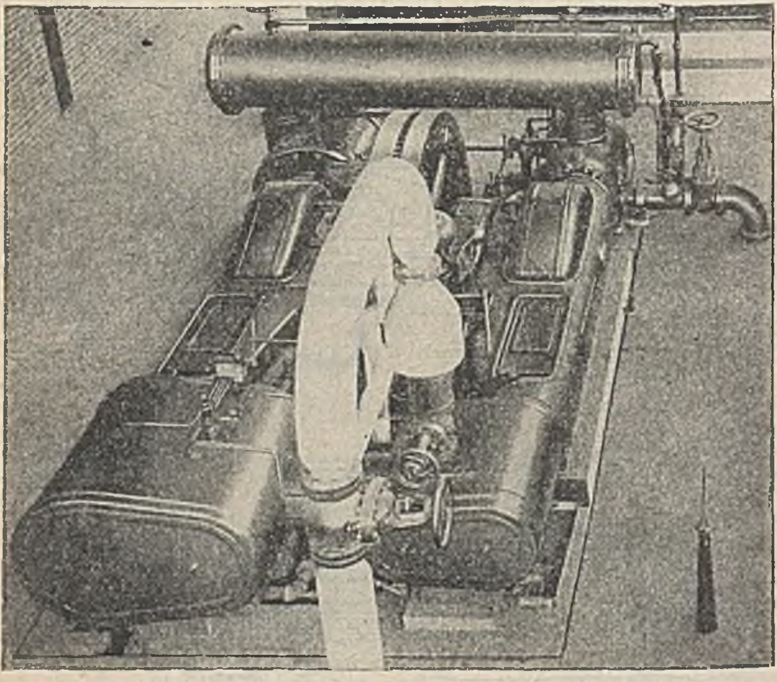


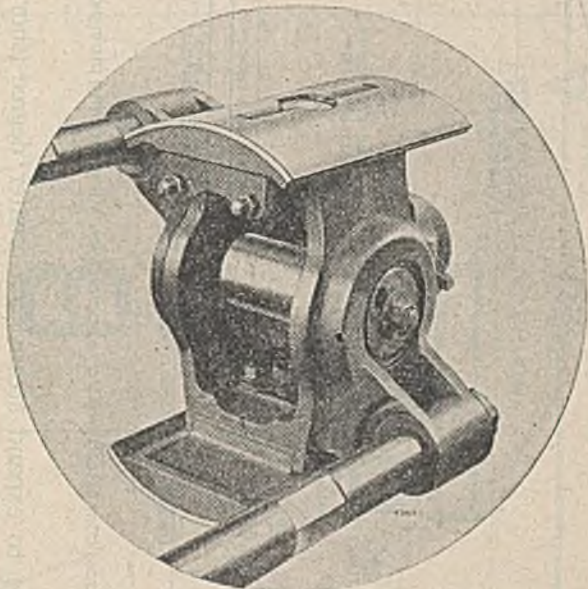
Fig. 18.—General appearance of direct connected compound steam driven ("four corner") two stage water cooled compressor.

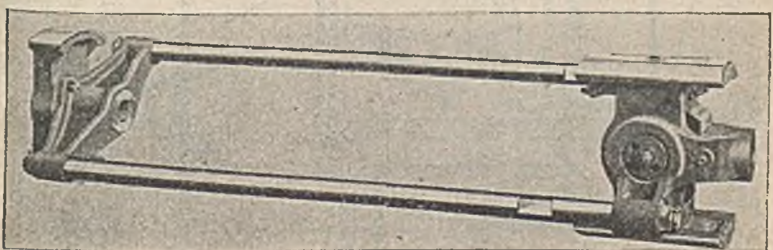
cylinder is decreasing. This tends to equalize the load at different points of the stroke.

Ques. What advantage in operation is obtained?

Ans. The phase difference in the two air cycles makes it possible to run at lower speeds.

Ques. What other advantage is obtained?





Figs. 19 and 20.—Straight line double cross head and tow rod construction with detail of the power cross head for the unit shown in figs. 16 to 18.

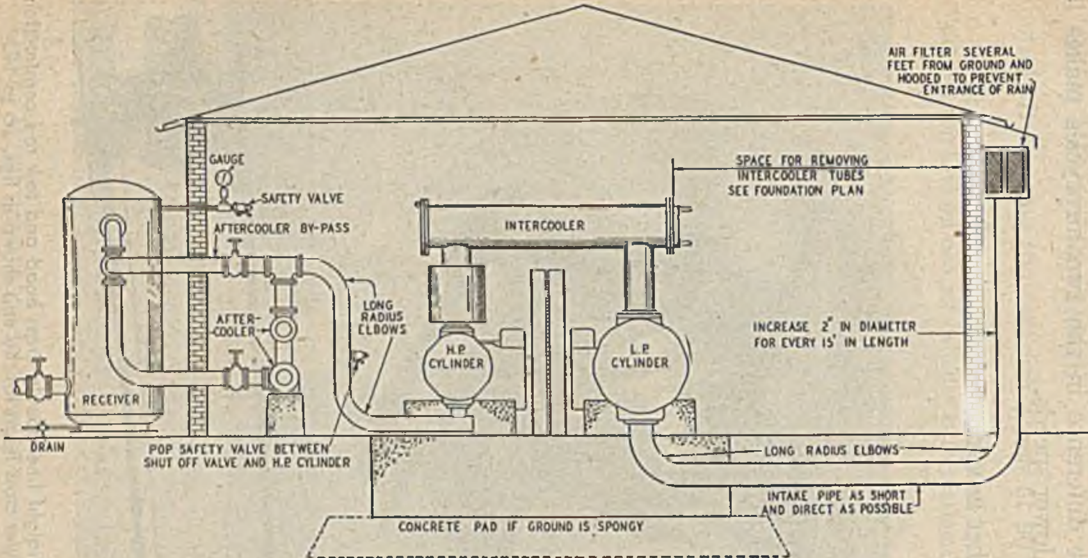


Fig. 21.—Installation of a duplex (cross connected) two stage compressor. View showing piping and auxiliaries.

Ans. It makes it possible to expand steam to a greater degree (shorter cut off) and secure the economy corresponding to the increased ratio of expansion.

Ques. What are the disadvantages of the duplex type?

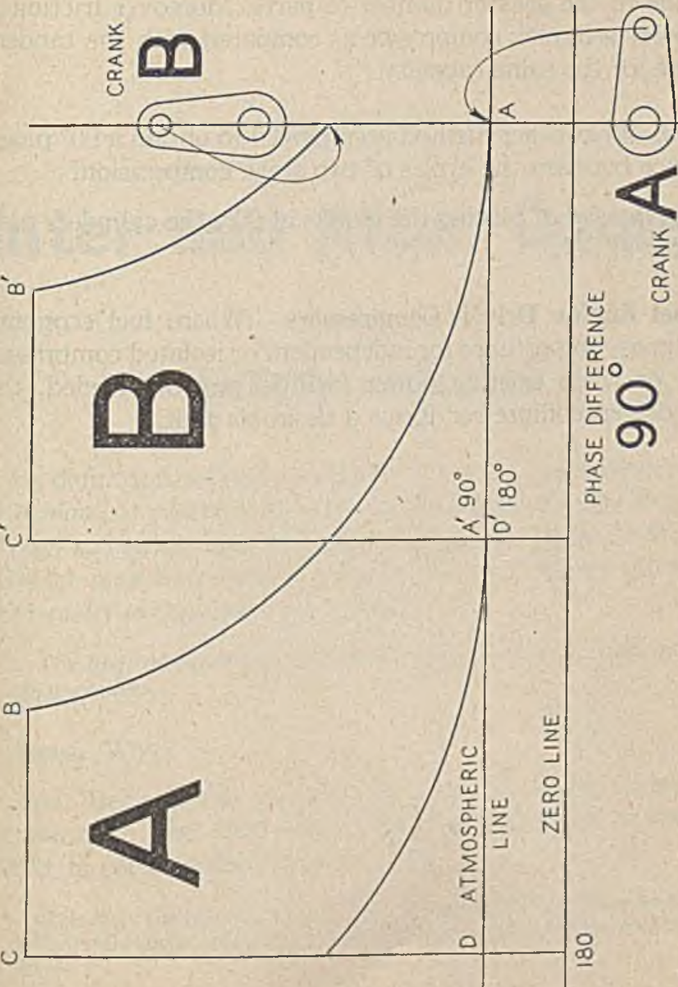


Fig. 22.—Diagram showing steam and air simultaneous distribution in duplex compressor with 90° sequence of cranks.

Ans. Increased cost of construction and installation and upkeep due to the greater number of parts. Moreover friction is greater for a duplex compressor as compared with the tandem machine for the same capacity.

Ques. What other method is employed to obtain a 90° phase difference between the cycles of two stage compression?

Ans. Instead of placing the cranks at 90°, the cylinders may be placed at 90°.

Diesel Engine Driven Compressors.—Where fuel economy is of primary importance for independent or isolated compressor plants, or when existing power facilities are overloaded, the Diesel driven compressor forms a desirable unit.

CHAPTER 49

Inter and After Coolers

1.
INTER-COOLERS

By definition, an **inter-cooler** is: *A species of surface condenser or economizer placed between the two cylinders of a two stage compressor so that the* **heat of compression** *generated in the first stage cylinder* **may be removed** *from the air as it passes through the inter-cooler to the second stage cylinder.*

The ***heat of compression*** is accordingly the reason for both ***inter-*** and ***after-coolers.***

Ques. Why?

Ans. Because the pressure of a gas increases with rise in temperature and this excess pressure results in a waste of power in compressing a gas.

Evidently the primary function of an inter-cooler is to conserve power by reducing through cooling the volume of air or gas before it enters the second stage.

Ques. What is the approximate saving?

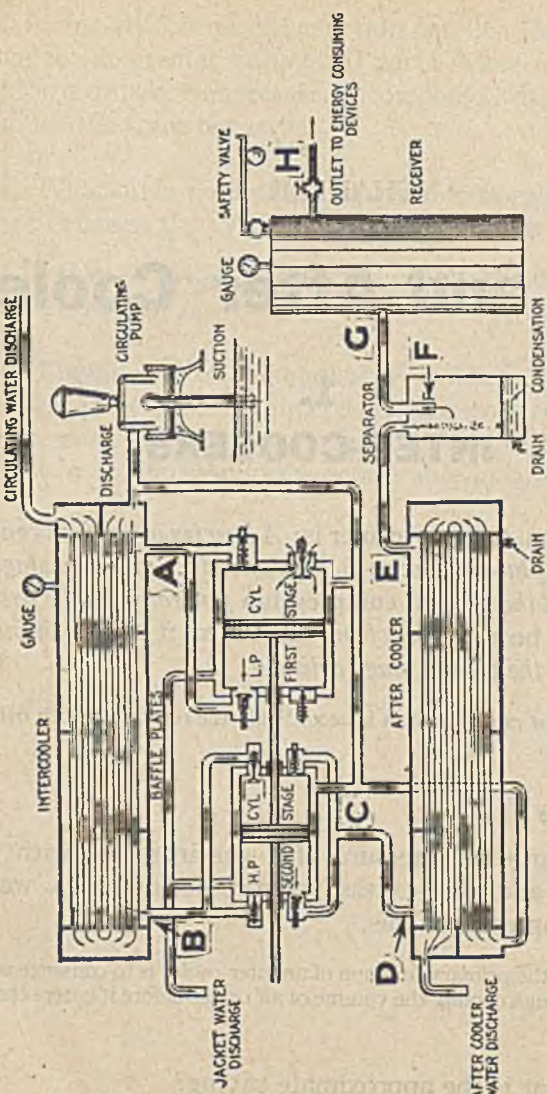


Fig. 1.—Elementary compound or two stage air compressor showing all the auxiliary apparatus essential for maximum efficiency in two stage compression. **The cycle of operation** is explained in the accompanying text.

Ans. Roughly each 10° Fahr. decrease in temperature between stages results in 1% saving in power output.

Ques. Mention another favorable result obtained by inter-cooling between stages.

Ans. Since it reduces the maximum temperature in the second stage cylinder better cylinder lubrication is obtained which minimizes valve troubles.

Ques. What objectionable action is brought about by inter-cooling?

Ans. Cooling causes *condensation* of some of the moisture passing through the inter-cooler.

Ques. How is the condensation disposed of?

Ans. A separator with float valve ejector is provided to get rid of the condensate.

Ans. What would happen if the condensate passed into the second stage high pressure cylinder?

Ans. It would wash away lubrication and induce rapid wear.

Two Stage Compression Cycle with Inter- and After-Coolers.—Fig. 1 shows an elementary two stage compressor with both inter-cooler and after-cooler, also all the auxiliary apparatus.

Referring to the elementary drawing, free air is compressed in the low pressure or first stage cylinder to a moderate pressure and discharged at **A**, into the inter-cooler where the *heat of compression* is carried away by the *cooling* or *circulating water* which flows from the *circulating pump* through the *inter-cooler* cooling tubes.

In this passage short circuiting is prevented by the series of baffle plates.

Passing out of the inter-cooler at **B**, the air at moderate pressure enters the high pressure or second stage cylinder and is then compressed to high pressure being discharged at **C**, into the after-cooler at **D**, wherein the heat of the second stage compressor is abstracted in precisely the same way as in the inter-cooler, the circulating water connection being clearly shown.

The highly compressed air now leaves the after-cooler at **E**, and passes to the separator.

The construction of this device as shown, causes the current of compressed air to ***suddenly change its direction*** 180°, thus causing any moisture or slugs of water that may be in the air ***to be hurled to the bottom of the separator by centrifugal force.***

Thus freed of any condensate, the dry air leaves the separator at **F**, and enters the receiver at **G**, wherein it is stored at the working pressure and is discharged through the outlet **H**, to perform various kinds of work such as driving rock drills, engines and all kinds of pneumatic tools.

The numerous arrows in fig. 1 show:

1. Course of air in traversing the apparatus.
2. Direction of piston movement at the instant depicted.
3. Direction of water flow; *a*, through the cylinder jackets; *b*, through the inter-cooler, and *c*, through the after-cooler.

The cycle may be briefly stated as follows:

1. First stage compression.
2. Cooling by inter-cooler.
3. Second stage compression.

4. Cooling by after-cooler.
5. Separation of condensate by separator.
6. Storage in receiver.

Ques. What should be noted about temperatures?

Ans. The final temperature in each cylinder will be the same if the work has been divided equally and the inter-cooler properly designed.

It will be very much lower than it would be if the compression were done in one cylinder.

For instance, in compressing air to 100 pounds pressure in a two-stage compressor, the air is compressed from atmospheric pressure to, say, 26½ pounds in the intake cylinder and is delivered to the inter-cooler at this pressure and at 240° Fahr.* (atmospheric temperature at 60° Fahr.). If all of the heat of compression be taken out by the inter-cooler, it is admitted to the high pressure cylinder at atmospheric temperature and is there compressed from 26½ pounds to 100 pounds and delivered to the receiver at 240° Fahr.

In a single stage compressor the air is compressed from atmospheric pressure to 100 pounds in one cylinder and reaches the receiver at 482° Fahr.*

Ques. What difficulties are encountered in an attempt to cool the air in the cylinder?

Ans. There is not sufficient time during the stroke to thoroughly cool the air by any available means.

Water jacketing is the generally accepted practice, but it does not by any means effect thorough cooling.

The air in the cylinder is so large in volume that but a fraction of it is brought in contact with the jacketed parts. Moreover air is a bad conductor of heat and accordingly requires time to change its temperature.

Again the piston drives it away from the jacketed surfaces so that the cooling effect is greatly reduced. This is especially true of large cylinders resulting in less cooling than with small cylinders.

*NOTE.—Radiation and cooling influence of water jacket not considered.

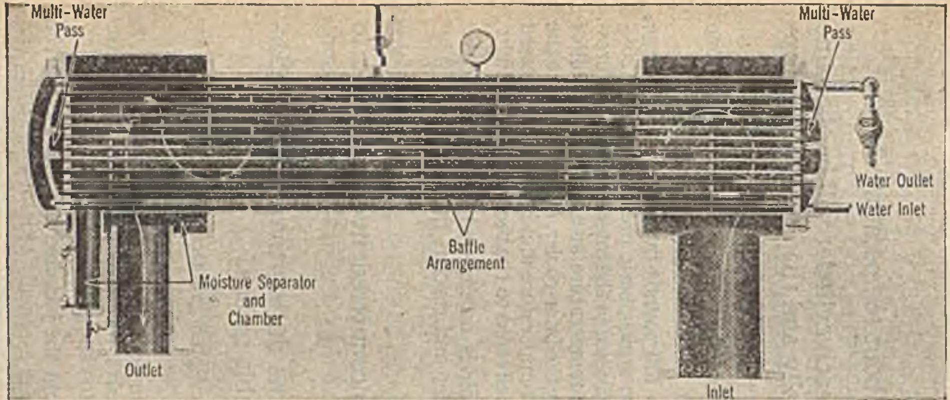


Fig. 2.—Typical inter-cooler. **It consists of** a nest of Admiralty metal tubes expanded into corrosion resistant tube plates at each end. Baffles effectively direct the air or gas across the tubes many times. The entire nest of tubes may be withdrawn as a unit for cleaning.

Ques. What should be inferred about compressors whose indicator diagrams show compression lines running away from the adiabatic?

Ans. Such lines should be regarded with suspicion, as they probably result from leakage rather than efficient cooling.

Such leaks will explain many isothermal compression lines.

Inter-Cooler Construction.—The inter-cooler, as already pointed out, is the one element which makes possible the superior economy of two stage compression over single stage.

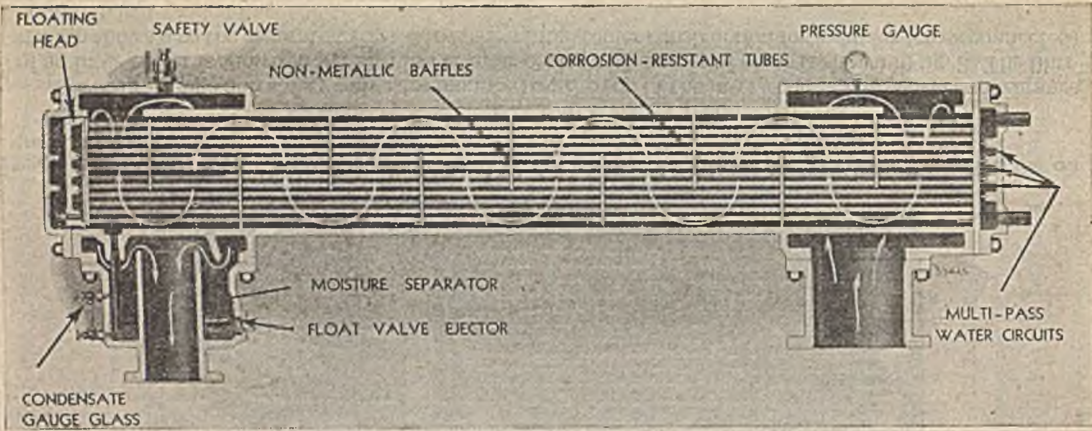
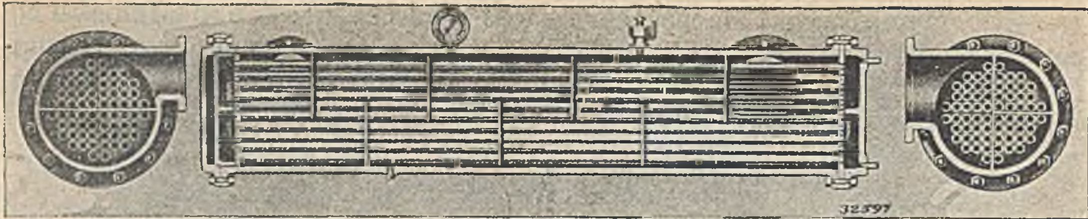


Fig. 3.—Sectional view of inter-cooler with names of parts. In this design cooling water flows through the tubes and the air over their exterior surfaces multi-pass.

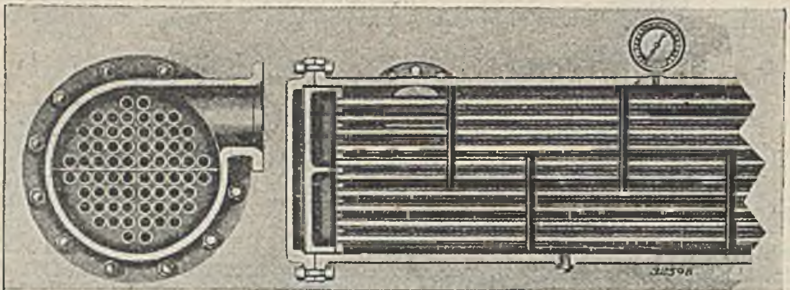
It is important that the inter-cooler be designed to bring about the greatest possible temperature change as the air passes from the low pressure cylinder to the high pressure cylinder.

Ques. Describe the construction of a typical inter-cooler.

Ans. It consists of a nest of special metal tubes expanded into corrosion resistant tube plates at each end. Baffles effectively direct the air or gas across the tubes many times, thus affording perfect contact with the cool tubes. The entire nest of tubes may be withdrawn as a unit for cleaning. Provision is made for tube expansion.



Figs. 4 to 6.—Sectional view through inter-cooler showing water passages. Water circulates through the tubes.



Figs. 7 and 8.—Sectional view through left end of the inter cooler (shown in figs. 4 to 6) showing the air connection to the high pressure cylinder.

A pressure gauge and safety valve are mounted on the top of the inter-cooler. The general appearance of an inter-cooler is shown in fig. 2. A larger inter-cooler with names of parts is shown in fig. 3. The illustration shows also the separator. Figs. 4 to 8 show inter-cooler construction for a duplex (cross connected) compressor.

Ques. What stabilizing effect has the inter-cooler?

Ans. The large receiver volume, formed by the inter-cooler and the connecting piping, results in a nearly uniform discharge pressure from the low pressure cylinder.

Ques. How are inter-coolers mounted?

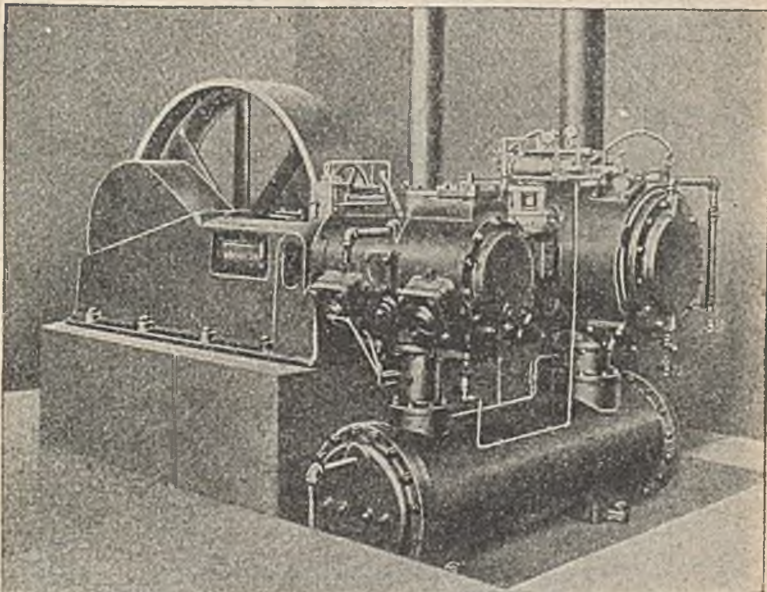


Fig. 9.—Belted compressor with under-mounted inter-cooler.

Ans. Compressors of 1200 cu. ft. capacity and larger have their inter-coolers mounted directly over the air cylinders. The inter-coolers of smaller machines are usually arranged underneath the cylinders.

Fig. 9 shows a compressor with under-mounted inter-cooler.

2

AFTER-COOLERS

By definition, an after-cooler is: *A species of surface condenser in which compressed air is cooled after compression.*

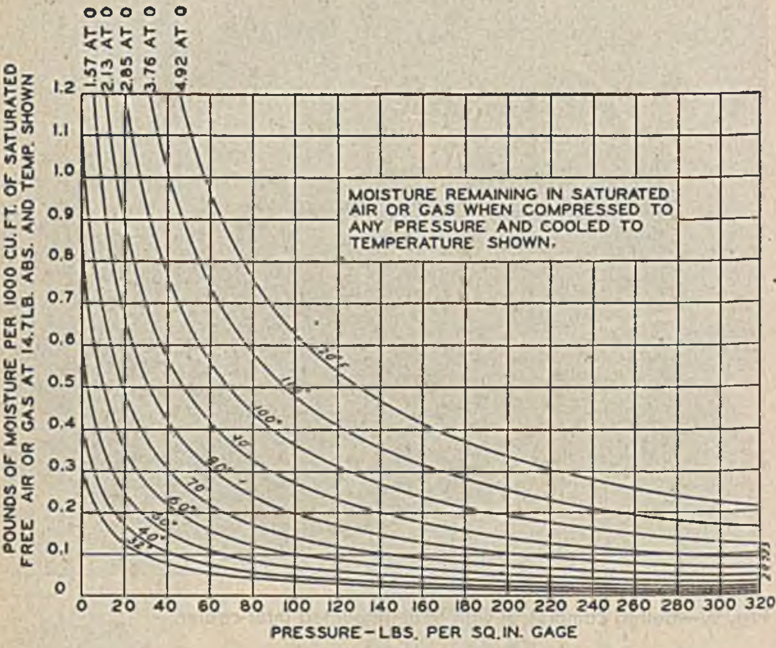


Fig. 10.—Air moisture chart. See accompanying text "After Cooling Principles."

After-Cooling Principles.—All free air and all manufactured gas contain varying amounts of moisture in the form of water vapor. The temperature and volume, and not the pressure, determine the moisture capacity of air or gas. This capacity

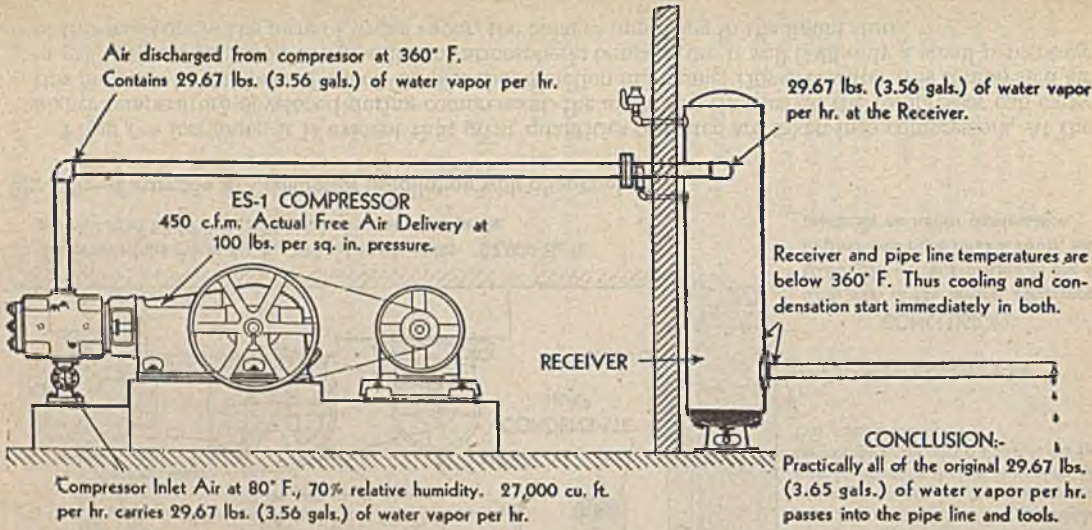


Fig. 11.—Example 1. Compressor installation without after-cooler.

increases as the temperature rises. For example, at 80° Fahr., 1000 cubic feet of saturated air, *i.e.*, air of 100 per cent relative humidity, will contain 1.57 pounds of moisture. At 100° Fahr., it will contain 2.85 pounds. One gallon of water weighs 8.33 pounds. Thus, it is apparent that air can contain a very appreciable amount of water.

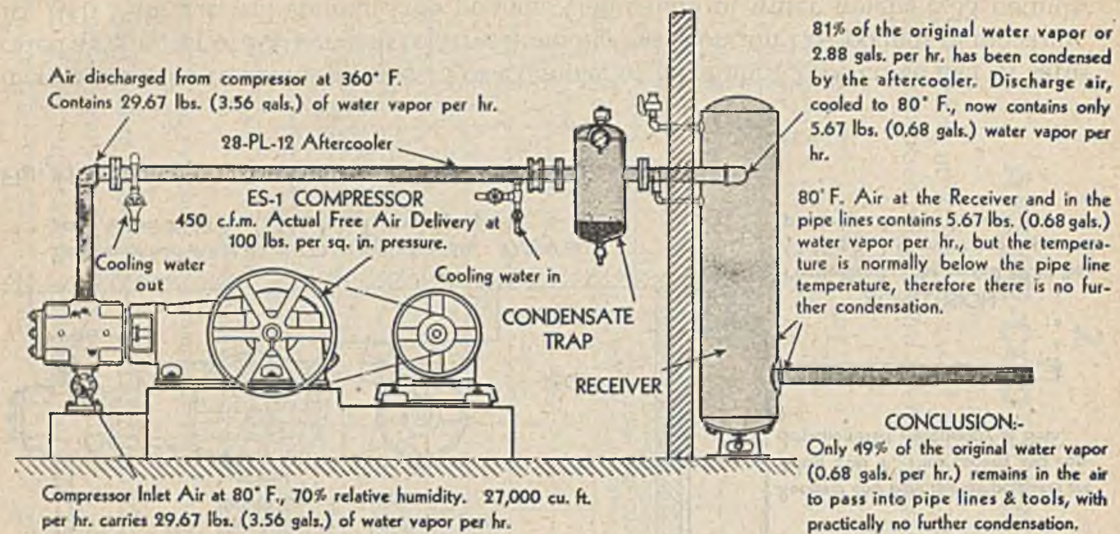


Fig. 12.—Example 2. Compressor installation with after-cooler.

From the foregoing, it is evident that great quantities of water are taken into compressors. At the higher temperature developed during compression, the air or gas that leaves the compressor can carry this moisture in the form of vapor, despite the reduction in volume. However, after this compressed air or gas becomes cooled in the pipe line to atmospheric temperature it will hold only a small percentage of this moisture in the form of water vapor, the balance appearing in the liquid state.

Example.—Saturated air at 80° at compressor intake (0 lb. gauge) contains 1.57 lbs. of moisture per 1000 cu. ft. Compressed to 100 lbs. and cooled to 80° with 65° water, the air holds only 0.20 lbs. per 1000 cu. ft. or 13% of the water vapor originally taken into the compressor. The rest has been condensed in the inter-cooler (if used) and the after-cooler. On this basis for a compressor of 1000 cu. ft. per min. capacity nearly 10 gallons of water per hour will be condensed. The advantages of after cooling are therefore evident.

Air cooled to within 15° Fahr. of incoming cooling water temperature is considered good practice. Any attempt to secure a lower air temperature (with the same water temperature) requires a much larger and more expensive cooler, and increases water consumption. These disadvantages more than offset the gain in extra condensation.

For example, note that 87% of the original moisture has been removed by cooling the air back to 80° Fahr. If the air had been cooled to 70° Fahr. instead of 80° Fahr., about 92% of the original moisture would condense, a gain of only 5% in the total condensate.

After-Cooler Construction.—When water is used as the cooling medium, air may flow:

1. Through the tubes, or
2. Outside the tubes

Fig. 13 shows a cooler known as a pipe line after-cooler in which the air passes through numerous tubes, being divided into thin streams.

In this small diameter long length tubes of proper velocities are developed for high heat transfer to the surrounding water. At the right end is seen the separator which is an important part. Fig. 14 shows a two-stage after-cooler in which the cooling water flows through the tubes and the air outside the tubes.

In this two-stage cooling warm inlet air or gas in the first stage shell is cooled by water discharged from the second stage. This permits final cooling in the second stage tube nest which is supplied with cold inlet water.

Ques. How are pipe lines affected by moisture?

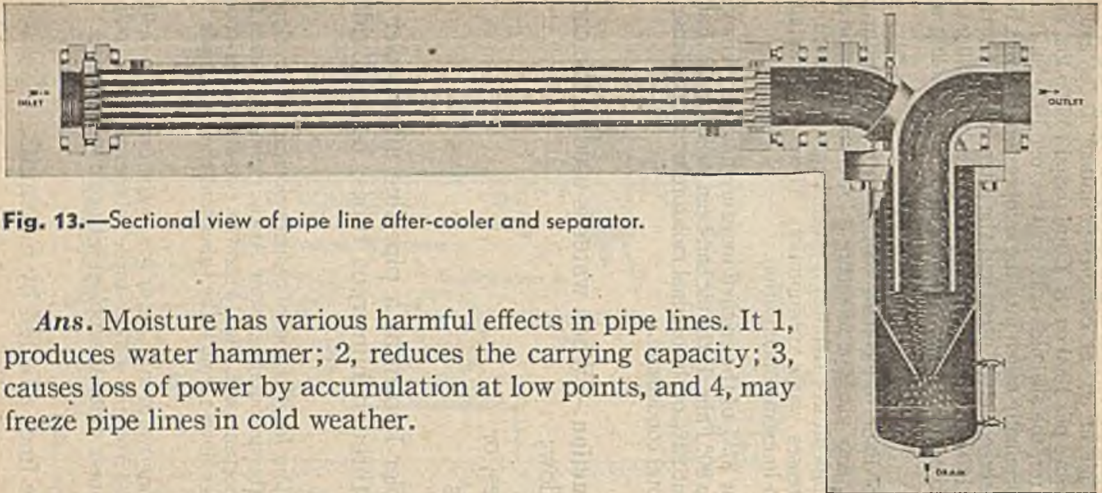


Fig. 13.—Sectional view of pipe line after-cooler and separator.

Ans. Moisture has various harmful effects in pipe lines. It 1, produces water hammer; 2, reduces the carrying capacity; 3, causes loss of power by accumulation at low points, and 4, may freeze pipe lines in cold weather.

Ques. What bad effect is encountered in pipe lines in the absence of an after-cooler?

Ans. Where hot air is passed directly from the compressor into pipe lines, the heat causes the lines to expand; this is followed when the compressor is shut down, by contraction. The repetition of this cycle daily induces leaky joints and consequent air losses even when expansion joints are employed.

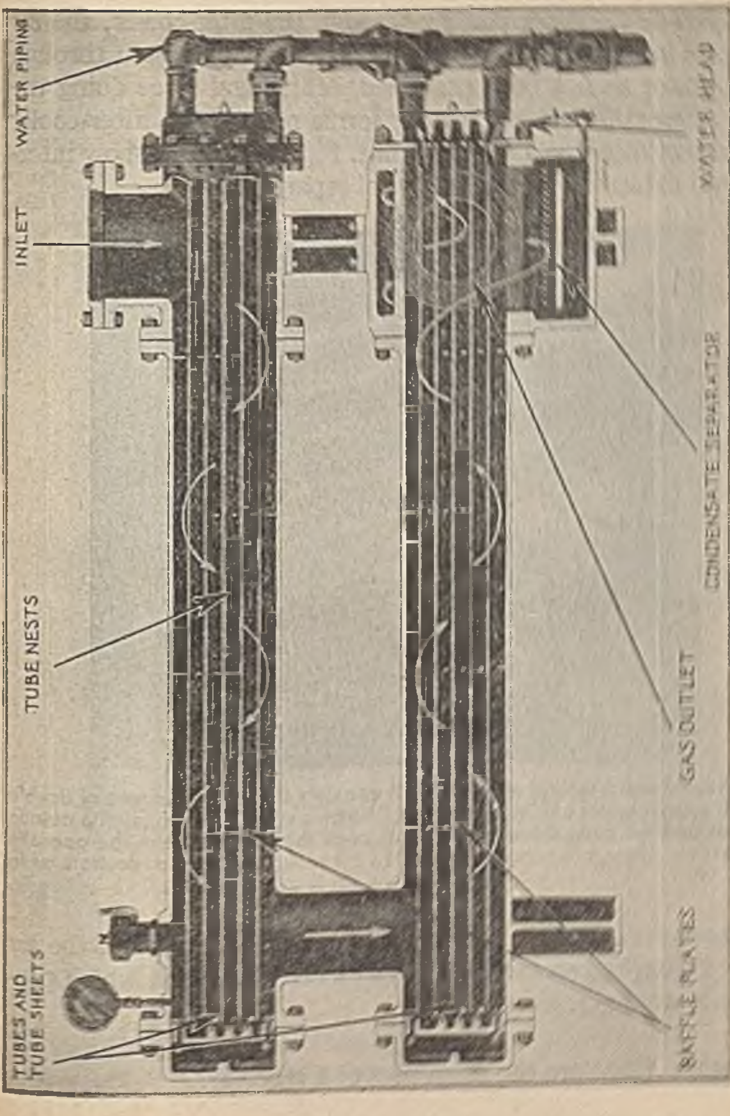


Fig. 74.—Two stage after-cooler.

Separators.—After passing through the after-cooler, the air must be freed from moisture. This is done by passing it through a separator as has been explained, centrifugal force doing the work. Sometimes the separator forms part of the after-cooler and sometimes it is a separate unit. Figs. 17 and 18 show internal and external views of a typical separator.

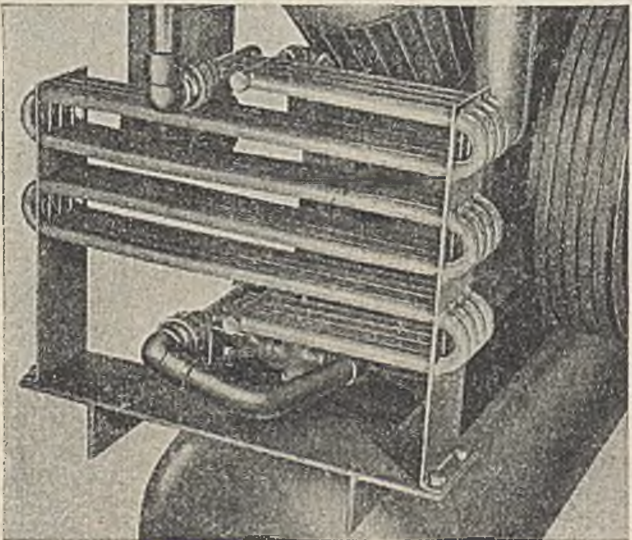


Fig. 15.—Exterior view of after-cooler. **It consists of** an arrangement of double coil, copper tubes, one inside another. Water is circulated through the outside tube, and the compressed air passes through the center tube in the opposite direction. These after-coolers are suited for tank mounted compressors as in fig. 16.

Receivers.—After compressing to the final pressure, the air passes from the separator into the receiver or tank which provides a storage reservoir.

Ques. What are the main uses of a receiver?

Ans. To reduce pulsation and provide adequate air storage capacity.

They do, however, provide additional advantages in the supplementary values of cooling the air and separating the moisture and lubricant from the air.

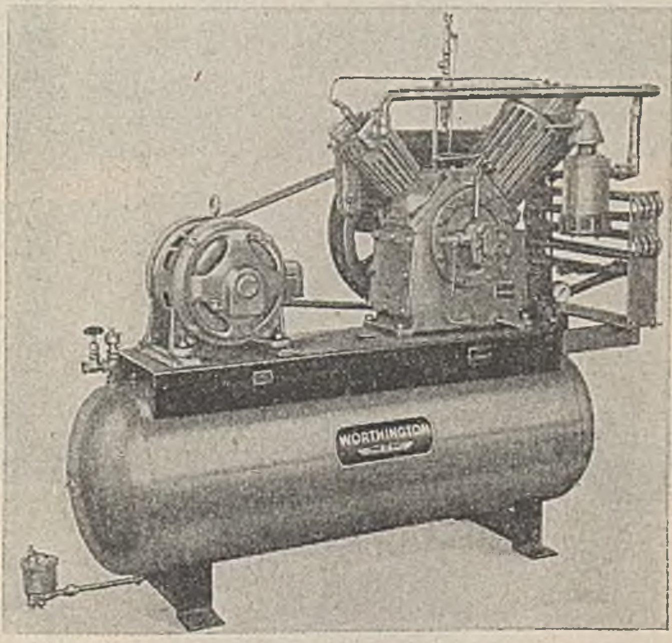
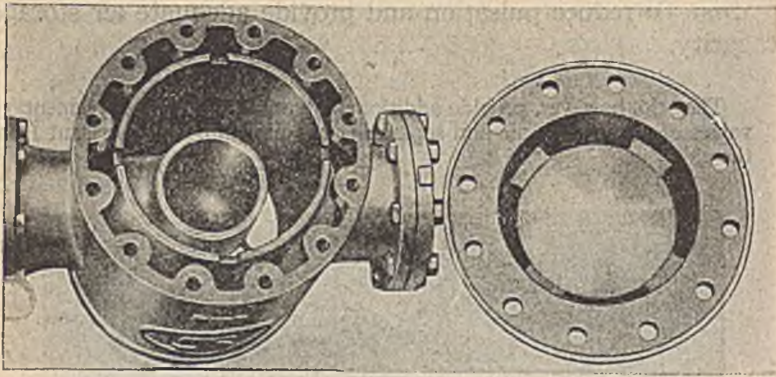
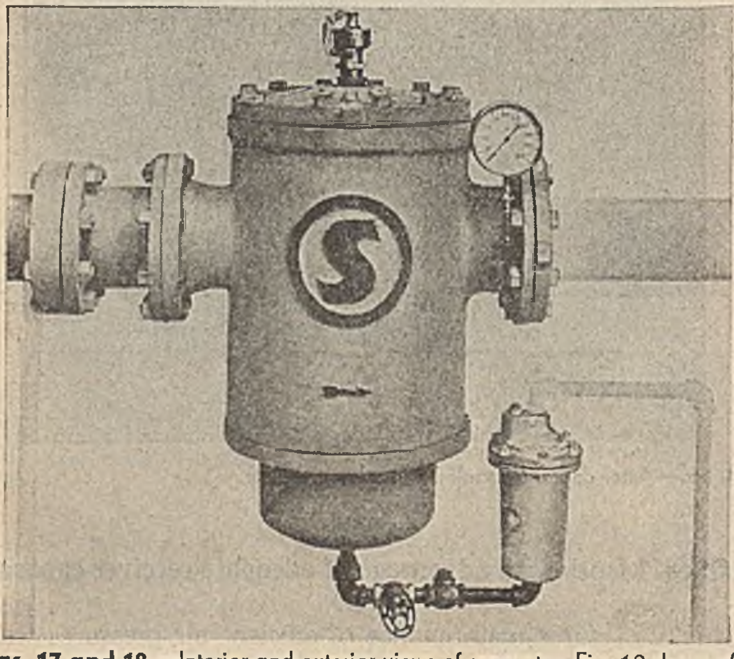


Fig. 16.—After-cooler on tank-mounted compressor.

Ques. Mention an advantage of adequate receiver capacity.

Ans. It insures maintenance of efficient air pressures when momentary demands are excessive.





Figs. 17 and 18.—Interior and exterior views of separator. Fig. 18 shows after-cooler with pressure gauge, cooling water and condensate drain.

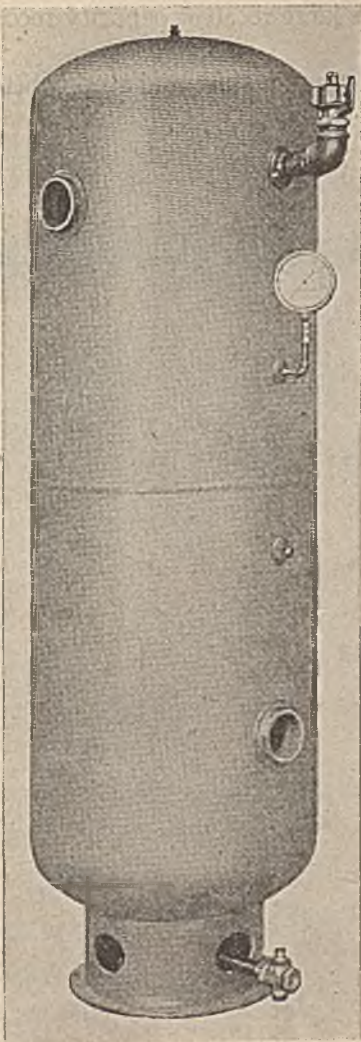


Fig. 19.—Typical receiver.

Ques. When is large receiver capacity recommended?

Ans. When automatic start and stop control is used on the compressor.

CHAPTER 50

Regulating Devices

In the many applications of compressors a fundamental requirement is to maintain a constant pre-determined pressure in the receiver, regardless of demand variations.

Numerous methods of control have been developed to meet this requirement. They may be classed:

1. With respect to speed
   - *a.* Constant
   - *b.* Variable
   - *c.* Start and stop

2. With respect to the steam end (on steam driven compressors)
   - *a.* Throttling governor with non-adjustable cut-off
   - *b.* Throttling governor with hand adjustable cut-off
   - *c.* Variable cut-off governor

3. With respect to the air end
   - *a.* Free air unloader
   - *b.* By-pass unloader
   - *c.* Variable clearance
   - *d.* Dual

Constant Speed Governor Control.—Although the usual function of a governor on a steam engine is to maintain a constant speed, but as applied to a compressor it functions to maintain a pre-determined constant pressure by varying the speed of the machine. This may be done automatically,

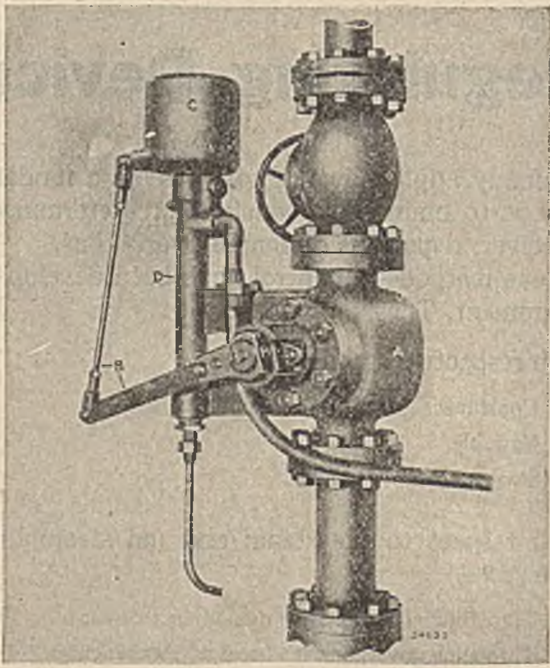


Fig. 1.—Steam throttling governor.

1. By throttling the steam supply.
2. By varying the cut-off.

Steam pressures in any steam plant vary more or less and accordingly this of itself will vary the speed of the unit. Therefore, a second and no less important function of regulation is to maintain the speed of the compressor at a predetermined rate regardless of fluctuations in steam pressures.

Throttling Governor Control.—A throttling governor with non-adjustable cut-off valve is used on the smaller machines because of the small capacities and low power involved.

Also where economy is of secondary consideration, or for other reasons, a throttling governor with hand adjusted cut off is suitable for large size units. A governor of this type is shown in fig. 1. As explained the governor controls the speed of the machine by throttling the steam supply.

In construction, referring to the illustration, the valve A itself is of the sleeve type which is rotated by the governor weight linkage B. A fine graduation of steam control is obtained by the V shaped ports in this sleeve valve.

Movement of the governor weight C, is accomplished by varying the pressure of the oil under the plunger D. Speed changes to meet varying demands on the compressor output or compensation for varying steam pressure, is accomplished exactly as in the cut-off governor, except that the governor throttles the steam instead of adjusting the cut-off.

To obtain maximum economy for average conditions with this governor, the valve gear cut-off adjustment can be altered manually while the machine is running.

The safety stop is built into the throttle valve body. It is of the same type as the throttle valve and rotates within it. However, its operation is entirely independent of the speed controlling throttle valve.

A fly ball type governor is used on large units.

Variable Cut-Off Governor Control.—The type valve gear used with this method of control is the variable lap, or Meyer riding cut-off. See Chapter 47. On page 154 is shown a Meyer riding cut off gear of the piston valve type.

The principles of the variable lap gear are fully explained in pages 155 to 157. On units equipped with this automatic cut off the rotation of the cut off valve stem, which changes the cut off and thus varies the speed of the compressor, is accomplished, as shown in fig. 2, by the raising or lowering of the rack and weight B, which is a part of the governor.

Movement of this weight is accomplished by a change in the pressure of the oil under the plunger. The amount of this pressure variation is a resultant of two separate methods of regulation acting simultaneously.

The oil is supplied by means of a pump A, chain driven from the main shaft, which produces a flow of oil under pressure acting against the governor plunger.

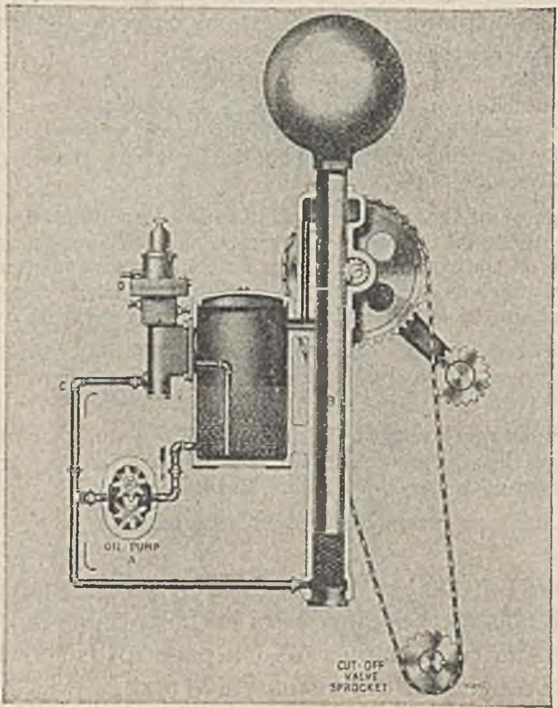


Fig. 2.—Steam cut off governor.

Any variation in steam pressure tending to change the speed of the compressor is at once reflected in an increase or decrease in this oil pressure, causing a movement of the weight and rack which rotates the throttle or cut-off valve rod in such a way as to restore the required speed.

Variations in the demand for air or gas also affect this oil pressure by means of a diaphragm operated valve D, which by-passes a portion of the flow of oil. The diaphragm is connected to receiver pressure.

The change in oil pressure, which is affected by these two distinct methods of regulation acting in unison, control the speed of the unit according to the load demand and compensates for varying steam conditions.

The governor is connected with the riding cut-off valve stem by chain transmission as shown in fig. 3.

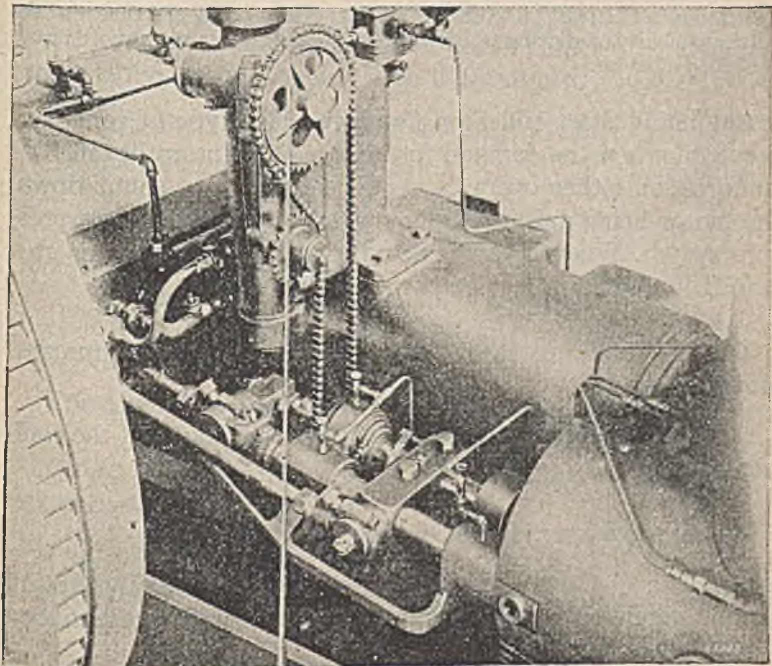


Fig. 3.—Connecting transmission or drive between riding cut off valve steam and governor. As seen it is a chain drive.

Ques. How does the governor function to facilitate starting?

Ans. It is necessary to lengthen the cut off and thus obtain the necessary torque for starting. It automatically adjusts the cut off of the steam valve for starting, then returns it to the proper operating position as soon as the machine is up to speed.

Ques. What provision is made to prevent over speeding?

Ans. In the event of accidental damage to the governor over speeding is prevented by an independent safety stop valve.

This must be reset each time the machine is shut down, and thus removes the possibility of sticking which might be caused by infrequent need or use of such a device.

Automatic Start and Stop Control.—This type of control is provided when the demand for air or gas is intermittent. The compression either operates at full capacity or is shut down, the motor being automatically started and stopped by a pressure switch. When no air is being used the compressor is at a standstill and consumes no power.

Ques. What important device operates as the compressor slows down?

Ans. A regulator unloads the cylinder and shuts off the cooling water.

Ques. How long is this condition maintained?

Ans. Until the motor is started and again running at full speed. This results in a saving in electric current and cooling water.

Dual Control—This system is a combination of constant speed and automatic start and stop control. It provides constant speed control while there is a steady and large demand for air or gas and permits switching to automatic start and stop control during periods of intermittent or small demand.

Free Air Unloaders.—By definition, an unloader is *an automatic device that varies the amount of air or gas being pumped.* If the pressure in the receiver build up, due to the compressor supplying air when none is being used, a relief valve must blow, thereby resulting in a loss of air, and consequently, of power.

Several methods of unloading the compressor as commonly used are employed, so that when completely unloaded, no air is delivered to the receiver, practically no work is done in the cylinders, and consequently, only power sufficient to overcome the friction of the machine itself is consumed.

The methods of unloading commonly used are:

1. Inlet line unloading valve.
2. Inlet and discharge unloader.
3. Inlet valve kept closed.
4. Inlet valve kept open.
5. Discharge valve kept open.
6. Adjustable compression stroke.
7. Variable clearance volume.

These methods accomplish the same thing in different ways.

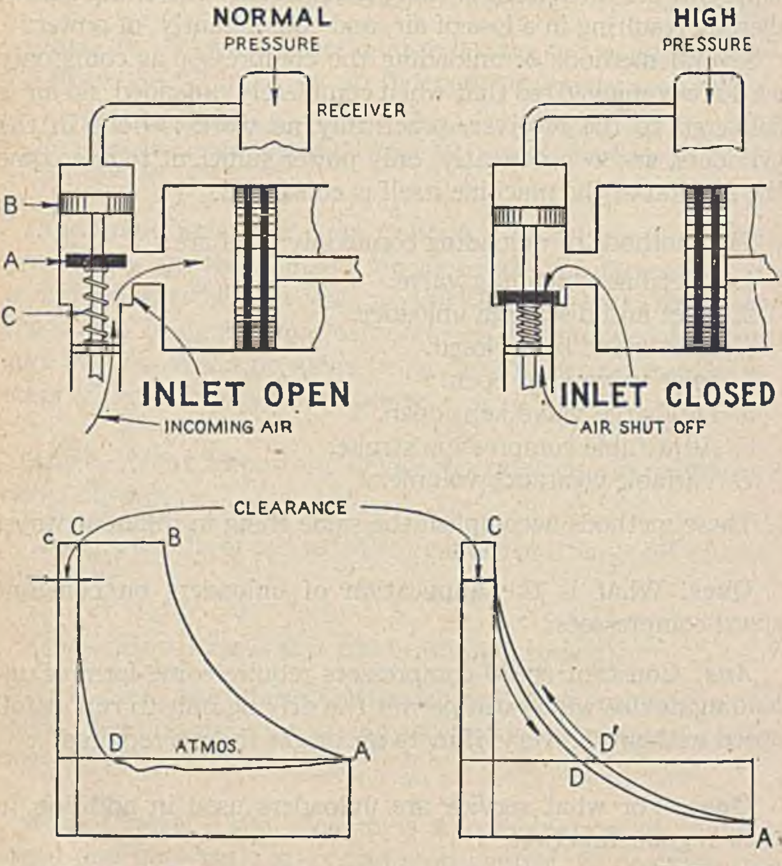
Ques. What is the application of unloaders on constant speed compressors?

Ans. Constant speed compressors require some form of unloading device which will permit the driving unit to run at full speed without delivery of more air or gas than is required.

Ques. For what service are unloaders used in addition to their regular function?

Ans. Unloaders are used also to facilitate starting a compressor, permitting temporary operation without doing the full amount of work at the start.

Unloader Operation



Figs. 4 and 5.—Elementary free air inlet unloader showing operation of the automatic control valve. Fig. 4, inlet open; fig. 5, inlet closed.

Figs. 6 and 7.—Diagrams corresponding to operation in figs. 4 and 5 as explained in the text.

Ques. With what kind of drive is this especially important and why?

Ans. It is especially important with certain types of electric motors that fail to develop full torque until full speed is reached.

Inlet Line Unloader.—This type of unloader *automatically opens and closes the inlet line under pressure variations* in the receivers.

The elementary drawings, figs. 4 and 5, show the basic operating principle of air inlet line unloader. The essential elements of the unloader is valve A, fig. 4, which admits or shuts off free air to the cylinder. On the same spindle is a piston B, which is acted upon on one side by the air pressure from the receiver and resisted by a spring C.

In operation when the pressure is low in the receiver the spring overcomes the air load on the piston and forces the valve open. With the inlet line open free air is drawn into the cylinder and the compressor pumps up the pressure in the receiver. When the pressure reaches a pre-determined point, it, in acting upon the piston pushes the valve shut which shuts off the air supply and accordingly no more air is pumped into the receiver until the pressure in the receiver drops below the pre-determined working pressure.

The diagrams, figs. 6 and 7, will help to illustrate the operating cycle, showing the different distribution with valve open and shut. The diagram, fig. 6, shows normal operation with the unloader valve open. Here a charge of air has been taken in at A, compression takes place from A to B, BC represents discharge at constant pressure into the receiver. Note at the end of the stroke the clearance space cC, is filled with air at the discharge pressure.

On the admission stroke the clearance air quickly expands to atmospheric pressure as indicated by CD, and at that point the compressor inlet valve opens and admission takes place from D to A.

Now, when the air pressure in the receiver rises to the predetermined point the unloader valve is moved shut excluding entry of any more air to the cylinder.

With air in the clearance space at receiver pressure, it expands during the stroke as indicated by the curve CDA, fig. 7. That is, the expansion passes the atmospheric line at D, and continues to a low pressure (but not zero pressure) which is reached at A. It will be noted that the compression curve AD′C is higher than the expansion curve CDA. This is because the heat of compression elevates the compression curve AD′C and heat transfer and more or less leakage causes the expansion curve CDA to lie below.

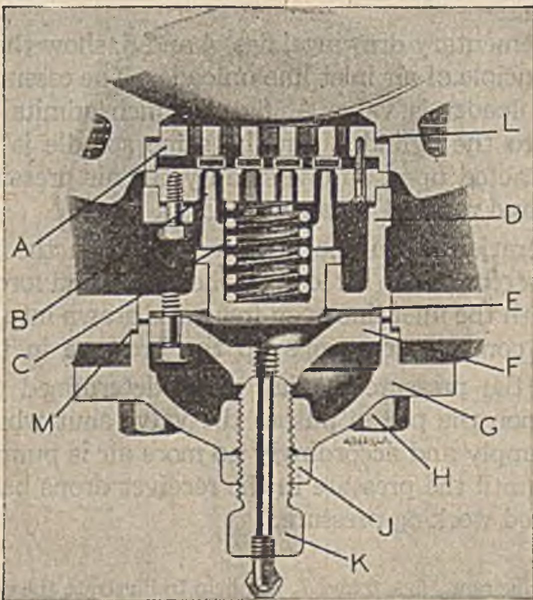


Fig. 8—Channel valve free air unloader assembled in cylinder. The free air unloader assembly includes the inlet valve complete. The inlet valve complete is bolted to the unloader body and can easily be removed for inspection or replacement after the unloader assembly has been removed from the cylinder. When re-assembling a valve to the unloader where used, be sure the valve center bolt nut is locked, and that the pusher B, and spring C, are in place. If it become necessary to replace a spring C, always use one of the same size and having the same compression as the one originally furnished in order to insure proper operation of the unloader.

Ques. How does the actual unloader differ from the elementary device just described?

Ans. In the actual unloader there is a small piston which is actuated by the pressure in the receiver against the resistance of a spring. When the pressure in the receiver becomes too

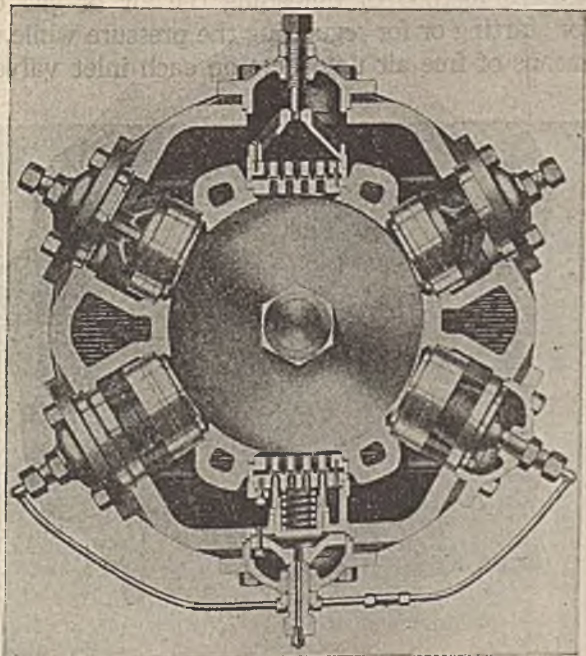


Fig. 9.—Cross section of compressor cylinder showing the arrangement of channel valves and free air unloaders.

high, the piston is moved sufficiently to open a small port which admits air to a plunger valve of larger area. This plunger valve is forced against its seat, thus preventing further admission of free air to the cylinder until the pressure in the receiver drops below the maximum pressure for which the mechanism is set.

The compressor being unable to draw in any air, does no work in the compressor cylinder, and hence delivers no air to the next stage or to the receiver.

A standard method of unloading the air or gas cylinder, either for starting or for regulating the pressure while running, is, by means of free air unloaders on each inlet valve. Fig. 8

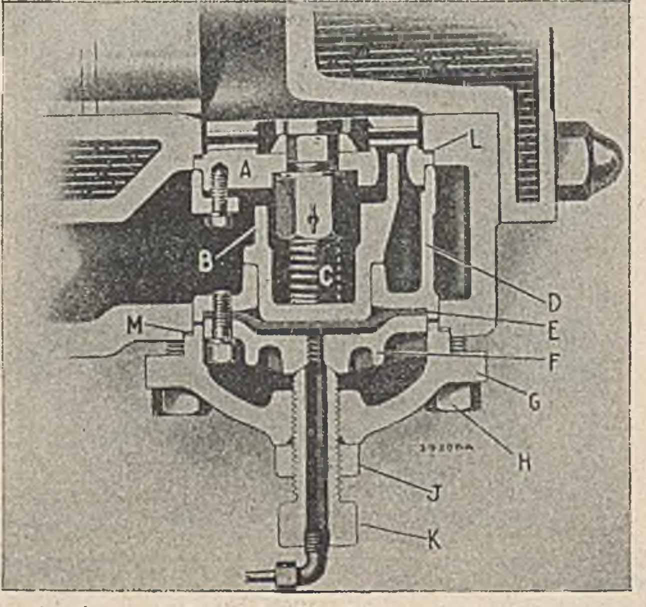


Fig. 10.—Another type of free air unloader.

shows the construction of a channel valve free air unloader assembled in a cylinder. The unloaders are controlled by the regulators described later.

The construction of free air unloaders is such that when air is admitted to the under side of the diaphragm E, through the hollow inlet cover set screw K, the fingers on the valve pusher, passing through the valve seat ports, raise the inlet valve off its seat until the pressure under the diaphragm is released.

When the unloaders hold the inlet valves off their seats, the air or gas cylinder is vented to the inlet passage, the air or gas passing alternately in and out of the cylinder without being compressed.

Ques. What happens when the receiver pressure rises to a value for which the auxiliary valve has been set?

Ans. The auxiliary valve functions to admit live air or gas to the free air unloaders.

Ques. What is the effect of this?

Ans. The valve cover G has a machined recess into which fits the inner cover F, of the unloader assembly. Thus when the unloader assembly is being put into the bottom of a cylinder, unscrew valve cover set screw K, and set the whole assembly in the outer cover G, so that inner cover F fits into outer cover. Place valve seat gasket L, on the valve seat and cover gasket M, on the cover, and carefully raise the outer cover G, with the unloader assembly into position. After the valve seat and cover find their fits in the cylinder, insert and tighten the valve cover tap bolts. Now screw up on the valve cover set screw K, forcing the valve seat A, solidly onto its gasket, and then lock the set screw in position by means of the lock nut J.

Free Air Unloaders Controlled by Auxiliary Valve.—This type of regulation is standard where the compressor is driven at constant speed from any source of power, and where the demand for air or gas is fairly constant. Since the compressor runs at constant speed it is necessary to alter the output of the compressor cylinder when the demand for air changes. This is accomplished by using free air unloaders on the inlet valves so that the cylinder will operate either at full capacity or at no capacity. These free air unloaders are controlled by the auxiliary valve, which is mounted on the cylinder as shown in fig. 11.

The auxiliary valve is connected to the unloaders by means of copper or brass tubing with suitable connectors. A plug valve is furnished in this piping, as shown, so that the operator can throttle the air or gas to the regulators and quiet their operation. Do not close the valve too far because the unloaders will fail to operate and the receiver pressure may rise to an unsafe value.

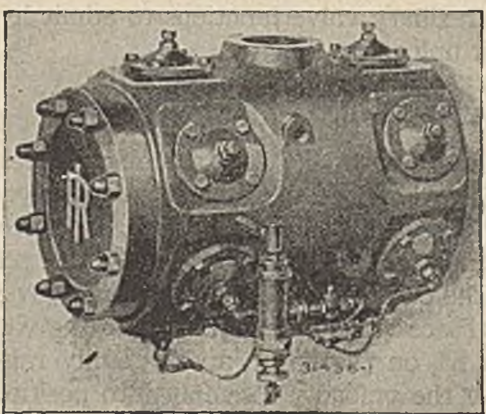


Fig. 11.—Auxiliary valve mounted on cylinder.

A separate pipe of ½ in. pipe size or preferably larger should connect the receiver to the top of the auxiliary valve. This pipe should have no low spots or pockets in which condensed water vapor or oil might collect and as few turns as possible.

Always use pipe free from scale and blow out all piping and tubing before connecting to regulators and unloaders.

Ques. What happens when the receiver pressure rises to a value for which the auxiliary valve has been set?

Ans. The auxiliary valve functions to admit live air or gas to the free air unloaders.

Ques. What is the effect of this?

Ans. The free air unloaders push all the inlet valves off their seats, allowing air or gas to pass in and out*through the inlet valves while the compressor piston continues to move back and forth.

After a continued use of the air or gas reduces the receiver pressure to a predetermined value, the auxiliary valve exhausts the unloaders, thereby allowing the inlet valve to function normally and the compressor to operate at maximum capacity. Thus, the compressor unloads and reloads to maintain the receiver pressure within the pressure range for which the auxiliary valve has been set.

Ques. How is the compressor unloaded for starting when there is pressure in the receiver?

Ans. Screw up on the wing nut at the bottom of the auxiliary valve.

Ques. What is the effect of this adjustment?

Ans. Air will be admitted to the free air unloader and thereby vent the compressor cylinder to the intake passage.

Ques. What should be done after the compressor attains full speed?

Ans. Unscrew the wing nut and the compressor will begin to operate in the normal manner.

Ques. What is used in place of the wing nut on compressors handling gas?

Ans. A three way valve in the line between the auxiliary

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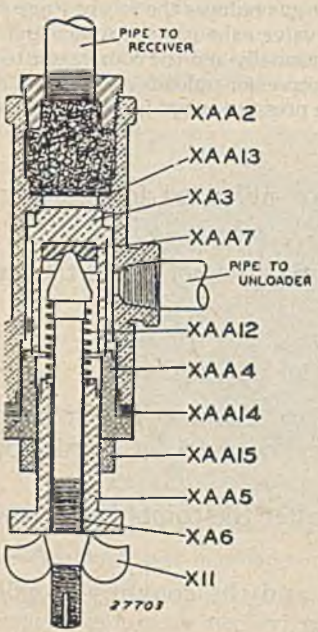
*NOTE.—If the belt slap and the idler pulley jump when the compressor slows down to a stop, unload the compressor by screwing up on the auxiliary valve wing nut before shutting off the motor or other form of drive. Do this as a matter of routine in stopping the compressor.

valve and the unloaders with one connection of the three way valve period to receiver pressure.

Auxiliary Valve.—This valve is used as a controlling valve for operating the main regulating devices of the compressor. It operates automatically with the variations of receiver pressure.



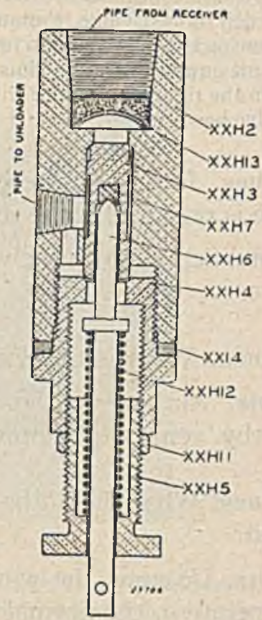


Fig. 12.—Type XAA auxiliary valve.
Fig. 13.—Type XXH auxiliary valve.

It is important that the air or gas supply for the auxiliary valve should come from the receiver and not from the delivery pipe of compressor, as pulsations in the latter even at some distance from compressor, may cause the auxiliary valve to work unsatisfactorily.

There should be no low spots or pockets in the pipe connections to and from the auxiliary valve, in which condensed water vapor or oil might collect.

This line should be ½ in. piping, or preferably larger, and should have as few turns as possible. The pipe should be as free from scale as can be obtained. Before connecting to the auxiliary valve and regulators it should be rapped with a hammer to loosen scale, and then thoroughly blown out with a good pressure of air.

Duplicate Part List for XAA and XXH Auxiliary Valves

Part No.	Name of Part	Part No.
XAA1	**Auxiliary Valve Complete** (includes Parts 2 to 15)	XXH1
XAA2	Auxiliary Valve Body	XXH2
XA3	Auxiliary Valve Bare	XXH3
XAA4	Auxiliary Valve Lower Valve Seat	XXH4
XAA5	Auxiliary Valve Spring Adjuster	XXH5
XA6	Auxiliary Valve Spring Guide	XXH6
XAA7	Auxiliary Valve Spring Guide Socket	XXH7
X11	Auxiliary Valve Hand Unloading Wing Nut	
XAA12	Auxiliary Valve Spring	XXH12
XAA13	Auxiliary Valve Filter Disc	XXH13
XAA14	Auxiliary Valve Range Shims (set)	XXH14
XAA15	Auxiliary Valve Spring Adjuster Lock Nut	XXH11

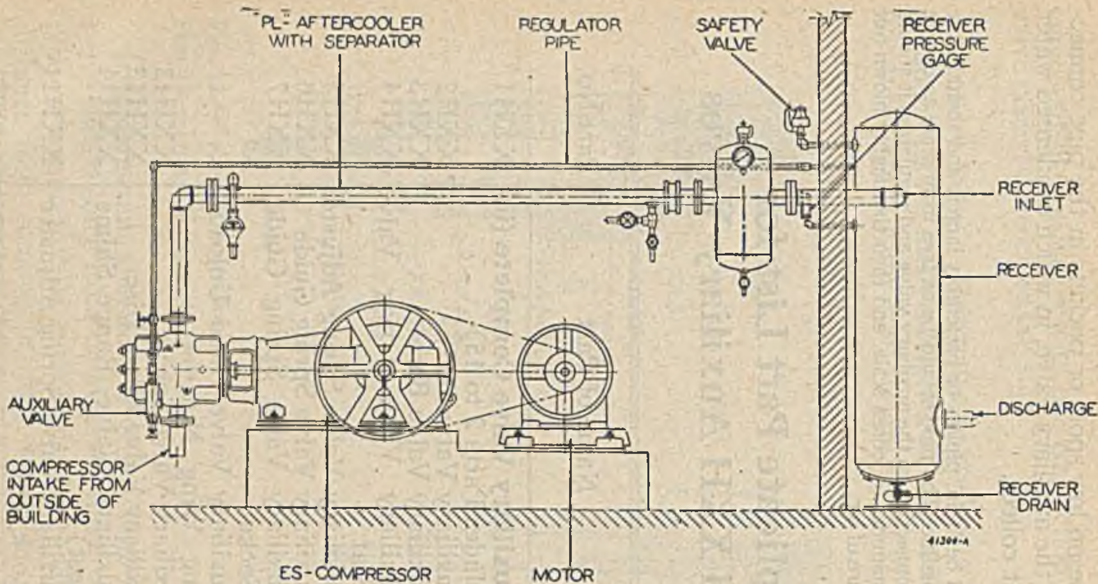


Fig. 14.—Diagram showing the piping arrangement for constant speed regulation.

Operation of Auxiliary Valve.—Figs. 12 and 13 show the type auxiliary valve here considered. The upper part containing woolen yarn filled filter is connected to the receiver (never to the discharge pipe) by means of a ½ in. or preferably larger pipe.

When the air or gas within air receiver reaches the pressure at which auxiliary valve is set to operate, the pressure overcomes the resistance of spring XAA12, and forces valve XA3 from its seat. As this valve opens, the increased area of the flange just below the upper valve seat becomes exposed to air or gas pressure and the downward force is greatly increased. This causes the valve to jump open quickly until stopped by the contact of its lower end with the adjustable valve seat XAA4

The flange is slightly smaller in diameter than the bore of the valve chamber and hence permits free passage of the air or gas by it and out through the side pipe to unloaders.

When the receiver pressure has fallen a predetermined amount, spring XAA12 throws valve XA3 back to its upper seat, shutting off the pressure from the unloader or unloaders. This permits the air in the unloaders to flow back through the side pipe, past lower auxiliary valve seat XAA4, and exhaust to atmosphere through the opening in the lower end of spring adjuster XAA5. In case of gas, the exhaust should be piped back to intake.

Adjustment of Auxiliary Valve.—The pressure at which auxiliary valve operates is adjusted by varying the tension of spring XAA12, by means of spring adjuster XAA5. The range of pressure between unloading and reloading points is adjusted by giving valve XA3, less travel to obtain a longer range, and more travel to obtain a closer range. The valve travel can be adjusted by removing or adding shims at XAA14. The lower valve seat XAA4 must be removed for this purpose. Turning the lower valve seat XAA4, only a fraction of a complete revolution makes an appreciable change in range; consequently this adjustment must be made very carefully.

When tested in the shop, the auxiliary valve is usually adjusted for a range of about 10% of the pressure at which the auxiliary valve is set. It is not advisable to cut the range below this amount, less required because of unusual conditions, as the pulsations in receiver are liable to cause the valve to work with less certainty when the range is very short.

On air compressors a hand unloading nut XII provides for unloading the compressor at will, as in starting the compressor when there is pressure on the line.

Setting up nut XII, overcomes the resistance of spring XAA12, and admits pressure to the unloaders. When nut XII is released the compressor takes up its load and the auxiliary valve will then automatically operate at the pressure for which it is set.

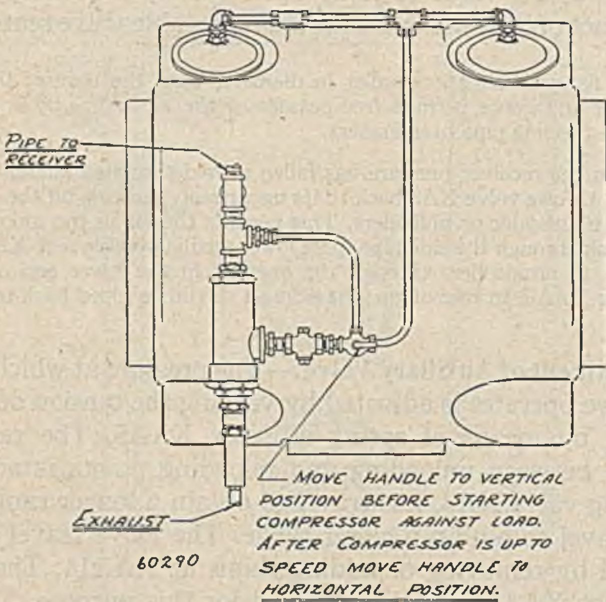


Fig. 15.—Arrangement of regulator piping for gas compressors with by-pass for unloading when starting.

The filter, which is above the upper valve seat, is filled with woolen yarn or a special sponge. This prevents any particles of dust or grit from being carried into the valve chamber, and removes the gummy deposit which sometimes comes from oil used in compressor cylinders.

Should the auxiliary valve fail to work properly at any time, take it apart and clean it thoroughly, putting fresh yarn into the filter. Use only genuine wool. Cotton will pack, and stop the air flow.

If the pipe from the receiver to the auxiliary valve be long, it may cause uncertain action and chatter. In this case, enlarge the pipe.

A small receiver placed in the regulator line just above the auxiliary valve will be an additional help. This receiver should be made of three-inch pipe and should be about 14 ins. long.

Variable Clearance Regulation.—In this method a number of clearance pockets are thrown into communication with the

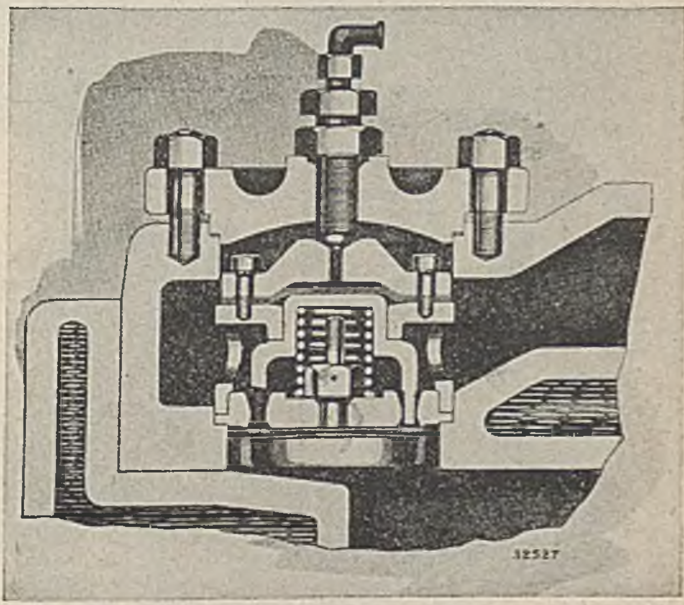


Fig. 16.—Free air unloading valve for 8 and 10 inch stroke compressors. The regulator controls the free air unloading valves holding open the inlet valves during the unloaded period. The single stage compressors have either three or five step control, while the two stage compressors have the three step control only. The free air inlet valves always operate in pairs; **i.e.** frame inlet valves and then the outer inlet valves, maintaining equalized load conditions on each side of the compressor. When the inlet valves are held open, the compressing stroke of the piston forces the air back through the inlet valves and passages.

ends of each air cylinder when the pressure in the receiver has reached the required pressure. When the piston returns, the air in the clearance pockets expands and partly fills the cylinder, thus preventing the opening of the automatic inlet valves until later in the admission stroke. When no air is used from the receiver, air from the clearance pockets may completely fill the cylinder and prevent the opening of the inlet valves, admitting no free air.

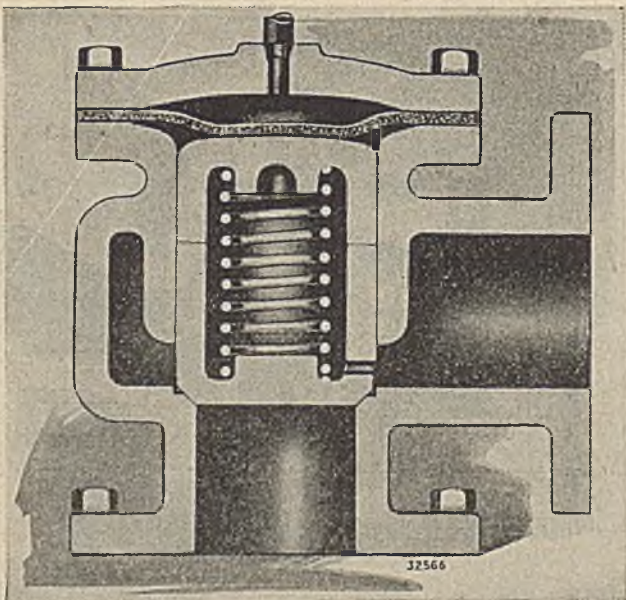
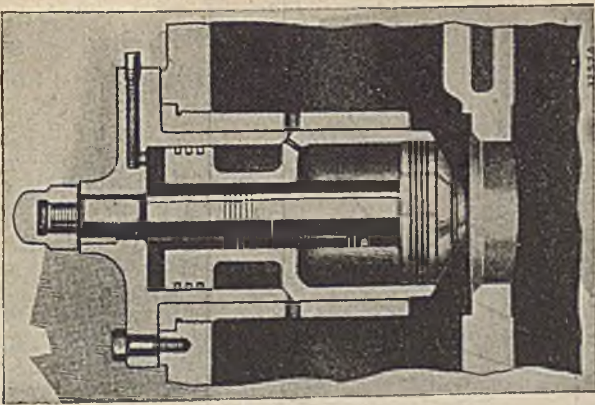


Fig. 17.—This type of low pressure unloader provides an exhaust to atmosphere to unload the inter-cooler and low pressure cylinder.

The pressure volume diagram, fig. 20, shows the effect of variable clearance volume upon the amount of air compressed. Curve 1 indicates compression adiabatically without any clearance. All the air compressed is discharged and there is no



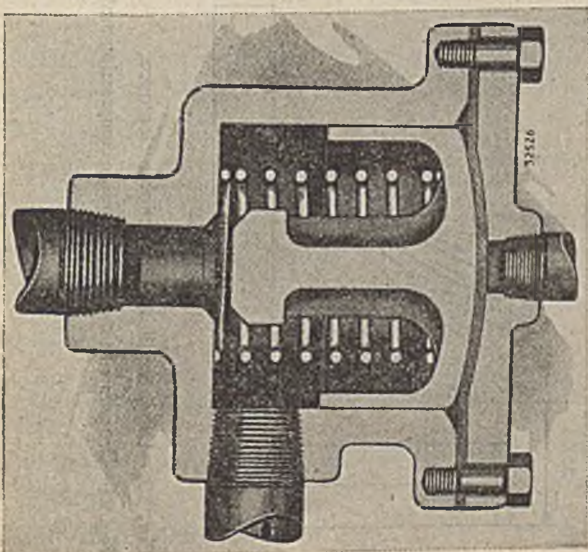
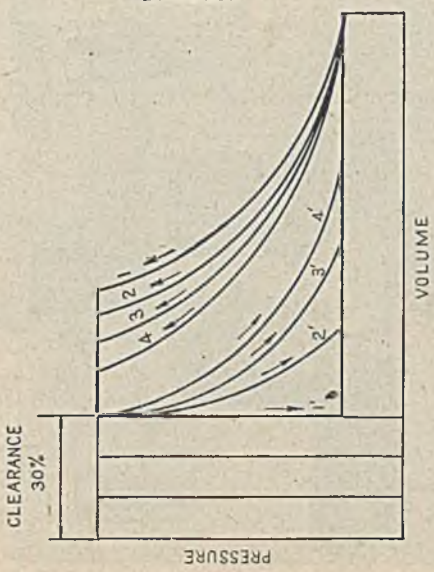


Fig. 18.—High pressure cylinder relief valve which unloads the high pressure cylinder.

Fig. 19.—Balanced clearance valve used on 12 and 14 inch stroke compressors. These valves connect the cylinder to the clearance pockets. The clearance valves are controlled by the regulator to reduce the capacity of the compressor when the demand for air falls off. The valves are operated by compressed air are cushioned by air when opening and closing, and are balanced at all times.



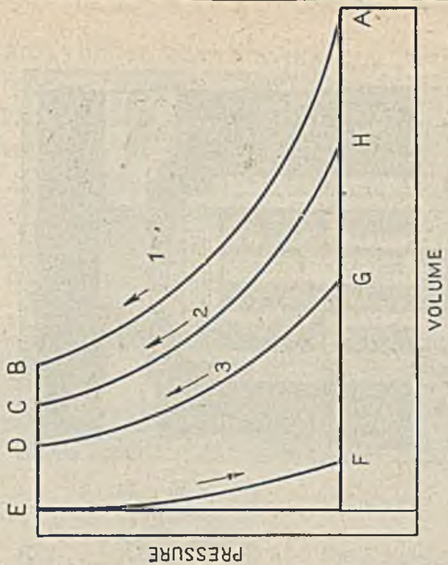


Fig. 20.—Diagrams illustrating variable clearance.

Fig. 21—Diagrams illustrating adjustable compression stroke method of regulation.

re-expansion line other than the vertical line 1′ indicating a direct drop to atmospheric pressure.

Curve 2 shows the compression curve with 10% clearance as do Curves 3 and 4 for 20% and 30% respectively.

With the increase in clearance volume, the expansion curve also changes, as there is more air to expand into the cylinder volume with large clearance than with small clearance. These curves are indicated by the numbers 2′, 3′ and 4′, respectively.

It is seen that in this particular diagram, with 30% clearance volume, only a very small portion of the capacity of the cylinder is permitted to enter the cylinder at each stroke, because the expansion of the clearance air fills the cylinder at atmospheric pressure to nearly three-fourths of its volume. Thus the amount of air compressed grows less as the clearance volume is increased. The clearance could be increased to such an extent that the Curve 4 for compression and Curve 4′ for expansion would coincide, thus no air would be delivered.

In this diagram the curves are for adiabatic compression. Clearance assumed at 5% is constant for all operating conditions and, therefore, the re-expansion line EF is the same for all compression curves.

The effect of holding the inlet valve open after the compression stroke has begun, is quite well defined in the curves numbered 1, 2 and 3.

Curve 1 indicates full compression stroke. Compression starting at A rises to B, and discharges from B to E.

Curve 2, intake valve held open for ¼ of stroke allows only ¾ of cylinder volume to be compressed. Starting at A, air is forced out of the cylinder from A to H, along atmospheric line.

At point H, the intake valve closes and compression rises H to C, and discharges C to E.

Curve 3 represents similar action with intake valve held open for ½ stroke. Consequently only ½ the volume is delivered to DE.

Five Step Variable Clearance Regulation.—An example of the application of variable clearance control is here given. In this method there are five approximated equal steps: full, ¾,

½, ¼, zero load, keeping the power input approximately in proportion to the amount of air delivered at the compressor discharge.

Ques. What provision is made in the cylinder to obtain variable clearance?

Ans. Each cylinder is equipped with four clearance pockets, two at each end as shown in fig. 22.

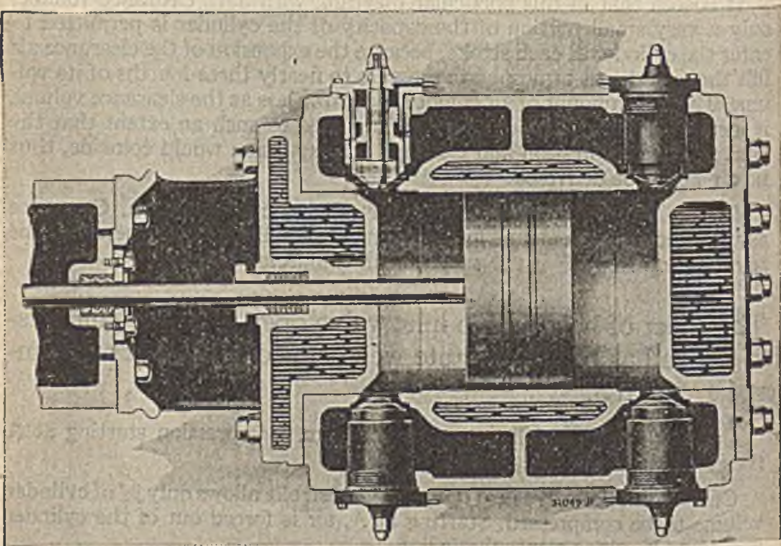


Fig. 22.—Longitudinal sectional view of cylinder showing clearance valves and pockets—two at each end of the cylinder.

For a two-stage machine, the volume of the clearance pockets in the high pressure cylinder bears the same relation to the volume of the low pressure clearance pockets as the cylinder ratio. On a single stage machine with two cylinders, the clearance pockets in both cylinders have the same volume. Each of these clearance pockets is equipped with a pressure operated valve.

The regulating device, which is automatically controlled from receiver pressure, is built with four pilot valves, which operate in succession to control the opening and closing of the clearance pocket valves in the proper sequence.

Ques. What happens if the compressor be running at full load and rated discharge pressure and the demand for air decreases?

Ans. The receiver pressure will tend to rise until at a predetermined point, the clearance regulator functions to open

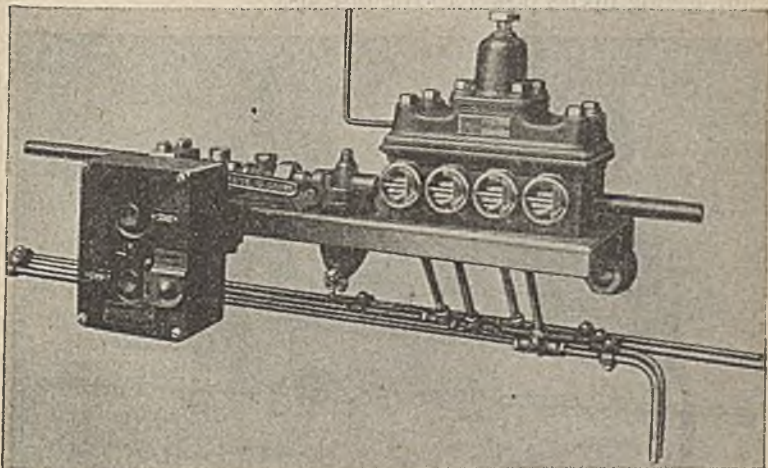


Fig. 23.—Five step regulator with push button starting station. The handle of the three-way cock shows behind the starting station. The air filter is shown next to the regulator. **In construction** a diaphragm connected to receiver pressure, operates a pilot valve which causes auxiliary valves to open and close to admit or exhaust pressure to or from the control valves in the cylinder. With five step clearance control the five step regulator shown operates all clearance valves. With three step free air regulation used on 8 in. stroke units, a similar three step regulator controls the unloaders. **In operation,** a slight rise in receiver pressure causes the first step of control to operate and reduce the compressor capacity. A further reduction in the demand for air results in a slightly higher receiver pressure and causes the next step to operate, and so on to no load, if necessary, or until the compressor output equals the demand for air or gas. The total increase in receiver pressure from full load to no load is very small. When desired, the discharge pressure may be varied by a screw adjustment on the top of the regulator.

the first set of clearance pockets, that is, one clearance pocket in each cylinder.

In this manner approximately 25% of the air being compressed will pass into the clearance pocket, cutting the capacity of the machine to 75% of its full capacity rating.

Ques. What happens on the return stroke?

Ans. The air thus trapped in the clearance pocket expands again, giving up its power to the pistons.

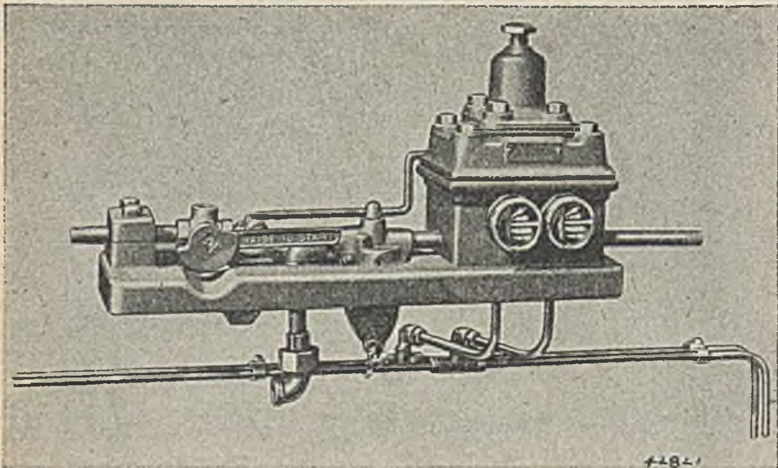
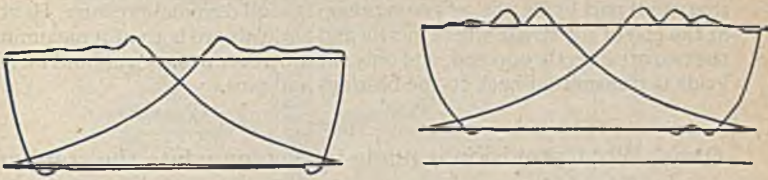


Fig. 24.—Three step regulator, with air filter and the three way cock (used for unloading during the starting period). Note the indicator provided for each step of unloading.

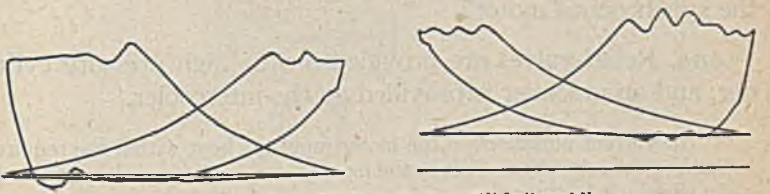
Ques. What happens if the demand for air continues to decrease?

Ans. The second set of pockets open, cutting the capacity of the compressor to 50%, etc., until, with all the pockets open, the machine is completely unloaded and no air is delivered to the receiver.

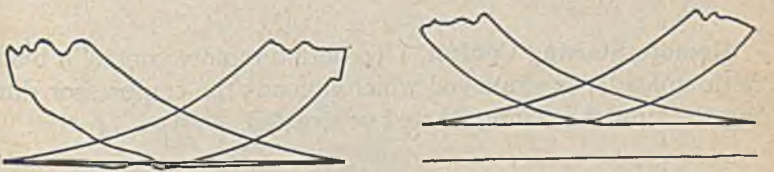
LOW-PRESSURE DIAGRAMS HIGH-PRESSURE DIAGRAMS



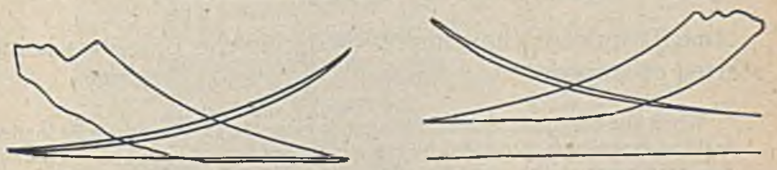
Full Load—100% Capacity; 100% Indicated Hp.



Three-Quarter Load—75% Capacity; 76 to 77% Indicated Hp.



One-Half Load—50% Capacity; 52 to 53% Indicated Hp.



One-Quarter Load—25% Capacity; 27 to 29% Indicated Hp

No Load—0% Capacity; 3 to 5% Indicated Hp.

Figs. 25 to 34.—Indicator diagrams showing operation of clearance control at five load points.

Reference to the characteristic indicator cards, figs. 25 to 34, show clearly that at all part loads, the air is compressed to full terminal pressure. Hence, at the end of the stroke where inertia and air loads are both at a maximum, the two are directly opposed, and only the difference between air and inertia loads is transmitted back to the bearings and pins.

Ques. What provision is made to accommodate the standard two stage duplex compressor to the starting characteristics of the synchronous motor?

Ans. Relief valves are provided on the high pressure cylinder, and an unloader is provided on the intercooler.

The current pulsations in the motor input are kept within the required limits by the fly wheel effect in the motor rotor, combined with a separate fly wheel when required.

Remote Starting Control.—To obtain remote control a magnetic unloader is employed which unloads the compressor during the time it is being started or stopped.

Ques. When and how does it work?

Ans. It unloads the compressor during the time it is being started or stopped.

When the magnetic unloader circuit is closed, the compressor is loaded and the capacity is controlled by the regulator. With this means of control, the compressor may be started or stopped from a push button station located, at any convenieht place.

Ques. What other important function may be performed by the magnetic unloader?

Ans. Combinations of the magnetic unloader and relay circuits may be used to automatically unload the compressor in case of line voltage drop, cooling water failure, etc.

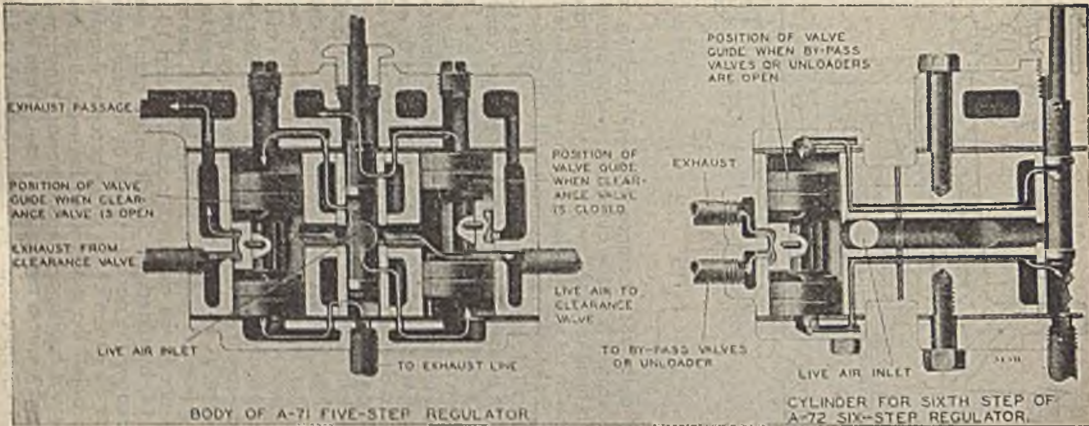


Fig. 35.—Sectional views showing operation of five step and six step clearance regulators. Air flow is indicated by arrows. Operating air pressure is admitted through a drilled hole in the regulator body to the auxiliary valve and to the middle of all valve guides. At full load the lever holds the auxiliary valve in its bottom position so that air is exhausted from the top of each valve guide and air pressure is applied to the bottom of the guide moving in upward as shown at the right hand side of the body. Seating of the valve guide against the leather gasket prevents leakage of air pressure past the guide to the exhaust ports. With the guide in the upper position, the D, regulator valve uncovers the port leading to the clearance valve pipe line, applying pressure to the clearance valve piston and closing the clearance pocket.

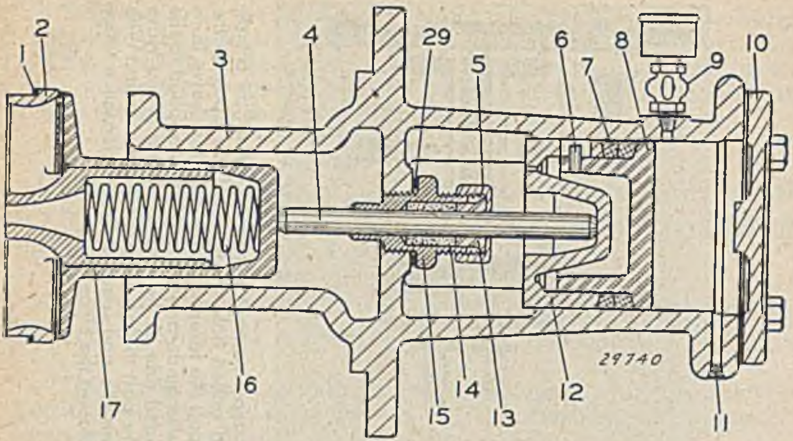


Fig. 36.—Sectional view of automatic clearance valve with locking screw and enclosed cap for gas producer. It is of the automatic, piston operated type. When special regulation is furnished, they may be of straight hand operated type or of automatic type with set screw for keeping the valve closed when so desired. The valve seat 2, fits into the inner cylinder wall with copper gasket 1 sealing the joint. The valve 17, is held on its seat by means of the piston and the valve stem 4. The piston is moved by means of air pressure controlled by the A-71 or A-72 clearance regulator. The air connection is at 11. Spring 16, opens the valve quickly when air is exhausted from the piston, prevents fluttering as the valve closes, and retains the valve seat 2, in the cylinder wall. The piston consists of two pieces, a bull ring 12, and a piston head 8, which slide into one another and hold a double ring of soft packing. A safety dowel pin 6, which is driven in the piston head, fits into a hole in the bull ring and permits just enough relative movement of the two parts for two rings of packing—no more and no less. This packing is of special material, which was chosen as a result of careful experimenting. The soft packing rings require no "wearing in" as in the case of metallic piston rings. Because of the construction of the piston, the soft packing is squeezed firmly and evenly against the cylinder wall of the valve cap as soon as air pressure is put behind the piston, thereby immediately sealing it against air leakage. A light cup grease should be used to lubricate the packing and to keep it in good condition. Stuffing box 15, seals the valve stem against cylinder pressure in the clearance pocket. Gland 13, is pulled up against soft packing 14, by gland nut 5. Pull up on gland nut 5, only enough to prevent air leakage. The packing should be replaced when it must be pulled up so tight that the action of the valve becomes sluggish. Do not put fingers in the cap to adjust the gland when the regulation system is in operation. Use proper wrench furnished with compressor. When necessary to repack stuffing box, shut off air from regulating system. When the clearance valve is hand operated, a separate screw stem, provided with a hand wheel,

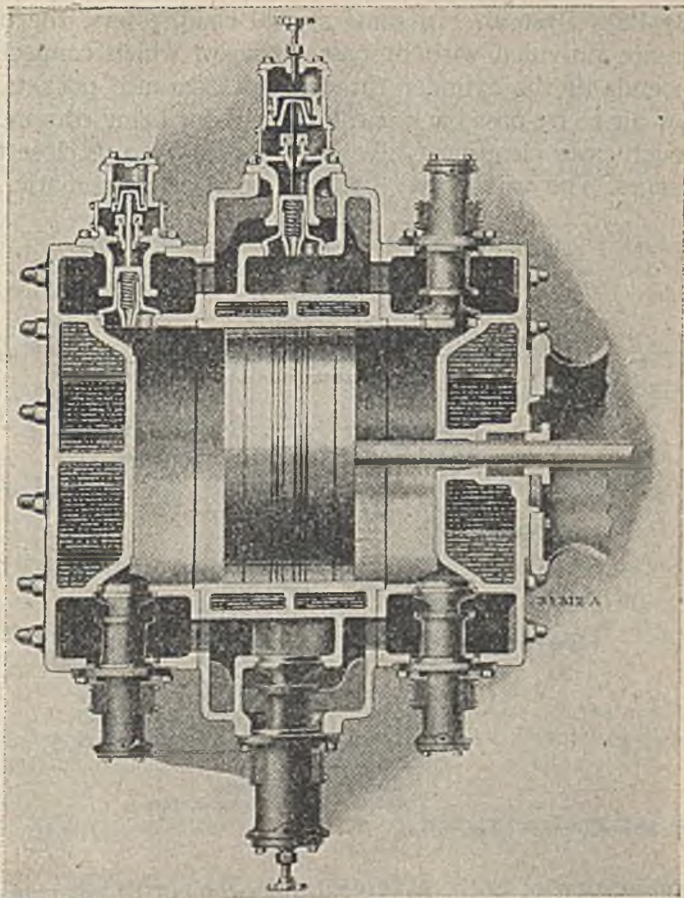


Fig. 37.—Sectional view of cylinder and heads, showing arrangement of clearance pockets, clearance valves, and by-pass valves on a low pressure gas compressor.

Fig. 36—Text continued.

screws through cover 10, and pushes against stem 4. In such cases the piston is omitted. When the automatic valve is furnished with a set screw for locking the valve closed, a separate screw stem, which screws against piston head 8, is used.

By-Pass System.—In some special compressors, the cylinders are provided with by-pass passages, which connect the two ends of the cylinder through the clearance pockets and allow air to by-pass back and forth without being compressed. In such cases clearance valves are used to open or close these passages. The valves may be hand operated, automatic, or a

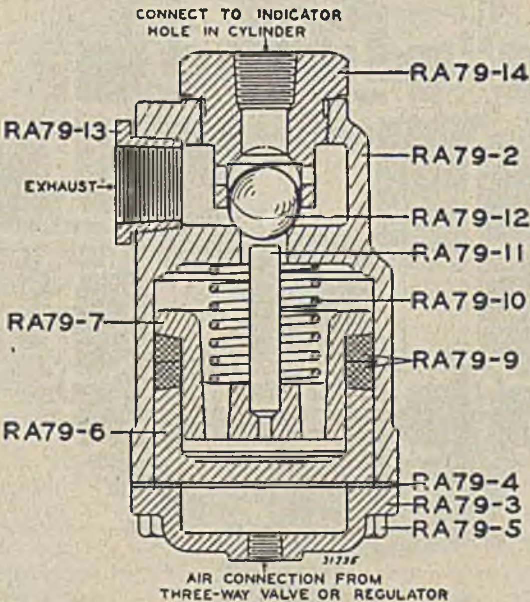


Fig. 38.—Cylinder relief valve.

combination of both. A typical arrangement of clearance and by-pass valves is shown in fig. 37.

Ques. What is this by-pass valve used for on single stage compressors?

Ans. To unload for starting.

Ques. What should be noted about the closing of by-pass valves?

Ans. They will not close automatically when there is no pressure in the receiver. It is therefore necessary to close the by-pass valves by hand until enough pressure is built up in the discharge line to hold them closed.

The locking set screws or hand wheels should be unscrewed after pressure is built up so that the valves will function automatically to unload the compressor when necessary.

Cylinder Relief Valves.—The high pressure cylinder of standard two stage compressors are provided with two automatic cylinder relief valves as shown in fig. 37.

Ques. How are these relief valves connected into the regulating system?

Ans. They are so connected that during the starting period, they open at the same time as the unloader vents the intercooler to atmosphere and the clearance valves open all clearance pockets.

Thus, their only function is to relieve the high pressure cylinder of any air which may be trapped in the cylinder bore and which may leak back from the discharge line through the discharge valves.

Ques. What is the action as the compressor comes up to speed?

Ans. The high pressure cylinder acts as a single stage compressor and takes in air from the inter-cooler at nearly atmospheric pressure. The cylinder pressure will not build up to more than approximately 25 lbs. per sq. in. because the clearance pockets will completely unload the cylinder at this pressure.

Ques. Describe the piping arrangement of the relief valve.

Ans. The hook-up is shown in fig. 39.

In fig. 38, the piston is similar to the clearance valve piston. The bull ring, RA79-7, contains a stem, RA79-11, which raises the ball valve, RA79-12, from its lower position in the body, RA79-2, when the piston raises.

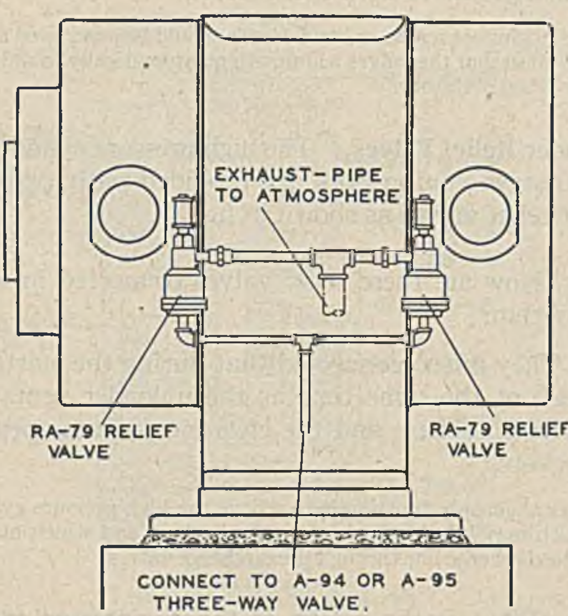


Fig. 39.—Piping arrangement of cylinder relief valves on cylinder.

The ball valve seat, which closes off the cylinder relief openings, is in the removable relief valve head, RA79-14. There is one relief valve for each end of the high pressure cylinder, each valve being attached to the indicator connection at that end by means of pipe fittings leading from the indicator hole to the relief valve head, RA79-14.

The opening in the side of the body between the two-ball valve seats is piped to atmosphere and is the exhaust opening of the relief valve. In the case of gas compressors this should be piped back to inlet. Be sure the exhaust connection from the RA-79 relief valves, if connected together, is large enough so as not to build up pressure and keep the ball valves from seating when the compressor is running. See fig. 39.

The bottom side of the piston is connected to the line supplying air to the regulator body from the starting unloader A94 or A95.

Ques. Describe the operation of the relief valves?

Ans. It may be studied by referring to figs. 38 and 39.

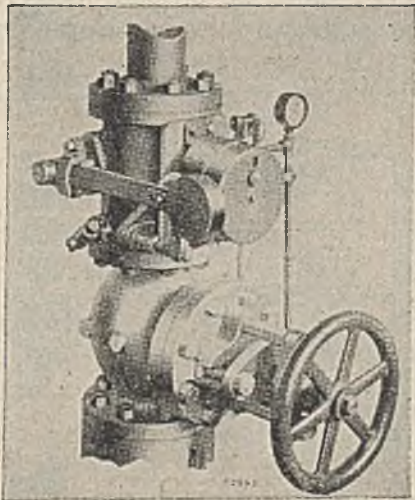


Fig. 40.—Safety stop valve.

Without any outside pressure, the piston would be pushed down on cover RA79-3 by spring RA79-10, and ball valve, RA79-12, would rest in its lower position in the body. This will be the position of the piston and valve during the compressor starting period after the three-way valve exhausts pressure from the regulator and the unloaders. Thus, when starting, any air trapped in the bore of the high pressure cylinder and any air which leaks back through the discharge valves will pass out through the RA-79 relief valves into the atmosphere.

After the three-way valve is turned to the running position, the pressure on piston RA79-6 overcomes the spring pressure and pushes ball valve, RA79-12, on its upper seat in the relief valve head, RA79-14.

As long as the three-way valve is in running position, the ball valve remains on its seat and cuts off the indicator connection from atmosphere. This allows the high pressure cylinder to compress air at its normal discharge pressure.

Safety Devices on Large Steam Driven Compressors.—In the type machine here considered, a positive, weight actuated safety stop, located in the main system line, is operated by a hydraulic tripping device. Oil pressure in the lubricating oil system locks the shut-down valve in the open position, while a

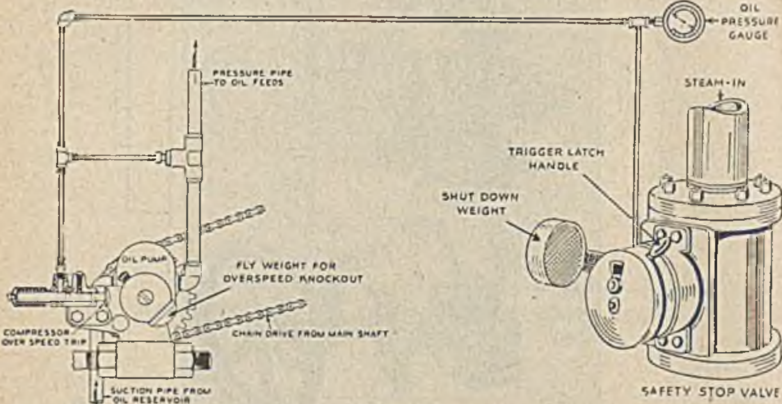
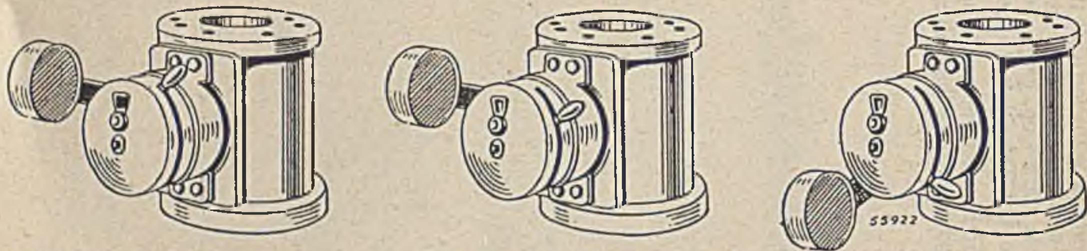


Fig. 41.—Safety stop valve and operating mechanism.

centrifugal knock-out plunger valve, also in the lubricating oil system, releases this pressure by opening a by-pass valve when the unit exceeds a predetermined speed. The release of pressure, caused by the action of the knock-out, instantly trips the drop valve, which cuts off all steam to the unit.

Fig. 40 shows the safety stop valve, and fig. 41 its connections. Figs. 42 to 44 show starting, running and shut-down positions, respectively, of the stop valve. It is apparent that failure of the lubricating oil system *for any reason* will shut down the compressor.

OPERATION OF SAFETY STOP VALVE



Figs 42 to 44.—Operation of safety stop valve 1. **Starting position.** Before the compressor can be started, the shut down weight must be raised and latched by raising the trigger latch handle. The red spot indicates a lack of oil pressure, as is the case while the machine is idle. **2, running position.** As the oil pressure rises, the trigger latch handle drops into running position. The red spot warning disappears. The trigger will now function if oil pressure fail. **3, shut down.** When the compressor stops, or if for any other reason oil pressure fail, the trigger latch handle releases the shut down weight and closes the shut off valve. It must be reset before the machine can be started.

Of particular importance is the fact that the tripping mechanism at the stop valve functions whenever the compressor is stopped. It must be reset before the machine is started up, giving protection against sticking or freezing due to lack of use. This relates to automatic steam cut-off units.

Machines with throttling control are equipped with a similar safety stop incorporated in the governor throttling valve.

CHAPTER 51

Installation

The subject of installation has been covered at considerable length with respect to pumps in the first section of this book and since the procedure is largely the same for both pumps, and air compressors, the author will not waste any space in useless repetition of matter common to both.

For such items as location, excavation, foundations, pouring of concrete, erecting, grouting, etc., see Chapter 51.

Compressor Intake.—Pipe the intake of an air compressor cylinder to the outside of the building and keep it above the ground level. Locate the end of the intake pipe so that steam, water, dust or other waste being discharged from pipes near it, cannot be blown or drawn into the compressor intake.

Ques. What is the primary intake requirement?

Ans. Keep the intake dry.

Ques. What provision should be made if the intake air contain any dust or other abrasive material?

Ans. A filter must be installed.

Such materials if carried into the cylinder will cause rapid wear of the piston and cylinder and will promote the formation of carbon on the valves. Correct size of filter and correct installation are important. Consult a reputable filter manufacturer.

Ques. If impossible to install an intake air filter, what should be done?

Ans. The intake pipe should be run some eight or ten feet above the ground.

Ques. What provision should be made for the top of the intake pipe?

Ans. The top should be hooded or turned down and protected by a wire screen so that rain and large particles of dirt cannot be drawn in.

Ques. If impossible or impractical to run the intake to the outside of the building, what should be done?

Ans. The air may be taken into the cylinder directly from the room.

However, as the room temperature is usually much higher than the temperature of the air outside the building, the compressor cannot deliver as much work.

Ques. What should be noted about intake air temperature?

Ans. Although the power required to compress a cubic foot of air does not depend upon the intake temperature, the quantity of air delivered to the tools does depend upon this temperature.

The lower the compressor intake temperature in relation to the temperature of the air at the tools, the greater the amount of energy delivered to the tools or work.

Ques. What should be noted about moisture in the intake air?

Ans. Moisture will wash away the cylinder lubricant, and if large quantities should accumulate in the intake pipe and be drawn into the cylinder, the non-compressibility of the liquid may cause breakage of parts.

Ques. How may moisture be avoided?

Ans. Arrange the piping so that moisture may be drained off before it is drawn into the cylinder.

Ques. What should be noted about long intake air pipe lines?

Ans. The longer the line the larger should be the pipe to avoid undue friction and resulting lowering of admission pressure.

Compressor Intake Pipe.—The size of the intake connection on the compressor cylinder is indicated on the foundation plan. The intake pipe should be at least this size and it is a good rule to increase it one pipe size for every 10 ft. in length.

A slow intake pipe velocity reduces friction and consequent pressure loss, and also tends to reduce air pulsations. Make the intake pipe as short and direct as possible and use long radius elbows where bends are necessary.

Ques. What is the proportion of air intake pipe for direct servicing more than one compressor?

Ans. The cross-sectional area of the main pipe or duct must be at least as large and preferably larger than the combined area of all the individual intake pipes. In branching from the main intake pipe avoid sharp corners.

Ques. What kind of intake pipe is used?

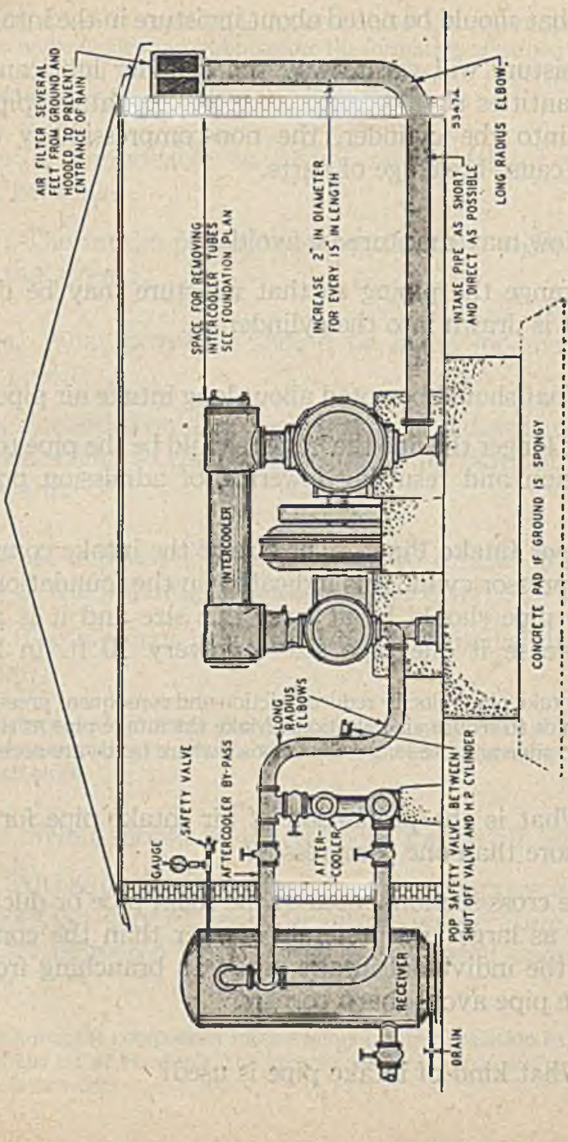


Fig. 1.—Arrangement of intake piping and filter.

Ans. It may be of cast iron, black or galvanized steel pipe or any other material which will not crack and which will keep out moisture and dirt.

Ques. What is used for larger sizes?

Ans. For larger sizes or when several compressors have a common intake duct, glazed vitrified pipe with cemented joints, all bedded in concrete makes a good construction.

Ques. Why?

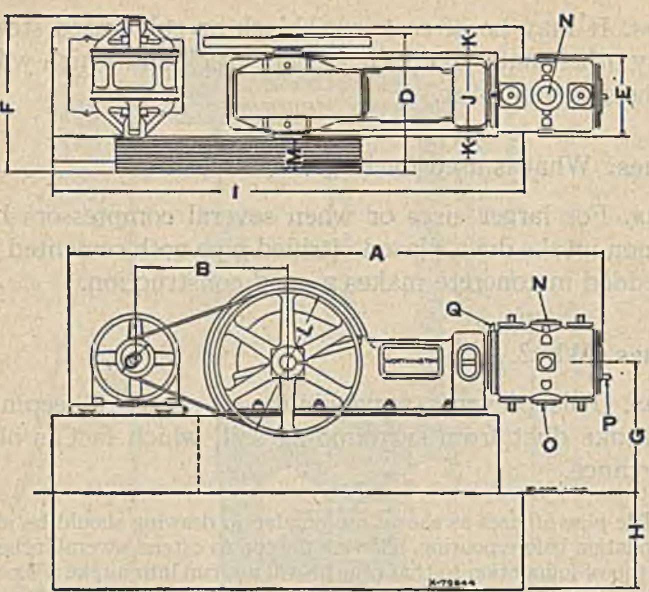
Ans. This prevents a possibility of moisture seeping into the intake duct from surrounding soil, which fact is of great importance.

Tile pipe of sizes as shown on foundation drawing should be placed in foundation before pouring, allowing the top to extend several inches above the top of foundation so that cement will not run into intake when pouring grout. No connection is made between intake pipe and cylinder but machined surfaces on four sides of cylinder and sole plate make a dust-tight joint.

Fig. 1 shows suggested arrangement of intake piping and filter with sump to prevent any water seepage from surrounding soil or excess water carried with air into intake during wet weather, being carried into low pressure cylinder.

If a concrete duct be used as an air intake, it must have a smooth, hard interior surface, for if the concrete crumble or disintegrate, it will be carried into the compressor cylinder and cause rapid wear. Painting the interior of a concrete duct with a good waterproof paint is advisable.

Where compressors are used in or adjacent to chemical plants, the air or gas taken into the compressor often contains acid fumes, which attack iron and steel, causing corrosion and wear of the valves, pistons and cylinders.



Figs. 2 and 3.—Plan and elevation of electrically driven compressor with dimension indicated to correspond with the table below.

TABLE OF DIMENSIONS
SINGLE BELT DRIVEN COMPRESSOR

Size	A	B	C P.D	D	E	F	G	H	I	J	K	L P.D.	M	N Suct.	O Dis.	P Inl	Q Water Out.
10 x 10	12' 6"	53"	9.1"	40½	22"	41¾	2' 6½"	1' 9"	11' 3"	20"	6"	37"	9⅝"	3½"	3½"	1"	1½"
11 x 12	13' 9"	51"	9"	47⅞	2' 3"	47"	2' 10¾"	2' 0"	11' 9"	2' 0"	6"	39"	12⅜"	4"	4"	1"	1½"
13 x 14	15' 3"	56.8"	13.2"	53 5/16"	2' 6"	55¼"	3' 2½"	2' 6"	12' 4"	2' 4"	6"	47"	11 13/16"	5"	5"	1"	1½"
15 x 16	16' 9"	66.3"	13"	64 5/16"	2' 6"	58 5/16"	3' 9½"	3' 0"	18' 4"	2' 9"	6"	52"	14 11/16"	6"	6"	1"	1½"
11 x 10	13' 0"	53"	9.1"	40½"	2' 3"	44"	2' 6½"	1' 9"	11' 3"	20"	6"	37"	9⅝"	4"	4"	1"	1½"
13 x 12	14' 3"	51"	9"	49⅞"	2' 6"	51½"	2' 10¼"	2' 0"	11' 9"	24"	6"	39"	14⅜"	5"	5"	1"	1½"
16 x 14	15' 0"	56.8"	13.2"	56 5/16"	2' 9"	58 5/16"	3' 5¼"	2' 6"	14' 4"	2' 4"	6"	47"	14 11/16"	6"	6"	1"	1½"
18 x 16	17' 0"	66.3"	13"	68½"	3' 0"	66½"	3' 9½"	3' 0"	18' 1"	2' 9"	6"	52"	19"	8"	8"	1"	1½"
20 x 16	17' 0"	66.3"	13"	70 13/16"	3' 5"	70"	3' 9½"	3' 0"	18' 10"	2' 6"	6"	52"	20 5/16"	10"	10"	1"	1½"
13½ x 10	13' 3"	53"	9.1"	42½"	2' 3"	46"	2' 6¾"	1' 9"	11' 3"	20"	6"	37"	11⅜"	6"	6"	1"	1½"
15 x 12	14' 0"	51"	9"	51⅞"	2' 8"	59⅝"	2' 11¾"	2' 0"	13' 9"	24"	6"	39"	16⅝"	6"	6"	1"	1½"
18 x 14	15' 0"	56.8"	13.2"	56 5/16"	3' 0"	58⅝"	3' 4¼"	2' 6"	14' 4"	2' 4"	6"	47"	14 11/16"	8"	8"	1"	1½"
22 x 16	17' 0"	66.3"	13"	70 13/16"	3' 5"	70"	3' 9½"	3' 0"	11' 3"	2' 9"	6"	52"	20 5/16"	10"	10"	1"	1½"
15 x 10	13' 0"	50.2"	9.1"	40½"	2' 6"	44"	2' 6¾"	1' 9"	11' 3"	20"	6"	40"	9⅝"	6"	6"	1"	1½"
17 x 12	13' 9"	48.5"	9.1"	50⅞"	2' 10"	52½"	3' 3¼"	2' 0"	13' 9"	24"	6"	42"	15⅜"	8"	8"	1"	1½"
20 x 14	16' 0"	65"	13"	54¾"	3' 5"	56⅞"	3' 4"	2' 6"	15' 0"	2' 4"	6"	69"	13¼"	10"	10"	1"	1½"
22 x 14	16' 0"	65"	13"	56 5/16"	3' 5"	58 5/16"	3' 9½"	2' 6"	14' 10"	2' 2"	6"	69"	14 13/16"	10"	10"	1¼"	2"

If this condition be known to exist, the manufacturer of the compressor should be informed when the machine is purchased, so that proper provision can be made to take care of it.

Ques. What precautions should be taken with the intake line?

Ans. An intake line to compressor should be thoroughly cleaned before the machine is first put in operation to remove accumulations of pipe scale, sand, grit or other foreign objects thoughtlessly placed in the line during the work of installing the compressor.

The free air intake on an air compressor should never be installed or located at or near a gas pop relief valve, gas or gasoline vapors of any kind, oil vapors, or any other gas or vapor which will form an explosive mixture. The drawing in of these vapors into a compressor might very easily cause a disastrous explosion.

Ques. What shape intake should be avoided?

Ans. Avoid intake pipes of rectangular cross section, particularly when wood or metal is used for the duct.

Ques. What should be the condition of intake pipe?

Ans. The intake pipe must be clean, free from all scale and foreign matter when it is installed. Never put a valve in the intake pipe of an air compressor with atmospheric intake.

However, should a valve be necessary a small receiver should be placed between the valve and the compressor cylinder to reduce pulsations in the intake pipe. The receiver will collect dirt and moisture and should, therefore, be drained periodically.

Closed Systems.—For compressors on booster or gas service, a valve on the intake pipe may be necessary, according to operating conditions.

Ques. What provision should be made if a valve be used, and why?

Ans. A small receiver should be placed between the valve and the compressor cylinder to reduce pulsations in the intake pipe. The receiver will also act as a moisture remover and should be drained regularly.

Ques. What are the requirements in the case of a gas compressor which handles a very dirty or tarry gas?

Ans. Some means of scrubbing the gas before it enters the compressor should be employed. The scrubber should be used in addition to the receiver, or some method of combining the receiver and the scrubber may be found more practical.

Compressor Discharge Piping.—The discharge pipe connects the discharge opening of the compressor cylinder with the after-cooler and receiver. Its size must be at least as large as the connection on the cylinder, it should be short and direct, and have as few elbows as possible, and where the latter are necessary they should be long radius elbows.

Ques. What provision is made for overhead piping?

Ans. It must be well supported so that there is no strain on the compressor cylinder.

Ques. Why should the discharge pipe be not installed in the intake duct?

Ans. In the first place the area of intake for entrance of air would be reduced and secondly because the temperature of the intake air will rise and consequently cause a loss in volume and work delivered by the compressor.

Ques. What should be placed in the discharge circuit between the compressor and receiver?

Ans. The after-cooler.

Water Piping.—Compressor cylinders are provided with inlet, outlet and drain openings in the water jacket. The

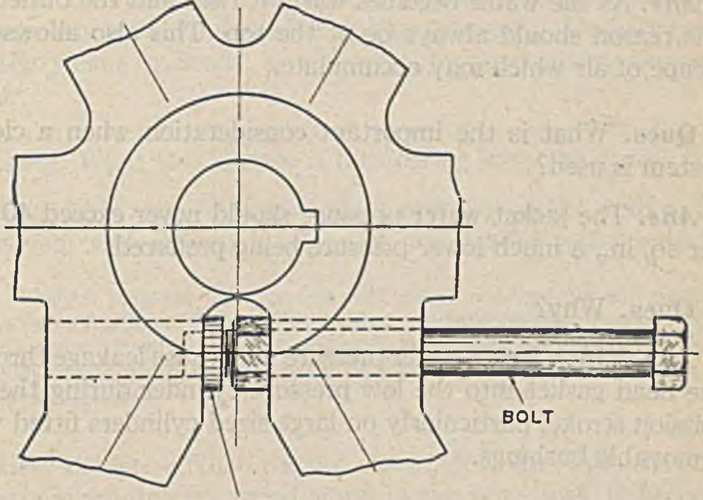


Fig. 4.—Special steel plug and bolt for assembling or removing the wheels. Compressors are always shipped with the wheels removed from the shaft, and they should be put on in the following manner: To secure a gripping fit, the bore of the wheel hub is a few thousandths of an inch smaller than the shaft diameter. For this reason the plug which is furnished must be used in putting on and taking off wheels. Remove the hub clamping bolt. Place special plug in slot so that the boss on plug fits in bolt hole as shown. Hold nut in slot and turn bolt until end touches special plug and nut rests against opposite wall of slot. Use wrench on outer end of bolt to tighten. As the bolt is turned, the inner end will exert force against plug, thus spreading split hub so that wheel can be placed on shaft or removed as shown. Wheels will be in proper position when the end of shaft projects 1/8 in. beyond the face of the wheel hub on the 7 in. stroke and 3/16 in. on the 9 in. and larger sizes. The 7 in. stroke and smaller compressors are usually equipped with heavy belt wheel only, the fly wheel being omitted. For details see foundation plan furnished with the machine.

controlling valve should be placed on the water inlet, the outlet open, the water falling into an open pipe end or funnel so that it can be seen at a glance whether water is passing through the jacket.

Ques. Why should the water outlet always be at the top?

Ans. As the water becomes warm it rises and the outlet for this reason should always be at the top. This also allows the escape of air which may accumulate.

Ques. What is the important consideration when a closed system is used?

Ans. The jacket water pressure should never exceed 40 lbs. per sq. in., a much lower pressure being preferred.

Ques. Why?

Ans. A high jacket water pressure may cause leakage through the head gasket into the low pressure cylinder during the admission stroke, particularly on large sized cylinders fitted with removable bushings.

Ques. How is the water piping arranged on the compressor?

Ans. The circuit is arranged with the cold water entering the intercooler first and then carried to the cylinders so that the warmer water is used in cylinders to prevent sweating on the cylinder walls.

Ques. What should be the temperature of the water at the outlet?

Ans. Enough water is circulating when the water at the outlet is lukewarm (80° to 100° Fahr.).

Where the cooling water is cold, it is often necessary to merely crack the water inlet valve to maintain the proper water discharge temperature. This may lead to trouble and it is advisable to install a by-pass of smaller pipe around the main inlet valve to get better regulation of the water flow.

Ques. What precaution should be taken when compressor is exposed to freezing temperature?

Ans. Be careful to drain the cylinder jacket thoroughly if compressor is to stand in a freezing temperature. If water freeze in the jacket or heads, it will certainly crack them sooner or later.

Ques. What will happen if the cooling water be dirty?

Ans. Mud will be deposited in cylinder jacket, and unless removed, will ultimately obstruct the flow of water entirely.

Clogged passages will interfere with proper cooling and will result in excessive heating of valves. This heating will cause deposit of carbon which will make the valves leak. Clogging will also cause possible damage to the cylinder and piston and will result in a hard-baked coating of mud in the water passages.

After Cooler.—When the hot, moist air or gas from the compressor discharge is carried along in the pipe lines, it becomes cooled by radiation and the moisture in it is deposited in the pipe. This causes "water hammer" which may develop leaky joints and may freeze and burst pipes in cold weather.

Wet air is very detrimental to pneumatic tools, switch and signal systems, or any open work such as quarrying, etc., as it washes away the lubricant or freezes when expanding in valves and ports, thereby greatly interfering with the operation of the tools. It is, therefore, very advantageous in most installations to use an after cooler to condense the moisture before it gets into the pipe lines.

Ques. Where should the after-cooler be located?

Ans. As near the compressor as possible, placed between the compressor and receiver.

Full size pipe connection between compressor and after cooler should be used to take care of pulsations in the pipe line.

Ques. In installing the piping what precautions should be taken?

Ans. Make sure there is no strain on the pipe flanges. Properly support the piping. Also provide for expansion of piping.

Location of Valves.—The location of the valves on the intake and discharge of the gas compressor would at first appear to be a minor consideration. This is not always the case.

Ques. Why should an intake and discharge valve be provided?

Ans. To block off or cut off the compressor when it is necessary to work on the compressor without shutting down the rest of the plant.

Ques. What should be noted as to location of intake and discharge valves?

Ans. The location depends upon: 1. The size of the valve and space available. 2. The location of headers. 3. The discharge relief valve. 4. Other piping and space considerations of the individual installation.

For the operation of the valves, that is, the opening and closing when putting the compressor in service or taking it out, the most convenient location is above and below the compressor. The size of the valves prevents this location on larger cylinders and where room is a factor. The intake valve is generally located on top of the building. The pop relief valve on the discharge must *always* be placed *between* the compressor and the discharge valve, and when the discharge valve is located inside the compressor room near the compressor, then the pop relief valve is located inside the building. This means that the relief valve would discharge or pop gas into the engine room and form a dangerous explosive mixture. This condition may offer a problem of additional piping in the engine room. The discharge valve of the compressor is frequently placed outside the engine room and near the discharge header.

Air Receiver.—The air receiver or tank should be placed as close to the compressor (or after cooler, if one be installed) as possible so as to keep the discharge pipe short.

Ques. What important point should be noted here and why?

Ans. Never place a valve in the line between compressor and receiver, unless a safety valve be installed between this valve and the compressor, as there is a possibility of starting the compressor with this valve closed. In such a case, an explosion will result, since the air cannot escape.

Ques. What provision should be made when the compressor is to be connected into an air main common with other compressors and why?

Ans. A safety valve should be placed between the compressor and the first valve in the pipe line. This will safeguard the compressor in case of failure to open the valve before starting up.

Ques. Where should the receiver be placed if the compressor be located near the wall?

Ans. Place the receiver outside the building where it has an opportunity to radiate some of the heat.

Ques. What provision is made for draining the receiver?

Ans. A drain cock is placed near the bottom and the receiver should be drained from time to time.

Exciter Belt.—Where the compressor is driven by an electric motor if a belted exciter be used, the pulley on the exciter shaft should be very carefully lined up with the pulley on compressor shaft. After the exciter is in line, the driving belt should be made to proper length. It is always preferable to make an endless belt instead of using leather lacers or wire fasteners so that there will be no jarring as the belt passes over the pulley, as this sometimes causes hunting of the motor.

Rotation.—In a compressor, as in a steam engine, it is advisable to rotate the wheels so that the pressure of the cross head shoe will be down upon the lower guide.

This prevents any tendency to lift, and also renders lubrication more efficient. For this reason the machine should be run with the top of the wheels moving toward the cylinder as shown by arrows on foundation plan supplied with each machine.

CHAPTER 52

Lubrication

Lubrication of a compressor is the most important point in its care and operation. Lack of proper lubrication will result in damage to many parts. Moreover, use of low grade oil will greatly shorten the life of the compressor and result in lower efficiency.

Use nothing but high grade oil meeting the specifications given for each application.

Crank Case Lubrication.—In crank case lubrication systems all the oil delivered to the bearings and running gear is returned to the oil sump in the bottom of the crank case, which is entirely enclosed so that the oil cannot escape. The same oil is thus used repeatedly.

A circulation of oil which floods the bearings is maintained by various devices acting as a pump, such as:

1. Oil dipper.

2. Crank disc.

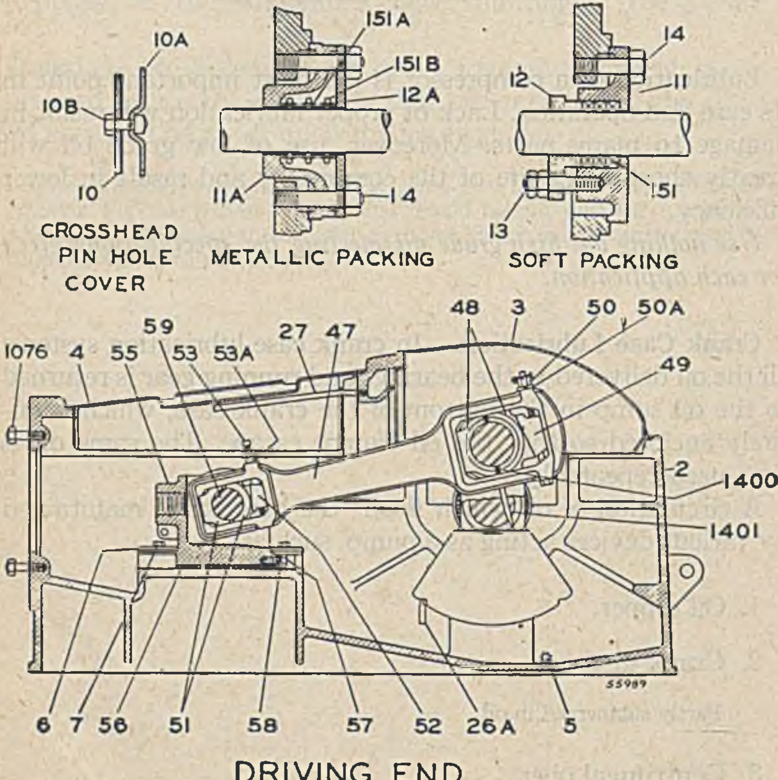
   Partly submerged in oil

3. Centrifugal oiler.

The oil dipper system has a dipper attached to the crank shaft counter weight as shown in figs. 4 and 5, part 26A and which dips into the oil in the crank case at each revolution.

Ques. How does it work?

Ans. A portion of the oil runs into holes supplying the main bearings and drains back into the crank case sump, causing a continuous circulation of oil through the bearings.



Figs. 1 to 4.—Bed plate connecting rod and cross head parts; oil dipper lubrication.

Bedplate, Connecting Rod and Crosshead Parts

Part No.	Name of Part
1	**Bedplate Complete with Metallic Packing** (Includes Parts 2 to 10B, 11A, 12A, 14, 151A, 151B, 1063 to 1067 and 1073 to 1076)
1A	**Bedplate Complete with Soft Packing** (Includes Parts 2 to 10B, 11 to 14, 151, 1063 to 1067 and 1073 to 1076)
2	Bedplate Bare (Includes Parts 8, 8B, 9, 27, 1073 and 1076)
3	Bedplate Oil Guard
4	Bedplate Cover
5	Bedplate Drain
6	Bedplate Guide Bar
*6A	Bedplate Guide Bar Liner
7	Bedplate Guide Bar Tap Bolt
10	Crosshead Pin Hole Cover and Gasket
10A	Crosshead Pin Hole Cover Yoke
10B	Crosshead Pin Hole Cover Bolt
11	Partition Stuffing Box (Soft Packing Type)
11A	Partition Stuffing Box (Metallic Ring Type)
12	Partition Stuffing Box Gland (Soft Packing Type)
12A	Partition Stuffing Box Cover (Metallic Ring Type)
13	Partition Stuffing Box Gland Stud and Nuts (Soft Packing Type)
14	Partition Stuffing Box Stud and Nut
27	Crosshead Oiler

Part No.	Name of Part
46	**Connecting Rod Complete** (Includes Parts 47 to 53A)
47	Connecting Rod Bare
48	Crank Pin Box (2 pieces)
49	Crank Pin Box Wedge
50	Crank Pin Box Wedge Bolt, Nut, and Cotter Pin (Includes Part 50A)
50A	Crank Pin Box Wedge Bolt Lockwasher
51	Crosshead Pin Box (2 pieces)
52	Crosshead Pin Box Wedge
53	Crosshead Pin Box Wedge Set Screw and Nut
53A	Crosshead Pin Box Wedge Set Screw Locking Sleeve
54	**Crosshead Complete** (Includes Parts 55 to 59)
55	Crosshead Bare
56	Crosshead Gib
57	Crosshead Gib Tap Bolt
58	Crosshead Gib Tap Bolt Lockwasher
59	Crosshead Pin, Washer, Tap Bolt and Lockwasher
151	Partition Stuffing Box Packing (Soft Packing Type)
151A	Partition Stuffing Box Oil Wiper Ring (Includes Part 151B)
151B	Partition Stuffing Box Oil Wiper Spring
1076	Bedplate Flange Stud and Nut
1400	Crank Pin Box Shims
1401	Connecting Rod Oil Scoop

Parts marked thus (*) not shown on cut.

Some of the oil is caught in a pocket inside the crank case and is delivered by a pipe in a continuous stream to the cross head pin, cross head guide and piston rod, flooding these parts.

Ques. What provision is made where the piston rod passes through the frame or crank case to prevent heavier crank case oil working along the shaft into the compressor cylinder?

Ans. A stuffing box is provided.

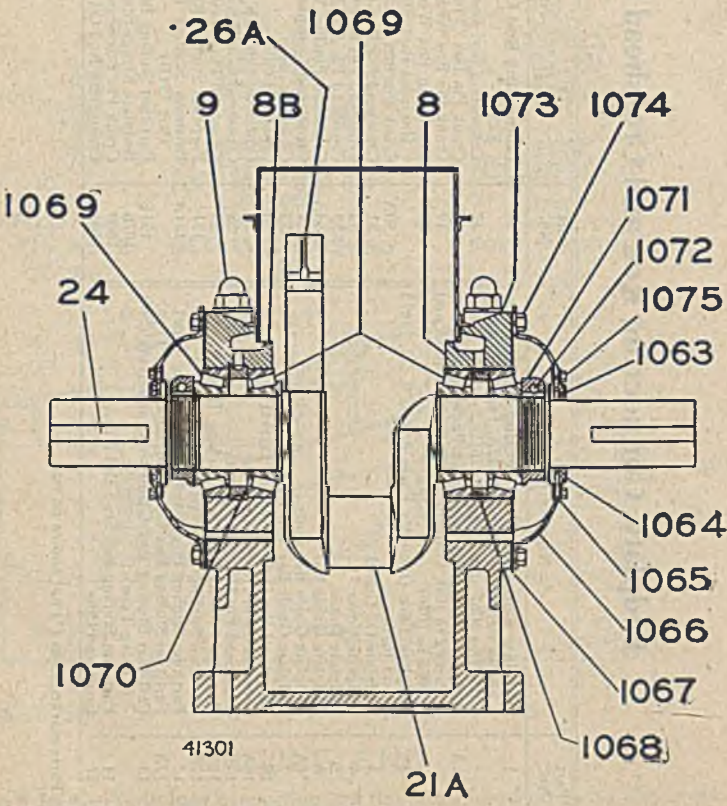


Fig. 5.—Crank shaft main bearing and wheels.

Crankshaft, Main Bearings and Wheels

(7″ Stroke and Larger Sizes)

Part No.	Name of Part
8	Main Bearing Cap (Right-hand) Right- and left-hand caps are identical for 7-inch stroke
8B	Main Bearing Cap (Left-hand)
9	Main Bearing Stud and Nut
20	**Crankshaft Complete** (Includes Part 21A, 24, 1068, 1069, 1070, 1071 and 1072)
21A	Crankshaft (Includes Part 26A)
26A	Oil Dipper
1063	Crankshaft Felt Washer
1064	Crankshaft Felt Retainer
1065	Crankshaft Felt Retainer Gasket
1066	Main Bearing Side Cover
1067	Main Bearing Side Cover Gasket
1068	Roller Bearing Spacer Ring (Right-hand)
1069	Timken Roller Bearing (Four required)
1070	Roller Bearing Spacer Ring (Left-hand)
1071	Shaft Collar and Oil Ring (Includes Part 1072)
1072	Shaft Collar and Oil Ring Clamping Bolt and Nut
1073	Main Bearing Stud Lock Washer
1074	Main Bearing Side Cover Tap Bolt
1075	Felt Retainer Tap Bolt
	WHEELS
22*	Flywheel (not furnished on 7″ stroke Compressor) (Includes part 25A) for ES machine
22*	Plain Flywheel (Includes part 25A) for FS machine
23*	Belt Wheel (Includes part 25) for ES machine
23*	Flywheel (Includes part 25A) for Adjustable-Speed Governor on FS machine
24	Belt or Flywheel Key
25*	Belt Wheel Stud and Nuts
25A*	Flywheel Stud and Nuts
1173*	Texrope Sheave (Includes Part 25)

Parts marked with an (*) Asterisk are not shown on cut.

Ques. What kind of oil is used for crank case lubrication?

Ans. For the crank case only use a high grade motor oil recommended by a reputable oil company. Use S.A.E. 30 or S.A.E. 40 oil for normal operating conditions. S.A.E. 20 oil will be required in freezing temperatures.

Ques. How much oil is required to fill each crank case of various size bed plates?

Ans. Ratings as given by one manufacturer are: 14 in. stroke, 4 gals; 16 in. stroke 5 gals.; 18 in. stroke 9 gals.; 21 in. stroke, 13 gals.; 24 in. stroke 40 gals.; 27 in. stroke 50 gals. These ratings, however, do not take into account the amount of oil which would be splashed over the surfaces of the crank case.

Ques. Why should the oil be perfectly clean and free from any moisture or foreign matter?

Ans. Aside from the ill effect of abrasive foreign matter, its presence may develop an emulsion by constantly splashing on the bearings.

Ques. Describe the crank disc pump system.

Ans. Oil is carried on the outer rim of each crank disc to the top where it is diverted into an oil boat by an oil wiper mounted on a stud in the oil boat.

This system is shown in fig. 6. The oil boat, which is bolted to the main bearing cap, has four compartments. All of the oil is poured into one main compartment from which it overflows to each of three supply compartments.

From one supply compartment the oil is carried through a cored opening in the oil boat, to the main bearing. One supply compartment is piped to a sight feed oiler at the cross head guide which lubricates the cross head guides and the cross head pin bearing.

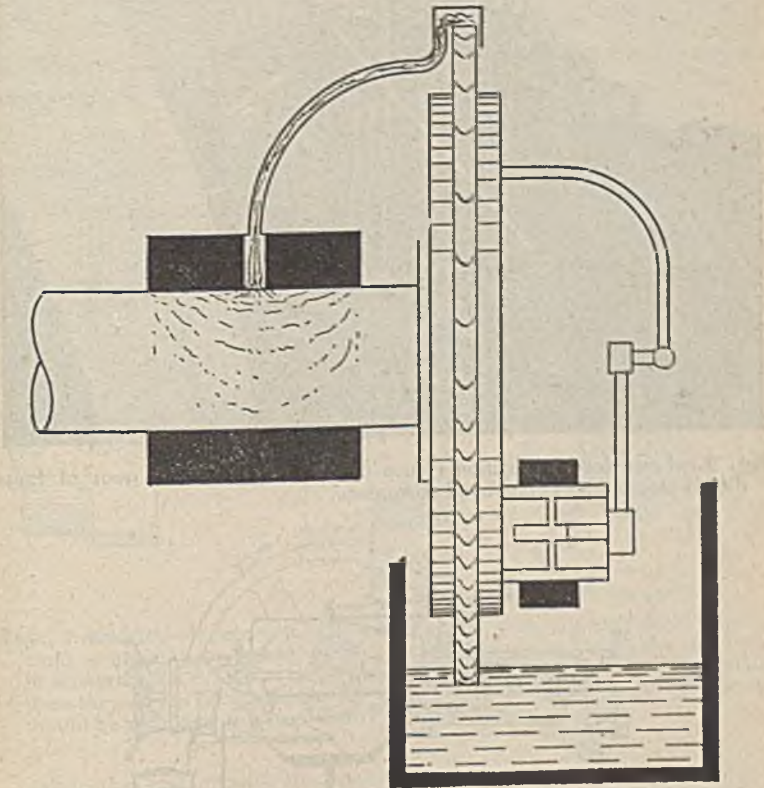


Fig. 6.—End view of crank case showing detail of crank disc "pump" lubricating system.

The other supply compartment is piped to a centrifugal oiler which carries oil to the crank pin bearing.

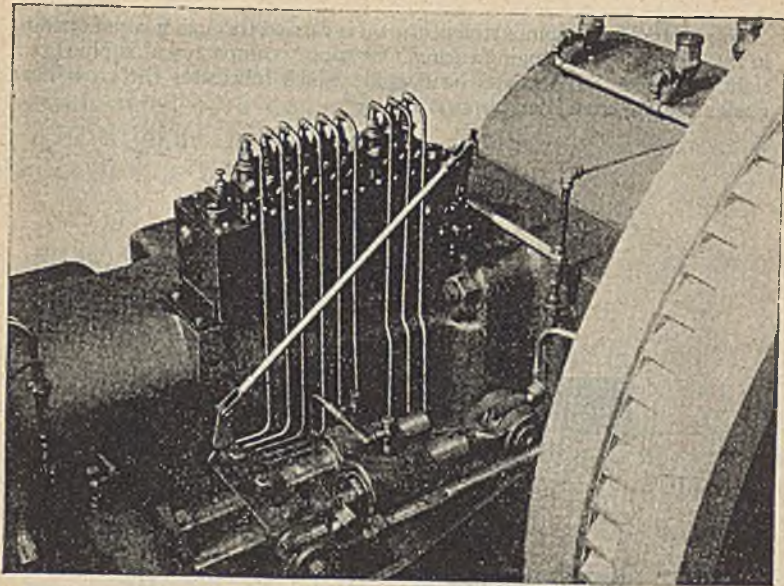


Fig. 7.—Force feed lubrication system driven from the valve gear of large duplex steam driven compound compressor.

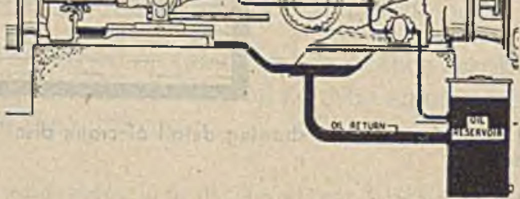
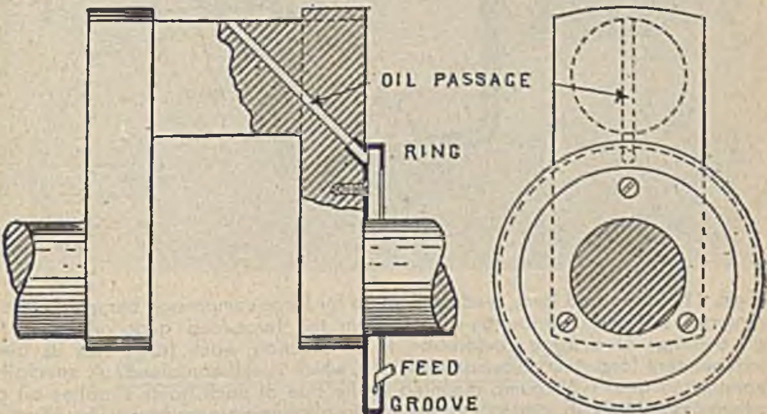


Fig. 8.—Elevation showing reservoir and oil circuits of the lubrication system shown in fig. 7.

Ques. What is the particular feature of this system?

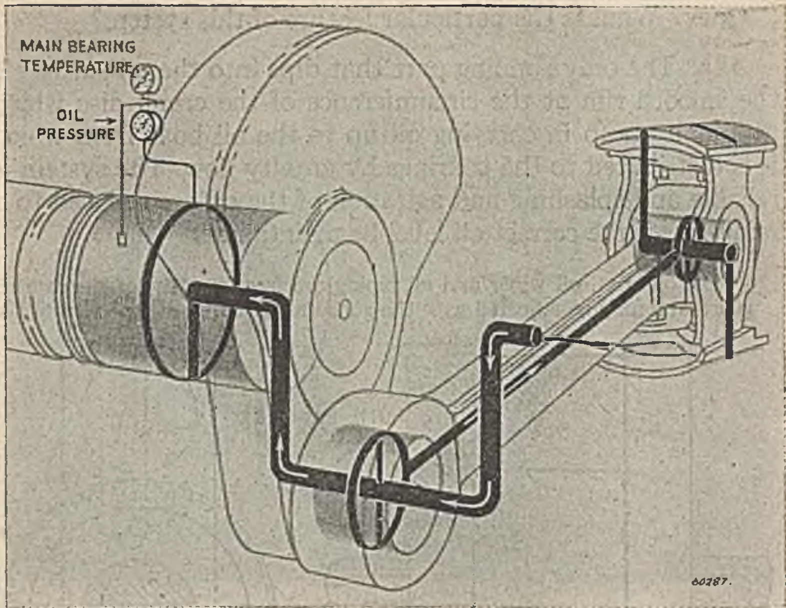
Ans. The only running part that dips into the oil surface is the smooth rim at the circumference of the crank disc which acts as a pump in carrying oil up to the oil boat from which it is distributed to the bearings by gravity flow. The system is reliable and splashing and agitation of the oil are reduced to a minimum if the correct oil level be maintained.

The oil boat, oil wiper and all piping are free from the crank case oil guard and may be adjusted with the guard removed. For inspection of the lubrication system while the machine is running, hand hole covers are provided at the main bearings, at the top of the crank disc, and in the cover at the cross head.



Figs. 9 and 10.—Ring centrifugal crank pin oiler. This type is adapted to center crank engines. An oil passage leads from the grooved ring to the crank pin. In operation the oil which drops into the groove, is carried off by centrifugal force through the oil passage to the crank pin. In construction, the oil passage should be of liberal size to prevent clogging.

Ques. Explain how centrifugal force is employed in forcing a circulation of oil.



Figs. 11.—Standard force feed lubrication for large compressor, beng a skeleton diagram showing distribution of oil from the force-feed gear oil pump to all bearings of a large compressor. In construction, each frame has its own independent force feed lubrication system which is self-contained. A specially constructed gear type pump mounted in the side of each frame supplies oil at approximately 25 lb. per sq. in. pressure to all running gear parts. The pump is driven by the crank pin through a full floating telescopic arm which prevents binding from any possible mis-alignment. A fine mesh cylindrical filter surrounds the pump and filters all oil before going to the bearings. The pump draws the oil from the sump in the crank-case through a strainer which protects the pump gears from damage caused by dirt, scale or any foreign material which might otherwise be drawn into the inlet pipe. Oil leaving the pump gears pass to the outside of the cylindrical filter, through the filter, and thence through the hollow pump drive shaft and telescopic drive arm to the crank pin. From the crank pin, oil is distributed through drilled passages to the main bearing and also through a drilled hole in the connecting rod to the wrist pin. From the wrist pin oil is carried through drilled holes in the cross-head to the top and bottom cross-head shoes as shown. A relief valve mounted on the oil pump safeguards the lubricating system against excessive oil pressure. The extra capacity of the oversize pump, normally by-passed back to the crank case sump, assures maintenance of oil pressure indefinitely.

Ans. To illustrate a centrifugal oil ring system is shown in figs. 9 and 10. Although it receives its supply by drip feed and not by submersion, it brings out the centrifugal principle.

In operation, centrifugal force tends to throw the oil from the center of rotation; hence it presses against the bottom of the groove and is forced through the duct to the bearing, thus lubricating the pin.

Force Feed Lubrication.—Large compressors (24 in and larger) are shown in figs. 7 and 8. The system comprises a gear

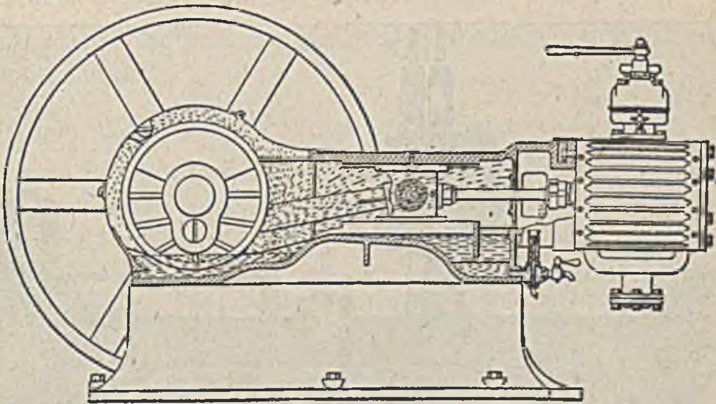


Fig. 12.—Typical horizontal steam engine showing splash lubrication system. Oil is placed in the enclosed frame in sufficient quantity to submerge the connecting rod at its lowest point of travel, causing it to splash the oil over the various bearings. It should be noted that the piston rod stuffing box is outside the enclosed portion of the frame, thus keeping the oil separate from any water of condensation, which may leak past the stuffing box—an important point in this system.

type oil pump, chain driven from the main shaft; is mounted in the left hand frame and supplies oil at approximately 20 lbs. per sq. in. pressure to the bearings. This pump draws oil from the main oil sump in the left hand frame, through a removable strainer.

The strainer serves to protect the pump gears from damage, caused by dirt, scale or any foreign material which might otherwise be drawn into the pump intake pipe.

Ques. How does the system operate?

Ans. Oil under pressure is forced through a filter and then through an oil cooler assuring clean, cool oil for all bearings. From the filter and cooler the oil is pumped through a header to separate feeds to each main bearing and crank pin bearing. Oil to crank pin is carried through a swivel joint and crank oiler.

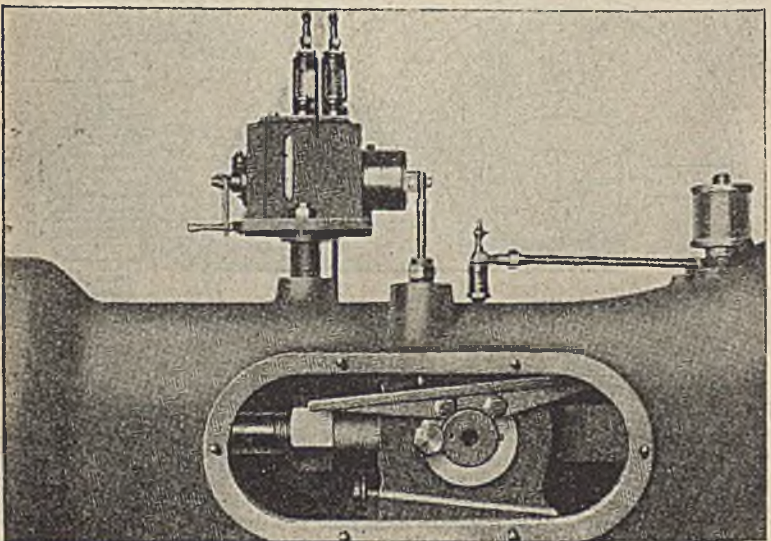


Fig. 13.—Force feed cylinder lubricator.

From each crank pin bearing oil is piped along the connecting rod and through a swivel joint to the cross head pin bearing. Oil is also carried from the cross head pin through drilled passages in the cross head to both top and bottom cross head shoes.

Ques. What safety device is provided?

Ans. A relief valve installed in the oil line between the pump and filter safeguards the lubricating system against the building up of excessive pressure.

Ques. What is the normal capacity of the oil pump?

Ans. The oil pump normally delivers considerably more oil than is required for the bearings.

Ques. What becomes of the excess oil?

Ans. It is by passed through a relief valve and returned to the oil sump.

Ques. What safety provision is made in case of low pressure?

Ans. An automatic pressure switch connected to the oil header is arranged to stop the compressor if oil pressure drop below 8 lbs. per sq. in.

This is an important safeguard as it protects the bearings against damage from oil failure. *This pressure switch must not under any consideration, be blocked up or tampered with to prevent its normal operation.*

Ques. How is the force feed system applied to larger units such as "four corner" machines?

Ans. An oil reservoir, located in a pit between the compressor frames, provides oil storage capacity, supplementing the oil sumps cast as part of the frames. A motor driven, centrifugal pump takes oil from this reservoir and pumps it, at approximately 20 lb. per sq. in. pressure, through an oil filter and oil cooler to a header supplying oil to the bearings.

Lubrication for Air Cylinders.—The cylinders are supplied by force feed lubricators delivering oil to the top of the cylinder bore, as shown in fig. 11.

The cylinders should be lubricated with the best grade of air cylinder oil, which is to-day made special for air cylinder lubrication by all the reputable oil companies.

Ques. What precaution should be taken in the lubrication of air cylinders and why?

Ans. It is advisable to use as little oil as possible, as an excess quantity of oil tends to form carbon on the valves, making them leaky which in turn causes excess temperature in the air cylinders. Improper and excess lubrication might cause an explosion in the air cylinder, discharge line or receiver.

In general the following is approximately the amount of oil required for air cylinders:

12 to 17 ins diam.	3 to 4 drops per min.
18 " 24 " "	4 " 6 " " "
25 " 33 " "	6 " 8 " " "
35 " 45 " "	9 " 12 " " "

As figuring oil by drops is only approximate, the following table lists the quantity of oil which is required for the larger size compressors, for a period of ten hours:

Stroke	*Oil Required In 10 hrs.*	*Stroke*	*Oil Required In 10 hrs.*
14 ins.	.50 pints	24 ins.	.90 pints
16 "	.55 "	27 "	1.05 "
18 "	.65 "	30 "	1.20 "
21 "	.75 "	36 "	1.35 "

Ques. Do the rates just given hold for new compressors?

Ans. No. When a new compressor is first started, double the previous quantity of oil for a short time, but when the machine is running in proper shape, these quantities of oil are the maximum required for good operation and should be reduced as much as possible whenever the circumstances permit. Many compressors are operated on less oil than shown in the table.

CHAPTER 53

Operation

The compressor having been installed and piped in accordance with the instructions in the chapter on Installation, carry out the instructions following before starting the compressor.

Preliminary Procedure.—1. Clean up the floor and foundation, and remove all cement, dirt and dust from the exterior of the compressor.

2. Remove the oil covers on the frame. Wipe out thoroughly every portion of the interior and oil basin to insure a clean interior, free from dust and dirt which may have entered during shipment and installation

Clean the slush and dirt from the piston rod.

3. Blowout all oil piping, passages and pockets with a jet of air.

Use a bellows if no other means be available. This part of the compressor cannot be too clean. Thoroughly inspect the compressor.

4. Compressors shipped assembled require no adjustment of the parts before starting because they are operated previous to shipment under full speed and air pressure, and are properly adjusted before leaving the factory.

However, the compressor should be carefully inspected to see that nothing is loose or has been tampered with during shipment or installation.

In addition to the instructions given in the chapter on Lubrication, the following suggestions should be noted:

Starting the Compressor.—1. Rotate or bar the compressor over a few times by hand to see that it works freely and that everything is clear. This is an important precaution as accidents have resulted by the starting of a newly erected machine without first having turned the compressor through at least one revolution.

2. When starting for the first time, or after overhauling or installation of new parts, run the compressor without any pressure for an hour or so.

If the compressor is to pump directly into the line, see that it is kept unloaded during this period, or open a valve in the discharge line so that no pressure will be built up.

3. Turn on a full supply of cooling water to the compressor cylinder jacket.

4. See that there is plenty of oil in the cylinder lubricator and that the oil in the bed plate is of proper height.

With force feed lubricators for the cylinder, break each of the lubricator piping joints nearest the cylinder and pump the lubricator by hand until oil appears at the broken joints. This shows a clear passage and insures oil at the cylinder just as soon as the compressor starts.

This procedure is recommended for starting after long shut downs. Usually after one day shut downs, pumping the lubricator by hand is sufficient, and after shorter shut down periods this is usually not necessary.

5. Start the compressor and make sure that the tops of the wheels rotate toward the cylinder.

6. On steam driven compressors, open the drain cocks on the steam cylinder and steam chest, and drain the steam pipe above the throttle valve until it is warmed up.

Then open the throttle very little and let steam blow into the steam cylinder until it is thoroughly heated up, turning the compressor over very slowly so that steam blows first into one end and then into the other.

When well warmed up, give it a little more steam and let it run slowly awhile, gradually bring it up to speed.

Regulating the Cooling Water.—A liberal supply of cooling water is needed to absorb the heat of compression. Otherwise this heat increases the power required to compress the air and renders cylinder lubrication less effective.

The actual amount of cooling water to be supplied to each jacket varies with different conditions of water temperature and compressor load. In general, sufficient water must be circulated through cylinder jackets so that the discharge water feels warm to the hand (approx. 100° to 110° Fahr.).

The intercooler requires two gallons of water per minute for every 100 cubic feet of piston displacement varying according to the temperature of the intake water. With sufficient cooling water the end of the intercooler connecting with the high pressure cylinder should feel cool to the hand.

If the water discharging from the cylinders be too cold, it may cause a condensation of moisture along the interior of the cylinder wall and impair lubrication.

As more water is required for the intercooler than should be used in the cylinders, the drain valve at the bottom of the cylinders can be opened slightly so that just enough water will be carried through the cylinder to keep the temperature of the water leaving the cylinder warm to the hand or approximately 100 to 110° Fahr.

Do not permit water to circulate through the water jackets and intercooler after the compressor has been shut down as moisture will form on the cylinder walls and destroy the cylinder bore polish.

Ques. What precautions should be taken after running without any pressure for an hour or so?

Ans. If the compressor show no signs of heating in the bearings or stuffing boxes, the pressure may be increased gradually to the working point.

Ques. What is the next precaution that should be taken?

Ans. Watch the regulators (and governors on a steam driven machine) and never leave the machine until satisfied that they control the compressor properly.

Make sure that the safety valve will start "blowing" at a pressure just a few pounds higher than the working pressure.

Ques. What should be done after the compressor has run for an hour or so and is thoroughly warmed up?

Ans. Go over the exterior of the machine and tighten any loose nuts. Gaskets squeeze up a little when new. Do not use too much wrench pressure on the valve cover set screws as this may spring the valve seats and cause leakage and excessive heat.

Ques. What next should be done?

Ans. Adjust the flow of cooling water through the compressor cylinder jackets, so that the outlet is just lukewarm (between 80° and 100° Fahr.).

Ques. What general attention should be given when the compressor is running?

Ans. The compressor should be watched closely at the start, particular attention being given to the crank case. Even though the compressor has been carefully cleaned, there is sure to be more or less dirt remaining which will be washed out by the circulation of the oil. All of this dirty oil must be removed.

The bottom of the crank case should be wiped out again, all sediment removed and a fresh supply of oil put in. This will remain clean for a longer time; but when it becomes too dirty, it should be removed.

Ques. What should be done with oil when removed?

Ans. The oil removed may be filtered and used again.

Always pour oil over the bearings and fill trough and oil boats where furnished before starting compressor after the crank case has been drained and cleaned or after a long shut down.

Ques. How often should the oil be removed?

Ans. Once the compressor is in operation and the interior thoroughly cleaned in accordance with the foregoing instructions, it should not be necessary to change the oil more often than once a month.

Compressor Operating Suggestions

All Compressors.—1. Drain the receiver frequently and regularly. An accumulation of oil in the receiver may cause an explosion of oil vapor. A trap makes an ideal way of automatically draining the receiver.

2. Test the safety valves on the receiver and other places occasionally by means of the hand lever or by raising the air or gas pressure to the proper blow-off point.

3. Keep a supply of cool water flowing continuously through the water jacket of the air cylinder while the compressor is running.

4. Keep the piston rod packing well set up, but not so tight as to cause excessive friction or score the rods.

5. Keep compartment between piston rod stuffing boxes well drained.

6. Usually 1 to 5 drops of oil per minute are sufficient for lubricating the compressor cylinder. Too much oil will form carbon on the valves.

7. Maintain a constant level of good clean oil in the crank case.

8. Have regular times for inspecting the lubrication system, inlet and discharge valves, also for cleaning the interior of the compressor cylinder and water jackets.

9. Inspect bearings occasionally, keeping them properly adjusted. Do not let a bearing pound.

10. Keep all bolts tight.

Gaskets will squeeze up a little with time. Do not use too much wrench pressure on valve cover set screws, as this may spring the valve seats and cause leakage and excessive heat.

Belt Driven Compressors.—For machines connected by belt drive, note the following suggestions:

1. Inspect belt regularly.

2. Keep belt surfaces clean.

3. Apply dressing to leather belt which is too dry.

4. Apply proper amount of dressing. Enough dressing in winter is too much in summer.

5. Take grease out of belting which is too oily.

6. Do not expect too much from a leather belt when subjected to temperature in excess of 100° Fahr.

Steam Driven Compressors.—Note the following with this type of drive. 1. Drain all low places in steam lines. Provide automatic traps. Never connect a trap to more than one point as a slight difference in pressure will prevent draining more than one point.

2. Drain steam cylinder by opening drains as soon as the machine is stopped.

To Shut Down Compressor.—First read chapter on Regulating devices.

After shutting down, shut off the cooling water, and on steam driven machines open the steam cylinder and steam chest drains.

If the compressor is to be idle for some time in freezing weather, be sure there is no leak of cooling water or steam into the cylinders.

Extra precautions should be taken in freezing temperature because a cylinder or head will surely crack if water be allowed to freeze in it.

If the shut down be a long one, it is advisable to remove packing from the stuffing boxes as it is apt to corrode and pit the piston rod when packing is left in the boxes.

CHAPTER 54

Maintenance

By definition, maintenance is: *The act or process of keeping anything, especially machinery, in a state of efficiency and good order.*

The term maintenance broadly comprises:

1. Inspection.
2. Cleaning.
3. Care.
4. Adjusting.
5. Renewing.
6. Repairs.

The successful operator must not only understand the necessary conditions of working and control, but he must know how to meet the numerous disorders and mishaps that may be encountered as those arising from:

1. Faulty construction.
2. Careless or ignorant handling.

Disorders arising from careless or ignorant handling are those due to:

1. Insufficient lubrication.
2. Faulty adjustments.
3. Racing.
4. Overheating.
5. Wear or breakage of parts.

INSPECTION

Regular inspection should be made at definite intervals at which time any necessary corrections may be made, such as replacement of worn parts, new packing, adjustment for wear of working parts and cleaning air valves and crank case.

Every Day.—Check oil level in crank case and lubrication.

CLEANING

Machinery should be clean to begin with—and kept clean. Dust and almost any foreign matter contains abrasive particles which coming into contact with any of the working parts accelerates wear, resulting in lost motion, loss of compression, noisy operation, etc. It is of extreme importance that everything be kept immaculately clean.

Every 100 hours of operation.—Drain the sludge from the bottom of the oil filter by removing the drain plug. Air pressure may be applied at the top of the filter to help blow out the sludge. A fitting is provided to receive a standard tire fitting chuck.

Every 500 hours of operation.—Drain the crank case. While the oil is drained, clean the strainer (located in the bottom of the crank case) and wash out the bottom and corners of the crank case with kerosene or gasoline to remove any sediment that may have settled.

The strainer can be cleaned by disconnecting from the copper tubing and removing the strainer. Refill the crank case with new oil. Check the valves for valve and valve spring wear, carbon formation, etc.

The presence of any deposit on the valves indicates that either the intake air is dirty or that too much oil or unsuitable oil is being used.

All ports and passages should be examined and any obstructions such as carbon and sticky oil removed.

Clean the filtering element of the oil filter—remove top nut, outer shell, and lift out filtering element. Scrape the sludge from the element and wash in kerosene or gasoline.

Clean the oil cooler. Remove the water lines to the cooler, and then remove the reducing bushings on each end by unscrewing from the cooler.

Remove the cooler from the frame of the compressor.

Wash the cooler in kerosene or gasoline to remove sediment in the cooler. If the tubes show signs of scaling they should be cleaned by pushing a rod or wire brush through the tubes.

These recommendations as to time are considered as average.

In some cases a closer check must be maintained and in others a longer time may be allowed between checks. On installations where the unit is in an ideal location a good grade of oil used, and the filters cleaned regularly, the oil can be left in the unit for periods of 2,000 hours or longer. A check by the operator should determine the proper periods for cleaning and changing oil and oil filters.

CARE

The term care is here used in the sense of maintenance, that is, in the constant running of an engine certain supplies such as:

1. Oils.
2. Packings.
3. Gaskets, etc.

must be kept on hand and applied and adjusted at the right time, and with the right technique. Much has been said on such items in other parts of the book and no repetition will be made here, but instead additional instructions will be given.

ADJUSTING

Natural Wear.—No machine as yet has been invented which will operate without adjustments or occasional repairs. Friction imposes wear that even the best lubrication cannot eliminate, and breakdowns from many and varied causes are inevitable. Careful handling will prevent most shutdowns. A needed minor repair may soon develop into a major repair. Putting off until

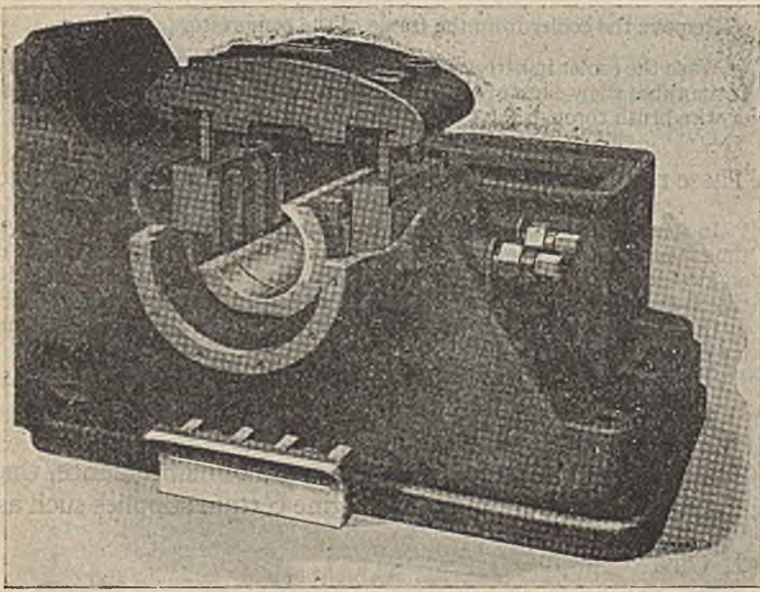


Fig. 1.—Main bearing. In this design the main bearing can be removed without disturbing the shaft or rotor.

to-morrow what should be done to-day has caused many an engine compressor to be wrecked or to wear out before its time.

Ques. Which are the major wearing parts of a compressor?

Ans. 1, Piston rings; 2, cylinder or liner; 3, valves; 4, piston rod.

The piston also receives wear, but in most cases the rings take the brunt of the wear. The valve plate cover cap screws and other bolts and nuts receive wear not from operation but from taking out and replacing.

The close observation of the compressor is essential to see that the worn parts do not become too worn and cause additional damage.

Adjustment of Main Bearing.—For illustration, consider the type shown in fig. 1. Here the main bearings consist of three boxes, two side and one bottom, and may be adjusted by means of the two large set screws in the bed plate. This adjustment is important and calls for good judgment.

Ques. What is the first thing to do?

Ans. Take up on adjusting screws uniformly and tight, then back off set screws a half flat and hold adjusting screw and tighten jam nut.

Ques. What is the result of this adjustment?

Ans. It will give a somewhat loose bearing and there should be a slight pound when the machine is first started up.

If bearing fail to become quiet after machine is warmed up from two or three hours running, take up very slightly on adjusting screws, being careful to take up the same amount on each screw. After any such adjustment watch bearing carefully to make sure it does not heat up.

Ques. What precaution should be taken?

Ans. It is far better to make two or three slight adjustments to get proper bearing clearnace than to pull bearing up too tight and burn it out.

Ques. What is the procedure when it is necessary to inspect or renew the boxes?

Ans. Take off the caps, loosen the set screws and slide the oil ring along the shaft. Jack up the shaft just enough to remove pressure from the bearing. The side boxes can then be removed sideways, turned and lifted out. The bottom box should be moved in the same manner until the ribs line up with slots in the bed plate, then turned around the shaft and lifted out.

There is a plate in front of quarter box against which the set screws bear. This plate is to fill up the gap between bed plate and bearing box so that if set screw should get loose no excessive movement of shaft can occur, and damage motor windings. (This is a direct connected electric driven machine being considered.)

Ques. What should be placed between the main bearing cap and the bearing jams of the bed plate?

Ans. Liners.

Ques. Describe the adjustment.

Ans. The bearing cap should be adjusted by means of these liners so that there is about 0.004 in. clearance between the cap and the top of each side box.

A main bearing for a large machine is shown in fig. 2. Referring to the illustration, positive lubrication of the bearings is provided through oil basins in the top box of each bearing. These are filled from the frame lubrication system.

Sight cups outside the machine show the oil flow to these basins, and removable cover directly over the bearing provides an additional means of inspection.

Adjusting Roller Type Main Bearings.—On the machine being considered, the main bearings are over size, double row, Timken tapered roller bearings, which are carefully and properly set up before shipment. If unusual operating conditions necessitate adjustment of the main bearing, proceed as follows: Take off wheel at the bearing to be adjusted. Remove the stamped steel oil guard and gasket and remove the bearing cap.

The adjusting nut is now wholly accessible. Loosen the clamp bolt and by means of a bar of steel, a hammer and the cast notches in the nut itself, it is very easy to tighten up the bearings.

A fine thread on the nut permits close adjustment.

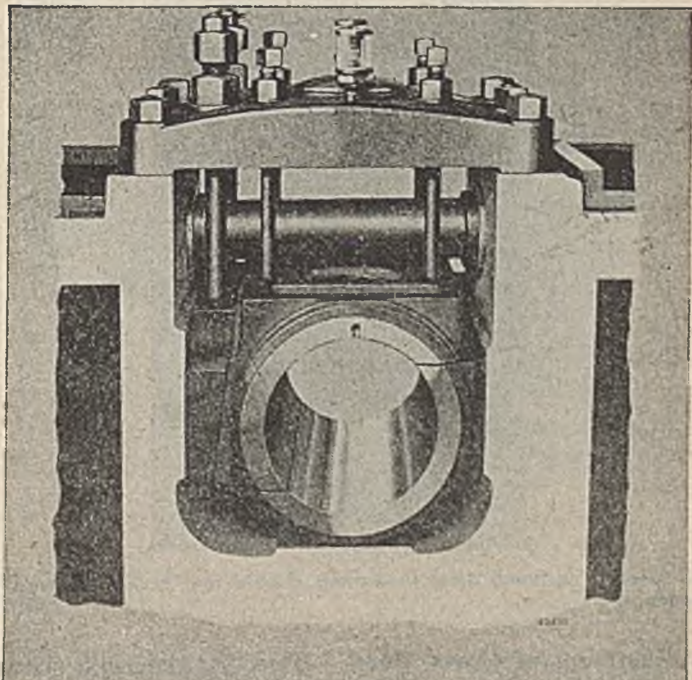


Fig. 2.—Large compressor main bearing. Bearing adjustments are made by means of a slow taper wedge moved by pull bolts and locked by jack screws outside the bearing cap. The bearing permits additional adjustments in all directions.

Ques. How is it known when the bearing is properly adjusted?

Ans. It is properly adjusted when there is no perceptible end play and the compressor turns over quietly and freely by hand.

Fig. 3 is a sectional end view which illustrates the double row Timken tapered roller bearings.

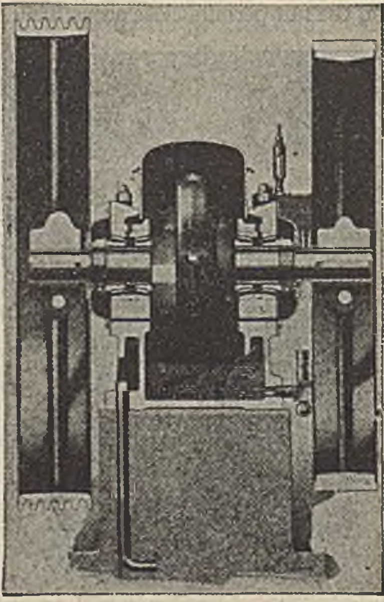


Fig. 3.—End sectional view illustrating double row Timken tapered roller bearings.

Adjustment of Cross Head.—Consider the type shown in fig. 4. It is as seen, of the box pattern, made of cast steel with wedge adjusted top and bottom shoes.

Ques. What is the regular name for shoes?

Ans. Gibs.

Ques. How is the cross head shown in fig. 4 adjusted?

Ans. Loosen tap bolts A and A, and then turn adjusting screws C, which will move wedges and force shoes outward. When in correct position, be sure to draw up all tap bolts tight.

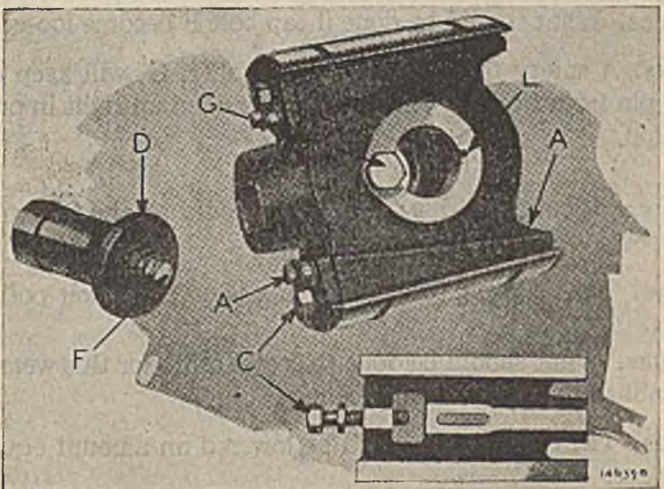


Fig. 4.—Cross head details. Box type with wedge adjustable gibs.

Of course this does not mean to take a six-foot wrench and strip the threads.

Ques. What should be done before cross heads have been worn in?

Ans. Care should be taken to allow enough clearance to keep the gibs from heating.

Ques. Describe the clearance setting.

Ans. Start with about .012 in. clearance and when the bearing surfaces have been worn in, take up the clearance to approximately .008 in.

The cross head pin here shown is hardened, ground and held in cross head by tapered fit with washer D, tap bolt and lock washer, and kept from turning by pin in slot E.

Ques. What should be done if tap bolt F become loose?

Ans. A safety tap bolt and lock washer, G, will keep cross head pin from working out until pin is pulled up tight in proper position.

Ques. What happens when wear occurs at the piston and cylinder wall?

Ans. The piston naturally sets in a relatively lower position.

Ques. What should be done to compensate for this wear and reestablish the original alignment?

Ans. The cross head should be lowered an amount equal to the wear.

Ques. How is this adjustment made?

Ans. First loosen tap bolts and back off wedge at bottom until piston rod is level, then pull in wedge at top to give correct clearance in cross head guide. Be sure tap bolts are tightened up securely after adjustments have been made.

When the adjustment has been completed, check the piston rod to make certain it runs true by placing an indicator on it and barring over the compressor a few turns.

Adjustment of Connecting Rod.—The connecting rod here considered is of the solid end type with wedge and screw adjustments as shown in fig. 5.

There are two adjustments:

1. Crank pin bearing.
2. Wrist pin bearing.

On the connecting rod is a scoop rolled into the bottom of it which dips into the oil with every revolution and forces the oil into the crank pin bearing.

With this arrangement it is necessary to keep the oil in the bearing and, therefore, there are shims between the two halves of the bearing.

Referring to the illustration (fig. 5), the wedge part 49 now acts as a lock for the bearing and maintains a fixed clearance between the bearing and the crank pin.

This clearance is set at approximately .003 in. when the compressor leaves the shops and with this clearance the machine will run a long time before it becomes necessary to take up.

Ques. Explain how to reset the bearing.

Ans. Remove the cotter pin and the castle nut 50A. Then take out the wedge and turn crank shaft so that the wedge half of the box moves forward to take up the space created by the removal of the wedge. This leaves space to take out the shims between the bearing halves. The shims are made up of .002 in. laminations and, therefore, it is possible to make a close adjustment. In peeling off shims be sure to peel off the same number of laminations from both shims.

Ques. How is the bearing reassembled?

Ans. Dip shims in heavy grease and push against back half of crank pin box, the grease will keep the shims in position while the front half of the box is being pushed back into position. Insert the wedge and lock the bearing assembly by pulling up tight on the wedge bolt and putting cotter pins in place, maintaining .003 in. running clearance in the bearing.

The crank pin box can be replaced without touching the main bearings or removing the connecting rod. Remove nut 50A and wedge bolt 50, from outer end of rod, which will allow crank pin box wedge 49, to be taken out at the side, the crank having been placed on the forward or outer center.

The outer crank pin box can then be drawn forward from pin and taken out at the side.

Turn the crank in the direction of rotation about one third of a turn, to push the connecting rod up and back, and then give the crank a slight reverse rotation to bring crank pin toward the outer center again and separate the crank pin from the rear crank pin box. The latter can then be removed by drawing forward slightly and taking out at the side.

Ques. Describe adjustment of the crank pin box in the connecting rod.

Ans. Caution. This adjustment is simple to make, but it must be done carefully. A loose bearing will knock and a tight bearing will heat up.

The correct way to adjust the crank pin bearing is first to remove the cotter pin 50B, and loosen the castle nut 50A. Then turn wedge bolt 50, by means of wrench applied at the head of the bolt 50 on the under side of the connecting rod, and pull up until the bearing grips the crank pin.

Back off the wedge bolt one quarter turn and tap the wedge bolt to free wedge. This will finally give proper running clearance.

Pull up castle nut to lock position of wedge at the same time hold bolt with wrench to prevent wedge bolt turning and insert cotter pin.

Ques. Describe how to adjust the wrist pin bearing.

Ans. It is adjusted by means of wedge 52, and set screw 53. Lock nut 53B, is first loosened, so that screw 53, can turn in wedge 52. Screw 53, bears on lower wall of connecting rod end so that screwing it in raises wedge 52, and takes up bearing clearance.

To adjust the bearing correctly, the set screw should be screwed in until the bearing grips the cross head pin, then back off set screw one quarter turn and free wedge by tapping set screw. This procedure, if followed will provide the correct running clearance.

Ques. What should be done after adjusting the wrist pin bearing?

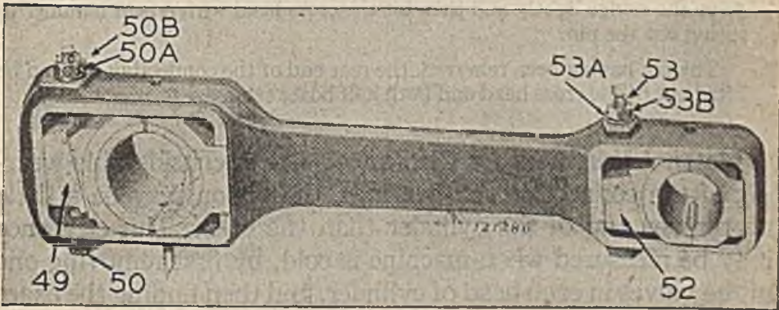


Fig. 5.—Connecting rod of the familiar type having wedge adjusted bearings at both ends.

Ans. After adjusting, lock all parts with lock nut 53B. There is a sleeve 53A, between lock nut 53B and wedge 52, enabling set screw 53, and wedge 52, to be locked together by nut 53B.

Ques. How is the wrist pin box removed while the connecting rod is in position in the machine?

Ans. Loosen lock nut 53B, and take out set screw 53, thereby freeing wedge 52. Take off the wrist pin hole cover from the side of the bed.

Remove the washer, screw the tap bolt in the pin again, and then move the cross head so that by unscrewing the tap bolt the latter will jam against a rib so provided in the side of the bed plate.

Put a light strain on the tap bolt by unscrewing it. Then hit the cross head with a lead hammer, causing the wrist pin to spring loose. The pin may be removed through the hole in the other side of the bed plate.

On some smaller compressors the wrist pin is secured by a washer and bolt running through the pin.

To remove the pin on such machines, first remove the bolt and washer.

Place a nut or similar object on the end of the wrist pin opposite the hole in the frame and lightly drive the wedge (used in removing the compressor wheels) between the nut and the rib on the side of the frame.

Do not hit the wedge with the purpose of driving out the pin. Simply drive the wedge lightly and then hit the cross head with a lead hammer to spring out the pin.

The pin having been removed, the rear end of the connecting rod can be lifted out of the cross head and both half boxes removed.

Piston Adjustment for Clearance.—After considerable wearing of the connecting rod bearings, the piston may be a little nearer one end of the cylinder than the other. This clearance may be measured when machine is cold, by first removing one of the valves in each head of cylinder, and then finding the exact clearance by barring over compressor, using lead wire between piston and inside face of cylinder head which will be flattened out to amount of clearance when piston has been moved to end of stroke.

The piston rods may then be screwed in or out to adjust clearance by loosening piston rod jam nut and screwing back to end of thread. Then with special cross head jam nut wrench move piston rod in or out as may be required. Never use wrench or similar tool on the piston rod.

Clearance at the frame end should be about 1/64 in. less than at the back end. After proper adjustment lock piston rod jam nut against the cross head.

Piston Rod Oil Wiper Rings.—On some machines the stuffing box in the partition between crank case and frame head yoke contains a set of metallic wiper rings of soft metal. These rings are split and held to the piston rod by means of garter spring. The rings have a special edge for wiping the oil off the piston rod and have grooves for draining. The oil then flows back into the crank case.

The oil wiper rings are designed to wipe the rod practically dry, leaving just enough oil film to lubricate the cylinder packing.

Ques. What attention should be given to these rings?

Ans. Clean the wiper rings periodically.

Nicks or dents in the rings or a scored rod prevents a tight seal.

Ques. What should be done in the case of broken springs?

Ans. They should be replaced at once.

Ques. How are wiper rings assembled?

Ans. In assembling, stagger the joints.

Ques. What should be done to newly installed wiper rings?

Ans. They should be spotted to the piston rod to insure them hugging the rod.

Fig. 6 shows typical design of partition plate oil wiper rings. Note that there is only one correct way to install these rings. Refer to the illustration. The total end clearance of the wiper rings should not be more than .003 in. More clearance will cause the rings to act as a pump instead of wiping the oil from the rod.

Ques. What provision is made with force feed lubrication?

Ans. The partition plate stuffing box is fitted with tangential cut, cast iron oil scraper rings. These rings are specially designed to take care of the larger amount of oil carried by the piston rod on machines equipped with force feed lubrication.

Ques. How should partition plate and scraper rings be installed?

Ans. As shown in fig. 7.

Servicing Stuffing Boxes.—The reader should review the extended treatment given to packings and stuffing boxes in

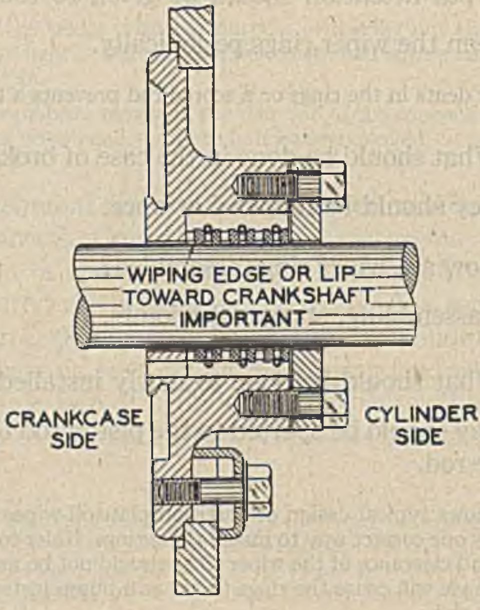


Fig. 6.—Partition plate oil wiper rings.

the first section of this book, as there will be no duplication here.

The compressor cylinder front head stuffing box seals the rod against air or gas leakage into or from the cylinder. On standard air compressors this box, shown in fig. 10, is packed with a number of rings of fibrous packing.

A sufficient amount of packing is furnished for this box and any others that are to be so packed. This packing is cut to proper lengths, and the packing for each stuffing box is wrapped in a separate package. If the labels be preserved, additional packing can be ordered by giving the dimensions or other markings on the package.

Pack the box as follows: 1. Move back gland 71 (fig. 10).

2. Make sure that bushing 155 is located at the inner end of the box.

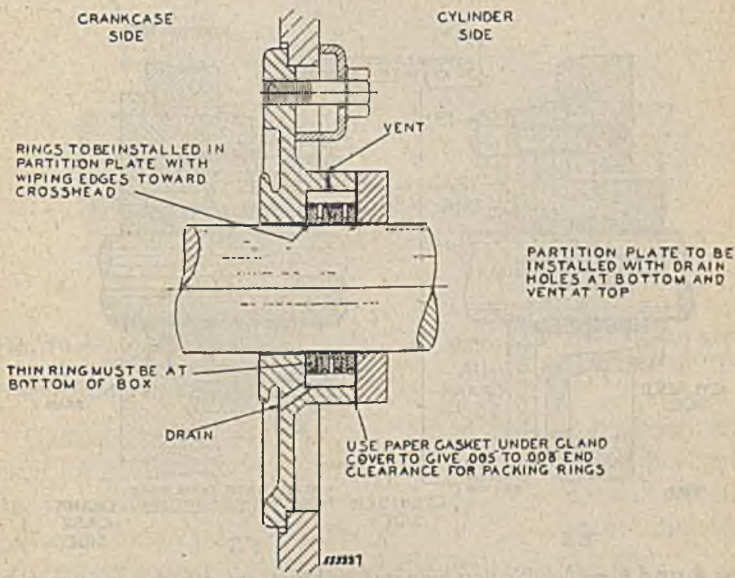


Fig. 7.—Partition plate oil wiper rings for force feed lubrication.

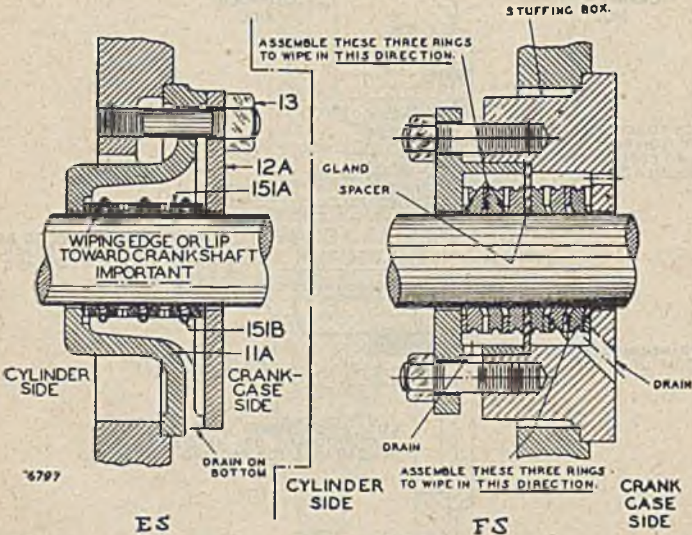
3. Dip each ring of packing into a good grade of crank case oil and insert one ring at a time, pushing it well into place. The joints of succeeding rings must be staggered.

4. Put the gland 71 in position and draw up evenly on the gland bolts until they are snug, then back off the nuts and tighten them again with the fingers only.

5. Always adjust the stuffing box glands while the compressor is running. Tighten the nuts only enough to prevent leakage. Further tightening results in undue friction, wear on the packing and possible scoring of the rod. Keep the nuts set up evenly so that the gland is square with the rod.

6. When a stuffing box leaks excessively after wear, and leakage cannot be stopped by pulling up lightly on the gland nuts, repack the box with new packing.

On steam driven machines attention must be given to steam cylinder and steam chest stuffing boxes. On such machines



Figs. 8 and 9.—Metallic ring type stuffing box located in the partition plate where the piston rod passes through the crank case. On some gas compressors this may be of the soft packing type, as shown in fig. 10.

where the piston rod passes through the steam cylinder heads, and where the valve rod passes through the steam chest cover, the stuffing boxes are packed with fibrous packing similar to the air or gas cylinder front head box.

Ques. How is the packing applied to the box?

Ans. 1. Move gland back. 2. Dip each ring of packing into a good grade of crank case oil and insert one ring at a time, pushing it well into place. The joints of succeeding rings must be staggered. 3. Put the gland in position and draw up evenly on gland nuts until they are snug, then back off the nuts and tighten them again with the fingers only.

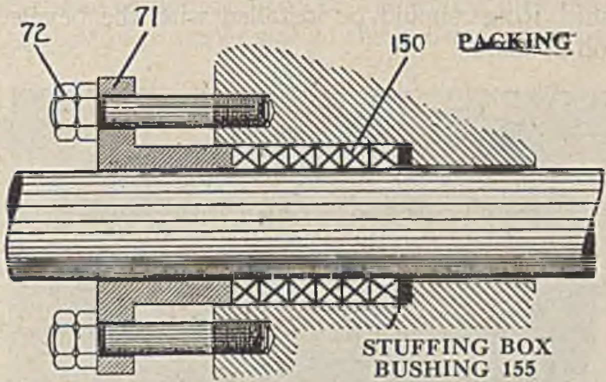


Fig. 10.—Stuffing box for air cylinder of single stage compressor. This general construction is used for steam cylinders, some two and three stage compressor cylinders and sometimes for the crank case partition plate.

Gaskets.—Any graphited asbestos body sheet packing can be used for gaskets between air cylinders and heads. Rubber packing is not satisfactory as the heat and oil in time will soften it. Material for air head gaskets should be one-sixteenth inch thick to provide proper clearance between piston and heads.

The gaskets under the valve seat and also under the cover are of special type consisting of a copper covering over an asbestos core. This makes an ideal gasket and can generally be used over many times. Should they be damaged and others not available, substitutes can be made out of soft copper wire with ends soldered, or of ½ in. cord packing.

If gaskets be irregular in size, fit over valve seat if tight inside; and into recess in cylinder head, if tight outside.

Heavy grease or shellac is a convenient substance if there be anything wanted to stick gasket in place. It is advisable to carry a complete extra set of gaskets obtained from the compressor manufacturer. They are inexpensive and much better than any substitute.

Oil Sealing Rings.—Split ends covers on the frames are fitted with split oil sealing rings to prevent leakage of oil along the main shaft. Rings should be installed with the beveled joint at the top of shaft.

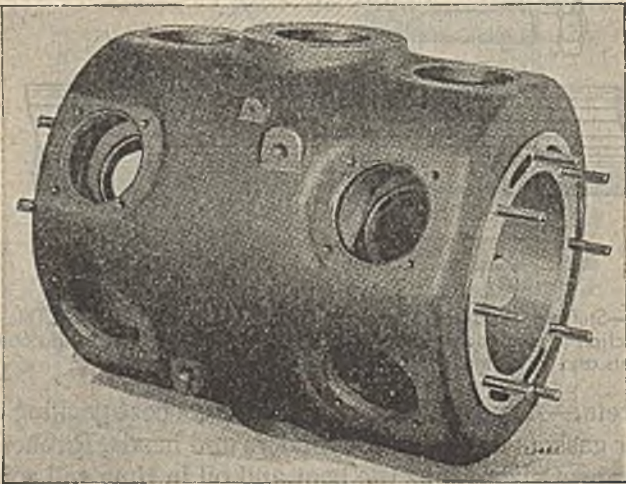


Fig. 11.—Compressor cylinder with valve passages at ends.

To insure proper application and to prevent damage to the sealing lip of the ring, the installation should be accomplished in the following steps:

1. Oil seal rings should not be heated before application.

Preferably they should be installed at room temperature (70 to 80° Fahr.).

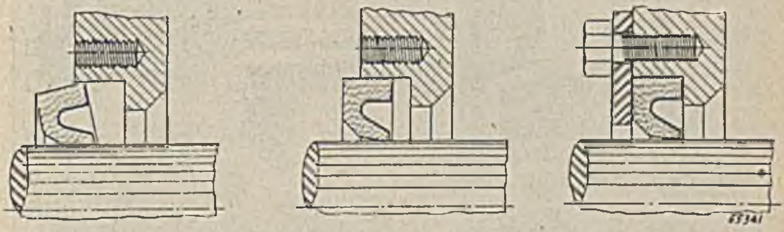
2. See that recess is thoroughly cleaned and that all burrs and sharp, cutting edges are removed.

3. Apply grease or oil to the shaft.

4. Place the ring around shaft at point near the recess into which it is to be installed.

5. Compress the ring slightly by squeezing the sealing lip to the body of the element.

6. Start the compressed end of the sealing element into the recess at the top or upper side of the housing, inserting the lip



Figs. 12 to 14.—Installation of oil seal rings.

first, as shown in fig. 12, and gradually work the heel or back of the element into place, keeping the seal compressed while so doing.

7. Continue this process around the entire periphery of the shaft until the sealing element is inserted in the recess as shown in fig. 13.

8. Seat the element in the recess, as shown in fig. 14 by tapping lightly, using care to prevent damage to the lip.

9. Apply cover plate, bolting tightly into position to compress oil seal ring into recess.

10. It is important that oil sealing rings should not be too tight on shaft as this would cause excessive heating.

If rings be found to be too tight, they should be relieved slightly with a smooth file.

Piston and Rings.—On machines 14 to 18 in. stroke, typical construction is to have the piston a press fit on the piston rod, being locked by a recessed round nut. In other designs the piston is a taper fit on the rod, being held by a large nut which is locked with a set screw.

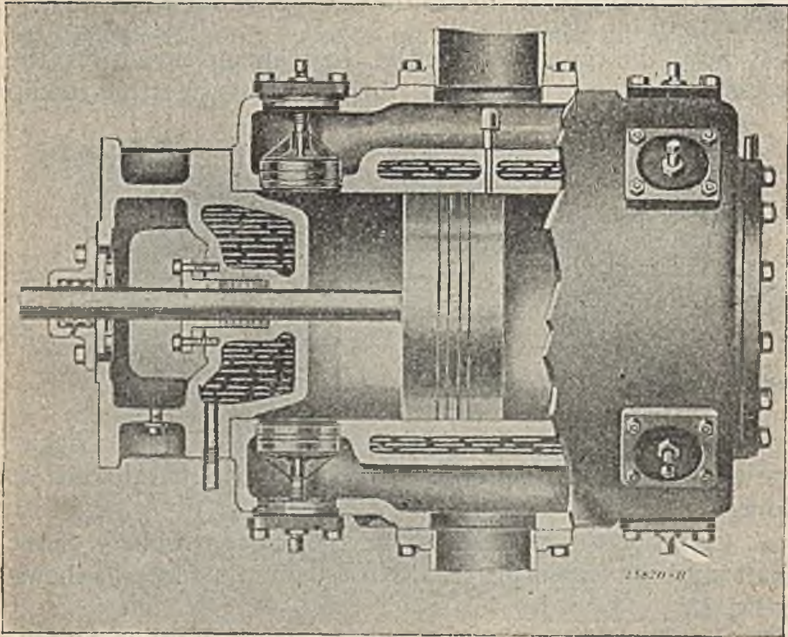


Fig. 15.—Compressor cylinder with valve passages in the cylinder casting.

The piston rings if of the three-piece type will include two outer sealing rings and one inner or spring ring. One outer ring has a right-hand angle joint and the other has a left-hand angle joint.

Ques. What should be noted about piston rings?

Ans. They tend to rotate when in operation.

Ques. In this connection what is the important procedure in installations?

Ans. When installing make sure that each three piece ring has one right hand and one left hand outer ring so rings will rotate in opposite directions and thus avoid the possibility of the splits lining up and allowing leakage through the joint.

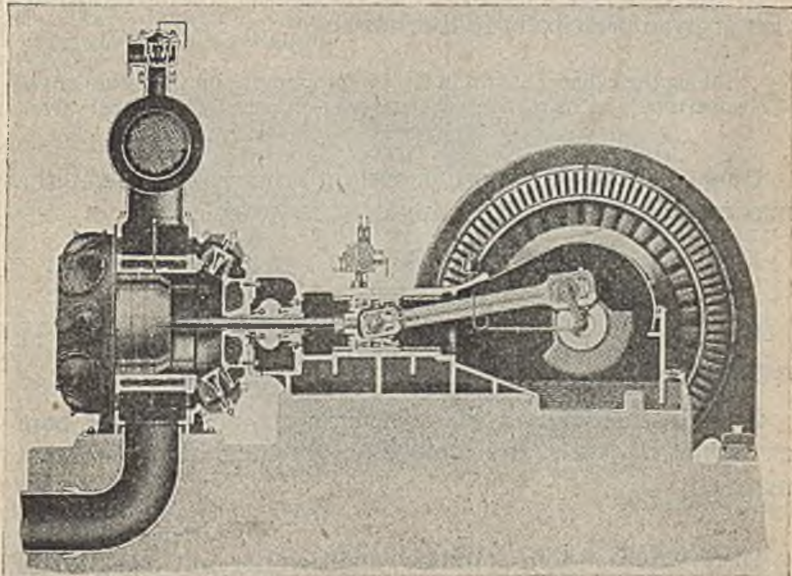


Fig. 16.—Cross sectional view of cylinder with valves arranged radially in barrel.

Care and Adjustment of Compressor Cylinders.—The cylinder with its valves, piston, piston rod and heads, represents the heart of the compressor and, therefore, should be periodically inspected. There are numerous designs of air cylinders and the service procedure will depend in part on the particular cylinder being serviced.

The valves may be in the heads or at the ends of the cylinder. The accompanying illustrations show variations in design.

Considering for instance a cylinder of the type shown in fig. 16 with valves in head, it is necessary only to remove the cap nuts on the back head to remove the cylinder head.

In the care and adjustment of compressor cylinders, attention is given principally to the valves.

Taking the cylinder shown in fig. 15, for example, its valves are of the channel type. The instructions following relate principally to channel valves.

Ques. What is the most important requirement to get the maximum efficiency out of any compressor cylinder?

Ans. The admission and discharge valves must be tight.

Ques. What inspection and attention should be given at regular intervals.

Ans. Inspect the valves, valve passages, and cylinder bore at regular intervals, and remove any accumulation of foreign matter.

Dirty Valves.—Valves in this condition not only cause a loss in capacity of the compressor, but can become the source of other troubles.

These troubles can be eliminated by a periodic check of the valves and valve passages.

No definite time for cleaning the valves can be given as this must be judged by the operator.

When the compressor is first started, try checking the valves once a month and if found to be particularly dirty, locate the cause.

A few causes of dirty valves and the remedy for each are listed here:

1. Dirty intake air.

   Remedy: Install air intake filter.

2. Excess oil or the improper grade.

   Remedy: Consult oil company for correct compressor oil and feed only the amounts recommended.

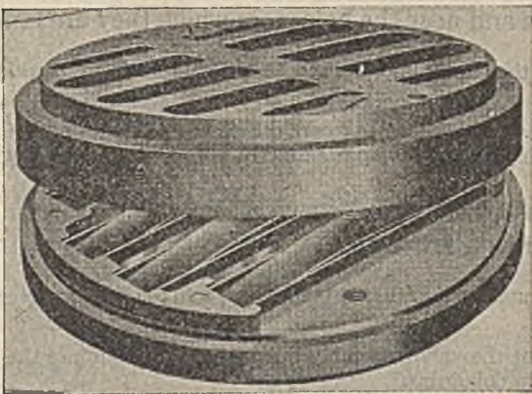


Fig. 17.—Channel valve with stop plate raised to show valve channels and valve springs in place.

3. High temperature of the air.

   Remedy: Check cooling water circulation. Check valves for leakage.

To Remove Valves from Cylinder.—In removing valves from the cylinder the same method may be used for each type. Each type valve assembly, including crab or free air unloader assembly when used, is held in place by the valve cover set screw.

1. Shut down the compressor and lock the switch if possible

(or shut off steam) so the compressor cannot be started accidentally before work is completed.

2. Close the valve in discharge line (to receiver).

Blow off air in the inter cooler of two-stage compressors and between the compressor discharge and valve in discharge line.

If the compressor have free air unloaders, the pressure must also be released from the regulator.

3. Obtain a piece of chalk and as the valve assemblies are removed from the cylinders place corresponding marks on the assemblies and near the hole from which they are removed.

Later they must be returned to the same hole from which they were taken.

When taking the valve apart use plenty of chalk marks to assure the re-assembly of the valve parts exactly as they were originally.

4. Loosen the valve cover set screw lock nut and unscrew the set screw a few turns.

Then unscrew the four tap bolts which hold the cover in place, and remove the cover.

Lift out the valve assembly but be sure not to injure the ground-joint surfaces in doing this.

When taking out an inlet valve, which is assembled with the free air unloader, take care that the free air unloader assembly does not fall out. Lower the cover slowly so that the unloader assembly will follow with it. If the gaskets be not damaged they can be used again.

Cleaning Valves.—Any dirt or carbon in the valve ports can usually be cleaned out without taking the valve apart. Brush the valve parts carefully so as not to damage the seating surfaces. Rinse the valve thoroughly in kerosene or safety solvent to remove the loose particles. Soaking the valve overnight in kerosene oil or safety solvent followed by a stiff brushing or light scraping will remove all carbon.

Ques. What precaution should be taken after the valve is thoroughly dried?

Ans. Replace it in the same hole from which it was removed.

Ques. What precautions should be taken with respect to the cylinder?

Ans. Never use kerosene, gasoline or coal oil in air cylinder to clean it out. This is a very dangerous practice and should be prohibited.

Instructions for Various Valves.—The following paragraphs give separate instructions for the correct assembly for each type of valve.

Channel Valves.—The channel valve complete consists of the seat, stop plate, channels and the springs. All parts are the same for the inlet and discharge valves except for drilling (and consequently the assembly) of the seat and stop plate.

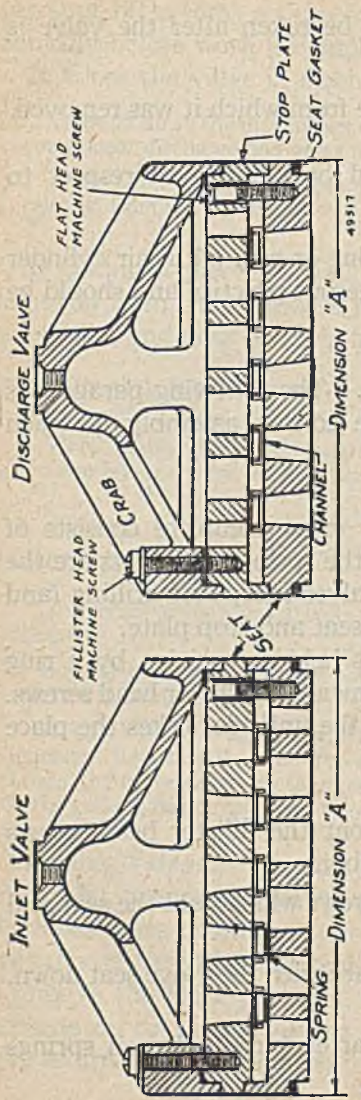
Normally the valve assembly is held in position by a ring type crab fastened to the valve by means of fillister head screws. Where a free air unloader is used, the unloader takes the place of the crab.

To take valve apart—1. Take out the fillister head screws and lift off crab or unloader assembly.

2. Take out the two flat head screws which hold the seat and stop plate together.

3. Lay valve on a clean flat surface with the valve seat down, as shown in fig. 17.

4. Turn the stop plate one quarter turn to loosen springs and then lift it off.



Figs. 18 and 19.—Channel valve assemblies for inlet and discharge.

5. Remove each spring and its channel separately, clean them, and then place them exactly as taken out without turning end for end.

In this way the original seating surfaces will be maintained.

6. The joint between the seat and stop plate is machined smooth and must not be damaged in handling.

When reassembling inlet valves, as a further precaution against leakage, a thin coating of red lead or similar sealing compound should be applied to this joint.

Any sealer should be allowed to set thoroughly before running the compressor.

7. Replace the stop plate and flat head screws and tighten up.

Replace the crab or free air unloader assembly and the fillister head screws. See that lock washers are under the fillister head screws and that the screws are well tightened.

8. Press each channel separately back against the stop plate to make sure each works freely.

To Replace Valves in Cylinder.—Before replacing valves in a cylinder, inspect the valve seat gasket and also the valve cover gasket. If the gaskets be damaged, replace them with new ones. These are special gaskets consisting of a small copper covering over an asbestos core. This makes an ideal gasket and generally can be used many times.

Should they be damaged and others not available, substitutes can be cut of 1/16 in. sheet asbestos packing. This, however, should be used only in case of emergency. It is advisable to keep one or two standard seat and cover gaskets in stock at all times. The type of valves here shown do not need to be ground in. When new, there is apt to be a slight leakage at first, but after running a short time, the valves become tight and remain so through their life.

In replacing valves in cylinder, proceed as follows:

1. Put each valve assembly in the same cylinder hole from which it was removed, place the valve so that it will rest squarely on the cylinder seat gasket.

In putting channel valves in the cylinder turn the valve assembly so that the valve ports are parallel with the piston rod.

2. Place the valve cover on the cylinder, being sure its gasket is squarely in place, and draw down the opposite cover nuts evenly and in turn so as not to tilt the cover.

3. Tighten down on the valve cover set screw (this should require about two turns) drawing it snug to hold the valve on its seat.

To prevent leakage along the thread, a turn of solder or fuse wire is placed around this screw and set down into a recess around the thread by means of the set screw locking nut.

4. To aid in assembling a channel valve in the bottom half of a cylinder when free air unloaders are not furnished.

A special rod threaded on one end is furnished to slip through the hole of the hollow set screw. Screw the rod into the valve crab which is tapped for the rod. Raise the valve into position in the cylinder and hold it there temporarily by means of the rod.

Then slip the cover set screw (with its cover) over the rod and into place while still holding the valve in position by means of the rod. Bolt on the cover, screw the set screw against the crab, then unscrew the rod and remove it.

Replace the pipe plug in the end of the cover set screw.

5. After all valves have been replaced, turn the compressor over one complete revolution by hand to see that everything is clear.

6. Open the valve in the discharge line and the one in the line to the regulator. Start the compressor in the regular manner.

7. After the compressor has run a few days the valve cover set screws should be examined and tightened a little if necessary.

Sometimes the valve seat gasket will squeeze up somewhat when first put into use, and if not followed up the seat will become loose.

Free Air Unloaders.—A complete understanding of the installation and operation of the free air unloader should be obtained by the operator in order to maintain an efficient regulating system.

To Remove the Free Air Unloader.—1. Disconnect the outside regulator connection to the cover set screw.

2. Remove the outside valve cover as previously described.

3. Remove the valve and unloader assemblies together.

4. Take out the two fillister head screws holding the unloader assembly to the valve assembly.

The only care which the free air unloader may require is possibly the replacing of the gasket or diaphragm at rare intervals.

General Instructions for Channel Valves.—The spring and channel widths are held to very close manufacturing tolerances to provide the correct clearance. This fit must be changed.

Channels and springs that are improperly made will not have this cushioning feature which is essential to the quiet operation and durability of the channel valve.

Refacing Worn Channel Valve Seats.—When the seats become worn or damaged to a point where the valve is no longer tight, they should be refaced to a smooth surface and new channels installed.

Of the various ways in which this facing may be done, the following is recommended to give the best results:

1. Remove the guides and dowels by lightly tapping a sharp edged flat chisel under the guide to loosen.

The dowels are a snug fit in the seat, but purposely left free enough for easy removal of the guide. Mark guides for replacing.

2. Using micrometers (or make gauge as shown in fig. 21) measure the distance from face of seat to shoulder A, on which stop plate rests.

This dimension is important later as it must be reestablished and it must be held to a close tolerance.

3. Place the seat in a lathe, being careful to grip the seat just tight enough to hold without causing distortion.

Remove enough stock from the face of the seat to provide a smooth surface at the port edges where the channel rests.

Take a light final cut and then use emery cloth (stretched over a flat, true plate) to remove tool marks.

4. Face off the shoulder A, on which stop plate rests to the micrometer reading (or gauge) so that this distance will be the same as when seat was originally made.

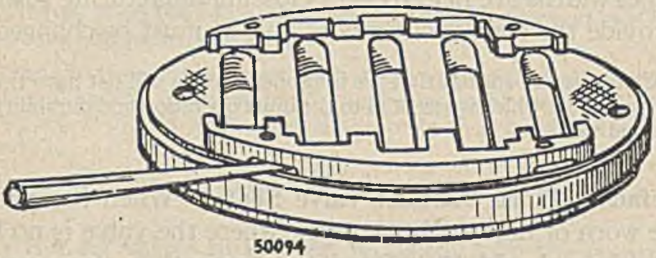


Fig. 20.—Lifting guide with flat chisel.

5. Place guides on the seat in the same position as before removal.

Drive in dowels to hold guide snug. If one of the dowels has been damaged, it can be replaced by a piece of drill rod or cold rolled steel of the same length.

6. Re-assemble valve using new channels.

The old channels, having become seated to a worn surface, will not be tight on the re-faced seat. The cost of new channels is very nominal.

When handling the valves, use care not to damage the seating surfaces. Before replacing parts make sure they are clean and smooth, as leaks at some of the joints will cause overheating, and cut down the compressor efficiency.

Type Y Valve.—In general, the type Y valve consists of a valve plate with guide arms riveted to it and held on its seat by several helical springs. Inlet valves, normally on the bottom of the cylinder, should be assembled in the cylinder so that the guard D, fig. 23, is toward the cylinder bore. Discharge valves should have the valve seat A, nearest the cylinder bore.

The valve assemblies are held in position by a central set screw passing through the valve cover. To prevent leakage along the thread, a turn of solder or fuse wire is placed around this screw and set down into a recess around the thread by means of the set screw locking nut.

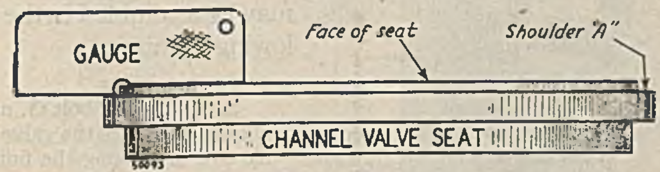


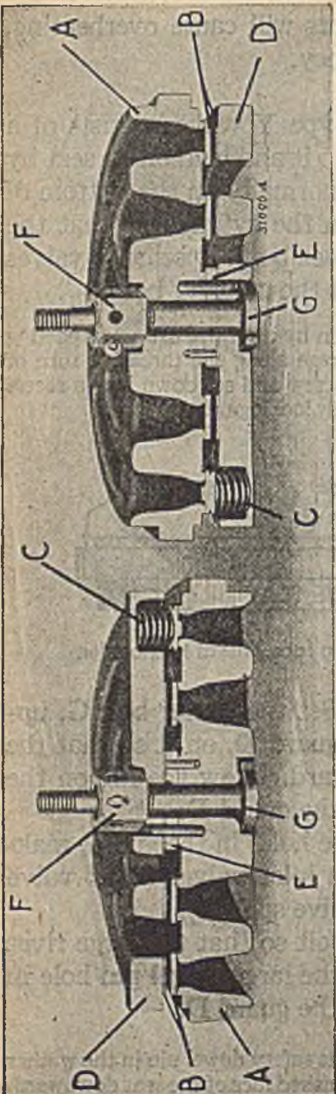
Fig. 21.—Gauge for maintaining distance from face of seat to shoulder.

To Assemble an Inlet Y Valve.—Set the center bolt G, upright on a table and place the cast guard D, on it so that the holes for the valve springs face upward. Allow the lug on the center bolt head to fit in the slot in the guard D.

Stand all of the valve springs in the holes in the guard, making sure that all springs are for the inlet valve. (Inlet valve springs are weaker than discharge valve springs).

Next slip the valve plate on the bolt so that the large rivet heads rest on the valve springs and the large dowel pin hole in the valve plate lines up with that in the guard D.

Put the washer E, on the bolt, having the safety dowel pin in the washer pointing upward. Place the valve seat A, finished face of the seat downward,



Figs. 22 and 23.—Assembly of type Y discharge valve (fig. 22) and inlet valve (fig. 23).

on the assembly so that the dowel pins fall into place. Then screw on nut F, and after tightening it, lock it in place by means of the cotter pin.

Discharge valves can be assembled in the same manner and then the center bolt removed and reversed so that the flat head fits into bottom of the valve seat. However, if more convenient, the discharge valve may be assembled in the following manner:

Set the center bolt G, up on a table and place the valve seat A, over it, having the finished face of the seat facing upward and the lug on the bolt head fitting into the slot in the bottom of the seat.

Put the washer E on the seat so that the dowel pin on the washer fits into the seat.

Place the valve plate on the assembly so that the rivet heads face upward.

NOTE.—In some type Y valves, washer E, figs. 22 and 23, js not used, its place being taken by having this port cast integral with the valve seat.

After standing the discharge valve springs on the valve plate rivets, put the guard D in place so that the springs fit into their recesses in the guard and the dowel pin fits into the hole provided in the guard. Then screw on the center bolt nut, tighten it and secure it with the cotter pin.

To Replace Y Valves in the Cylinder.—Follow instructions as outlined under the same topic for channel valves. In addi-

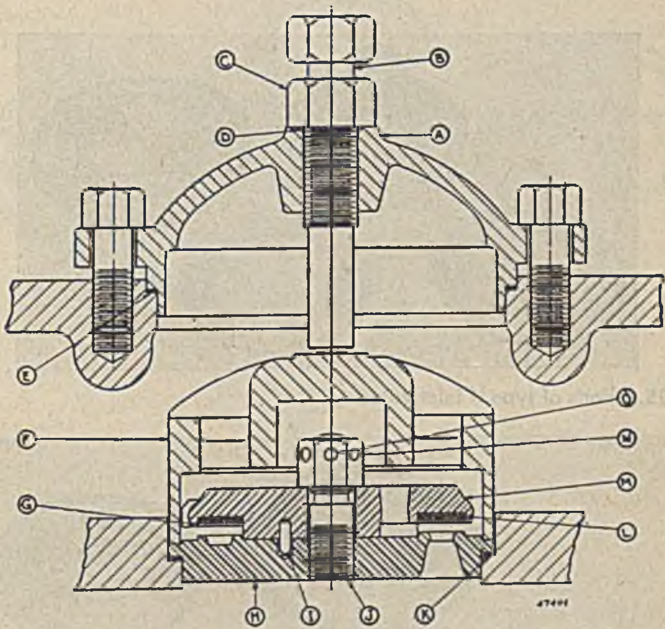


Fig. 24.—Type K discharge valve showing crab and valve cover complete.

tion, an added precaution must be taken when tightening the valve cover set screw; *Do not force the valve set screw in too tightly because excessive pressure will spring the valve seat and cause leakage and excessive heating.*

Rogler Valves.—These valves are shown in figs. 27 to 29. They consist of flat discs of special steel clamped upon valve seats and held in position by spring arms formed in one piece with the valve itself.

In construction, the valve **F**, fig. 27, which has integral spring arms, M, is held on its seat A, by stop plate H, which is made of steel for inlet valves.

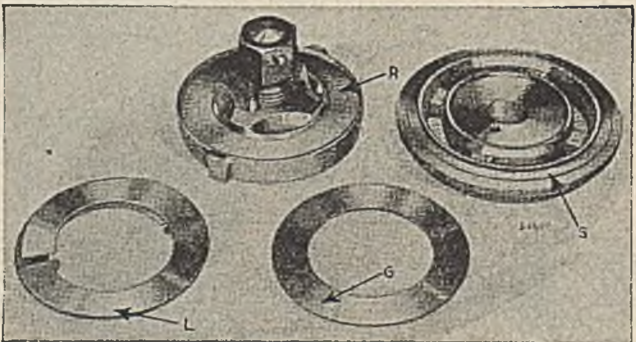


Fig. 25.—Parts of type K inlet valve.

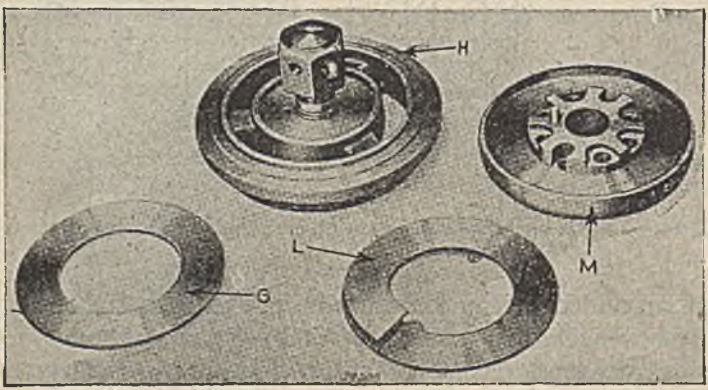


Fig. 26.—Parts of type K discharge valve.

The stop plate has spring arms which press upon the valve F, limiting the opening movement and closing the valve at the end of the stroke.

Stop plate H, is backed up by a buffer plate D, which is identical with the stop plate H, except that the spring arms are not cut through.

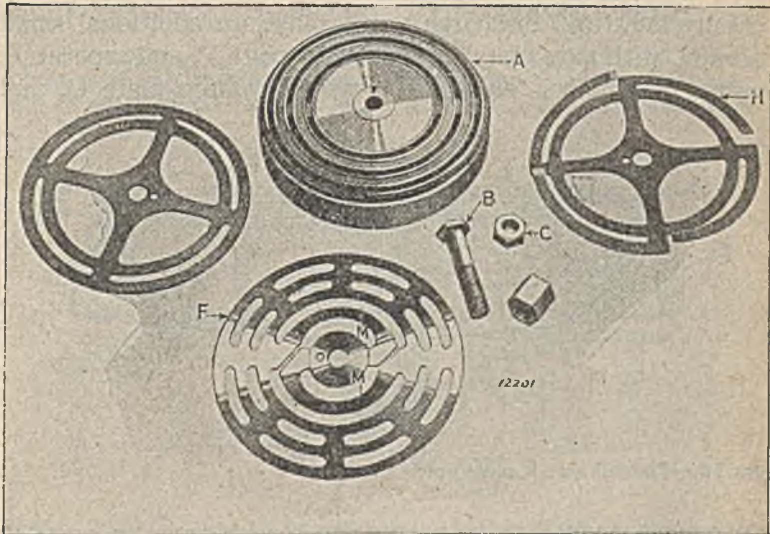


Fig. 27.—Parts of type Y (plate) valve.

D, serves as a stiffener for the stop plate H. In assembling the valve parts, the stop plate H, should always be placed on top of the valve washer G, fig. 28, so that the spring arms will rest on the valve plate and hold it to the valve seat A.

The discharge valves are equipped with cast guards J, fig. 29.

These have pockets for spiral springs K, which hold the valve on its seat when closed. The valve plate F has electrically welded to each face of its central portion, steel washers F and G, which separate the valve from its seat A, and from the stop plate H, on inlet valves, or cushion plate L, and cast guard J on discharge valves.

The sum thickness of washers E and G determines the lift of the valve.

The dowel pin, shown on seat A, insures proper assembling of valve parts which are held together by bolt B and nut C, and locked together with jam nut T.

Some inlet and discharge valves have an additional small valve N, with stop plate O, and buffer plate P, and separating washer Q, which is electrically welded to stop plate O, and

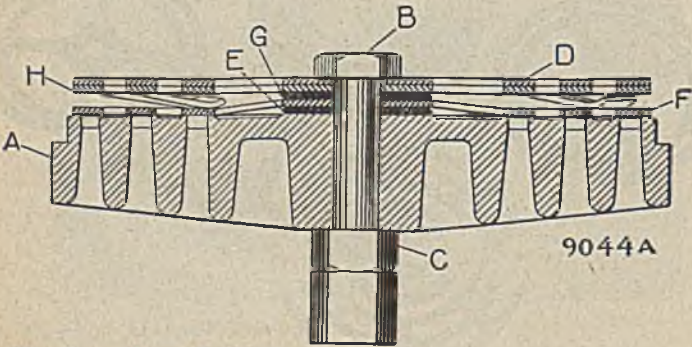


Fig. 28.—Parts of type R plate valve.

separates the small valve parts from the larger valve parts on these valves.

The sum thickness of washers R and S, which are electrically welded to valve N, determine the lift of the small valve.

The valve does not need to be ground in, but when new, there may be a slight leakage at first. After running a short time, they become tight and remain so.

Type K Valves.—These valves can be distinguished from Y valves by the valve plate and springs. The valve plate is a simple, flat annular ring while the valve springs are of the flat disc type.

Ques. What precaution should be taken in assembling the valves?

Ans. They must always be assembled in the cylinder so that the center bolt does not project in the bore of the cylinder.

Inlet valves must have a stop plate R, fig. 25, next to the cylinder piston, while discharge valves must have the valve seat H, fig. 26, toward the cylinder piston. The valve assembly is held in position by the valve

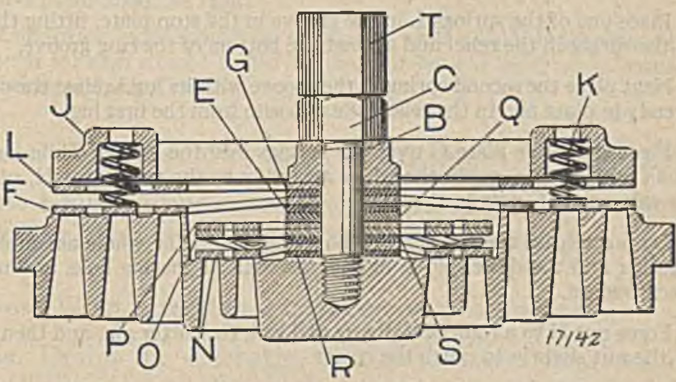


Fig. 29.—Plate valve with double valve disc as used on large size compressors.

cover set screw bearing against the crab which fits over the valve assembly. In this arrangement the valve cover set screw does not bear against the valve center bolt.

To Remove a K Valve from the Cylinder.—Follow the same procedure as outlined under similar heading for Channel Valves.

To Assemble an Inlet K Valve.—Parts of which are shown in fig. 25, place the stop plate R, on a table so that the center bolt stands upward. Place one of the springs L, in the groove in the stop plate R, fitting the lug on the spring in the relief and against the bottom of the ring groove.

Next place the second spring with its lug against the spring already in place and in the relief opposite from the first lug.

Place the valve plate G over the springs and while holding intact with fingers, place the valve seat S on the assembly. Be sure the small dowel pin I, fig. 24, fits into its mating hole. Screw on the nut N, force it to a tight seat, insert and lock the cotter pin, and then back off the nut slightly to pinch the cotter pin. To assemble a discharge valve, place the stop plate M on a table as shown in fig. 26.

Place one of the springs L in the groove in the stop plate, fitting the lug on the spring in the relief and against the bottom of the ring groove.

Next place the second spring in the groove with its lug against the spring already in place and in the relief hole opposite from the first lug.

Place the valve plate G over the springs into the groove while holding intact with fingers, apply the whole assembly to the valve seat H, placing the valve plate G next to the seat.

Do not release the pressure of the springs until the whole assembly fits together and the dowel pin I, fig. 24, fits with its mating hole, and nut N is screwed on.

Force nut N to a tight seat, insert and lock the cotter pin, and then back off the nut slightly to pinch the cotter.

Cleaning Cylinder Air Passages.—Never use kerosene or coal in an air cylinder to clean it out. This is a very dangerous practice and must be absolutely prohibited.

Ques. What is the most satisfactory method of cleaning the air passages?

Ans. Take out the valves and remove any hard carbon by light scraping. Any dirt or accumulation of foreign matter may then be removed with safety solvent.

Ques. What is the effect of oil?

Ans. Oil tends to collect in clearance pockets and they should be periodically cleaned and washed out with safety solvent which can be removed through the drain hole at the bottom of each pocket. Clearance pockets should be drained regularly.

In some cases where it is not practical to shut down the compressor for thorough cleaning or where it is known that there is no hard carbon, an accepted method of cleaning the cylinder is to fill the force feed lubricator with strong soap suds or soda water and allow it to be pumped into the cylinders for about an hour.

Lubricator feeds should be wide open while this is being done and if enough solution cannot be fed by the lubricator a hand pump should be added to each feed line.

Ques. What should be noted with respect to soap solutions?

Ans. Soap solutions should not be pumped into the cylinder any longer than absolutely necessary for cleaning.

Ques. What attention should be given to the drains?

Ans. Drains in discharge line, after-cooler and receiver should be opened to drain off accumulation of oil and water. *Oil should always be fed again before shutting down*, to prevent polished interior surfaces rusting.

This method is effective if there be no hard carbon present. The frequency of cleaning and amount of solution to be pumped through the cylinders depends on the amount of dirt and soft carbon accumulation and conditions of operation.

Cold Weather Precautions.—Whenever the compressor is exposed to a freezing temperature while not in operation, drain the water from the jackets and inter-cooler in order to prevent injury from freezing. Drain valves will be found at the side of the cylinder for draining the jackets. The inter-cooler can be drained through the valve provided for that purpose at the water inlet connection.

If the compressor is to stand idle for any length of time, remove piston rod packing. If this be not done, the packing will cause rods to rust very rapidly, causing trouble later.

Cleaning Cylinder Water Jackets.—If the circulating water be dirty, mud will be deposited in cylinder jackets and inter-cooler water heads and will ultimately obstruct the flow of water entirely unless care be used to prevent such an accumulation.

Clogged passages will interfere with proper cooling, which will result in possible damage to the cylinders and pistons, and a hard baked-on coating of mud in the water passages.

The cylinder heads and inter-cooler heads should be removed occasionally and the water jackets and passages inspected. If any mud deposit be found, it should be thoroughly cleaned out and the jackets completely flushed out with water.

Regulating Devices.—Owing to the length of this chapter, the servicing of regulators will be covered in Chapter 50 on Regulating Devices (Pages 229 to 268).

CHAPTER 55

Blowers

By definition, a blower is: *a form of air compressor for delivering large quantities of air at low pressure.*

Blowers may be classed as:

1. Positive displacement.
2. Rotary. { single stage / multi stage

Ques. What is a positive displacement blower?

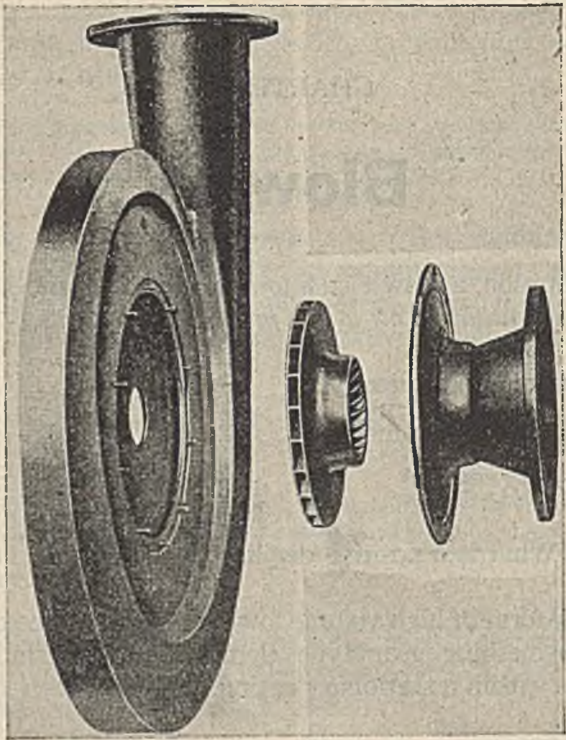
Ans. A form of high volume low pressure compressor having two inter-meshing gear type elements rotating in opposite directions within a stationary casing.

Ques. What is a rotary blower?

Ans. Nothing but a form of centrifugal pump designed to pump air instead of water.

Ques. What is its principle of operation?

Ans. When the shaft is rotated, the effect of centrifugal force upon the air within the impeller causes its compression and at the same time induces it to flow through the impeller.



Figs. 1 to 3.—Construction details of a single stage blower consisting of an impeller supported on a shaft and surrounded by a diffuser with a suitable casing. The impeller will by centrifugal force, entrain the air at its center (the eye) and discharge it at its outer circumference at an increased pressure. This pressure may be called centrifugal pressure. The impeller also sets the air moving at approximately the same velocity as the peripheral velocity or tip speed of the impeller. Hence the work delivered by the driver appears in the air discharged from the impeller in two forms of energy, namely pressure energy and velocity energy. The diffuser is so designed that the air flowing through it is gradually reduced in speed and the velocity energy is recovered in the form of increased pressure. Approximately 95% of the energy supplied by the driver appears as pressure energy and velocity energy in the air leaving the impeller. About one-half of the available energy is in the form of centrifugal pressure, while the other half is in the form of velocity.

Ques. What is the action of the air in passing through the impeller?

Ans. It is accelerated and the increase in velocity is a form of energy convertible into additional pressure.

Ques. How is the conversion produced?

Ans. By the orderly distribution of the air either in a bladed or open diffuser or in a volute or scroll surrounding the impeller.

Ques. What is a single stage blower?

Ans. One in which the entire pressure rise is obtained with a single impeller.

Ques. How much pressure do such machines develop?

Ans. Only a few pounds.

Ques. What is a multi-stage blower?

Ans. One having a number of impellers mounted on the same shaft connected in series in a common casing.

Multi-stage blowers (and maybe they should be called compressors) are sometimes provided with means for cooling between stages and may even be completely water jacketed.

Ques. How do the impellers of centrifugal blowers receive the air (or gas)?

Ans. They may be either single or double flow.

If there be an inlet opening at both sides of the impeller so that the gas flows toward the impeller from opposite directions, the impeller is known as the double flow type.

With a single flow impeller of the enclosed type the pressure upon opposite sides of the impeller is unequal. Thus, it is necessary with this type of impeller to provide some sort of balancing device or thrust bearing. With a double flow impeller, the pressure is the same on both sides and the machine will be practically balanced without the aid of balancing devices or thrust bearings.

Ques. After all just what is a centrifugal blower?

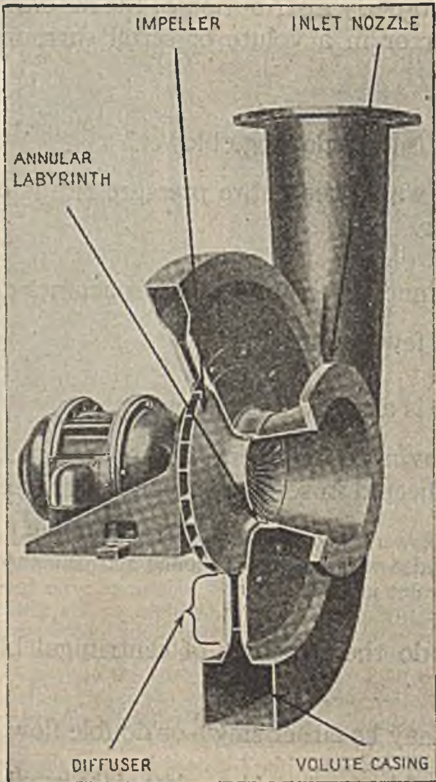


Fig. 4.—Assembly of motor driven single stage blower shown dis-assembled in figs. 1 to 3.

Ans. It is simply a centrifugal compressor designed for low pressure discharge.

There is no distinct line of demarcation between the two, but for the sake of convenience, the author in this chapter calls machines built for discharge pressure under 35 lbs. blowers and those built for discharge pressures above 35 lbs. compressors.

Ques. What are single stage compressors with a fan type impeller called?

Ans. They are commonly known as fan blowers.

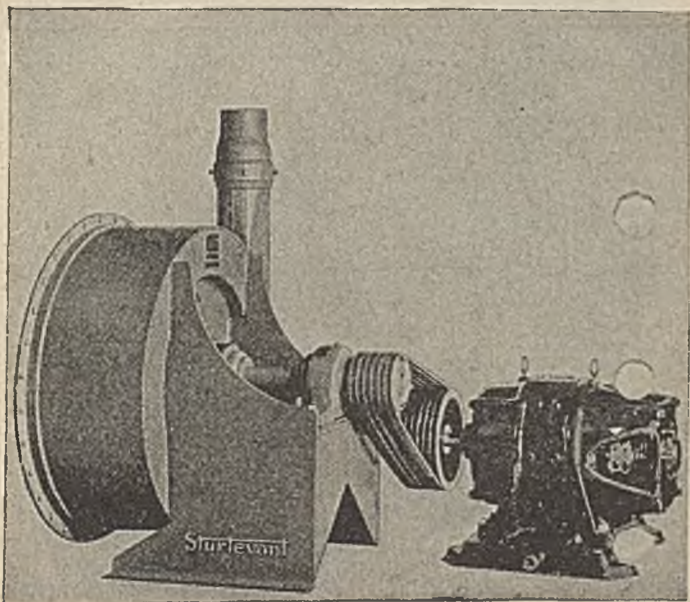


Fig. 5.—Typical centrifugal blower with multi-belt electric drive. With low motor speeds and small volumes such a machine will operate at much higher efficiency than a direct connected unit, and takes up less than half as much space. While this machine has an intermediate drive, it is a flexible one and consequently there are no misalignment difficulties.

Ques. What are they particularly adapted for?

Ans. They are adapted to cases where large volumes of air are wanted at low pressures ranging from a few ounces to ½ lb.

Ques. What are thecharacteristics of centrifugal blowers (so called compressors)?

Ans. The pressure rise within a so called centrifugal compressor is, as before stated, the result of the centrifugal force

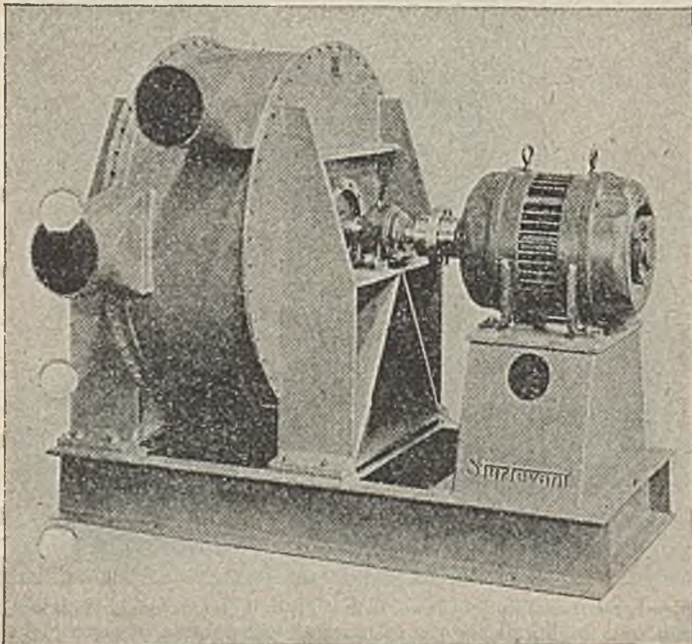


Fig. 6.—Typical centrifugal blower direct connected to electric motor. This is used on larger units (four bearing and coupling type).

produced when the impeller is rotated. The centrifugal force is a function of the impeller peripheral velocity and of the gas density; therefore, for any given speed and density, a centrifugal compressor should maintain approximately constant pressure for wide variation in capacity.

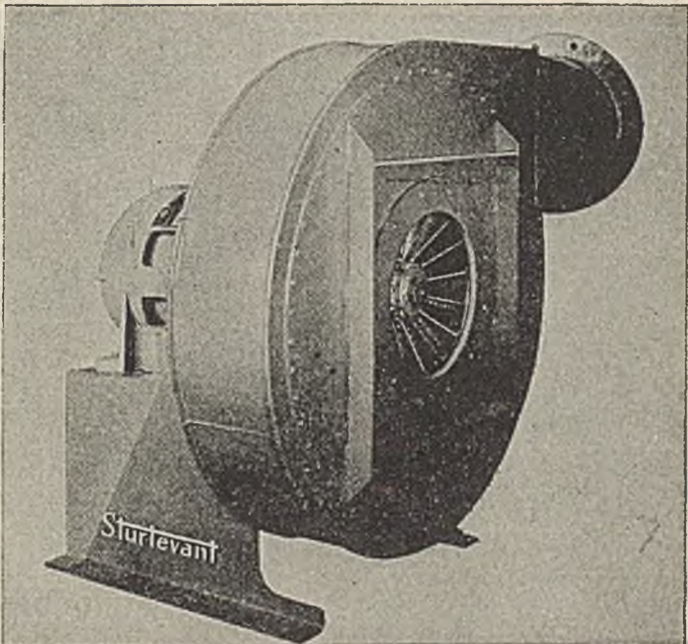


Fig. 7.—Typical centrifugal blower usual overhung wheel type.

However, several factors operate to change the discharge pressure slightly at other than normal capacity.

First, the design of the passage or passages, which convert into pressure the velocity obtained by the travel of the gas through the impeller. This conversion is most efficient at the rated condition of the compressor and is less efficient at smaller or larger capacities.

Second, the outlet velocity from the impeller and, with it, the velocity head available for conversion to pressure varies with the capacity.

Third, entrance losses into the impeller are least at the rated or designed condition of the compressor.

Fourth, there is a leakage loss from the high pressure side to the low pressure side because there has to be a small mechanical clearance between the rotating and stationary parts.

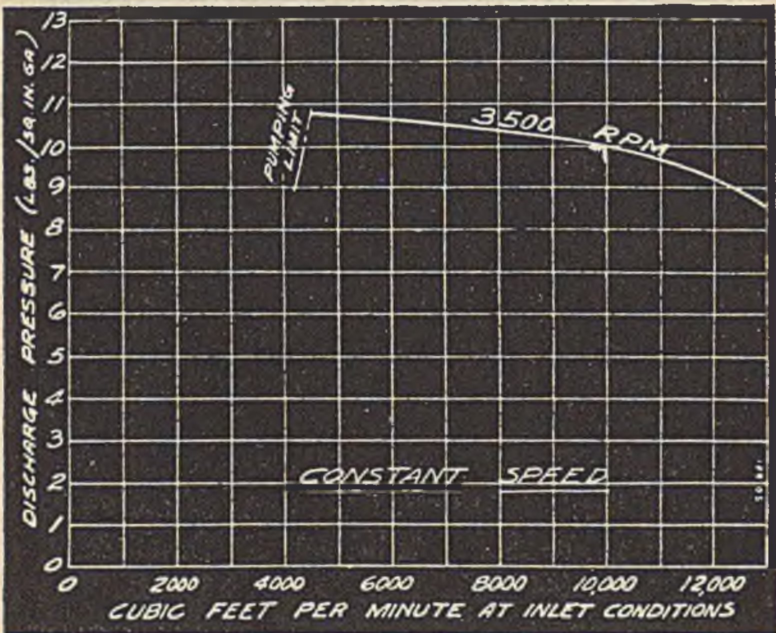


Fig. 8.—Typical pressure volume characteristic of a constant speed blower.

Finally, as in an ordinary pipe line, the frictional loss varies with the square of the velocity. For all these reasons, the discharge pressure of a centrifugal so-called compressor running at constant speed varies with the capacity.

Ques. Describe the characteristics in more detail.

Ans. It drops off gradually from its maximum (which usually obtains at some fractional rating) until, at some point beyond the maximum rating, it finally reaches zero. The relation between pressure and capacity, when represented graphically, is called the pressure volume characteristic of the blower or compressor. This pressure volume characteristic is of great value in judging the application of a given machine to a given purpose.

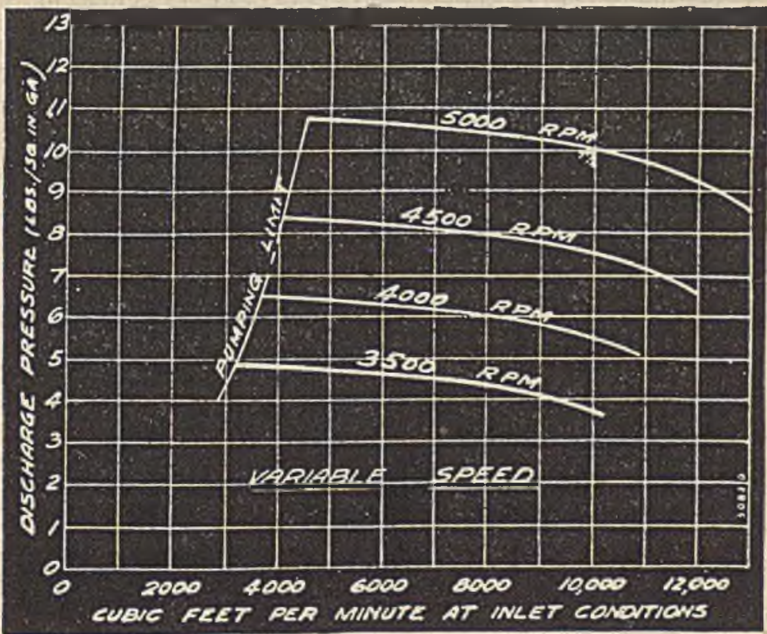


Fig. 9.—Typical pressure volume characteristic of a variable speed blower.

According to "Compressed Air Data" a typical pressure volume characteristic of a constant speed blower is shown in fig. 8 and of a variable speed blower in fig. 9.

Ques. What should be noted about every speed and pressure of a centrifugal compressor?

Ans. There is a certain minimum volume below which the machine does not operate properly.

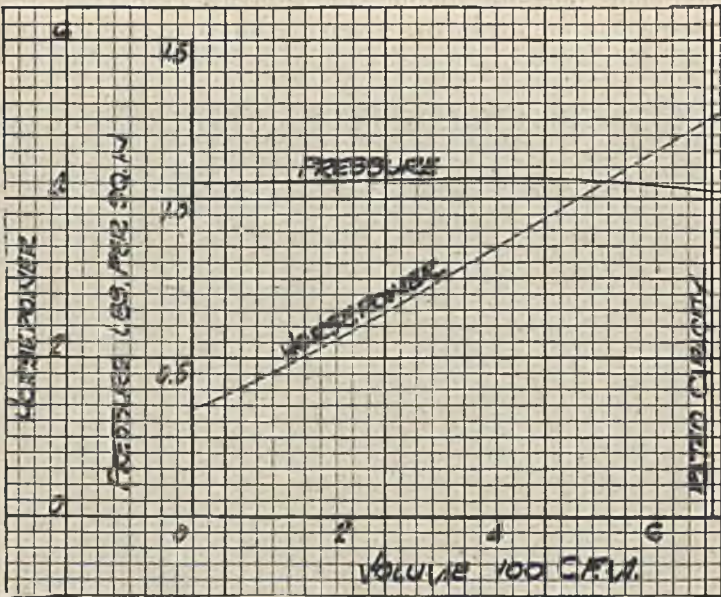


Fig. 10.—Performance curves of typical centrifugal blower at 1750 r.p.m.

Ques. What is this volume called?

Ans. The pumping point.

Below this volume its delivery of air or gas becomes irregular or unstable, reversing itself at frequent intervals with a characteristic noise known as "pumping." The pumping point is largely determined by the shape of the impeller blades and the design of the diffusor.

Ques. What is the characteristic of this action?

Ans. Due to this action of the centrifugal blower, it is impossible to build up a dangerous discharge pressure.

If the pressure increase, the compressor merely begins to "pump." No safety valves are necessary.

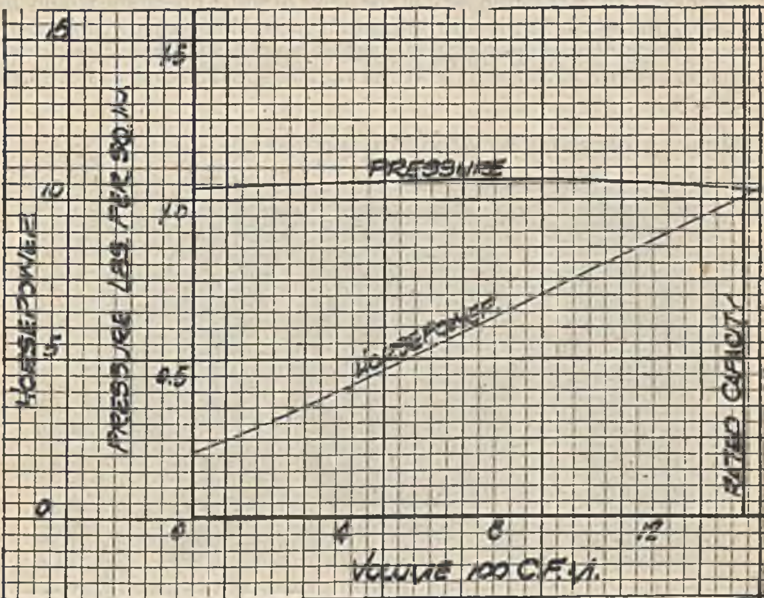


Fig. 11.—Performance curves of typical centrifugal blower at 3450 r.p.m.

Ques. What should be noted about two blowers operating in parallel, if they have similar characteristics and why?

Ans. Multi-port check valves are recommended for installation in the discharge lines of each compressor to prevent reverse flow through the idle compressors which would cause them to run backwards.

Ques. What kind of regulation should be given to steam turbine driven blowers?

Ans. The regulator should be connected to the steam turbine to vary the speed.

It can be seen from fig. 9 that an infinite number of pressure volume conditions can be obtained, within the limits of stable operation, simply by varying the speed. In the case of constant speed blowers and compressors, regulation can be obtained by throttling the intake or discharge of the blower. This is not as economical of power as variable speed control with a steam turbine.

Ques. How is better economy secured?

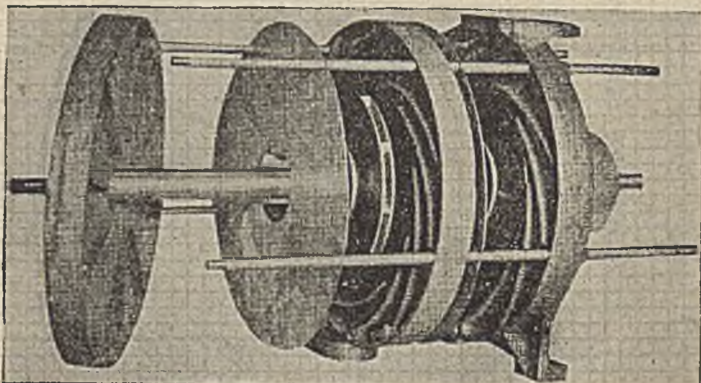


Fig. 12.—Dis-assembled multi stage blower showing impeller and stationary casings with guide passages.

Ans. Other methods of regulation are available.

One type works on the regenerative principle. The velocity energy which is released by the throttling of air or gas in adjustable guide vanes is recovered at high efficiency by a turbine element on the blower shaft.

Ques. What are the applications of turbo blowers?

Ans. Too many to enumerate here.

CHAPTER 56

Super-Chargers

An inherent defect in the gas engine is its failure to properly perform the first stroke of its working cycle. During this first or admission stroke, the fuel mixture must be pumped from the carburetor into the cylinders, but in this operation a full charge is never delivered, the amount of charge decreasing as the engine speed increases.

On account of the very high speeds at which engines are now run, this decrease in the amount of charge has become so marked, that there is a considerable loss of power, that is, at high speeds the power is much less than it would be if full fuel charges were admitted.

For instance, in the horse power and torque curves, fig. 1, representing results obtained in testing an eight cylinder engine, it will be noted that the power does not increase at the same rate for the high speeds as for low speeds, and finally above 3,600 *r.p.m.* there is a loss of power with increasing speed. This indicates that 3,600 *r.p.m.* is the speed limit for the valve area provided.

Note also how the torque decreases for speeds above 1,800 *r.p.m.* This indicates a gradual decrease in the amount of charge admitted, resulting in a gradual decrease in the mean effective pressure of combustion available to produce torque.

During admission the gas engine acts as an air pump and as such is a poor performer because of the large amount of clearance due to the combustion chamber. On account of this clearance, the degree of vacuum created during the admission stroke is less than it would be otherwise. Aside from this there are several other conditions as mentioned in the chapter on Gas Engine Principles that tend to reduce the amount of fuel mixture admitted. They are:

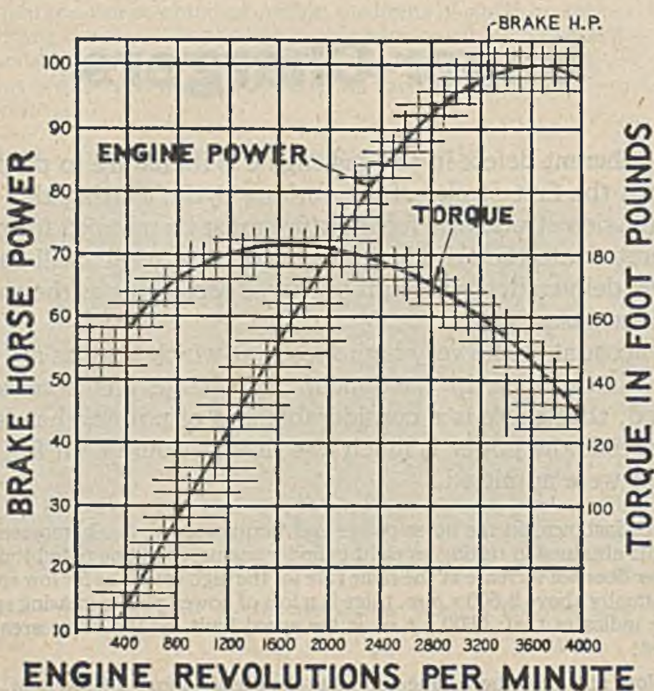


FIG. 1.—Horse power and torque curves for eight cylinder automobile engine.

1. Inertia of the mixture
2. Pre-heating
3. Manifold friction
4. Wire drawing

5. Heating
6. Leakage

These various conditions are illustrated in fig. 11 of Chapter 5 Audel's Automobile Guide, by the author. To overcome these difficulties and to greatly increase the power of an engine of given size, super-chargers are used.

Ques. What is a super-charger?

Ans. A charge booster, or form of blower for increasing the pressure of the mixture in the manifold in order to increase the amount of mixture entering the cylinders during the period of admission.

FIG. 2.—Root positive displacement super-charger. Inside the casing are two gears mounted on two shafts, one driving the other. In the blower chamber attached to these shafts are two double rotors set at an angle of 90° with each other.

Ques. What is a charge?

Ans. The amount of mixture admitted to the cylinder and combustion chamber during the period of admission.

Ques. What is a full charge?

Ans. A charge admitted at atmospheric pressure.

Ques. What is a super-charge?

Ans. A charge admitted above atmospheric pressure.

Ques. What are the effects of a super-charge?

Ans. Higher compression, considerably higher initial pressure of combustion and during expansion resulting in a higher mean effective pressure with corresponding increase in power.

Ques. What is the effect on gas engine operation at high elevations?

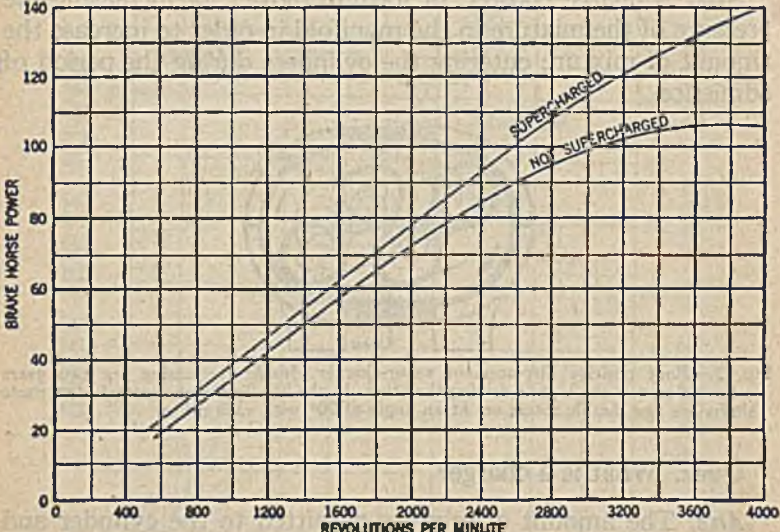


FIG. 3.—Performance curves with Roots super-charger for eight cylinder (300 cu. ins. displacement) engine.

Ans. On account of the decrease in atmospheric pressure, less mixture is admitted to the cylinders resulting in decrease in power.

On account of this, super-chargers are used on airplanes for high elevation operation.

Elementary Super-Charger.—The elements of a super-charger are shown in fig. 4. As shown a centrifugal blower is connected in the inlet manifold between the engine and carburetor, the rotor of the blower being operated from a side shaft through a worm and wheel gear.

The wheel is driven by chain or other gear from the crank shaft and runs a little faster than the crank shaft, while the

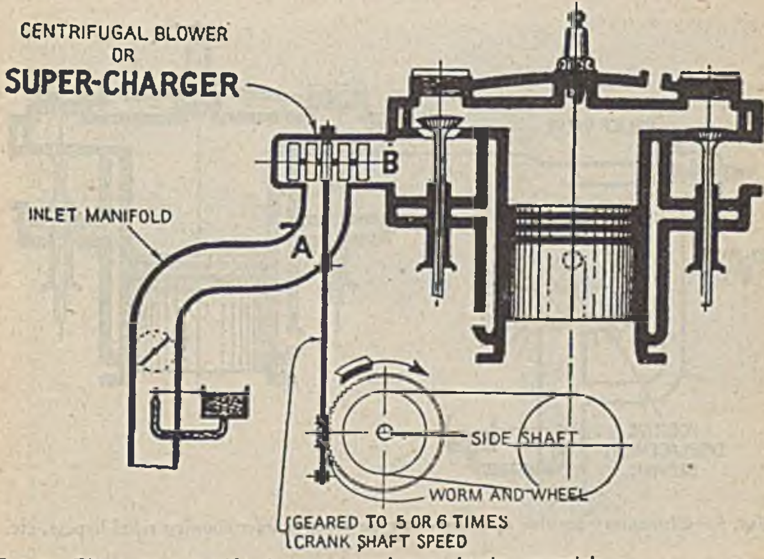


Fig. 4.—Elementary centrifugal type super-charger showing essentials.

blower shaft is speeded up considerably by the worm gear, attaining speeds of 20,000 *r.p.m.* or more.

In operation, the mixture is drawn in at **A**, by the rotor of the blower and builds up a pressure above that of the atmosphere at **B**. Accordingly when the inlet valve opens on the admission stroke, the mixture is forced past the valve so that even during the pre-admission period before the piston has quite reached the top dead center, the mixture is flowing into the combustion chamber and cylinder, the motive force to move the mixture being much greater than that due to inductive exhaust during the post-exhaust period of exhaust valves timed for late closing. The result is that as the piston traverses its admission stroke the incoming mixture is able to follow it at a pressure greater than atmospheric at highest engine speeds.

It must be evident then that with a super-charge, that is, one whose pressure is greater than atmospheric, the charge on the next stroke is compressed to a higher pressure than for ordinary

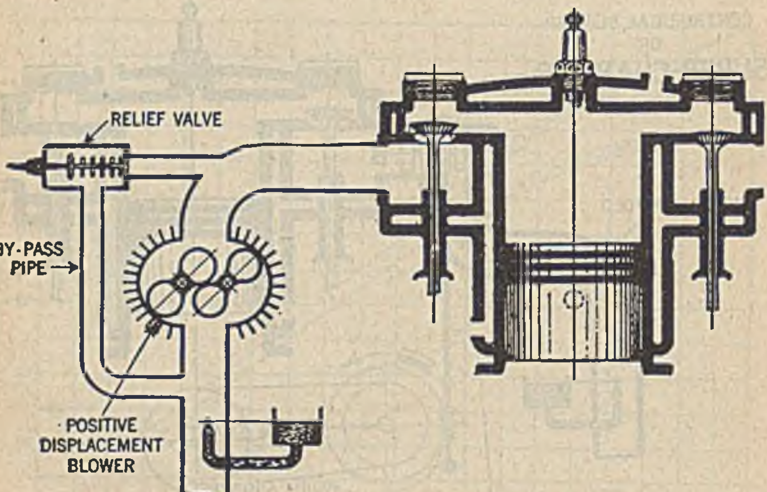


FIG. 5.—Elementary positive displacement type super-charger showing relief bypass, etc.

operation without a super-charger. Moreover, as already stated the initial and mean pressures of combustion are higher thus increasing the power.

Ques. How much admission pressure is obtained with a super-charger?

Ans. A few pounds above atmospheric pressure, more or less depending upon the speed of the blower.

Ques. Describe the super-charger used on the Graham car.

Ans. As shown in figs. 6 and 7, it consists primarily of a casing within which a rotor revolves at high speed. The centrifugal action of the rotor draws the mixture of fuel and air from the carburetor into the center of the casing and expels it under pressure at the rim or edge, forcing it through the intake manifold to the combustion chambers of the engine.

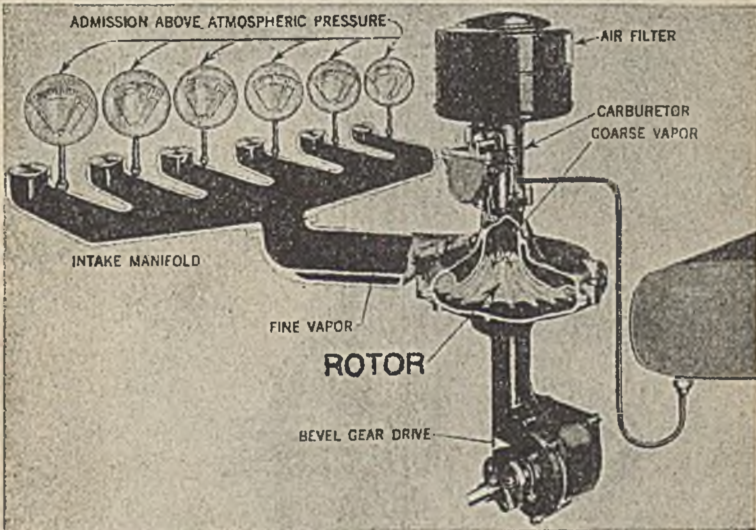


FIG. 6.—Graham super-charger showing details and connection to inlet manifold.

Both the rotor and worm wheel shafts are mounted on plain bearings. The lower part of the unit consists of a gear case enclosing worm gearing. No adjustment of either the bearings or the worm gearing is required.

All the bearings and the worm gearing of the super-charger are full pressure lubricated by the same oil used to lubricate the engine. A reserve supply of oil is automatically retained within the worm gear casing to assure adequate lubrication of the gears and bearings during the starting and warming up period.

The cover which forms the top half of the rotor casing is water jacketed. The cooling water controls the mixture temperature at desirable limits under partial and full throttle operating conditions.

The drive gears in the super-charger are a special cone worm gear which are driven from the crank shaft by two *V* belts with a step-up ratio for the rotor of 5.8 times crank shaft speed.

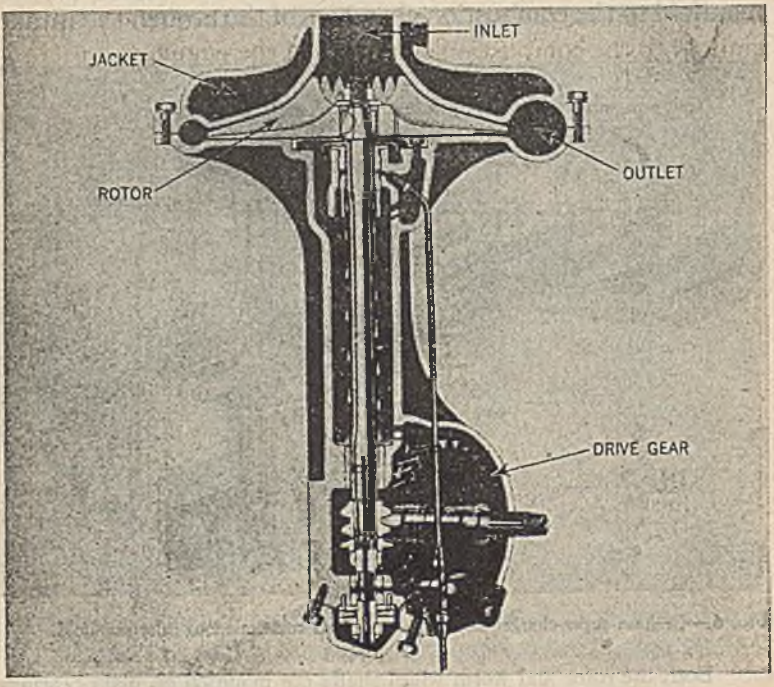
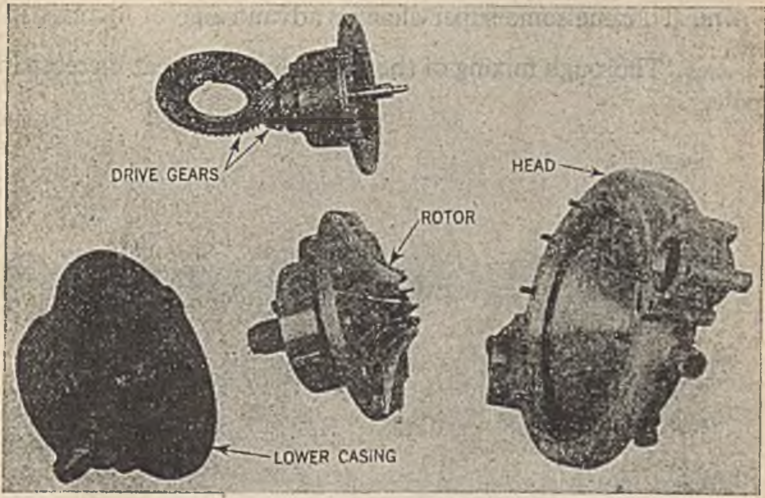


Fig. 7.—Graham super-charger showing rotor and drive mechanism, also method of lubrication.

The maximum horse power according to tests is produced at an engine speed of 4,000 *r.p.m.* The engine develops 85 horse power without super-charger and 120 with the super-charger.



Figs. 8 to 11.—Dis-assembled view of super-charger used on the Cord car.

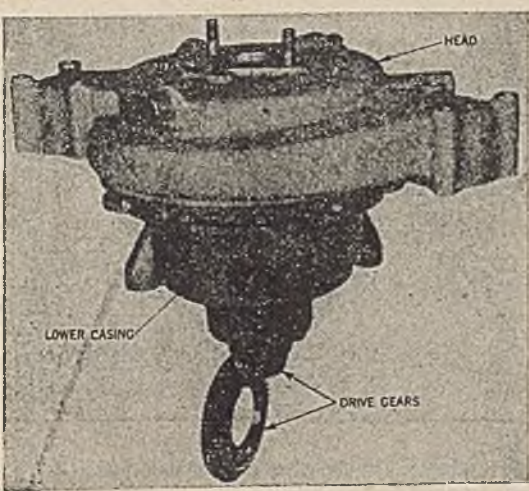


Fig. 12.—Schwitzer-Cummings super-charger as designed for the Cord car.

Ques. Name some super-charger advantages.

Ans. Thorough mixing of the mixture and a great increase in power.

CHAPTER 57

Pneumatic Hand Tools

The title of this chapter is used in a broad sense to indicate any tool employing compressed air as a power medium for its operation.

There is a great variety of tools operated by compressed air, such as (in part):

1. Drills.

    a. Rock

    b. Long stroke

    c. Multi vane

    etc.

2. Wood borers.
3. Wrenches.

    a. Quick operating

    b. Impact

NOTE.—In this chapter the author has broadly classed all tools using compressed air as a motive power as *pneumatic tools*, but it should be noted that manufacturers make a distinction between rock drills and the rest of the tools. However, they all depend upon compressed air as the motive power and strictly speaking all can be called pneumatic tools.

4. Grinders.
   a. Surface
   b. Sanders

5. Saws.
6. Hammers.
   a. Riveting
   b. Jam riveters
   c. Chipping
   d. Caulking
   e. Scaling

7. Diggers.
8. Sand rammers.
9. Backfill tampers.
10. Core breakers.
11. Concrete vibrators.
12. Hoists.

First will be treated *rock drills* and following that what the manufacturers call "pneumatic tools." Of course a rock drill is a pretty robust piece of mechanism built to stand super heavy duty and perhaps receiving more abuse than any other tool, and yet these drills are precision made tools, manufactured to very close tolerances and of the best materials, otherwise they would not stand up considering the terrific duty for which they are designed.

Compared with these rock drills all other tools are of comparatively light duty and that is the reason manufacturers do not include rock drills under the general term pneumatic tools.

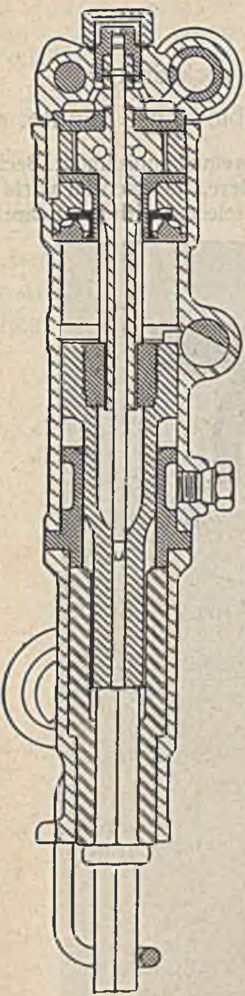


Fig. 1.—Section through a typical rock drill.

Rock Drills.—Considering this very important pneumatic tool the question is immediately asked

Ques. What is a rock drill?

Ans. A percussion tool in which a piston moves back and forth within a cylinder and strikes a drill steel which is loosely held in a chuck. The admission of the compressed air which operates it is controlled by a valve.

Ques. Describe the drill proper.

Ans. The drill steel ordinarily used has a small circular hole running through it longitudinally and is called hollow drill steel.

Ques. Why is the drill steel made hollow?

Ans. Either air alone or air and water combined may be forced through the steel to remove the cuttings from the hole.

Ques. With respect to these two methods of operation, how are the drills classed?

Ans. The are classed as *dry* and *wet.*

Ques. What is the speed of a modern rock drill?

Ans. They strike from 1600 to 2200 blows per minute.

Probably no tool in general service undergoes equal punishment. Because of the ceaseless pounding which it must endure, its essential parts are composed of high grade alloy steels. These are selected with great care and

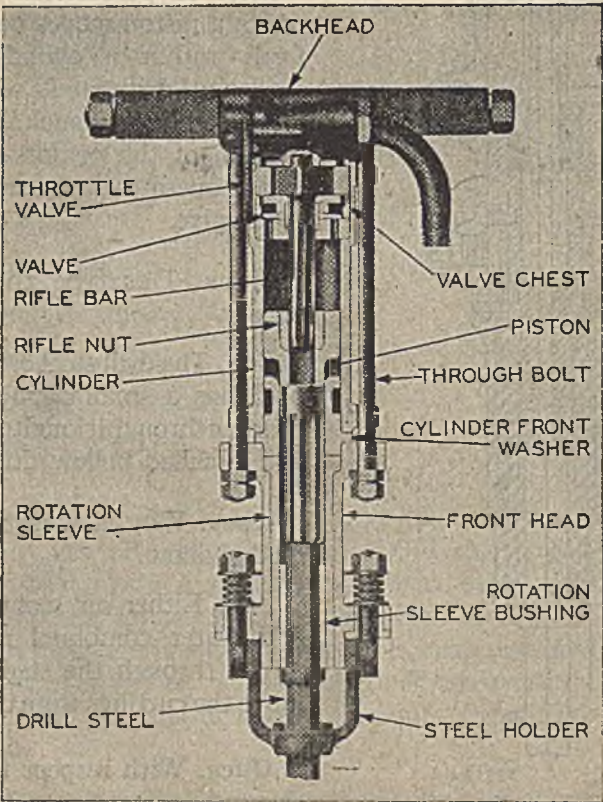


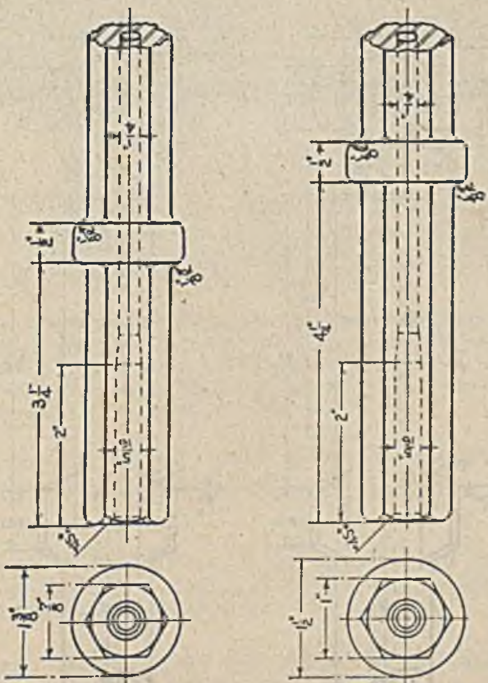
Fig. 2.—Cross section of a Jackhamer rock drill which shows the general construction and the relationship of the various working parts.

it is no exaggeration to state that the steels used for making razor blades are not of good enough quality to answer some of the requirements of rock drills.

Each part is formed from the particular steel that best meets its purpose and is heat treated to render it even more resistant to shock and wear.

Ques. What should be noted about the construction of rock drills?

Ans. Although they are designed to do rough work they are constructed with the same precision as though they were more delicate mechanisms.

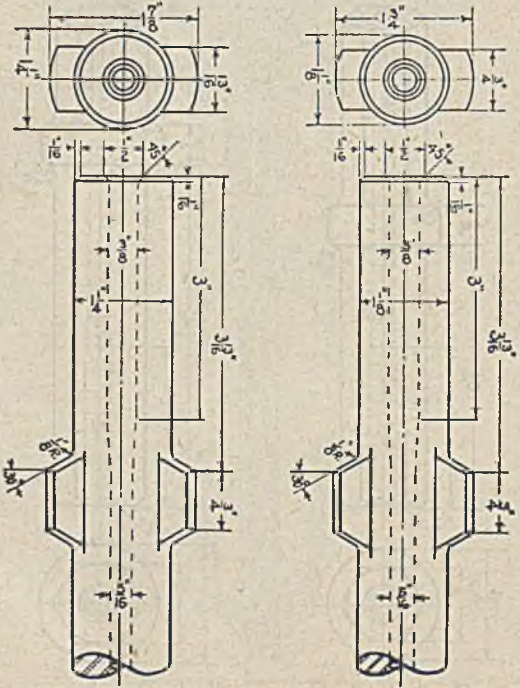


Figs. 3 and 4.—Shanks for Jackhamer drills showing dimensions of sizes generally used.

Clearances are regulated down to much less than one-thousandth of an inch. The entire process of manufacture is attended by painstaking care and thoroughness. The work is done by skilled mechanics in well equipped modern shops.

Fig. 2 shows the construction of a rock drill of the "Jackhamer" class. The illustration is a sectional view and the various parts of the mechanism are indicated.

Selecting the Right Drill.—There is a type and size of rock drill for each and every class of work and the most effective drilling results will be secured by selecting the correct tool for the job in hand.



Figs. 5 and 6.—Shanks for drifter drills showing dimensions.

The principal classes of drills and the general field of work of each are:

1. **Jackhamers.**—These are unmounted, self rotated, hand-held drills used principally for down hole work.

They are light enough to be handled readily by one man. They will put down holes as deep as 30 ft.

They are used for general excavation work such as trail building, road-building, foundation work, etc. Most Jackhamers are used dry.

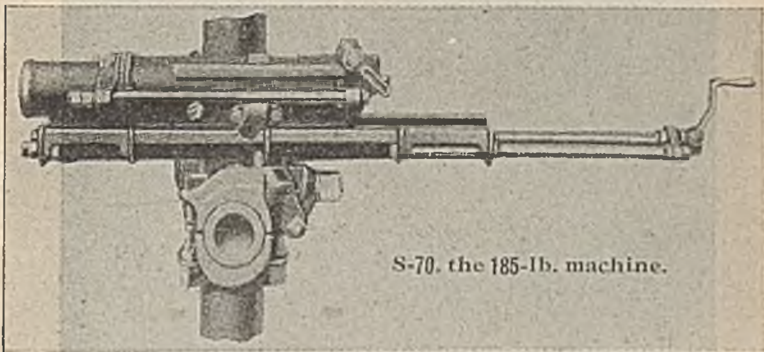


Fig. 7.—Drifter drill. While drifters are used principally in mining and tunneling operations, they are also used extensively for stoping in some parts of the worls. In driving large tunnels, heavy drifters have proven most satisfactory when mounted on a carriage. Quarrying operations have been speeded up by their use in quarry bars and wagon mountings.

2. **Drifters.**—These are mounted drills, designed for horizontal drilling and extensively used in mining and tunnel driving.

They are generally used wet. A drill runner and a helper ordinarily comprise a drifter crew. Several types of mountings are used, of which the most popular is the column and arm.

3. **Stopers.**—These drills are designed for drilling upward and gain their name from the fact that they are widely used in mine stopes.

Fig. 8.—A stoper drill in operation. This type drill is widely used in underground mining work.

Stopers are ordinarily wet drills.

4. Wagon Drills.—These are drifter type drills mounted in vertical guides on portable rigs.

They are suitable for heavy work and deep holes and have long been used in quarries. More recently they have been adapted for side hill road construction, particularly where the rock is difficult to drill with light, hand held drills.

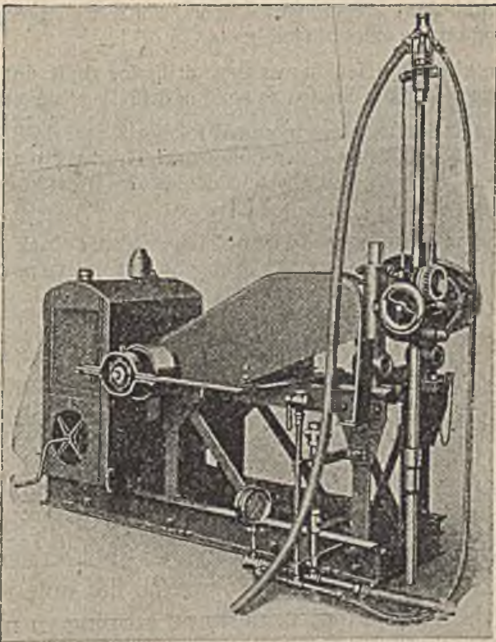


Fig. 9.—Calyx core drill. Used principally in exploration work, where a core of 1½ in. up to 72 in. diameter is desired. This drill uses, as the cutting medium, Calyxite (chilled steel shot). This material has been widely used throughout the world as a cutting medium for securing cores of materials of every description, from steel beams to the softest shale. These drills are built for steam, air, gasoline or electric motor, belt, chain, or any other type of drive. Machines can be equipped with augers, and special sampling of tools where accurate samples of the earth above bed rock are required, as for bridge and dam construction.

There has also recently been introduced a smaller wagon drill which utilizes a Jackhamer instead of a drifter.

This has automatic feed and return, which not only insures even drilling pressure, but which makes constant attendance unnecessary. This drill can be used for holes at any angle from vertical to horizontal.

Still other classes of drills might be cited.

For example, there are submarine drills, which are large, powerful tools for mounting on the sides of drill boats.

There are also large Jackhamer type drills for shaft sinking and other heavy drilling which are known as sinking drills, or "sinkers."

Sometimes Jackhamers are mounted for horizontal drilling and drifters are mounted on tripods for down hole work. Thus, there are innumerable types and combinations, but the four classes enumerated cover the drills most generally used.

Among these four classes are many sizes of drills. Each is fitted for its particular kind of work. They vary in weight, in rotative power, in blowing power, and in various other ways.

Determination of the most effective drill to use for any purpose depends upon the type of work, the character of the rock, the depth of drilling, the pressure of the air available, and other factors.

It pays to obtain the advice of experienced men when choosing rock drills

Rock Drill Mounting.—Mountings for Jackhamer drills have been found thoroughly practical for such flat hole work as tunnelling, brushing, drifting, underhand stoping in metal mines, and for breaking down coal, driving gangways or entries in coal mines, etc. They are made to hold either the dry or wet Jackhamers.

It is but an instant's work to clamp the standard Jackhamer in the carriage of the Jackhamer mounting. Therefore, a single drill may be used conveniently for the twofold purpose of drilling flat holes in the heading and down holes on the bench.

The carriage is moved by a feed screw which is supported on the arm of a column, on a horizontal shaft bar, or on a tripod, as the standard 5-in. cone upon which the shell rests will fit into any 5-in. Sergeant saddle or clamp.

The mountings are made in two types with lengths as follows:

1. Fixed cone shell mounting, 26 in. feed
2. Sliding " " " 20 " "
3. Sliding " " " 32 " "

Fig. 10.—Drifter drill with chain motor drive mounted on a quarry bar.

Ques. What is a wagon mounted drill?

Ans. An outfit (as shown in fig. 11) designed to do the work of two hand held drills.

Fig. 11.—Wagon mounted jackhamer—a one-man outfit. Useful on many road construction jobs, in foundation work for bridges, buildings, dams, etc., and in certain phases of quarry work. By means of a self-locking worm drive a small, four cylinder air motor automatically feeds the drill up to the rock exactly as with a hand cranked drill. This method prevents the drill jumping back, thus eliminating the use of unnecessary feed pressure which causes rotation binding. It also prevents the drill dropping on the operator, and keeps the drill from jumping forward in a soft spot or when a steel breaks. This self-locking feature also protects the air motor from thrust and shock.

Ques. Why?

Ans. Because:

1. It drills faster, as the automatic feed properly holds the drill down to its work at all times.

2. It drills just as fast at the end of the shift as at the beginning.

3. It permits the use of a more powerful drill with correspondingly less effort on the part of the operator.

4. It allows a 4-foot change of steel, the time spent in changing steels is cut in half.

5. The automatic feed frees the operator while the drill is running, enabling him to get his steel ready for the next change (under suitable drilling and rock conditions one operator can run two wagon mounted drills).

6. It pulls tight steels which are practically impossible to pull with a hand held drill.

7. It gives better control for fighting slippery ground.

8. It will drill holes at any angle, from vertical to horizontal.

9. There is no loss in efficiency, the drill is always held down, meaning lower air consumption per foot of hole drilled.

10. It is simple in construction and easy to operate and upkeep costs are low.

11. It is well balanced and easily handled by one man.

Ques. What is the application for submarine drills?

Ans. They are used in river and harbor improvement, deepening canals, removing submerged reefs, ledges, etc.

Ques. How are they mounted?

Ans. On a drill barge as shown in fig. 12.

Ques. For what service are long stroke drills used?

Ans. They are particularly suited for use where service is intermittent, such as for mill wright, construction, mine use, etc.

They are made both reversible and non-reversible. They have capacity for handling large work at very slow speeds, 20 to 35 *r.p.m.*

They are most commonly used for drilling, reaming, tapping, flue rolling, driving screw spikes, tightening bolts, etc. Their sturdy construction and power make them ideal for many miscellaneous types of work.

Frequently they are coupled to tools and act as air motors. They are being used in setting locomotive valves, in the operation of portable boring bars, in reboring cylinders, etc.

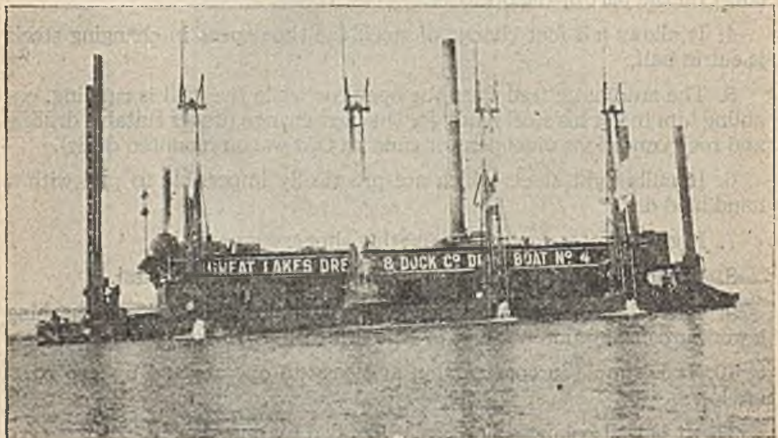


Fig. 12.—Submarine drills. The illustration shows submarine boat on a channel deepening job. The drill is hammer type submergible for work under water, close to the rock. It exhausts directly into the water in a bell shaped chamber which prevents water entering the working parts. The hollow drill steel is fitted with Leyner lugged shank and fits closely into the chuck. The rotation is effected by an independently operated motor which is above water and indicates the performance of the drill.

In machine shops and power plants they are adapted to such jobs as operating portable key-seating and oil grooving machines, grinding valve seats, cleaning boiler tubes, and operating mechanical scraper heads.

Figs. 14 and 15 show reversible and non-reversible long stroke drills.

Ques. What is the characteristic and application of multi-vane drills?

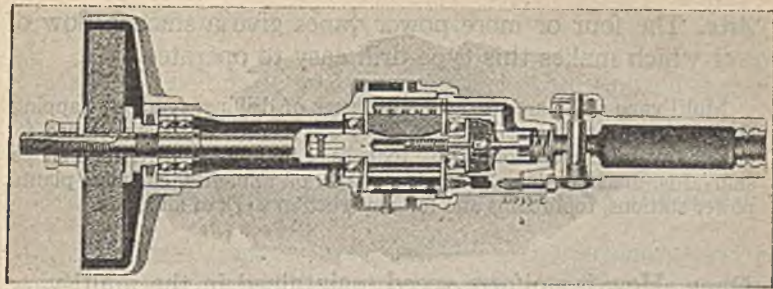
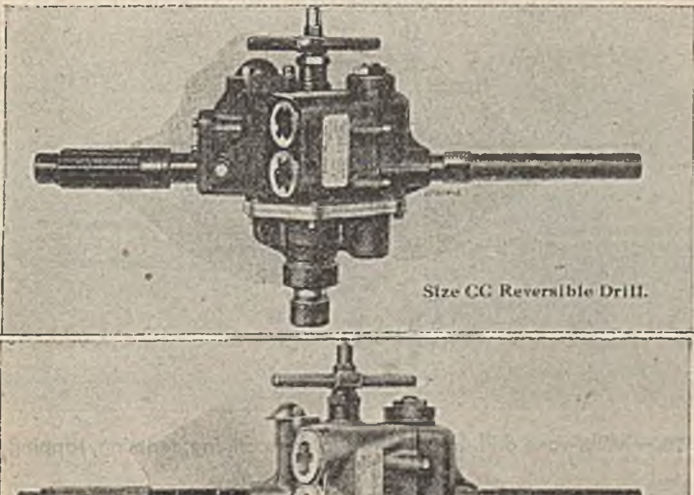
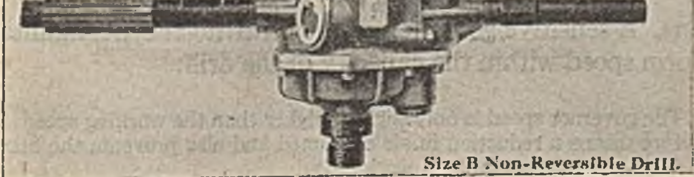


Fig. 13.—Multi-vane grinder. A sensitive governor makes it possible to operate grinding wheels at their most efficient speed and prevents over-speeding when running idle.



Size CC Reversible Drill.



Size B Non-Reversible Drill.

Figs. 14 and 15.—Reversible and non-reversible long stroke drills for intermittent or occasional service.

Ans. The four or more power vanes give a smooth flow of power which makes this type drill easy to operate.

Multi-vane drills are used for all classes of drilling, reaming, tapping, staybolt tapping, flue rolling, etc. They are hand held tools designed for one or two man operation. These tools have many uses in railroad shops, shipyards, machine shops, assembly plants, oil refineries, chemical plants, power stations, tool rooms and for construction work of all kinds.

Ques. How is uniform speed maintained in the multi-vane drill?

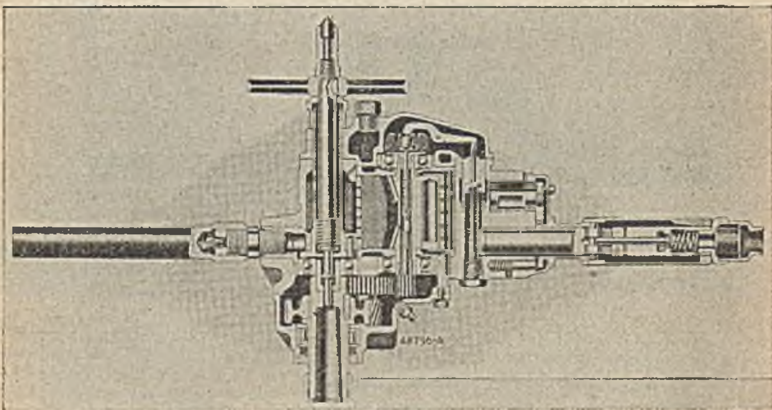


Fig. 16.—Multi-vane drill. Suitable for accurate drilling, reaming, tapping, etc.

Ans. A sensitive governor, of the centrifugal type, maintains uniform speed within the capacity of the drill.

The governor speed is only slightly higher than the working speed. This feature means a reduction in air consumed and also prevents the burning of drills and reamers on light work.

Fig. 16 shows the construction of a multi-vane drill.

Ques. What is an impact wrench?

Ans. A pneumatic tool that converts torque from a multivane motor into rotary impacts.

Ques. What is accomplished by these rotary impacts?

Ans. These impacts are applied to tighten nuts and cap screws to any desired degree or to remove "frozen" nuts which would otherwise have to be burned or cut off.

A typical impact wrench is shown in fig. 17.

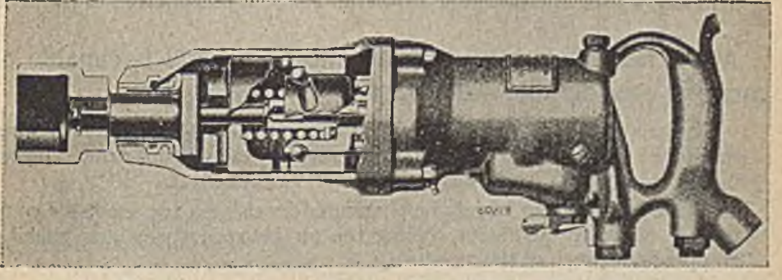


Fig. 17.—Impact wrench. The torsional impacts exert a more powerful turning effect than is possible with any portable torque type wrench. This type of turning action makes it possible to remove nuts which could not otherwise be taken off except by splitting with a chisel or burning with a torch. Impacting begins when resistance to turning reaches a predetermined value and continues until the throttle is closed or torque required to turn the nut falls below this point. Torque applied by the motor is but a small fraction of the turning effort exerted by the wrench and remains constant as long as impacting continues.

Preliminary Attention for New Drills.—Prior to putting the drill into service, the operator should remove the grease, blow out the air passages, and make certain that the drill is clean. Be sure that it is supplied with plenty of good lubricant.

Note the following points:

1. If possible use 80-90 lbs. air pressure. A drop of 5 lbs. air pressure reduces the tool's work about 10%.
2. Use size of hose recommended for tool.
3. Air must be clean, free of abrasives. Wash out lubrication frequently.
4. Air should be dry.
5. Lubricate frequently, using lubricant specified by manufacturer.
6. Blow out air line before attaching.
7. Couple with proper couplings.
8. See that driven tool fits socket or shank.

To remove any accumulated moisture as well as bits of rubber and other material, the air line should be blown out before it is connected to the drill.

Air is heated greatly during compression and is capable of carrying considerable water vapor.

As the air cools in the delivery lines, this vapor condenses. The resulting moisture is carried into the drills, where it washes away lubricant, freezes in valves and ports, and exerts a generally harmful effect.

These troubles can largely be eliminated by using a two stage air compressor, which provides for cooling the air between stages and thereby trapping out much of the moisture.

In almost all installations, it pays to install an after-cooler to remove moisture from the air before delivering it to the drills or other tools. Suitable drains should be arranged in the air lines and blown frequently.

How to Obtain Best Results in Drilling.—Best results can be obtained from drills by applying pressure through the handles and by using both hands.

Drill runners often throw one leg over the handle, or use a metal bar or piece of wood to obtain leverage in applying pressure. *These practices are not recommended*, as they lead to varying the pressure on the drill and also have a tendency to hold the drill off line with the drill steel.

The drill will do its best work if the contact between the piston face and the drill steel shank face is as large as possible.

This will result only when the drill is kept in line with the drill steel so that direct blows will be struck.

Do not exert enough pressure to slow up the drill.

A little experience will serve to determine the amount of pressure to put upon the drill to obtain maximum efficiency.

Be sure that the hole in the drill steel is open at both the shank and bit ends.

As air is directed through the hole from time to time to remove cuttings, obstructions in the steel interfere seriously with the blowing capacity.

Ques. What should be done when dry drilling?

Ans. The holes should be blown out frequently to remove cuttings from the hole and thereby prevent binding of the bit.

There is an attachment on the drill which permits passing the entire flow of air through the drill steel for this purpose. In ground that is slightly moist, there is a tendency for the drill cuttings to accumulate just above the bit and to become packed solidly against the sides of the hole through the action of the rotating steel. Such accumulations are known as mud collars and should be avoided. Sometimes a little water run in the hole will thin the consistency of the mud mixture so that it can be blown from the hole.

Ques. When should a wet drill be used?

Ans. When the ground is too damp to permit proper blowing of the cuttings from the hole.

Irregular Bits and Shanks.—If the drill steel be in proper condition, each blow will be transmitted to the rock being drilled. When the drill bit becomes quite dull, however, very little of the force of the blow is transmitted to the rock, and its full force must be absorbed by the drill steel and by the drill piston.

The repeated use of a dull bit and the resulting accumulation of shocks to the piston lead to premature failure of the piston and of other parts of the drill. Dull bits are also a leading cause of drill steel breakage.

Aside from these considerations, it is uneconomical to use a bit after it has become dull because it will then do little work.

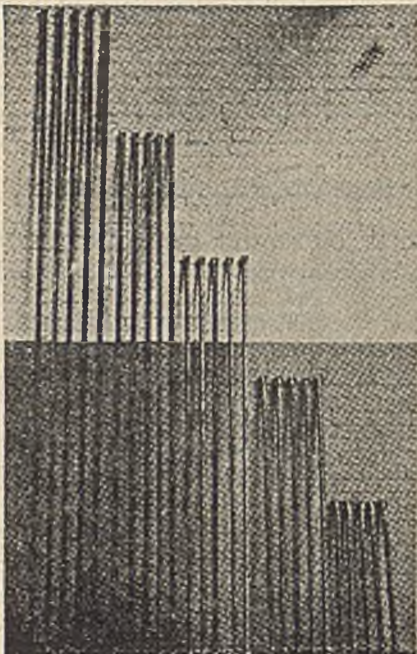



Fig. 18.—One shift supply of solid forged steel for one drill.
Fig. 19.—One shift supply of jack bit rods.

Care should be taken to renew the front cylinder washer of the drill when it becomes worn, as this part performs a very important function in connection with preserving the condition of the drill.

When the drill is raised while the air is still on, the shank no longer absorbs the full blow of the piston, and the front cylinder washer serves to build up air pressure which cushions the blow. This is accomplished by trapping air between the piston and the washer, and in order to do this the washer must be in condition to prevent air leakage. It is obvious that if the washer be allowed to wear beyond a certain stage, it will no longer

Fig. 20.—This piston has spalled at the edges from being used in a drill having worn chuck bushings.

fulfill this function and the piston will be allowed to exert a battering effect which will result in serious damage to both piston and washer.

Short drill steel shanks should be avoided.

If the length of the shank be materially less than normal, the piston will not strike the end of the steel with full force. This materially reduces the drilling speed of the drill.

Long shanks cause short stroking of the piston and are, accordingly, likewise to be avoided.

Standard lengths of shanks are given by the drill manufacturer.

Chuck Bushings.—Fig. 20 shows what happens when chuck bushings are allowed to wear unduly. The steel is no longer held in direct line with the piston. Instead of striking direct blows on the end of the drill steel, the piston will then strike the edges of the shank.

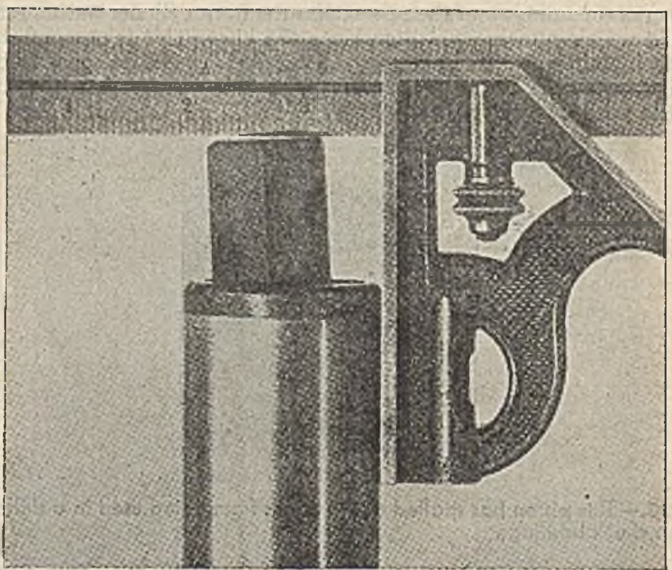


Fig. 21.—This shows the play of the drill shank that is permitted when the chuck bushings become worn.

Repetition of this procedure will soon cause spalling of the piston. Worn bushings is one of the leading causes of piston failure.

The striking of a shank on the edges will round them off and tend to produce a cone or point at the shank end.

If the same shank be then used in a drill having a good chuck bushing which holds its steel in line with the piston, the piston will strike direct blows upon this relatively smaller shank end. This in turn will promote breakage of the piston around the central opening.

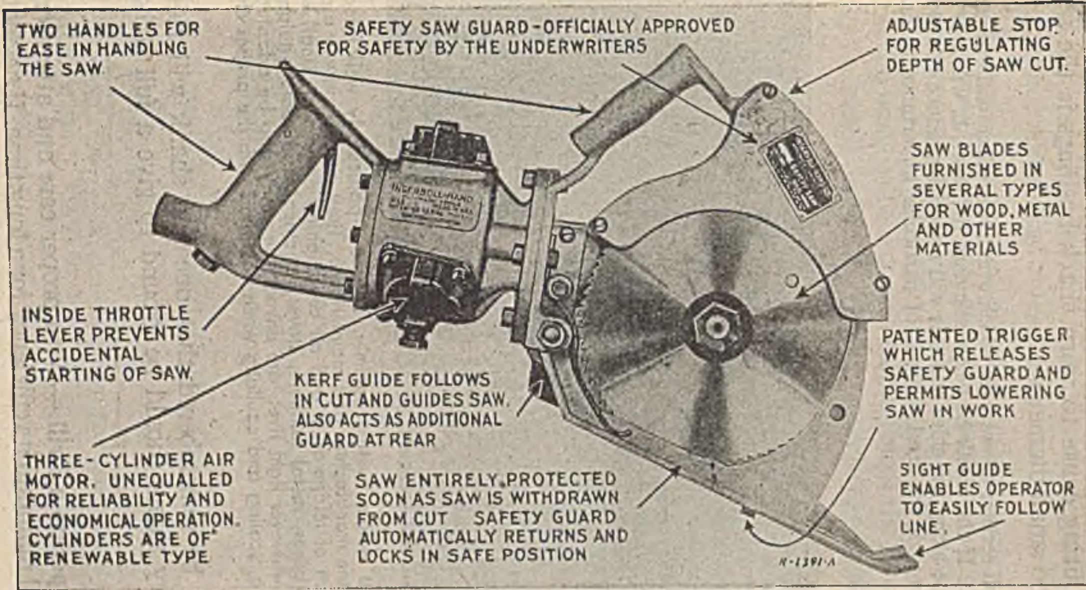


Fig. 22.—Safety type pneumatic saw. View with names of parts. Application is for trimming off sheet piling, cutting concrete forms and planking, ripping boards, trimming roofing, etc. The blades are easily removed and replaced. Cross cut, rip or combination cross cut and rip blades can be supplied. Special blades are obtainable for cutting hard rubber, fibre, bakelite, stone, metals, tile, cable and other materials.

A contributing cause to worn chuck bushings is the use of shanks of too small diameters.

The small section allows play which eventually causes enlargement of the bushing. If the shank be below standard diameter, it will also allow air to escape along the sides and this will materially reduce the blowing capacity through the steel. This is of particular importance in drilling deep holes.

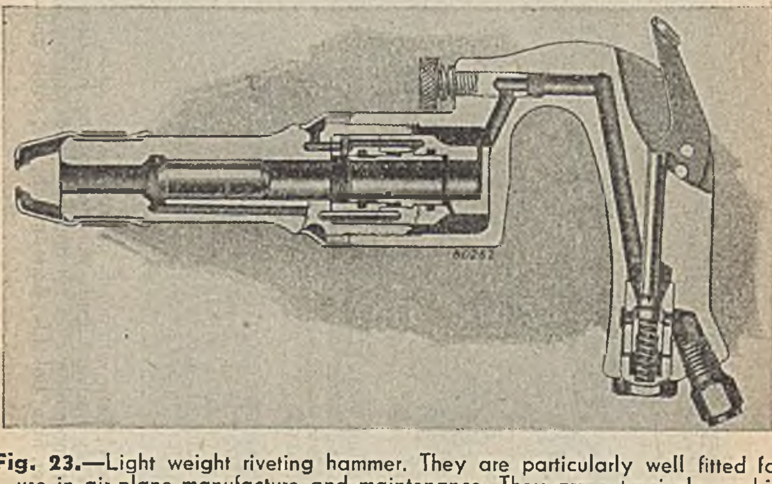


Fig. 23.—Light weight riveting hammer. They are particularly well fitted fo use in air plane manufacture and maintenance. They are extensively used ir the fabrication of refrigeration and air conditioning equipment, small wate craft, trucks, buses, metal furniture, stoves, electrical and radio equipment toys, in fact wherever light rivets are driven. They may also be adapted fo light chipping, scaling and caulking when equipped with the proper chisel

It should be remembered that one bad shank will ruin a piston. Every shank should be ground to give a full striking surface.

How to Operate a Drill.—With proper care and attention, a rock drill will maintain its efficiency for a long time. If neglected it will, as will any mechanical appliance, gradually fall off in performance and finally break down.

The wearing of one part affects others, so it is advisable to make replacements when they are needed and not to wait until major repairs must be made.

It is the purpose here to set down a few of the more important things to watch and to make a few practical suggestions which

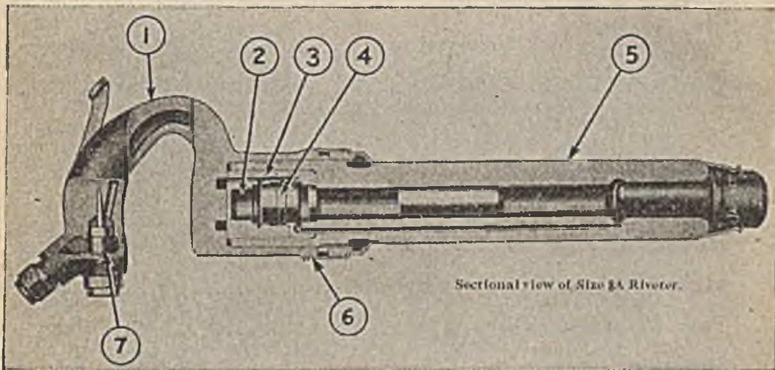


Fig. 24.—Riveting hammer. **The parts are:** 1, handle; 2 and 3, valve box; 4, valve; 5, barrel; 6, automatic spring locking device; 7, throttle valve.

will guide the man who knows little or nothing about drilling work. If he will follow them, he will lessen his troubles and increase his footage per shift.

Ques. What is the first thing to do?

Ans. Keep the drill well lubricated.

The importance of lubrication cannot be over-emphasized. Running at such high speeds, the surfaces of rubbing parts will develop friction heat in a few seconds of dry operation, which will not only destroy hardness but minute fractures will develop and failures of the parts will be hastened.

Ques. What is the indication that a drill is receiving sufficient lubrication?

Ans. Oil will appear on the drill steel shanks.

The following physical properties are representative of oils which have been found satisfactory under actual commercial service in the field, and all oils purchased to this specification shall conform to these requirements:

Flash point (open cup)	350° F.		min.
Viscosity at 100° F. (Saybolt)	450	Sec.	min.
(Universal)	700	"	max.
Viscosity at 100° F. (Redwood)	380	"	min.
	600	"	max.
Viscosity at 100° F. (Engler)	12.4°		min.
	19.0°		max.
Four point	15° F.		max.
Mineral acid neutralization number	.10		max.
Free fatty acid (% Oleic)	.40		max.
Steam emulsion number	300		min.

Good results can be had only from an oil selected especially to answer the requirements of rock drills.

It must resist washing away, must withstand low temperatures (in many cases below 32° F.) caused by sudden expansion of air in the tool, and must flow properly to all parts which need lubricating. A poor oil becomes sticky or stringy.

Ques. What are air line lubricators?

Ans. These are oil containers which are placed in the compressed air line leading to each drill.

Ques. How do they work?

Ans. The oil is picked up by the air and carried under pressure to all parts of the drill that require lubrication.

Ques. What results are obtained with a dirty drill?

Ans. It will work poorly and fail prematurely.

Ques. How are drills cleaned?

Ans. They are dismantled and the parts are cleaned with kerosene and then oiled as they are reassembled.

Fig. 25.—A four cylinder piston type wood borer drilling 1½ in. holes in 12 x 12 timbers on bulkhead construction work.

If any of the parts be found to have been scored because of inadequate lubrication, the drill should not be run further until the scored surfaces have been removed. This can be done by patiently rubbing the affected areas with fine emery cloth.

Sort and Segregate Drill Steel.—Drill steel is made in cross sections of several shapes. In some cases solid steel is used, but the steel employed for most operations has a hole through its center for passage of the air or water, or both.

For Jackhamer work ⅞ in. or 1 in. hollow, hexagonal steel is generally used.

Drifter type drills use hollow, round steel 1 to 1½ ins. in diameter. The most common size is 1¼ ins.

Ques. What should be noted in regard to selection of drill steel?

Ans. Selection of a suitable grade is important.

Ques. What are the requirements for a satisfactory steel?

Ans. It must not only resist the intensive pounding action to which it is subjected, but must also be capable of receiving innumerable bit forgings without developing weakness.

In hollow steel the hole should be centralized, and of the same size throughout its length. The surface of the inside walls should be smooth and even.

Indentations, furrows, or any other unevenness pave the way for the starting of fatigue cracks, which cause breakage and greatly reduce the life of the steel.

Ques. Before a drill runner starts work what should he have close at hand?

Ans. Adequate supply of drill steels, cut to proper lengths and bearing well sharpened bits of the correct sizes.

Much time will be saved by sorting the steels and segregating them into groups. When a steel becomes too dull for further service, it should be put aside where it will not become mixed with the sharpened steels.

If the drill runner become systematic in arranging his steels, he can select the correct change without confusion or delay.

When sharpened steels are delivered, they can be placed in their allotted positions quickly.

It pays to have plenty of drill steels on hand. "Waiting for steel" contributes to higher drilling costs on many jobs.

Air Delivery Lines.—Before starting work, the driller should make certain that the compressed air line is in condition to function properly. Some points to watch are:

1. See that the valve from the air line to the drill is fully open.
2. Make certain that there are no small connections in the line to restrict the flow of air to the drill.
3. Blow out the air line to the drill to eliminate any moisture that may have collected in it.
4. Do not connect the hose to the bottom of a tee manifold where moisture may collect.
5. Do not use long lines of small diameter pipe.
6. Keep all pipe lines and hose connections in good repair.

Ques. What should be noted as to leaks?

Ans. A great amount of compressed air is wasted through leaks which are so small as often to be considered unimportant.

Pipe lines, hose and valves should be inspected regularly for leakage.

Ques. How are large and small leaks detected?

Ans. Large leaks are detected by the sound of escaping air, or by the rush of air against the hand. Smaller leaks can be

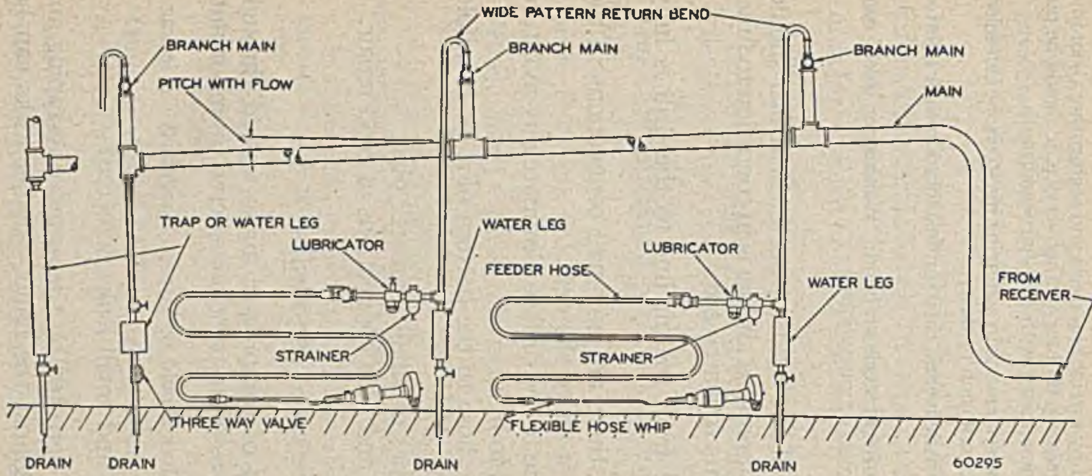
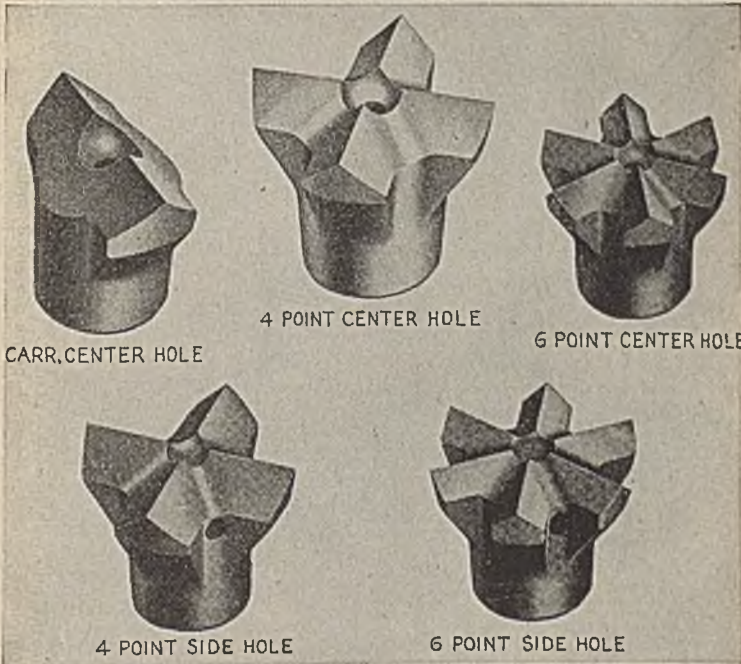


Fig. 26.—Proper layout for air piping system. Air mains and lines should be large enough to avoid excessive pressure loss under conditions of maximum flow. In air lines as in after-coolers, water is continually precipitated as the air cools. For this reason pipe lines should be provided with a means of draining or trapping out this water before it reaches the hose outlets. It is advisable to pitch the mains in the direction of air flow so that both gravity and air flow will carry the water to traps or water legs located at frequent intervals. These should be drained regularly and never allowed to become full and inoperative. Use of automatic traps will eliminate manual draining and the possibility of traps becoming full. However, traps which drain to sewers can waste considerable air unless regularly inspected and kept in good working order. Provision for quick and positive inspection of operation and for leakage should be made when traps are installed. Three-way cocks or stop and waste cocks in the drains are usually provided for this purpose. To aid in preventing condensed moisture reaching the tools, down pipes or hose connections should never be taken directly from the bottom of air pipes or mains. Connection should be made at the top of the main and a long radius return bend used.

found with the aid of a lighted candle, or by coating the joints with soapy water.



Figs. 27 to 31.—Various jack bits. Fig. 27, carr center hole, also made with side hole; fig. 28, four point center hole; fig. 29, six point center hole; fig. 30, four point side hole; fig. 31, six point side hole.

NOTE.—*Points relating to hose*: It will be found economical to use air hose of good quality. It should be tough enough to withstand the abrasion, kinking and general hard service to which it is subjected, and strong enough to carry the required pressure without expanding or developing weakness. Equally important, it should have a lining that will resist the deteriorating action of heat and oil. In this connection it might be mentioned that the life of hose will be prolonged by extracting heat, oil and moisture from the air by using a two stage compressor and an after-cooler. Wire wound hose was formerly in much favor, but recent improvements in cover compounding have made it possible to produce a rubber covered hose on which wire winding is unnecessary. When wire wound hose is crushed or otherwise deformed, the wire restricts the flow of air. When the wire becomes worn or broken, it must be handled guardedly to avoid cuts and scratches.

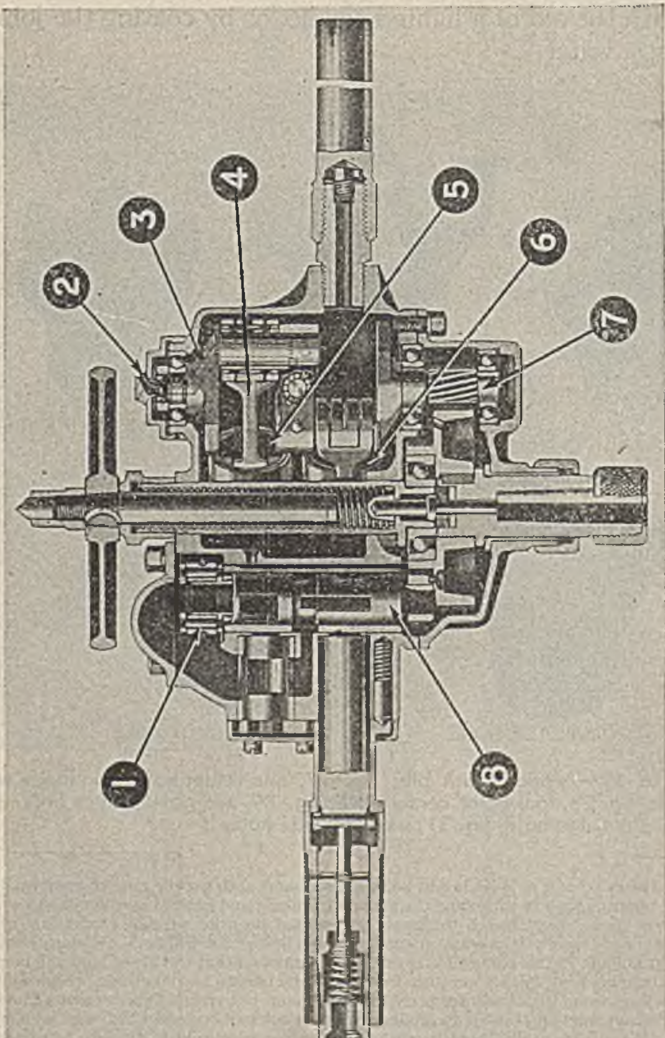


Fig. 32.—Long stroke piston pneumatic drill with governor control. The parts are: 1, governor; 2, case vent; 3, crank; 4, connecting rods; 5, pistons; 6, cylinder liners; 7, crank pinion; 8, main valve.

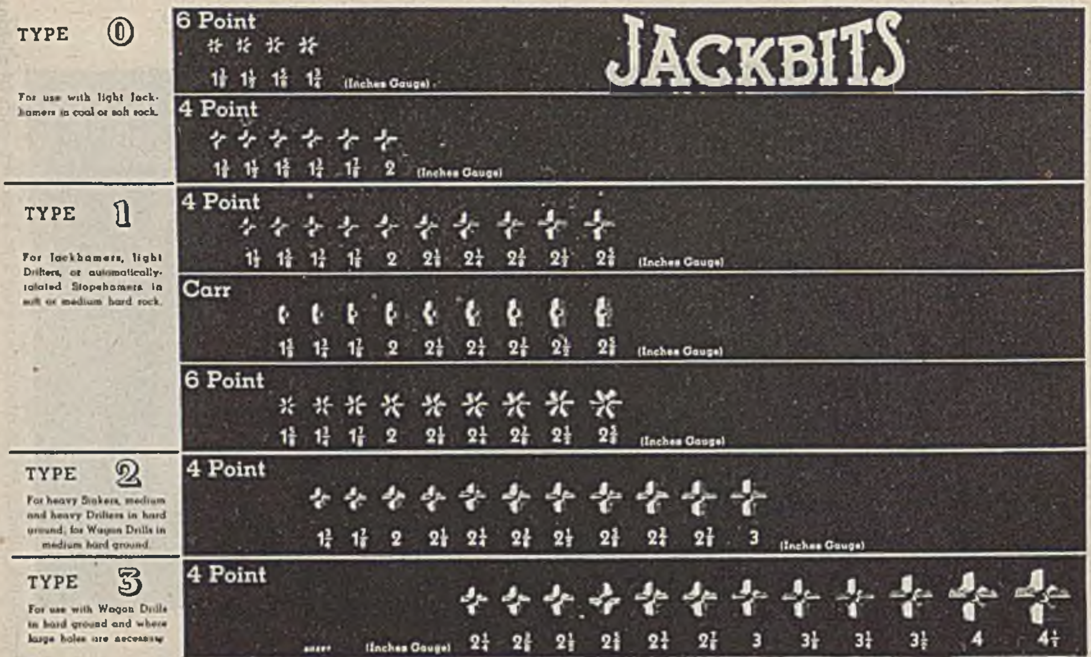


Fig. 33.—Jack bit chart.

Points On Bit Sizes.—In Jackhamer drilling it is customary for the convenience of the operator to change solid forged steels with every two feet of depth. As the abrasive action of the cuttings and of the sides of the hole serves to reduce the gauge, it is the established practice to use a bit of ⅛ in. smaller gauge with each succeeding change of steel.

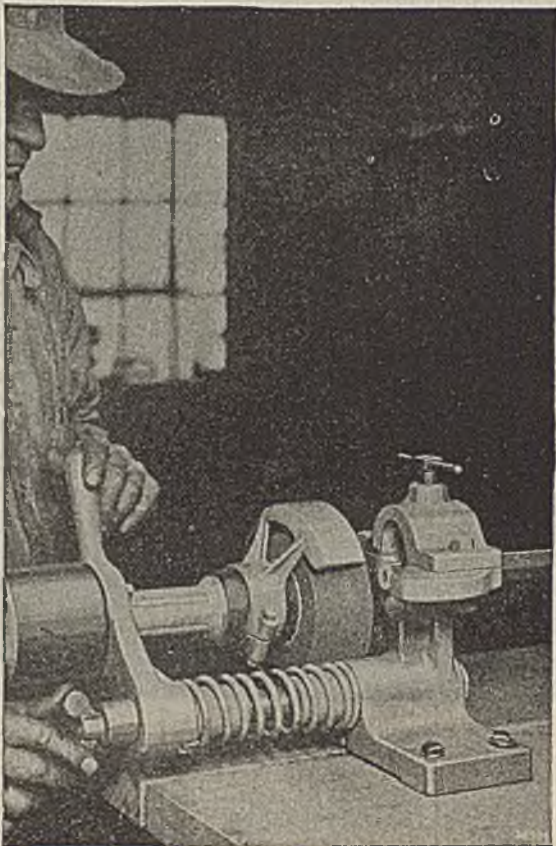
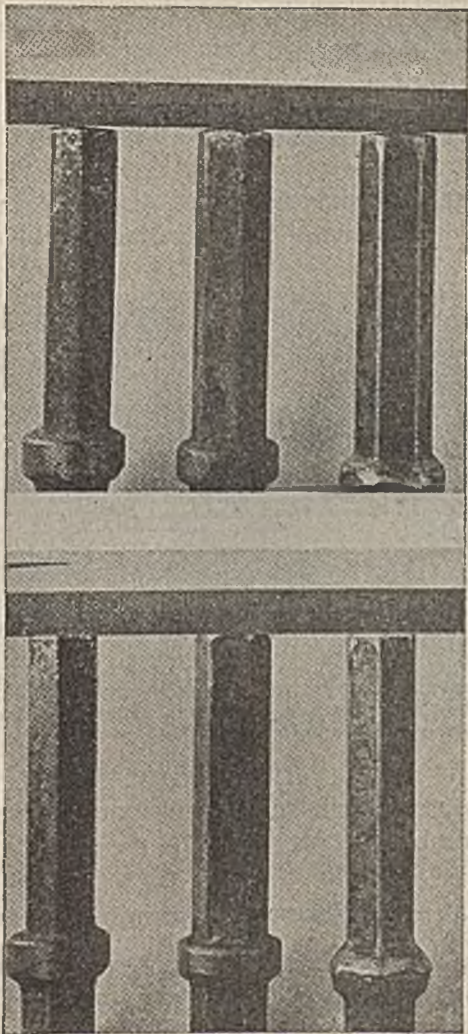


Fig. 34.—Shank grinder which also can be used to keep the ends of pistons true.



Figs. 35 and 36.—Three steel shanks before and after grinding operations.

This means that if you are drilling a 10 ft. hole and make four changes of steel, you will be using for the last two feet of the hole a bit ½ in. smaller in diameter than the starting bit.

For economy in drilling it is important to select the proper starting bit which will permit making the required number of steel changes and to bottom the hole at the size required.

Ques. What are jack bits?

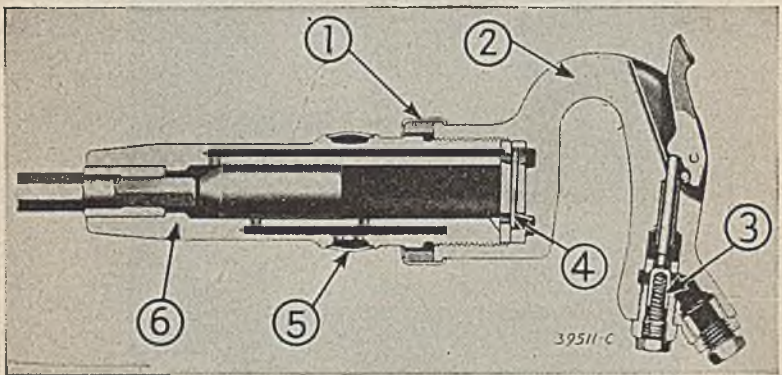


Fig. 37.—High speed "flapper valve" hammer. The parts are: 1, locking arrangement; 2, handle; 3, combination piston and poppet type throttle valve; 4, valve seat; 5, exhaust deflector; 6, barrel.

Ans. They are detachable bits which are screwed onto jack bit rods of suitable lengths.

Ques. What is the advantage of using jack bits?

Ans. The use of jack bits will often save considerable money by eliminating the necessity of changing the size of the bit and, therefore, enabling the hole to be drilled at the same diameter throughout its course.

In any rock which is not so hard as materially to reduce the gauge so as to lessen the cutting efficiency of the bit, it is possible to use the same bit for drilling the entire hole by merely successively transferring it to the several required lengths of jack bit rods as the work progresses.

Squaring Up Shank Ends.—The important bearing that the condition of the shank end has upon the life of a rock drill piston has been explained. Because of this, the same careful attention should be given to shanks as to bits. Neglect of shanks inevitably raises upkeep costs on drills.

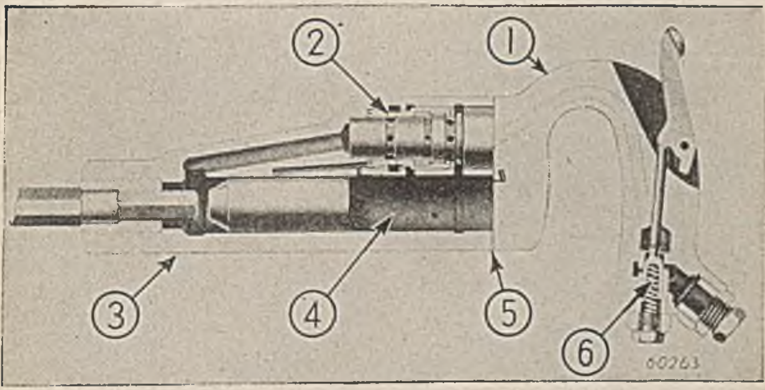


Fig. 38.—Offset valve hammer. The parts are: 1, handle; 2, valve; 3, barrel; 4, piston bore; 5, handle; 6, throttle.

Before being sent out of the shop, all shank ends should be ground true.

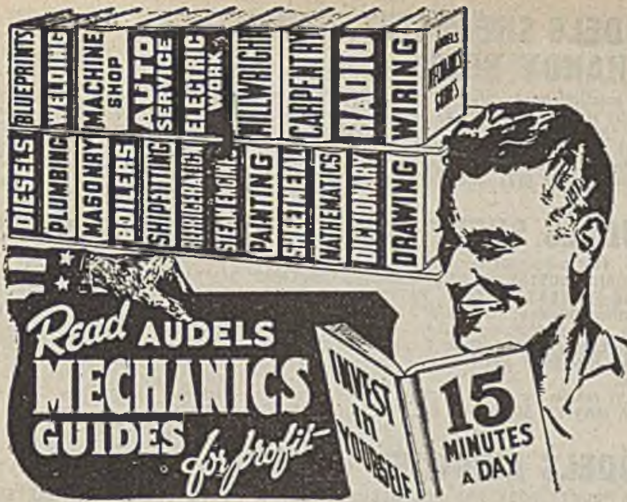
For this purpose there has been developed the air operated 4K shank grinder. It holds the drill steel at right angles to the grinding face and thus insures a true end surface.

This grinder may also be used for squaring up the faces of pistons when required and thereby securing longer life from them.

Care should be taken in grinding pistons to avoid burning or grinding checks. Not more than 1/8 in. should be ground from a piston.

If more be removed, there is danger of exposing the soft, tough core of metal that has not been case hardened.

Smaller pedestal type grinders may be used for these purposes, but the trueness of the face then depends upon the operator.



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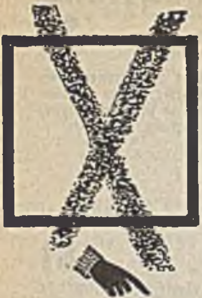

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