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GENERAL CRITERIA OF LUBRICANT SELECTION FOR FRICTION POINTS IN MINING MACHINES

Summary. The paper presents general criteria of lubricant selection for friction points in mining machines. The criteria are following:
- durability of friction point elements, most often determined by occuring destroying tribological processes,
- efficiency of a friction point, being a measure of dissipaled energy quantity,
- thermal condition determining stability of friction and wear processes and durability of a lubricant.

These criteria are essential in the range of optimum selection of the lubricant kind (liquid lubricant or grease), application of the best solution of a lubricant system and a properly efficient seal system.

It has been proved that the improvement of tribological points durability requires considering this matter in a system formulation of feedbacks between technological stages of designing, constructing and exploiting. Accepting of the criteria mentioned above is a necessary stage before the selection of rheological properties of a lubricant on the base of hydrodynamic or elastohydrodynamic lubrication theory.

1. INTRODUCTION

Improvement of planning, constructing and exploitation processes of the machines is the main task in technical-economical strategy of mining machines industry development nowadays, as well as in the nearest fifteen years. The analysis of outlays for the machine exploitation in Poland [1] that has been carried out in recept years shows that these outlays are relatively high.

In the period of the machine work the outlays for the machine repairs come to 60 - 80% of costs of production, and in the case of the machines working in particularly difficult conditions the outlays are several times (4 - 8 times) bigger than costs of production.

Problem of improvement of machine exploitation economy must be considered as a whole taking into account the necessity of a system formulation of feedbacks between the stages of designing, constructing and exploitation as it is presented in fig. 1. Fig. 1 is presented in step system of designing constructing and exploitation stages on purpose. Such presentation is to suggest different steps of difficulty occuring in the connection between the stages, and simultaneously to point at the range of necessary effort, especially while formulating instruction manual and gathering same information concerning the behaviour of practical features assumed by the designer, which are a system of the machine quality evaluation by a worker. We can risk to daim that correctly formulated instruction manual, compliance with it, and systematic gathering of information about practical features of the machine are the basis to improve the economy of means of production.

During the machine exploitation stage one of main problems is the matter of a proper selction and application of lubricants. Technical analysis of the causes of too early withdrawing the machines from the exploitation shows that in about 30% the cause of shortening of their assumed general durability are the damages resulting from the destroying tribological processes, i.e. connected with the excesseve wear of friction points. Particular big wear of frication point elements occurs when the machine works in a dusty environment with a big concentration of abrasive particles and with a presence of aggresive water. In such cases the most often causes (85 - 90%) of withdrawing the machine from work are the processes of tribological wear [2]. Influence of a lubricant and lubrication conditions on the elements wear is oboviousa. Also the following principle is obvious: <u>in every technical case, wherever it is possible, dry friction should be eliminated, boundary and mixed fiction should be limited, and fluid friction should be kept</u>.

From the presented in fig. 2 approximated wear intensity course of gear wheels of mining machine transmission gears in the function of exploitation time and different kinds of friction we can see that the wear at fluid, mixed and dry friction forms a sequence of proportions showing that the wear at dry friction is over 1000 times bigger than at fluid friction and thus illustrates the influence of lubrication, and underlines, first of all, technical-economical importance of this problem. To fulfill this basic principale of continues work of tribological point in the conditions of fluid friction in many practical cases is vary difficult, because of complexity of exploitation process and many limitations. Formulation of criteria and general pricipôes of lubricant selection and making use of detailed information conerning theory of friction, wear and lubrication allow us, with great probability, to achieve the aim: to decrease cosiderably the intensity of friction wear of cooperating elements.

2. GENERAL CRITERIA OF LUBRICANT SELECTION FOR FRICTION POINTS IN MINING MACHINES

General criteria of lubricant selection for friction poinst in mining machines are based both on the essence and the main aims of lubrication. The essence of lubrication means introducing between two surfaces of friction the third body called a lubricant - to change unprofitable external friction into friction inside the lubricant layer. Illustration of lubrication essence achieved by hydrodynamic efect is a Striebeck's curve (fig. 3).

The main aims of lubrication are:

1) increasing of surface durability of cooperating elements and of reliability of friction poinst,
2) increasing of friction point efficiency what is connected with decreasing of dissipated energy quantity (lasses) and means minimalization of coefficient of friction,
3) decreasing of a thermal state of a friction point, what means taking the heat from the contact area mainly with the lubricant stream and thermal conduction,
4) eliminating wear products from the friction area what assures stability of friction and wear conditions.

Thus, the basic criteria of lubricant selection for friction points are:

- durability,
- efficiency,
- thermal state.

2.1. Durability as a criterion of lubricant selection

Criterion of durability, understood as a friction point work time until boundry wear occurance must be referred to basic distroying tribological processes which are possible to occur.

Typical distroying tribological processes in kinematic pairs with silding motion are:

- adhesive wear,
- thermal-adhesive wear (seizing),
- abrasive wear; in pairs with rolling motion and rolling-sliding motion (except the ones mentioned above),
- fatigue spalling of the surface layer (pitting).

Adhesive wear is characterized by the classic Lorentz curve (fig. 4).

From the presented curves in fig. 4 we can draw the following conclusions:

- achievement of linear dependence of a wear as a time function in the fixed area ($t_2 - t_1$) is practically possible only in the conditions of sufficient and stable lubrication. Friction cooperation of the elements made of typical construction materials (steel, cast steel, cast iron, non-ferrous metals alloys) without any lubrication is impossible because of high adhesive affinity, what is connected with a considerable wear and quickly increasing non-linear characteristics,
- application of a lubricant with antiwear additions decreases the intensity of wear I_z, and in some cases of so-called group atoms transmission (so-called Garkunow's effect) the wearless state may occur,
- by application of a suitable lubricant the character of wearing-in can be considerably shortened (in the area 0, t_j) and changed.

Thermal-adhesive wear, called also seizing, is a process of avalanche adhesiv joining caused by the high temperature in the contact area which is a result of breaking of boundary layer of the lubricant and direct influence of microirregularities of both friction surfaces. Antiseizing properties of a lubricant are described by the notion of so-called lubricity and are determined by a proper test (usually in a 4-balls apparatus or Timken's apparatus). In the result of the test a seizing curve is determined, as in fig. 5, or a standard value called a load of seizing is determined.

Bacause during the minig machine exploitation a considerable nonstability of loading conditions occurs (strong stochastically variable overloads), there is a great probability of breaking of lubricant layers which separate cooperating surfaces, and occurence of seizing. Thus, in such a case there is necessity of application of lubricants with antiseizing additions. Yet, it should be mentioned that together with the increase of oil activation with some antiseizing additions, the load of causing pitting (table 1) decreases, and moreover the intensity of pitting increases (fig. 6). So, if the analysis of the conditions of friction point load does not point at the possibility of considerable overload occurence, the application of oils strongly activated with antiseizing additions is not necessary.

Wear of pitting type is connected with the process øf fatigue of surface layer in the result of repeated loads. This kind of wear is described, similarly to classic volume fatigue, by Wöhler's curve.

A course of pitting curves for transmision gear oils Hipol 10, Hipol 15 and Hipol 30 is presented in fig. 7.

In the figure we can see that togheter with the increase of oil viscosity the fatigue contact strength increases. This fact results both from the theory of contact load distribution in the oil viscosity function and from diminished oil penetration into thefatigue craks occuring from the friction surfaces. Thus the oil selection, when durability determined by the pitting process is taken into account, is based on the selection of optimum viscosity which guarantees bigger durability but also so-called hydraulic losses (losses of flow or mixing) that are possibly very small.

Abrasive wear is a characteristic tribological process consisting in microcutting of the surface by the hard elements of the abrasive material being present in the oil stream or on one of the friction surfaces. This kind of wear connot be a base for oil selection from the point of view of friction point durability. The process of abrasive wear, being a specific exploitation case, should be eliminated by applying effective seal to the friction point andoil filtration.

In friction points of the machines working in heavy industry, when variable character of exploitation conditions (dynamic and kinematic) are taken into account, the occurence of each kind of wear mentioned above in different periods in machnie exploitation is possible, as it ispresented in fig. 8 as an example.

2.2. Efficiency as a criterion of oil selection

Efficiency meant as a quotient of effective energy E_u to introduced energy E_{wk}

$$\eta = \frac{E_u}{E_{wk}} = \frac{E_{wk} - E_{str}}{E_{wk}} = 1 - \frac{E_{str}}{E_{wk}} \tag{1}$$

is a general measure of correctness of friction point work.

Assumnig that there are not any mistakes in designing and producing which could cause bigger losses of energy E_{str}, the total dissipated energy correlates directly to the coefficient of friction and hydraulic losses related to the lubricant forcing through and mixing.

In some references you can find the formulae connecting friction coefficient with friction parameters for given elementary tribological pairs such as: silde bearing, rolling bearing, in gear etc. Yet this problem is solved mainly to determine the thermal state of a friction point, and not to look for oil viscosity as a function of the assumed friction coefficient. Oil viscosity is selected from the point of view of stable conditions of fluid friction. But very important is to determine the optimum oil viscosity taking into account the friction conditions oil minimalization of hydraulic losses. Thus the quantity of hydraulic losses is one of the basic criteria decreasing the oil viscosity to the value that is necessary to keep the fluid friction.

From the flow theory [4] we know that losses of viscid fluid flow in pipes are dependent on Reynold's number defined as:

$$Re = \frac{\bar{v} \cdot d}{\nu} \tag{2}$$

where:

ϑ - flow velocity,

d - hydraulic diameter,

ν - fluid kinematic viscosity.

For the lammar low (Re $\leqslant$ 4000) the hydraulic losses coeficient C_s can be presented as:

$$C_s = \frac{64}{Re} \tag{3}$$

and for the turbulent flow (Re = 4000 - 8000) with Blasius formula:

$$C_s = \frac{0.316}{Re^{0.25}} \tag{4}$$

So, together with the increase, for example: duoble increase of lubricant viscosity causes:

- in the case of the lammar flow about 40% increase of the resistance of flow,
- in the case of the turbulent flow about 20% increase of the resistance of flow.

In similar dependences are the losses resulting from the movement of bodies of cylindrical or spherical shape in viscid environment. It the range of Reynold's number Re $> 10^5$ we can agree, according to Lamb [5], that the losses coeffiecient discribes the dependence

$$C_m = \frac{16}{Re} \approx \frac{50}{Re} \tag{5}$$

As we see from (3), (4) and (5), the hydraulic losses depend very much on the lubricant viscosity. Thus the selection of lubricant viscosity must be a rational compromise between the criterion of fluid friction behaviour determining a proper durability of the friction point and the criterion of minimum hydraulic losses influencing very much on receiving sufficiently high efficiency of the friction point and its lubrication system.

2.3. Thermal state of the friction point as a criterion of lubricant selection

Heat produced during friction and its abstraction define the thermal state of the friction point, and particularly the temperature in contact zone, the temperature of cooperating elements and the lubricant temperature. It is known that for the mineral oils when the temperature increases the viscosity dereases, and the bigger decrease gradient $\frac{d\eta}{dt}$ the smaller viscosity index IW. We can assume that for the most mineral oils used now in heavy industry the following approximate dependence (in the range of plus temperature) is obligatory:

$$\frac{\eta_T}{\eta_o} = c \cdot \left(\frac{T_o}{T}\right)^3 \tag{6}$$

where:

η_o, η_T - appropriate viscosity in reference temperature T_c and working temperature (usually $T_o = 40^oC$),

c - constant for given oil.

From (6) we can see that double temperature increase, for example from 40^oC to 80^oC, causes eight-times viscosity decrease. The increase of the friction point temperature causes the change of friction conditions, it means the increase of friction coefficient (fig. 9) and wear intensity (fig. 10).

The friction point temperature is defined by the main exploitation parameters, it means: load intensity, velocity and coefficient of friction, so the unit power of friction is:

$$N_T = \mu \cdot p \cdot \vartheta \tag{7}$$

where:

μ - coefficient of friction,

p - unit pressure,

ϑ - rubbing speed.

So the most often the thermal cirterion of lubricant selection is defined as a function of the parameters p and ϑ, as it is presented in fig. 11.

In the Figure we can see that the limit of grease application is considerably lower than of oils. It results from this that:

- at lubricating with oil the heat dissipation is by conduction (according to Newton's equation) and by convection in the fluid stream,
- at lubricating with grease the heat flow is worse than in the case of oil because of lower thermal conductivity.

Thus, taking the temperature criterion into account, we should apply oils and not grease in friction points with the high value of the product (p, ϑ).

3. CONCLUSIONS

The rational selection of a lubricant for friction points in mining machines should include two necessary, following each other stages:

- formulation of general criteria and their application to a given friction point determined by its geometrical features, material and expoitation conditions,
- selection of rheological proprieties of a lubricant on the base of the theory of hydrodynamic or elastohydrodynamic lubrication.

The general criteria determining reliability of friction points and, to some degree, reliability of winning and transport machines are: durability, efficiency and thermal state of friction point.

Durability is determined by the comrse (characteristics) of the main distroying tribological processes:

- adhesive wear,
- adhesive-thermal wear,
- surface fatigue,
- abrasive wear.

Efficiency, as a measure of dissipated energy quantity is a basic of evaluation of the friction point work correctness, it means both design solution of tribological pair and correctness of lubricant selection.

Thermal state of the friction point determines stability of friction and wear, and also durability of a lubricant.

Practical application of the general criteria of lubricant selection for friction points in mining machines is a compound problem of optimalization and requires carrying out of exact analysis making use of computers and experiments.

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Table 1

Comparison of teeth resistance to seizing and contact fatigue for different oils

Oil	Resistance to seizing		Fatigue contact strength
	testing in a machine with test gear wheels according to FZG		testing in a cylindrical frication machine
	seizing grade	seizing load N	limit of contact fatigue N/mm^2
Mineral oils withaut antiseizing additions:			
$\nu 50^{\circ}C = 60$ cSt	6	4000	1900
$\nu 50^{\circ}C = 115$ cSt	7	5500	2000
$\nu 50^{\circ}C = 250$ cSt	9	9000	2100
Mineral oils with anti-seizing additons of S-P type:			
Gear oil SAE 90 with weak active additon	10	11000	1800
Mineral oil No 1 + 4% of S-P additon	>12	>16000	1700
Gear oil SAE 80 with medium active addition	>12	>16000	1650
Gear oil SAE 90 with strong active addition	>12	>16000	1700

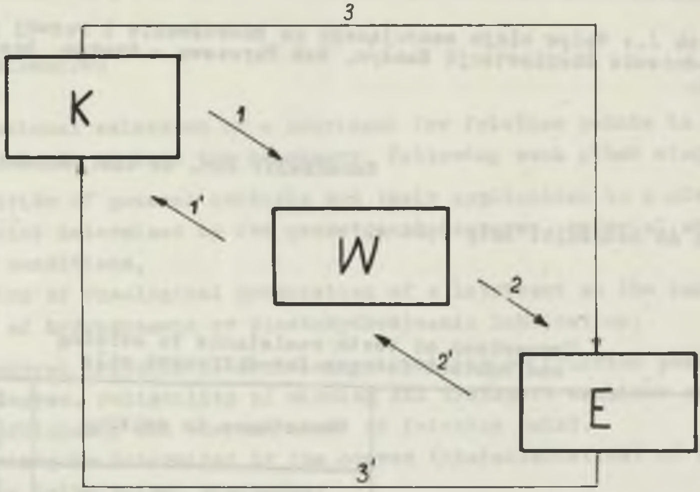

Fig. 1. System of feedbacks between the stages of designing K, constructing W and exploitation E

Rys. 1. System sprzężeń zwrotnych pomiędzy fazami konstruowania (K), wytwarzania (W) i eksploatacji (E)

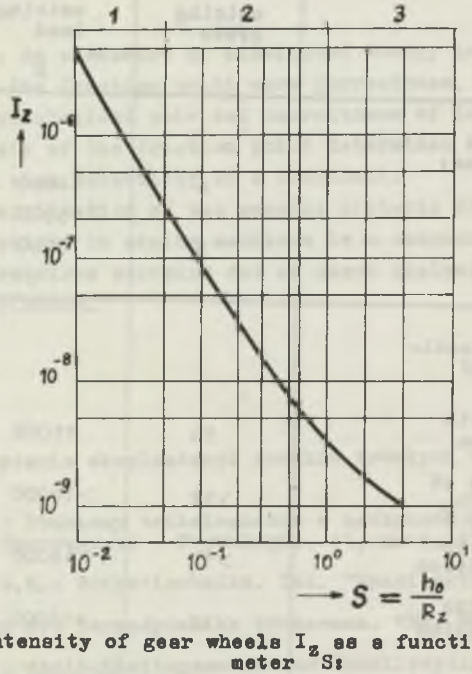

Fig. 2. Wear intensity of gear wheels I_z as a function of friction parameter S:

1 - dry friction, 2 - mixed friction, 3 - fluid friction

Rys. 2. Intensywność zużycia kół zębatych I_z jako funkcja parametru tarcia S:

1 - tarcie suche, 2 - tarcie mieszane, 3 - tarcie płynne

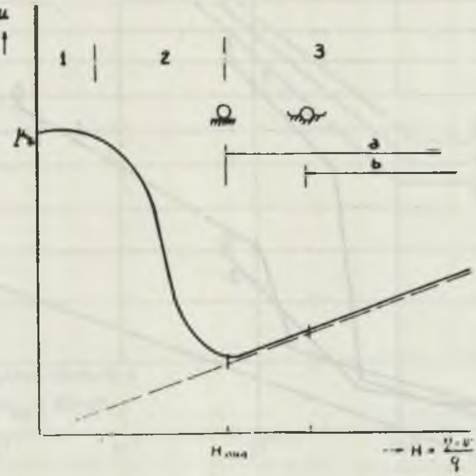

Fig. 3. Striebeck's curve - dependence of frictional resistance T as a function of Hersey's number H:
η - dynamic viscosity of grease, v - linear velocity, q - load intensity
Rys. 3. Krzywa Striebecka - opór tarcia T jako funkcja liczby Herseye H:
η - lepkość dynamiczna smaru, v - prędkość liniowa, q - obciążenie jednostkowe

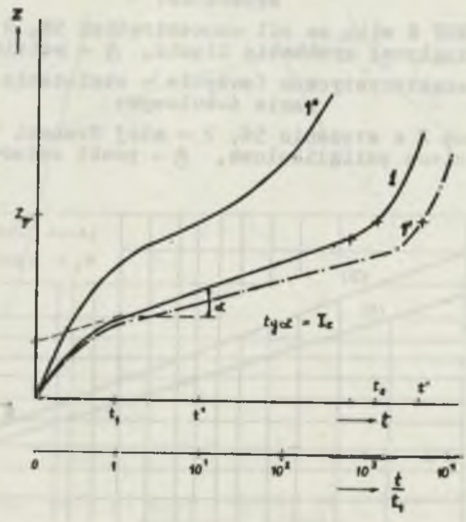

Fig. 4. Lorentz's curve - wear Z as a function of exploitation time t:
1 - wearing-in time, 2 - defined wear time, 3 - excessive wear time
Rys. 4. Krzywa Lorentza - zużycie Z jako funkcja czasu eksploatacji t:
1 - okres docierania, 2 - okres ustalonego zużycia, 3 - okres nadmiernego zużycia

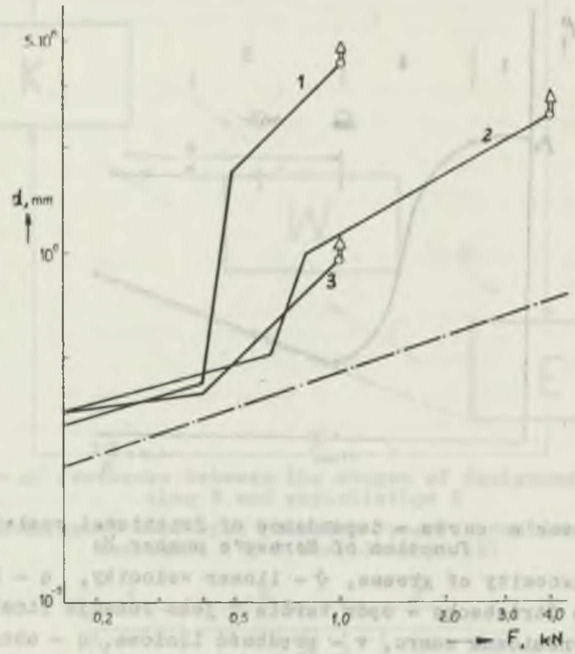

Fig. 5. Characteristic wear curves (wear - loading) determined in a 4-balls apparatus:

1 - emulsion EMULKOP R with an oil concentration 5%, 2 - oil Transol 75, 3 - polyglycol synthetic liquid, ♁ - seizing point

Rys. 5. Krzywa charakterystyczna (zużycie - obciążenie) wyznaczona na aparacie 4-kulowym:

1 - emulsja Emulkop R o stężeniu 5%, 2 - olej Transol 75, 3 - syntetyczna ciecz poliglikolowa, ♁ - punkt zatarcia

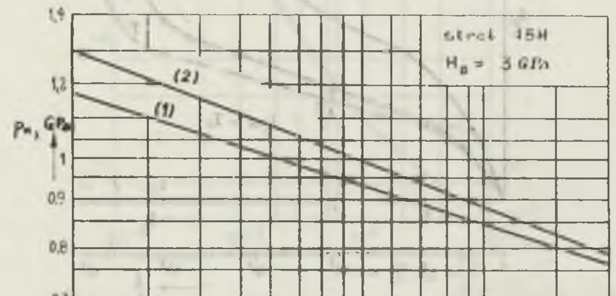

Fig. 6. Curves of fatigue contact strength for steel 45H thermally improved at lubricating with oil Hipol 15 (1) and with basic oil for this oil (2)

Rys. 6. Krzywe zmęczenia stykowego dla stali 45H ulepszonej cieplnie; smarowanie olejem Hipol 15 (1) oraz olejem bazowym (2)

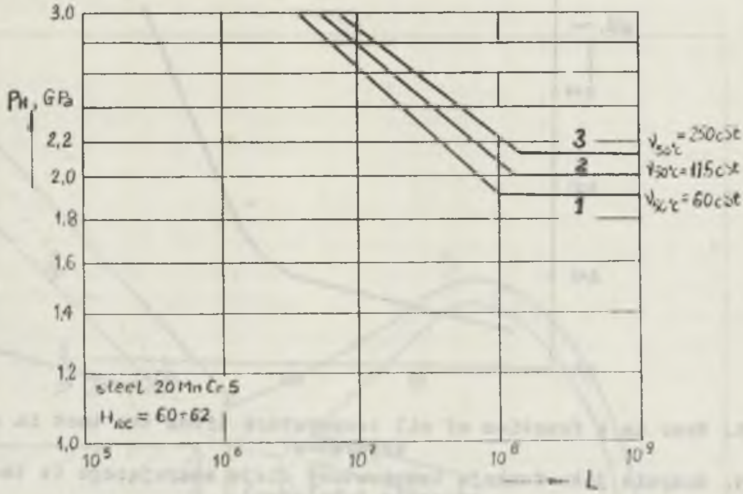

Fig. 7. Curves of fatigue contact strength for steel 45H thermally improved at lubricating with oils 60 cSt (1), 115 cSt (2) and 250 cSt (3)

Rys. 7. Krzywa zmęczenia stykowego dla stali 45H ulepszonej cieplnie oraz przy smarowaniu olejem o lepkości 60 mm^2/s (1), 115 mm^2/s (2), oraz 250 mm^2/s (3)

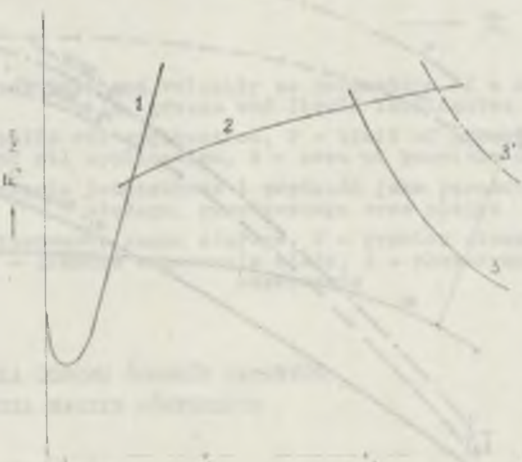

Fig. 8. Tribological wear of gear wheels as a function of loading M and perpheral speed

1 - adhesive wear, 2 - pitting, 3 - seizing (oil without EP additions), 3' - seizing (oil with EP additions)

Rys. 8. Rodzaje zużycia tribologicznego kół zębatych jako funkcja obciążenia M i prędkości obwodowej zębnika V:

1 - zużycie adhezyjne, 2 - zużycie pittingowe, 3 - zatarcie (olej bez dodatków EP), 3' - olej z dodatkami EP

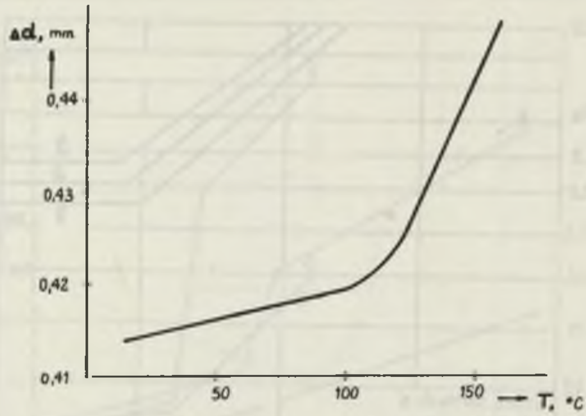

Fig. 9. Wear as a function of oil temperature (from the test in a 4-balls apparatus)

Rys. 9. Zużycie jako funkcja temperatury oleju smarującego (z testu na aparacie 4-kulowym)

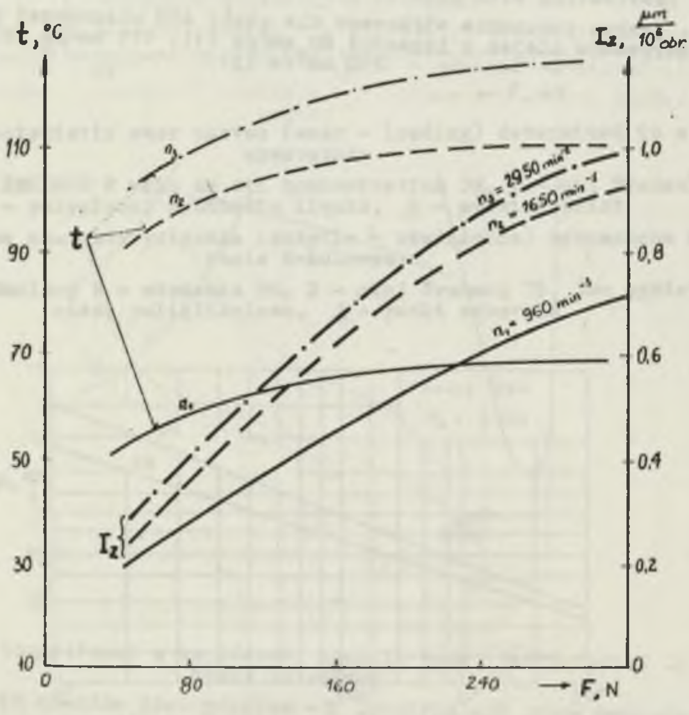

Fig. 10. Dependence of temperature t and intensity I_z on loadin F (from experiments in a frictiom machine of FALEX type)

Rys. 10. Zależności temperatury t i intensywości zużycia I_z od obciążenia F (z testu na maszynie tarciowej FALEX)

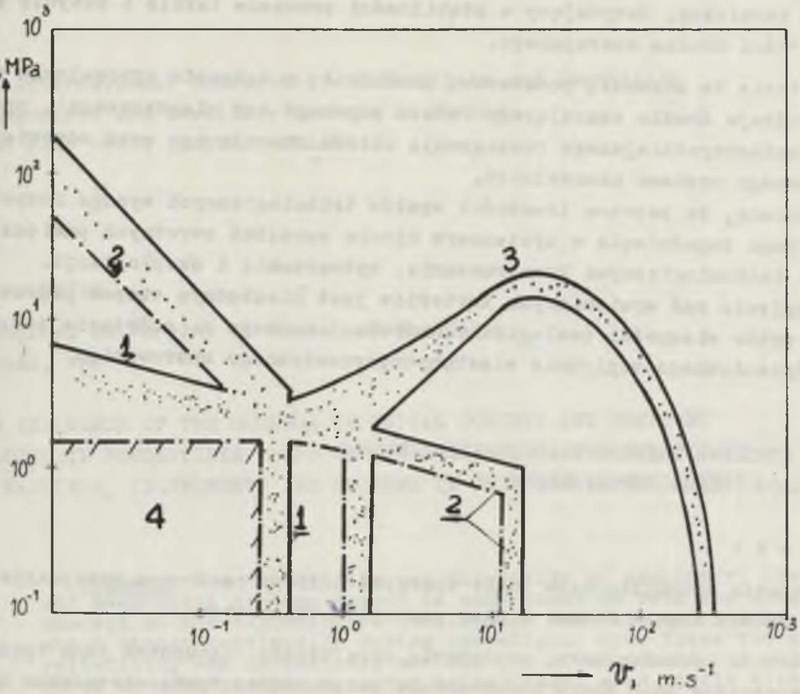

Fig. 11. Contact load and velocity as parameters of a selection of solid oils, grease and liquid lubricants:

1 - limit of solid oil application, 2 - limit of grease application, 3 - limit of oil application, 4 - area of possible dry friction

Rys. 11. Obciążenie jednostkowe i prędkość jako parametry doboru smaru stałego, plastycznego oraz oleju:

1 - granica stosowania smaru stałego, 2 - granica stosowania smaru plastycznego, 3 - granica stosowania oleju, 4 - obszar możliwej pracy bez smarowania

OGÓLNE KRYTERIA DOBORU ŚRODKÓW SMARNYCH DO WĘZŁÓW TARCIA MASZYN GÓRNICZYCH

S t r e s z c z e n i e

W opracowaniu sformułowano ogólne kryteria doboru środków smarnych do węzłów tarcia maszyn górniczych.

Kryteriami tymi są:

- trwałość elementów węzła tarcia, zdefiniowana najczęściej występującymi, niszczącymi procesami tribologicznymi,
- sprawność węzła tarcia, będąca miarą ilości energii rozproszonej,

- stan termiczny, decydujący o stabilności procesów tarcia i zużycia oraz trwałości środka smarującego.

Kryteria te stanowią podstawową przesłankę w zakresie optymalnego doboru rodzaju środka smarującego (smaru płynnego czy plastycznego), przyjęcia najkorzystniejszego rozwiązania układu smarującego oraz odpowiednio skutecznego systemu uszczelnień.

Wykazano, że poprawa trwałości węzłów tribologicznych wymaga rozpatrywania tego zagadnienia w systemowym ujęciu sprzężeń zwrotnych pomiędzy fazami technologicznymi konstruowania, wytwarzania i eksploatacji.

Przyjęcie zaś wymienionych kryteriów jest niezbędnym etapem poprzedzającym wybór własności reologicznych środka smarnego na podstawie teorii hydrodynamicznego względnie elastohydrodynamicznego smarowania.

ОБЩИЕ КРИТЕРИИ ПОДБОРА СМАЗОЧНЫХ МАТЕРИАЛОВ ДЛЯ УЗЛОВ ТРЕНИЯ ГОРНЫХ МАШИН

Р е з ю м е

В статье сформулированы общие критерии подбора смазочных материалов для узлов трения горных машин. К этим критериям относятся:

- прочность устойчивость, стойкость; срок службы элементов узла трения, определяемая чаще всего возникающими разрушающими трибологическими процессами;
- исправность эффективность, коэффициент узла трения, служащая мерой количества диссипированной энергии;
- термическое состояние, от которого зависят стабильность процессов трения, износ и стойкость смазочного материала.

Эти критерии являются основными предпосылками в масштабе оптимального подбора вида смазочного материала жидкой смазки или пластической , приема выгодного решения системы смазки и соответственно эффективной системы уплотнения.

Подчеркивается, что улучшение прочности трибологических узлов требует рассмотрения этой проблемы системным подходом к обратным связям между технологическими фазами конструирования, производства и эксплуатации.

Принятие же названных критериев - необходимый этап, предшествующий выбору реологических свойств смазочного материала, основанный на гидродинамической или эластогидродинамической теории смазки.