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contents

B. IVKOVIĆ

Tribological Approach to Cutting Processes

Applications of the existing and development of the new tribological knowledge in industrial systems in the world are present to a great extent. Without enough tribological knowledge it is not possible to realize the production with high productivity and small manufacturing costs, namely, it is not possible to be competitive enough at the market. The reliability of the production processes (production with minimum delays), as well as the quality of the product, depends upon, also, on the extent of the applied tribological knowledge in the phase of design of products and processes, and in the realization phase.

Conditions under which the contact is being realized between elements of the tribe- mechanical system by which the power and motion transmission is being performed, guiding of elements and information transfer, can be very different from the point of view of the contact zone loading and motion speed. However, conditions under which the contact between the tool and working piece material is being realized in tribo-mechanical systems in which the cutting is realized are very severe. Real contact surface in both basic tribo-mechanical systems (contact between shaving and tool front surface, and tool back surface and machined surface) is very small due to very high roughness of the shavings' surface and relatively high roughness of the machined surface. The small real contact surface causes the appearance of the very high pressures in the contact zone, as well as high temperatures. Pressures in the contact zone are as high as several tenths of GPa, and temperatures in certain points reach 1200*C in machining of the low machinability material with high cutting speeds.

In tribo-mechanical systems of this kind the great influence on development of tribological processes has also the very high activity of the newly machined surface towards the environment, namely, the coolants and lubricants. It is considered that only 10-12 s is needed for formation of some sort of chemical coating on the newly machined surface.

In such a hard conditions of the contact realization between the tool and the machined piece material, the high friction forces arise, as a consequence of the continuous forming and breaking of the frictional bonds in both basic tribo- mechanical systems. It is considered that about 20 % of total consumed energy in the cutting process is spent on overcoming the friction forces in the contact zones of tool and the machined piece material.

The process of the tool wear is, fundamentally, the process of the mass transfer of the tool material from the tool cutting wedge onto the shaving and machined surface. The consequence of this process is the tool cutting wedge tip deformation and phenomenon of its inability to further perform cutting.

Both processes, the friction process and the wear process, are realized in basic tribo- mechanical systems which in fact represent the sliding friction pairs. This means that solving the problem of very complex cutting process (control of the cutting processes based on knowledge about tribological characteristics of tools and cutting fluids and materials machinability) can be reduced to solving of the contact problem of the two solid bodies, one of which slides over the other with presence of lubricant and external loading.

Tribological characteristics of all the three elements of tribo-mechanical systems, in which the sliding is being realized under certain loading and with certain sliding speed, are determined, very frequently, by application of the "PIN on DISK" tribometers, on which is possible to realize all three types of contact geometry (point, line, surface).

Tribological approach to problems of cutting points to possibility of application of modern tribometers with the accompanying equipment for forming of data bases on tribological characteristics of tools and coolants and lubricants and materials' machinability. Investigations in this direction are being performed in Laboratory for Metal Cutting and Tribology at Faculty of Mechanical Engineering in Kragujevac for the longer period.

UDK 620.9.025.7.003

M. BABIĆ

Tribo-economic Aspect of Contact Surfaces Modification

RESEARCH

In this paper are presented some results of tribometric investigations of surfaces modified by different procedures. Based on the obtained results, it is real to expect that, by the contemporary procedures of the contact surfaces coating, one can contribute to savings of the high quality materials, not only through the increase of the life of the tribomechanic elements, but also through their substitution with the less qualitative materials with high quality contact layers.

Disagreement of the effects from the aspect of friction and effects from the aspect of wear imposes the necessity forar existence of the two criteria: frictional characteristics and the resistance to wear, where to these criteria different weight can be assigned, or the optimal solution can be searched for, based on both aspects. However, only the tribological aspect does not suffice in solving the particular tribological problem. Namely, in determination about the optimality of certain types of contact surfaces modifications, the unavoidable is also the economic aspect.

Keywords: Tribology, modification of contact surfaces

Taking into account that tribological losses, direct and indirect, in summary amount significantly erode national economies, and that they represent the retarding factor in development of technique in the world, the question of their lowering is very actual. Realization of that goal is directly related to improvement of tribological levels of technical systems, what, considering the nature of tribological phenomena, assumes, before all, improvements of tribological properties of tribomechanical systems contact layers. Due to that, from the total volume of tribological investigations, today in the world, about 40 % is related to the area of materials [1].

Since the quality tribological materials are, at the same time, expensive and scarce, and the tribological processes are located only in the thin surface layers, clearly the production of the critical triboelements integrally of the same material represents reality that should be overcome. The promising structure of triboelement along the cross section must be of the composite form, i.e., surface layer of high tribological quality on the lower quality base.

From that aspect, the area of contact surfaces modification, whether considering tribological coatings or changes of base material properties in the surface layer, is estimated as the area of great potentials.

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In this paper are presented some results of tribometric investigations of surfaces modified by different procedures, that are being performed within the corresponding project financially supported by the Ministry for Science and Technology of Republic of Serbia.

The established disagreement of effects from the aspect friction and the aspect of wear, requires necessity for existence of two criteria: frictional characteristics and wear resistance, where to these criteria one can assign different weights, or can seek the optimum solution from the aspect of both. Even when considering only the wear resistance, the necessary criterion is not uniquely determined.

However, the tribological aspect is not sufficient alone in solving the concrete tribological problem. Namely, in decision making about optimality of certain types of surface layers modification, the economical aspect is also unavoidable. Based on the developed input/output model, in which the input vector contains all costs of direct, indirect and past labor, the output vector contains reliability, working life and prices of the technical system, it is possible to determine the efficiency of the applied procedures of surface modifications.

This paper represents results of investigations conducted within the project financially supported by Ministry of Science and Technology of Republic of Serbia

1. TRIBOMETRIC CONDITIONS

Investigations are of the model type and they were performed on the computer supported tribometer. The contact pair geometry in this, due to a series of conveniences known in the world of tribometry, corresponds to one of possible combinations of the pin on disk or disk on disk type, Figure 1.

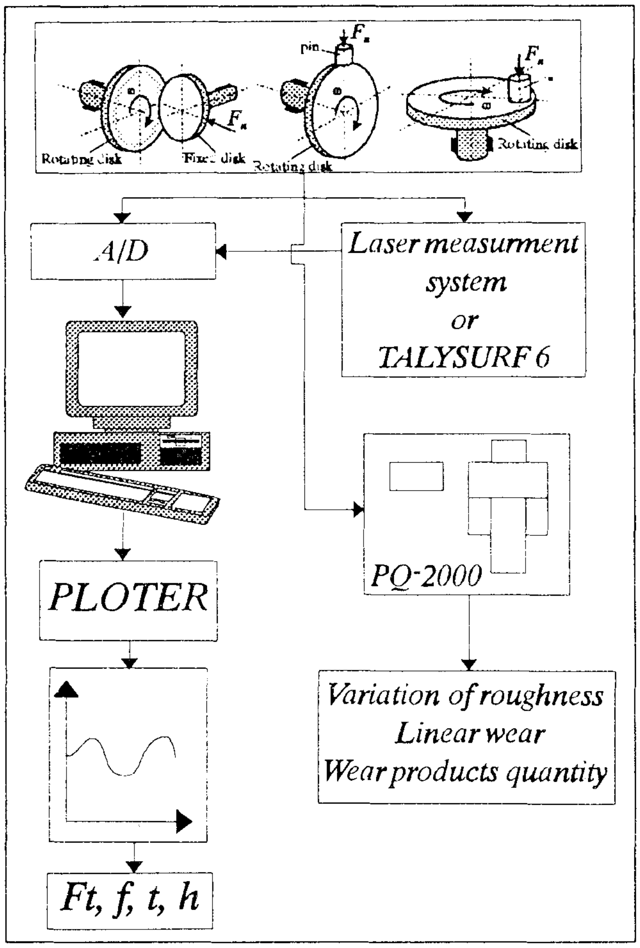

Figure 1. Measurment setup

In investigations, as the element whose contact surface modification is expected, is used the fixed element of the contact pair, which is, due to very small degree of covering exposed to significantly greater tribological loads. The types of modifications (PVD coatings, hard lubricants coatings and ion implantation) and base materials (substrate) are shown in Table 1. The investigated PVD coatings were applied to the front, previously ground surfaces of elements made of construction steels, at temperatures lower than 400°C. The applied procedures of deposition resulted in lower decrease of base material hardness (because of exposure to elevated temperatures of the PVD processes) and in unchanged level of the surface roughness with respect to the initial conditions.

As the moving element of the contact pair, in all investigations, were used disks of diameter of *82 mm*, and thickness of 8 mm, made of chromium-nickel steel Č.5420, in the cemented state, with the hardness 60 HRC. Contact surfaces of disks were machined by grinding under the same conditions ($R_a = 0.2\,\mu m$).

Table 1.

Type of modification	Modified layer	Base material
PVD Coatings	TiN ion plating TiN arc TiAlN TiCN ZrN	Č.5420 cemented Č.4320 cemented Č.4730 bettered Č 1430 bettered SAE 52100
Hard Lubricants	Based on: MoS2 PTFE WS2	
Ion Implantation	Ti+C Ta+C	

Series of repeated investigations were conducted in the conditions of sliding friction with the limiting lubrication, or without any lubrication, with varying the levels of the contact loading and sliding speed within the wide ranges.

2. RESULTS AND THEIR TRIBO-ECONOMIC ANALYSIS

2.1. Tribological effects of modification

Tribological effects of contact surface modification can be practically considered from the aspect of several fundamental questions, like: general influence of modification on the wear and friction resistance, influence of the type and modality of modification on tribological effects, possibility of substitution of the higher quality construction materials with the lower quality ones, at the expense of application of the adequate modification procedures, choice of tribological and also economical criteria for choice of the modification type, etc.

When considering the possibility of improvement of the wear resistance by application of the contact surfaces modification procedures, the most superior results were obtained with the hard coatings. This is related both to domain of the initial and stationary wear.

Effect of the TiN coating on the initial wear, during which the process of running-in is going on, is illustrated in Figure 2, by application of the roughness parameter R_a. It can be observed that the TiN coating application significantly contributes to lowering the wear intensity, and also the wear level, namely the level of changing of the initial topography of the contact surface due to running-in process. This is very important considering that the initial wear, in tribomechanic systems, can significantly change the constructionally set, and technologically provided conditions of contact surfaces leaning against each other (specially tolerances), what determines the quality and reliability of their functioning.

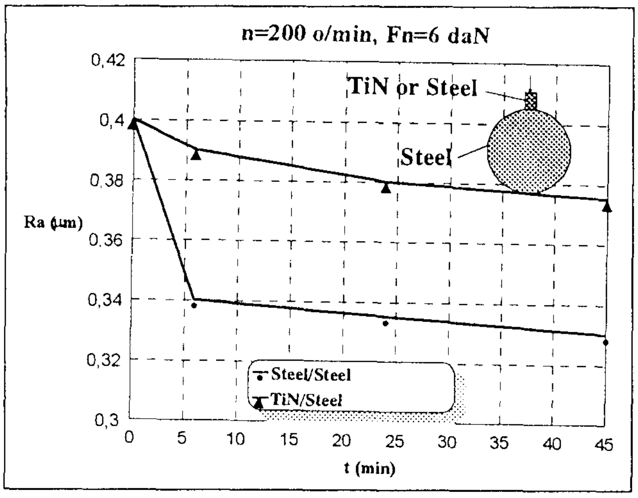

Figure 2. Roughness variation during the running-in period

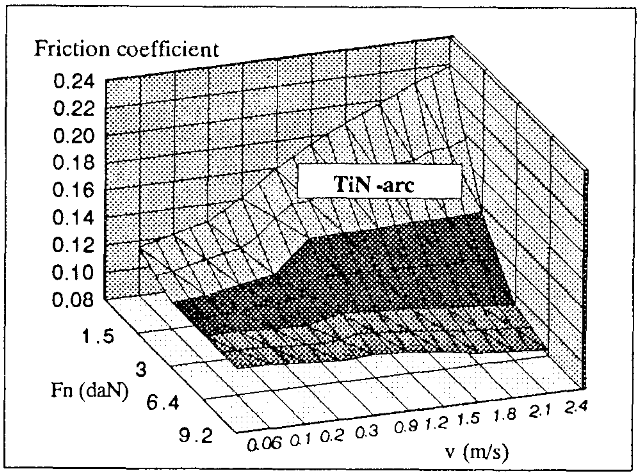

Figure 4. Friction coefficient as a function of normal load and sliding speed

From the standpoint of technical systems maintenance, it is specially important to improve the working life of the critical elements of tribomechanic systems. Practically those are elements that are operating in conditions of the Hertz contact, and with the very low degree of covering. Just for contact conditions like those, tribometrically obtained results show very positive effects of application of the hard tribological coatings (like the PVD TiN coating is), what is illustrated in Figure 3.

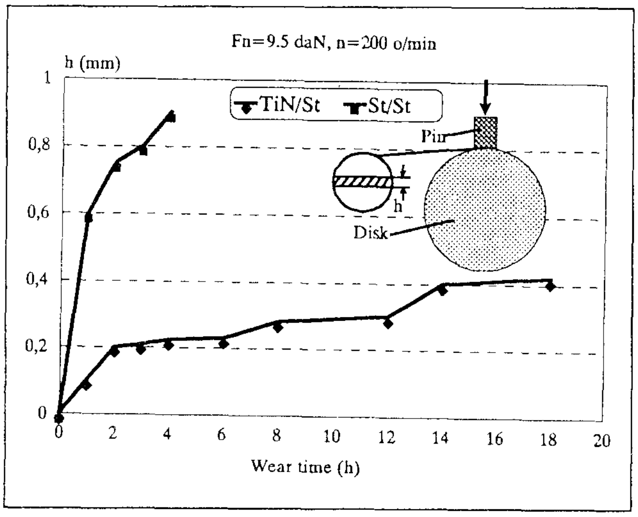

Figure 3. Wear curves of modified and unmodified surfaces

However, more complex consideration of tribological effects (especially frictional), of different types of contact surfaces modifications, imposes necessity of taking into account the fact that tribological characteristics of the modified surface do not represent its immanent property, but that they are only result of tribological interaction under certain contact conditions. Thus one can notice from Figure 4 that the TiN coating friction coefficient presents the function of the contact load (F_N) and the sliding speed (v).

With increase of contact loading the friction coefficient is decreasing, what is especially expressed for the large sliding speeds. The influence of the sliding speed is smaller and it is not unique. Such an influence can be expressed, with the high correlation degree, by the correlative dependence of the type:

$$f = C_v^x \cdot F_N^y$$

The tribological effects also represent the function of contact realization conditions (Fig. 5). It can be seen that at low sliding speed there is no positive effects of the TiN coating on the lowering the friction energy. However, with strictening of the contact conditions regime, the positive effects of the TiN coating become prominent.

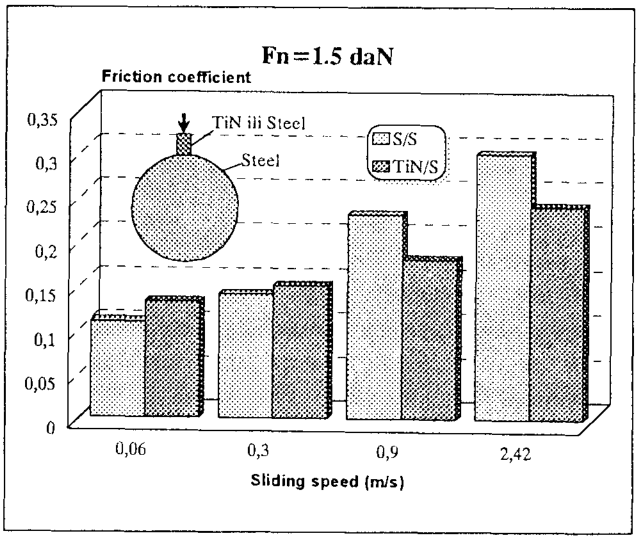

Figure 5. Friction coefficient for the modified and unmodified surfaces

Obtained results of investigations of different types of contact surfaces modifications can be used for comparison of their tribological effects, what creates the basis for choosing the optimal solutions for particular contact conditions. Those comparisons are possible both for effects of different modalities of one particular type of modification, and for essentially different types of modifications.

In that sense, the illustrative are comparative results of tribological parameters of different coatings (Figure 6), that are previously listed in the table. From the whole series of results obtained for different combinations of the contact conditions parameters, on these diagrams are shown results that are considered as typical, and by that also sufficient for comparison of tribological properties of investigated coatings.

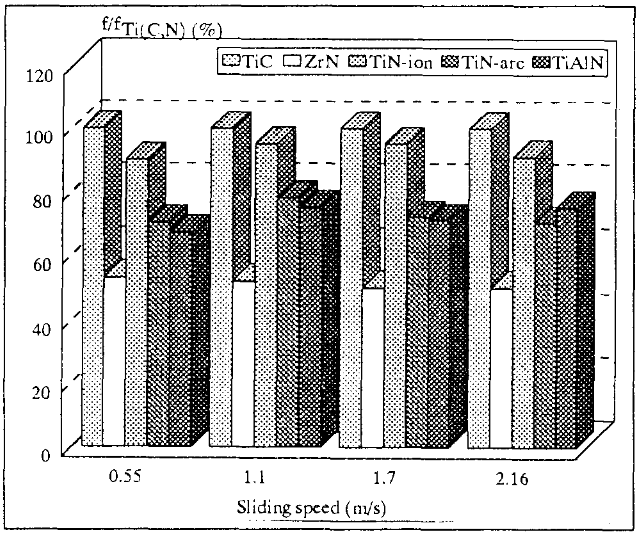

Figure 6. Influence of the PVD coating in the friction coefficient

From the relative ratios of the friction coefficients it can be seen that the best tribological properties has the ZrN coating, and the worst the TiCN coating, that was used as the reference for comparison. The friction coefficients of the other three coatings are within relatively narrow range. The largest differences of frictional characteristics were established in the area of small contact loads, while with the increase of the contact loads and the sliding speed the differences decrease. The high average level of differences undoubtedly points to the fact that by choice of the modification type one can significantly influence the frictional behavior of the modified surface. In that, the friction effects are the function of the conditions under which the sliding is occurring.

The development of the wear process on the tested specimens was monitored as the growing of the wear belt width on the nominally linear contact (for coatings until the moment of their complete destruction). The established relationships are presented by the percentage amounts of the corresponding wear coefficients for the friction conditions with the limiting lubrication (Figure 7) and without lubrication (Figure 8).

Thus, based on results like these, it is obvious that the PVD coatings strongly improve the wear resistance of the

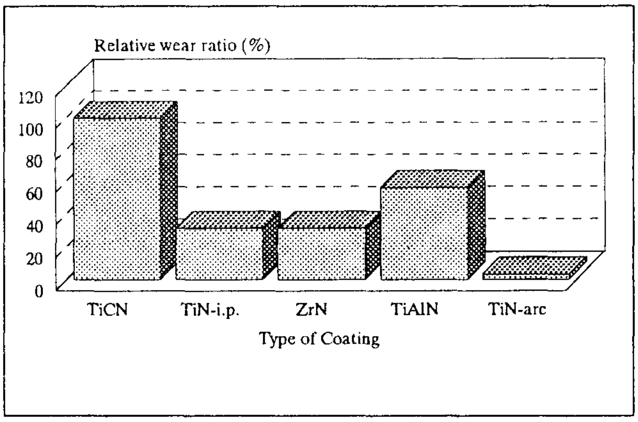

Figure 7. Influence of the PVD coating on the wear ratio in conditions of friction with lubrication

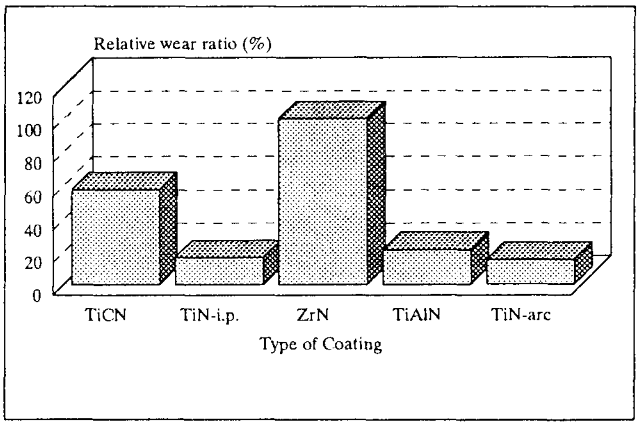

Figure 8. Influence of the PVD coating on the wear ratio in conditions of friction without lubrication

base material, but also that different coatings, or even that same coating obtained by different PVD procedure, have different contributions to that improvement. Simultaneously, it is obvious that the position of particular coatings on the rating lists, based on the friction coefficient, and based on the wear coefficient, is not unique. Thus, for instance, the ZrN coating has the best frictional properties, and, at the same time, the lowest wear resistance.

Examples of results, presented in Figures 9 and 10, show how different types of modifications of contact surfaces, influence the improvement of their tribological properties from the aspect of friction and the aspect of wear. As in the previous example, ranking based on these two criteria gives even completely opposite pictures about their contributions.

2.2. Choice of tribological criteria for evaluation of the modifications effects

The presented disagreement of effects from the aspect friction and the aspect of wear, requires necessity for existence of two criteria:

- frictional characteristics and
- wear resistance,

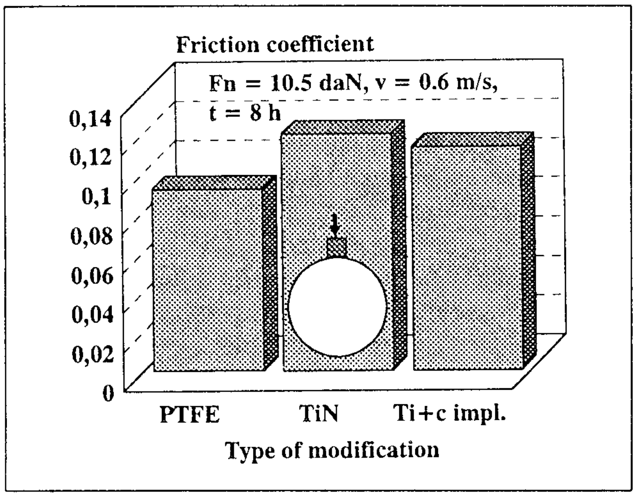

Figure 9. Influence of type of modification on friction coefficient

where to these criteria one can assign different weights, or can seek the optimum solution form the aspect of both.

Even when considering only the wear resistance, the necessary criterion is not uniquely determined.

Namely, the up to mow approach had a wear resistance, as a criterion for evaluation of the modified surfaces, but only of one element in the contact pair, and that was, understandably, the pin, which is tribologically highly jeopardized, considering the contact geometry (namely the degree of coverage). If one keeps in mind wear effects of the whole contact pair, then the picture can look completely different. Thus, in Figure 11, are shown comparative results of total wear of the contact pair, expressed by the *PQ* index, for the compared modification procedures. The *PQ* index represents the measure of the wear products concentration in the lubricating oil, that are the result of wear of both elements in the contact pair. The rating list, from the aspect of summary wear, looks completely opposite, with respect to one that comes out from the presentation in Figure 10. Thus, the smallest

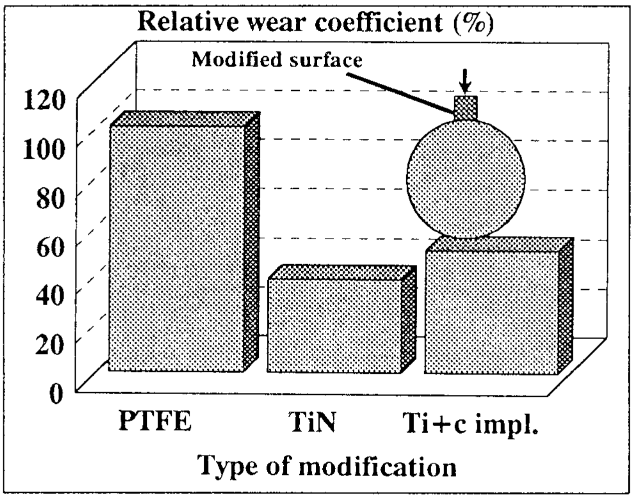

Figure 10. Influence of type of modification on wear ratio

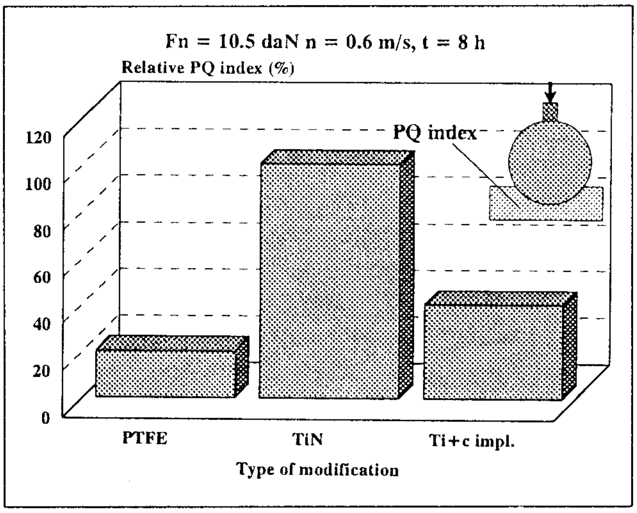

Figure 11. Influence of type of modification on PQ index

total wear corresponds to contact pair in which one of elements has the surface modified by the hard lubricant coating.

So, one of the most important questions is the question of criteria for evaluation of tribological effects of modification, where the criterion has to come out from the nature of the problem that is being solved. Thus, in tribomechanical systems, that are, by their structure characterized by existence of elements with small coverage degree, namely tribologically extremely jeopardized elements, the tribological problem is being solved by improvement of the wear resistance of those particular elements. Less interesting is the question of total wear, since the wear of other, noncritical elements, does not endanger the functioning of the system. And vice versa, in structures that are characterized by uniform tribological jeopardy of contact elements, i.e., where there is no expressed critical element, interesting is the question of decreasing the total wear.

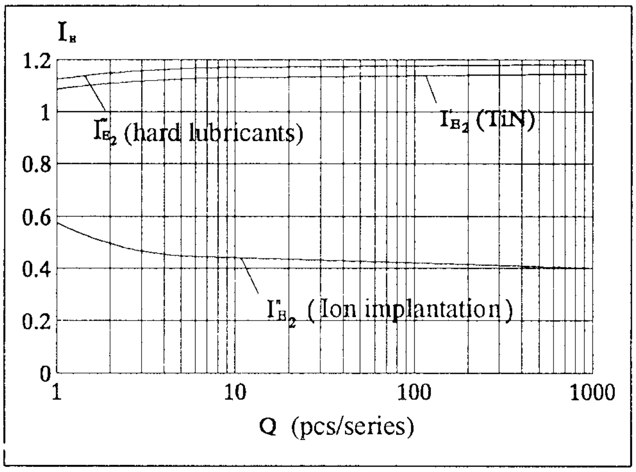

Figure 12. Efficiency indices for different types of modifications

2.3. Tribo-economic criterion for evaluation of modification effects

The solution of certain tribological problem assumes the more complex approach than just choice of tribologically most superior version of surface modification. Namely, in decision making about optimality of particular types of contact surfaces modifications, the economical aspect is inevitable. Based on the developed input/output model, in which the input vector contains all costs of direct, indirect and past labor, the output vector contains reliability, working life and prices of the technical system, it is possible to determine the efficiency of the applied procedures of surface modifications. Thus, in Figure 12 are shown relations between efficiencies modified and unmodified elements for the three mentioned types of modifications. It can be seen that the curves, as functions of number of pieces per series, are above 1. At the same time, the curve that corresponds to implanted surface is far below 1. That means that tribological effects of the implanted surface are not sufficient to make up for the high price of the procedure, and by that to ensure the increase of efficiency. Thus, the application of implantation makes sense only there where the specific requirements make it necessary, or at parts that are characterized by extremely low reliability, availability and the working life.

3. CONCLUSIONS

The investigated PVD coatings can contribute to significant improvement of wear resistance of tribologically critical elements made of construction steels. Positive effects of the PVD coatings, from the aspect of decreasing energy losses to friction, are mainly related to stricter contact regimes.

Considering the obtained results, it is real to expect that, with modern procedures of contact surfaces coating, one can contribute to savings of high quality materials, not only through increase of the working life of tribomechanical elements, but also through their substitution by the lesser quality material with high quality contact surfaces. The choice of criterion for evaluation of tribological effects of modification deserves special attention. In its definition, it is necessary to start from the nature of the tribological problem that is being solved on the real tribomechanical system. The final decision about acceptability assumes precise expression of economical effects.

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RESEARCH

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Some Observations on The Wear of Hard Metal Cutting Tool

In the paper milling tests were carried out with hard metal tools under different cutting conditions on normalised steel C 1730 (C60). The width of flank wear land, crater depth and crater width were measured by microscope and presented in graphical form versus cutting time. Worn surfaces were inspected by scanning electron microscope (SEM) as well. The SEM results show that under stable milling conditions a number of individual wear mechanisms are concurrently present. They are abrasion, adhesion, oxidation and micro chipping, along with such secondary features as plastic deformation, cracking and fracture.

Keywords: *wear of hard metal, tool wear mechanisms*

1. INTRODUCTION

Dependence of the tool-wear on cutting conditions, tool geometry, tool and workpiece material etc., was noticed long time ago. Tool wear has great influence on the machining process parameters such as: i) cutting forces; ii) machining power; iii) cutting temperature; iv) surface roughness.

Great attention was dedicated to investigations on wear of hard metal cutting tool. Investigated were parameters on rake and flank surface of cutting tool. Investigations and parameters of wear are defined by ISO standard.

During the investigations width of flank wear land was most frequently taken as relevant parameter of the tool wear [1]. Phenomena of groove wear on minor flank surface was noticed during turning long time age. Some investigators like Pekelhaning, Solaja in the 60's and 70's tried to explain this phenomenon [2, 3].

Width of flank wear on the major cutting edge and crater wear on the rake face mainly effect the fool life, but grooving wear on minor cutting edge modifies the profile of the nose of tool and causes surface roughness and workpiece dimensional accuracy change.

There have been many investigations concerning tool wear in face milling, but phenomenon of groove wear has not been noticed. During his investigations Kamm [1] considered the influence of cutting speed, feed and tool geometry on the tool wear. They tool wear on minor cutting edge as the average width of flank wear. More severe wear was observed on the minor cutting edge then on major cutting edge.

Investigations of tool wear mostly deal with the tool wear patterns, only small number with wear mechanisms. In the paper, beside tool wear pattern phenomenon, contribution is given to investigation of wear mechanisms.

2. EXPERIMENTAL INVESTIGATIONS

Milling tests were carried out on steel C.1730 (C60) in normalised condition, using 100x130x700 mm bars. As a tool the single tooth face milling cutter of 125 mm diameter, with plates of hard metal SPAN 12 03 ER, quality P25 were used. Geometrical elements of the tools cutting part were: normal rake 7° normal clearance 18° and cutting edge angle 75°. The investigations were carried out on the vertical milling machine, with driving power of 14 kW dry, without the application of cooling and lubricating agents.

The measurements of wear size on face surface and flank one of tool was performed on Zeiss measuring microscope with special fixture. Only crater depth *KT* measurements were performed on Schmalc microscope for surface roughness measurement. On the flank face average width of flank wear, on edges VBg and VB_s and one by one, width of grooves on the minor cutting edges V_{Bp}, versus cutting time were measured. On the rake face width of crater *KB* and depth of crater *KT* versus cutting time was measured. The wear of hard metal tool patterns, up to the initial stage of their breaking through was examined and correlated with wear mechanisms using a JEOL JSM 35 scanning electron microscope (SEM), operating at 25 kV.

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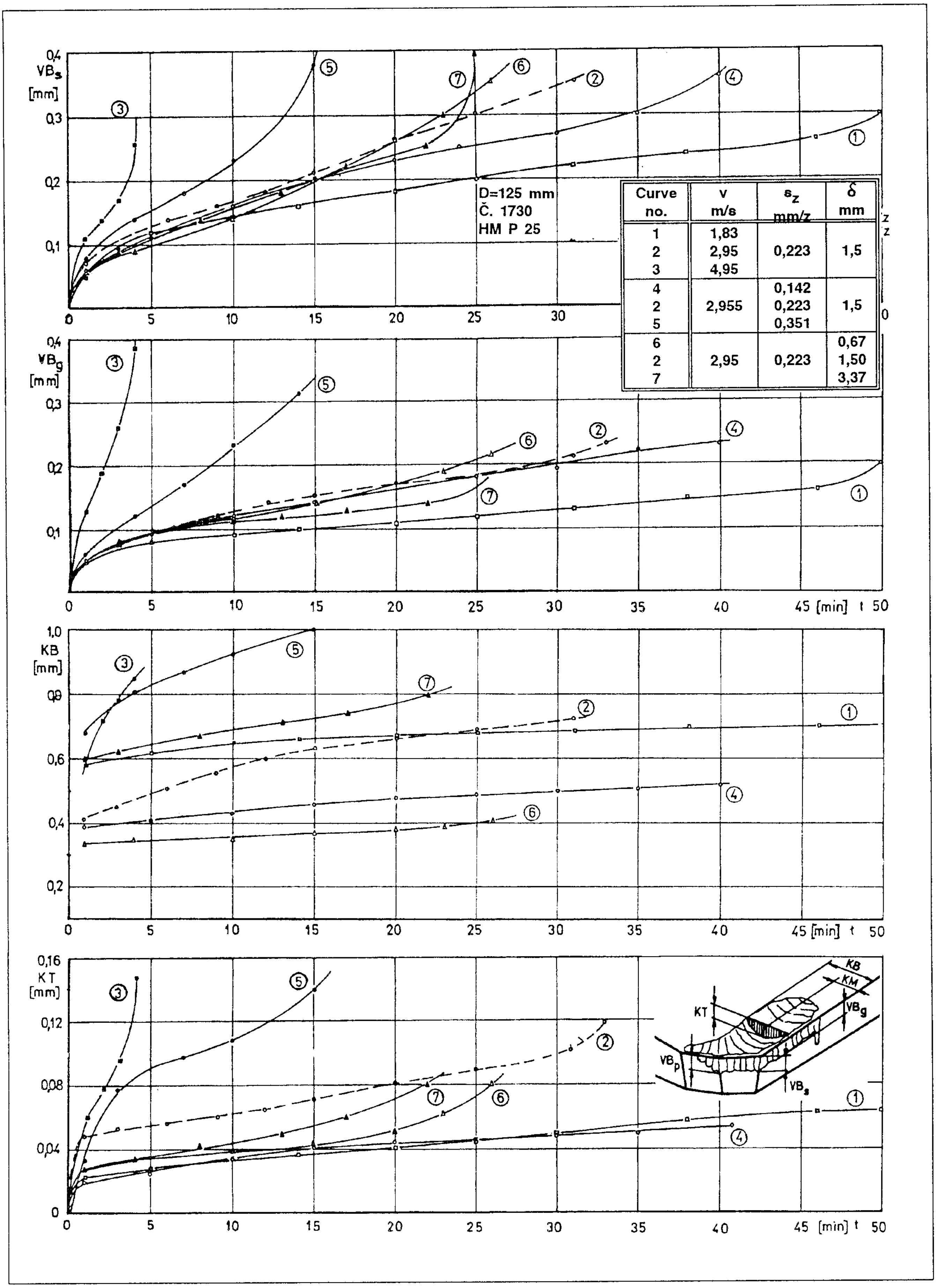
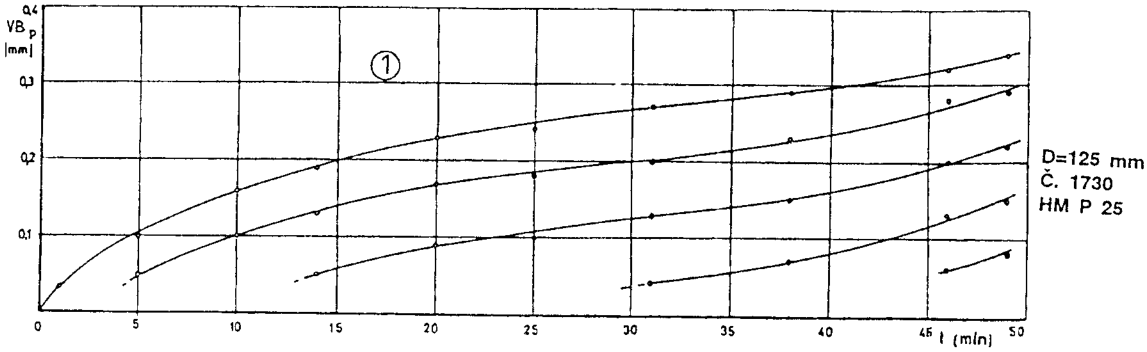
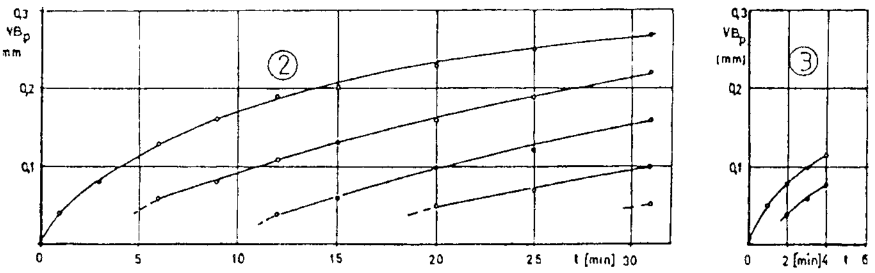


Curve no.	v m/s	s_z mm/z	δ mm
1	1,83		
2	2,95	0,223	1,5
3	4,95		
4		0,142	
2	2,955	0,223	1,5
5		0,351	
6			0,67
2	2,95	0,223	1,50
7			3,37

Fig. 1. Tool wear parameters versus cutting time





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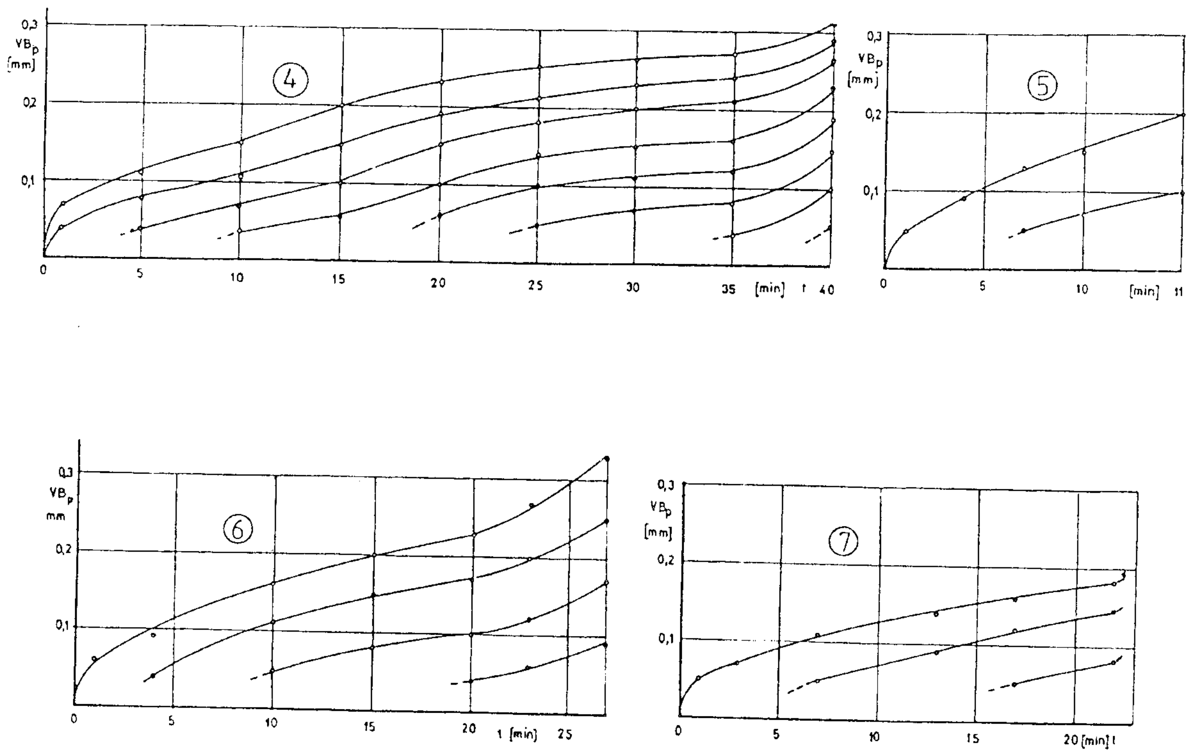

Fig. 2. Groove wear on minor cutting edge versus cutting time

3. RESULTS AND DISCUSSION

3.1. Analysis of tool wear patterns

The parameters of the tool wear on the major cutting edge and parameters of crater wear, versus cutting time for different cutting conditions are plotted in the Fig.1.

Width of grooves on the minor cutting edge *VBp* versus cutting time for different cutting conditions are plotted in the Fig.2. According to the measuring parameters of the tool wear *VBg*, *VBs* and *VBp* for tool life criterion *VB = 0.3* first is reached on cutting edge *VBs*.

In the investigated interval of cutting speed greatest influence on tool wear has cutting speed, curves 1, 2 and 3. Middle influence has feed, curves 4, 2 and 5. Cutting depth doas not influence the width of flank wear, as it can bee seen from curves 6, 2 and 7 it can be concluded for crater depth as well. Cutting depth influences the most crater width. Increasing depth of cut increase starting values of crater width *KB*, similar influence has feed. High values of cutting speed and feed cause quick changes in crater depth *KT* and crater width *KB* versus cutting time.

Grooves wear on minor cutting edge appears in face milling as well as in turning. In the face milling they are more visible because they are located on flat surface of minor cutting edge. Grooves on minor cutting edge appear at a distance equal to the feed and have different width. Grooves grow and appear new versus cutting time. All grooves have three characteristic phases as tool wear curves versus cutting time. In the beginning grooves have fast initial wear, than phase of stable wear and at the end fast phase of wear.

Every groove goes through all three phases, but the grooves that appear later, have the third phase shorter. It means that the third phase of wear on all grooves begin at the same time. The most significant inlluence has feed (Fig. 2). Low values of feed cause more grooves, curve 4, opposite to curve 5 with higher feed.

Influence of cutting speed can be seen from the curves' 1, 2 and 3. Because feed is constant, grooves are at the same distance. Cutting with higher speed causes smaller width of grooves, because cutting was stopped because of criterion *VBs = 0.3*.

Cutting depth influence is not significant what agree with influence on width of major cutting edge *VBs* and *VBg*.

3.2. SEM Metallography

The worn surfaces of hard metal cutting tool were observed by SEM as well. The results were summarised and would be presented as typical examples, to show that under stable milling conditions a number of individual wear mechanisms are concurrently present. They are: abrasion, adhesion, oxidation and micro-chipping along with such secondary features as plastic deformation, cracking and fracture. These mechanisms are present in all worn specimens investigated. The differences are either in intensity and time when they are appeared, or which mechanism is dominant.

The tool wear produced in specimen 1 is shown in Fig 3 (a, b). This is an example for an extreme case, where intense localised adhesion results in contiguous cracking and chipping of the rake and major flank surfaces and nose edge - the primary mechanisms of tool wear here. The loss of tool material due to adhesion and oxidation is also seen on the nose of tool - Fig. 3 b.

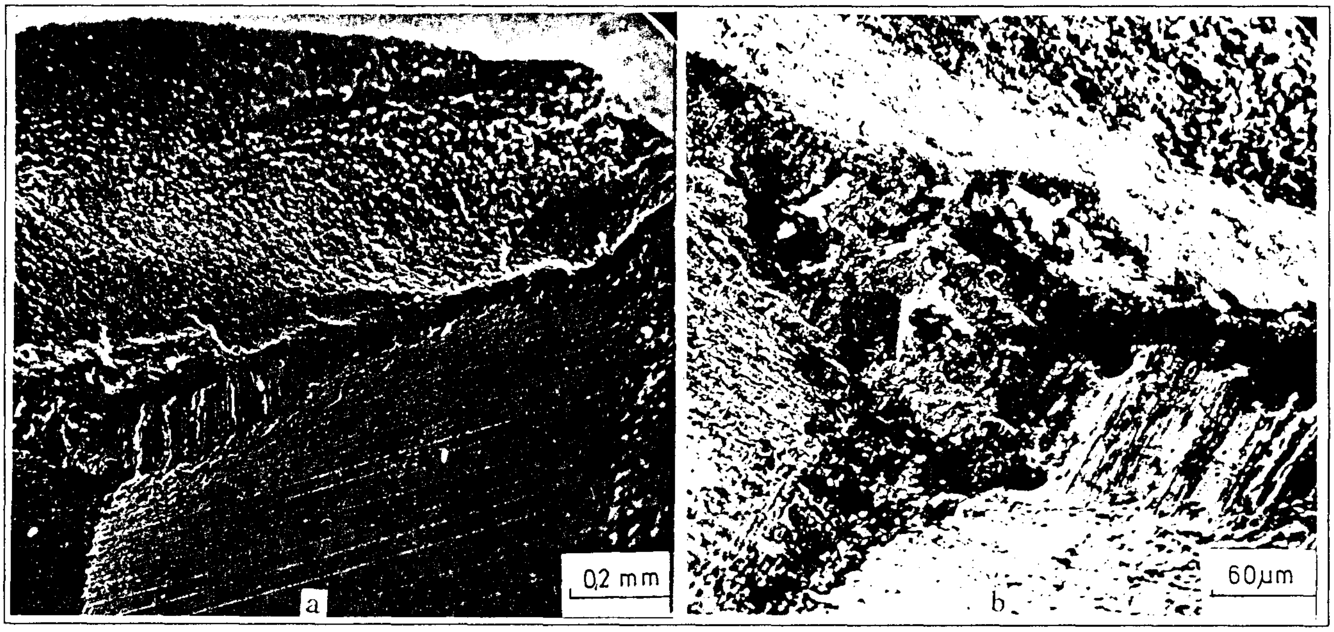

Fig. 3. Oxidation and adhesive chipping of a) major flank surface; b) nose of tool

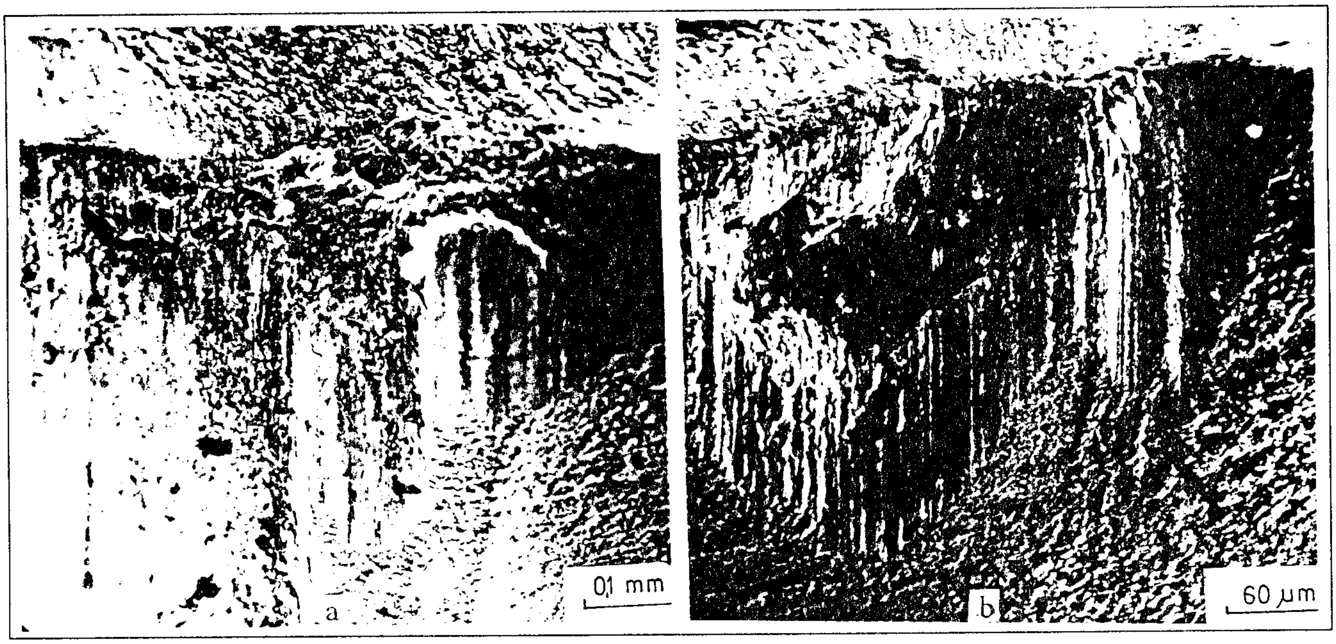

Fig. 4. Oxidation and abrasive wear a) craking across the nose; b) fracture on flank face

Fig. 4 (a, b) shows examples of the wear obtained in specimen 2. The products of sliding wear process by cracking across the nose of tool, deformed during cutting and fracturing of some fragments of tool material on flank face are shown in Fig. 4 (a, b).

The wear operating in these sliding regions is probably that which occurs under normal conditions, involving both abrasion and metal transfer and influenced by chemical interaction with the surrounding atmosphere.

Further, in milling where cutting is interrupted, crack can be initiated in the tool at the hottest position on the rake face (some distance from the edge), then spread across the edge and the flank face. The consequence of very high temperatures is oxidation wear on contact surfaces where oxygen can penetrate between contact surfaces and temperature is high enough so it appears on border of crater wear and width of flank wear Fig. 3 and Fig. 4.

Worn surfaces are characterised by distinct grooves of small and large size over both flank and rake surfaces. This intensity of abrasive wear is uniform over the flank, but shows variations on the rake. Deformation of tool nose with fairer and abrasive wear is also observed, Fig. 5.

The groove wear on the minor cutting edge is noticed in all specimens. Fig. 6 shows representative view of grooves on minor cutting edge during face milling .

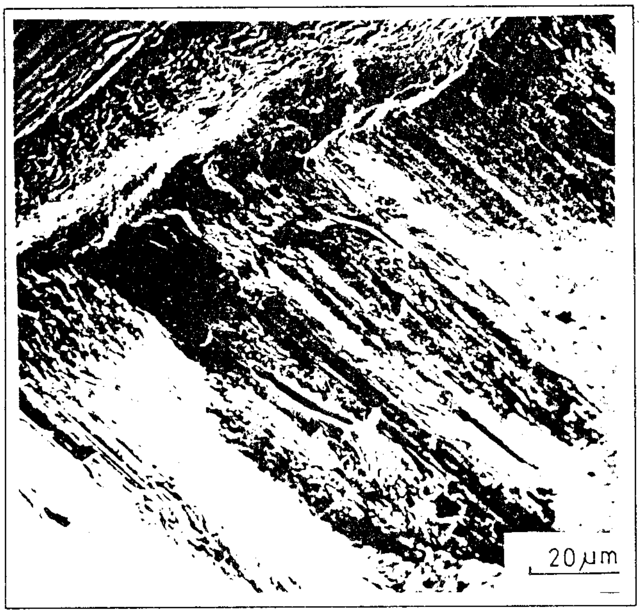

Fig. 5 Deformation and failure ot tool nose

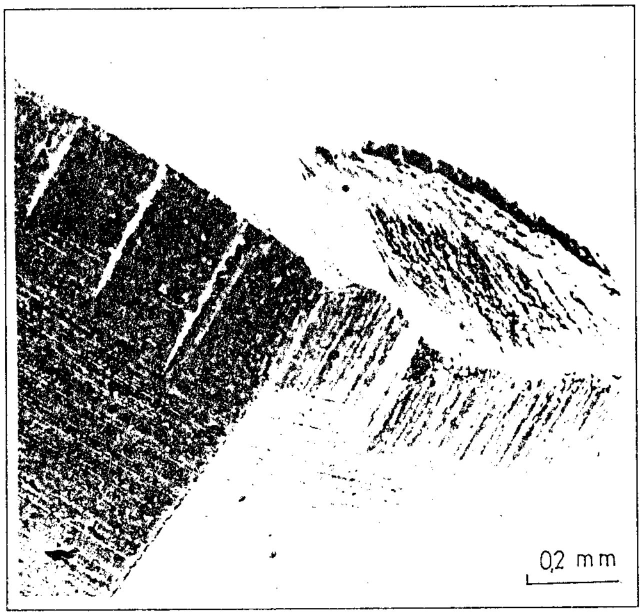

Fig. 6 Groove wear on minor cutting edge

4. CONCLUSIONS

On the basis of the former the following conclusions can be stated:

- Tool wear patterns on rake and flank surfaces versus cutting time are the most influenced by cutting conditions.
- Grooves wear on minor cutting edge of tool appears during milling.
- Cutting conditions have significant influence on groove wear.
- Wear mechanisms such as adhesion, abrasion, oxidation deformation, cracking, micro-chipping and fracture of hard metal, have caused wear patterns on tool.
- The SEM is suitable for investigation of tool wear mechanism.

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V. A. GODLEVSKI, V. N. LATYSHEV

The Cutting Lubricants for The Materials of Low Machinability: New Approach

RESEARCH

The nowadays using of cutting fluids in metal production meets more difficulties connected with insufficient activity of lubrication in operation zone, but at the same time excessive activity in interaction with working equipment and operating personnel. Especially it is important for cutting of low-machinability materials. To solve this problem it is recommended to apply the new elaborated technologies: external energy activation and temporary isolation of the lubricant's active components with the help of microcapsuled additives. Another way is to diminish the volume of cutting lubricants and to use "once only" products.

Keywords: *Machinability, cutting, lubricant,energy activation.*

1 INTRODUCTION

The cutting fluids (*CF*) became one of the important components of the machining technology. Especially this concerns the cutting of low machinability materials with high temperature strength: metals, alloys as far as non-metals. Under this condition we meet the main contradiction. On the one hand in this case it is necessary to apply a *CF* of high chemical activity (including *EP*-additives), on the other hand such compounds are very harmful for working personnel and contaminate the environment immensely.

One of the most promising decisions is to change the super-active chemicals to additives with increased adsorption ability (so called "mesogens") wich may built the effective structurized layers of boundary lubrication on the surfaces. But such worked materials as molybdenum, stainless steels and nickel-based alloys to regret often can't be machined without the *CF* containing phosphor, chlorin, sulphur and similar triboactive components. So if the application of *EP*-additives is inevitable, the special technique may be used of *CF*-application during cutting in order to eliminate the harmwfel influences of active chemical substances. Let us consider some of them.

2 CONTROLLED ACTIVITY PRINCIPLE

Have a look at the whole volume V of *CF* used on the certain machine-tool. It consists of the two unequal parts:

V_1 - "active" working part adjacent to cutting zone and
V_2 - the rest "passive" volume of CF (Figure 1).

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Both these volumes of liquids correlate approximately as $V_2 = 10^{-6} \dots 10^{-8}$ and for improving of their functions must be inffuenced separately. For these reasons it was proposed to use the two groups of methods:

- external energy activation and
- temperary isolation of the active components from environment.

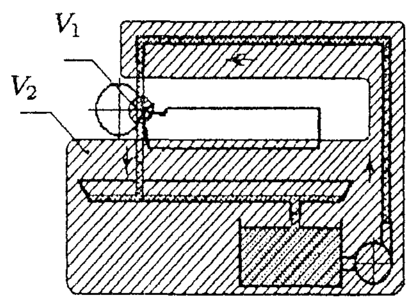

Figure 1: Active (V_1) and passive (V_2) volumes of CF on certain machine-tool

Each of these two approaches has the opposite purposes. The former stimulates the physico-chemical activity of *CF* promoting the more intensive interaction between *CF* and materials in cutting zone. The latter methods, on the contrary, must suppress the chemical activity in the region of "non-working" contacts of *CF* with elements of machine-tool, pipes, vessel's walls etc. This second purpose may be obtained by means of temporary confinement of the *CF* active components (e. g. *EP*-additives) into special capsules or microcapsules. The methods of technological influence on the functional volumes V_1 and V_2 are shown in Table 1. This conception of partial lubrication volumes control may be included into more wide approach connected with problems of properties optimization in the whole volume CF (Figure 2). We conducted the series of investigations to realize both of the above mentioned methods [1, 2, 3].

Table 1: The methods of influence on the functional volumes of CF

Initial state of CF	Object of influence	Purpose of influence	Mode of influence
Non-active	V_1 (cutting zone)	Activity increasing	External energy activation
Active	V_2 (non-working zone)	Activity decreasing	Microcapsulation of the active additives

2.1. The external energy activation

The external activation is conducted with the help of some physical (e.g. electric) fields applied to cutting zone. The survey of these numerous methods was given in some works [4, 5]. It would be better to localize this power influence in *CF*-volume V_1. As we see the electrical methods of activation are more convenient for this purpose [6]. The method of surface electrical charging to the most degree displays its local influence on *CF* in cutting zone, it is usable on both cutting and grinding operations. This method have a low energy consumption, it is not dangerous to use. Beside that it has a weak destroying action on the lubricating composition. The short review of such methods was made in work [6, section IX].

2.2. The microcapsulation of CF-components

We propose the application, in cutting lubrications compositions, the capsules containing mixtures as cutting fluid by machining [3]. The using of microcapsules filled by active substance (e. g. *EP*-additives) prevents the undesirable interaction of compontents with the machine details. At the same time the capsuled additives work only near the cutting zone, where the microcapsules are breaking. The size of particles can be varied from *0.001 mm* (nanocapsules, molecular clusters) to *10 mm* (ma-

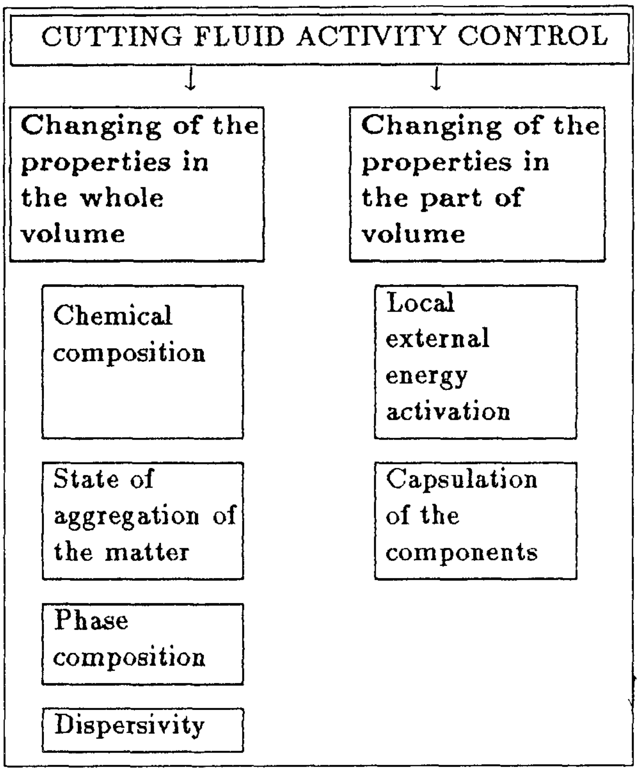

Figure 2: The general principle of CF-activity control

crocapsules). The different sizes of capsules have peculiarities of production and using in cutting process (see Table 2).

The cover of the capsules not only protects the confined additives from undesirable interaction outside of the cutting zone, but also measures out in doses of their content's access, brings down the harmful excretion to service zone and brings about some another useful effects. Let's enumerate them.

- The control of temperature level of additives access to cutting zone (by means of appointed heat resistance of the cover).
- The possibility of common action of noncomputable additives in one lubrication composition.
- The support of the constant concentration of additives (by controlled diffusion through the cover).

Table 2: The production technology and the usage peculiarities of the capsuled components of CF in connection with the size of particles

Size group	Macro-capsules	Micro-capsules	Nano-capsules
Mean size of particles, mm	> 5.0	0.001...0.5	< 0.001
Capsuling technologies,	Mechanical forming of the envelops and the following filing of it by the working substance	The film building on the phases interface in the disperse system	Synthesis of the spherical molecules of macropolymer with including of working substance in it
Peculiarities of capsuled additives using in CF-compositions	Capsules are swimming in the CF-vessel without access to cutting zone	Capsules are circulating in the whole volume of CF acting only in cutting zone	

- The lowering of the harmful and smelling excretions to environment.
- The improving of cooling process by using heat transport mediums inside capsules. (It is known the case of usage the capsules filled with special refrigerant [7].
- The possibility of additives concentration control by addition or filtration of capsuled components

Also it is interesting to consider the behavior connected with lubrication action of the microcapsuled *CF*. It was shown [2] that the apportionment of capsules contents may go according to different cutting condition as following:

- The mechanical destroying due to squeezing of the particles between tool and workpiece.
- The diffusion of the contents to outside through the capsules cover.
- The thermal destroying (like explosion) in the moment of touching of capsules to hot surfaces chip, tool or workpiece; the direct consequence of this may be quick evaporation of the capsules contents or thermal strength loss of the their cover.

Naturally, all the three processes can go simultaneously during cutting.

3. THE REDUCTION OF APPLIED AMOUNT OF CF

It is reasonable to propose that in order to decrease the harmful influences caused by lubricant in volume V_2, it must to diminish the volume as far as possible. This may be obtained with the help of "once only" lubricants. They may be brought into the cutting zone in the form of the liquid or grease droplets. It is better to bring it into contact periodically (when and where it is necessary). For this aim the special batch boxes can be used [8].

If the lubricating material has a low viscosity and can not be restrained near the cutting zone, a special technique must be used in order to hold the droplet in the necessary place. Known are the proposals to hold the small amount of *CF* with help of magnetic field (of course when the ferromagnetic lubricants are used) [9].

Ultimately, there exists the possibility to totally exclude the liquid lubricants from the machining process: to replace them with the solid or gas lubricants. For example the molybdenum disulphide containing pastes may be used. Also very promising is the formation with help of chemico-thermal method of the relativly mild solid lubrication layers. We showed that besides lubricating action these layers display also effective cooling of tools [10].

As very exotic method it is known the usage of air plasme installation where the corona discharge bring about the air environment to active state [11]. In this case we can realize cutting process without artificial lubricant at all. From ecological point of view this is the most advantageous decision.

4. CONCLUSION

Thus, in this paper was shown the new approach to the technology of cutting fluid application: the controlling of *CF* functional volumes activity. In this connection it is necessary to conduct a lot of additional investigation in order to solve many technical problem, answering to chemical, tribological and design questions.

It must be proposed a series of new materials for capsules, covering and it's content. Nevertheless, the conception of the "controlled activity" may be more helpful in the future from both technology and ecology point of view.

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RESEARCH

UDK 621.821.8:621.9.015.004.62

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An Alternative Solution To Logarithmic Roller Profile In Cylindrical Roller Bearings

By using a prestressing loading in elastic-plastic domain of a cylindrical roller a new profile is achieved for the roller generatrix. A fast numerical analysis based on the halfspace theory, has been developed to determine the pressure distribution along the roller centerline. The contact pressure distributions are further comparatively presented for diferent roller profiles and the superiority of the profile obtained by a suitable prestressing loading is pointed out.

Keywords: *cylindrical roller bearing; elastic-plastic deformation; nonhertzian contact.*

1. INTRODUCTION

In a cylindrical roller bearing with straigth roller generatrix, the stresses concentration occur at the roller edges, even under slight radial loads, without any misalignment or axial load component, and consequently the bearing would fail prematurely.

Therefore, one major objective in the design of the internal geometry of a roller bearing was to reduce peak stresses in the end parts of the roller - raceway contact.

Primarly, this purpose has been attained by using crowned and cylindrical - crowned profiles. However, optimum conditions would be achieved if the contact pressures were uniform along the contact length. In 1939. Lundberg solved the inverse problem [1], respectivelly he found the necessary small corrections to the radial profile of the roller required to obtain the uniform axial distributions of the contact pressure. This profile, known as Lundberg or logarithmic profile, that is correct at the design load only, is quite difficult to be manufactured even with the modern CNC machines. However, the direct problem of finding the contact stresses for a nonhertzian contact geometry is not yet theoretically solved.

2. PLASTICALLY DEFORMED PROFILE

If a cylindrical roller was subjected for a few cycles to a contact loading in elastic - plastic domain, some permanent deformation occur, especially at the end parts of the roller. The initial cylindrical roller with a straight generatrix becomes a cylindrical roller with a new, slightly curved generatrix. The contact geometry between a cylindrical raceway and a cylindrical rollers with plastically deformed generatrix, however is no longer a hertzian one.

The aim of the first stage of the experimental programe was to evaluate the new generatrix profile of the cylindrical rollers subjected to prestressing operation in elastic - plastic domain. A number of straight cylindrical rollers, similar to those used in a NU 306-E cylindrical roller bearing were selected. The main dimensions of a roller were: the diameter, $D_w = 11\,mm$; the length $L_w = 12\,mm$; the end radius, R = 0.5 mm. The rollers were subjected for 200 cycles to a contact stressing operation corresponding to one of the following maximum Hertz pressure: *3.0 GPa, 3.5 GPa, 4 GPa* and *4.5 GPa.* For each pressure the stressing operation was repeated for a number of five rollers. After that the new generatrix profile of each roller was measured using a Taylor & Hobson profilometer.

3. NUMERICAL PROCEDURE

Since the direct problem of finding the contact stress distribution for a non-hertzian contact geometry is not theoretically worked out, a fast numerical analysis, based on the flexibility matrices derived from the halfspace theory [2-4] has ben developed.

A plane rectangular domain chosen larger than the expected contact area is divided into rectangular elements, the pressure over each element being considered constant. To improve the accuracy without increasing computing time both the length and the width of the rectangular elements are established by using arithmetic progressions. The cartezian axes origin being chosen in the middle of the largest rectangle element, the smallest

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rectangle elements are situated at the borders of the contact domain, exactly where the stress concentration acts and the contact pressure varies sharply. A cylindrical roller with circular profile was chosen to compare the results obtained with this method to those obtained with the more time consuming finite element method. The contact geometty and the finite element results are those given by J. de Mul in [4]. The contact pressure variations along centerline presented in Figure 1. point out a very good agreement with the results obtained by the proposed flexibility method.

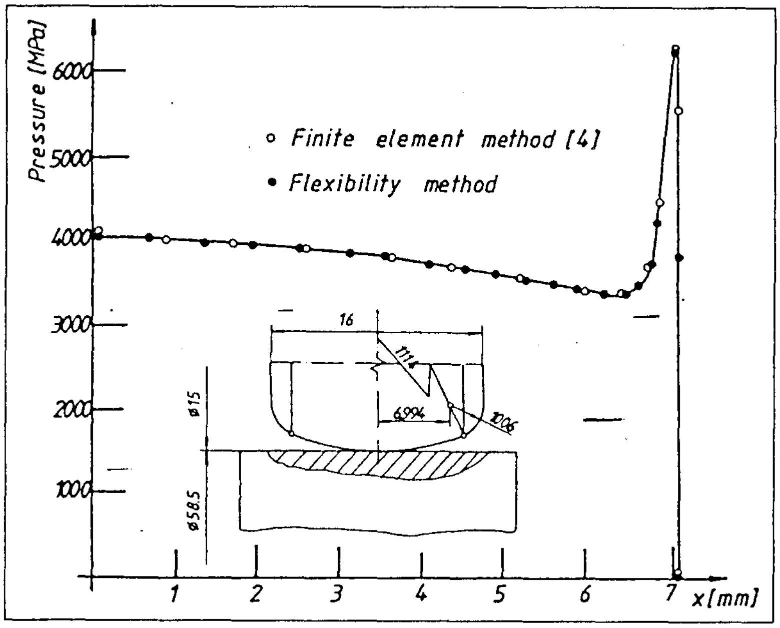

Fig. 1.

4. ANALYSIS OF PRESSURE DISTRIBUTIONS

The pressure distributions were analysed considering the contact geometry realized between the inner ring of a cylindrical roller bearing and the cylindrical rollers having both straight line generatrix and the generatrix profiles previously obtained through contact prestressing in elastic - plastic domain.

The inner racewav diameter was $D_i = 40.5\ mm$.

The pressure variations along roller centerline are presented in Figure 2 for a 2500 N load on the most loaded roller.

Usually, the external load acts unsteady, so that the influence of shocks and overloads have to be taken into account when the suitable prestressing regime has to be choosen. In Figure 3. the pressure variations along centerline are presented for the same rollers as in Figure 2. but considering an overload force of *5000 N.* The profiles obtained by prestressing at lighter loads, corresponding to *3.0 GPa* and *3.5 GPa* maximum Hertz pressure, caused relative important increase in stress level at the end parts of the contact, while the profiles obtained by prestressing at heavier loads, corresponding to *4.0 GPa* and *4.5 GPa*, maintained a low level of the end part stresses.

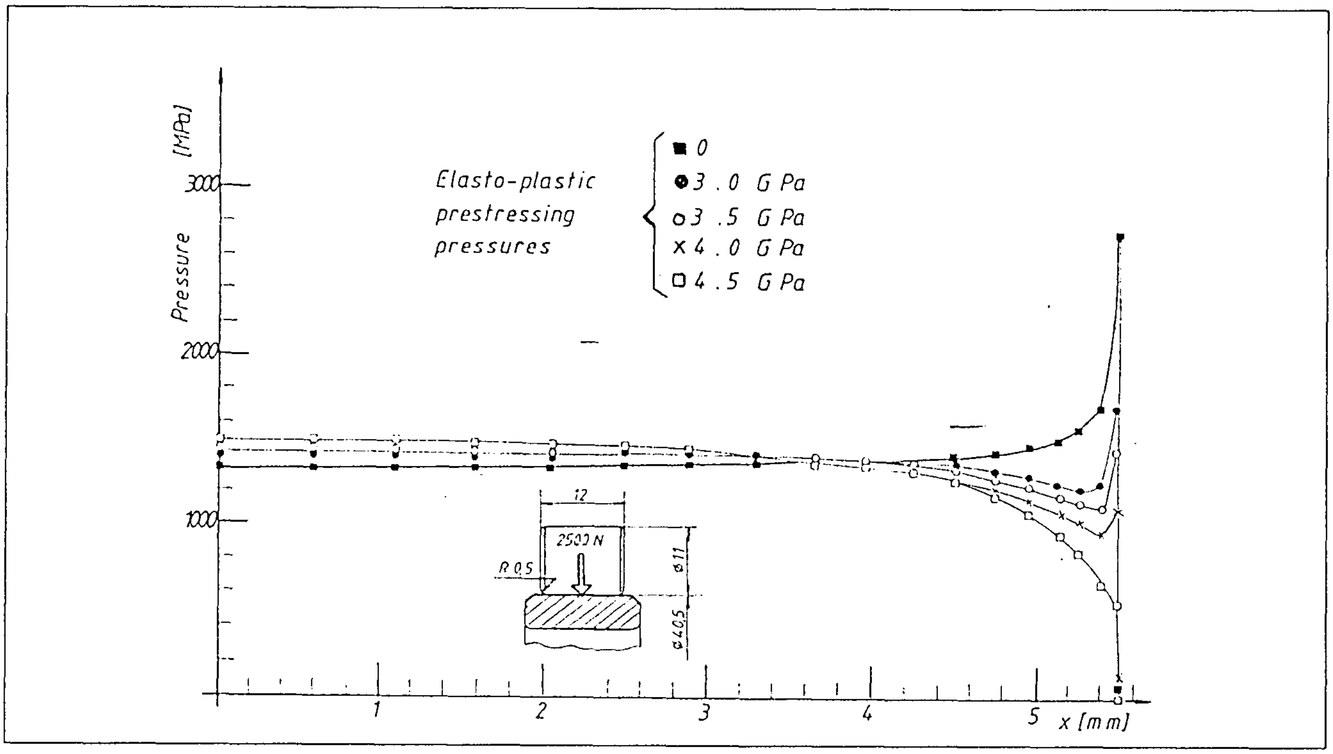

Fig. 2.

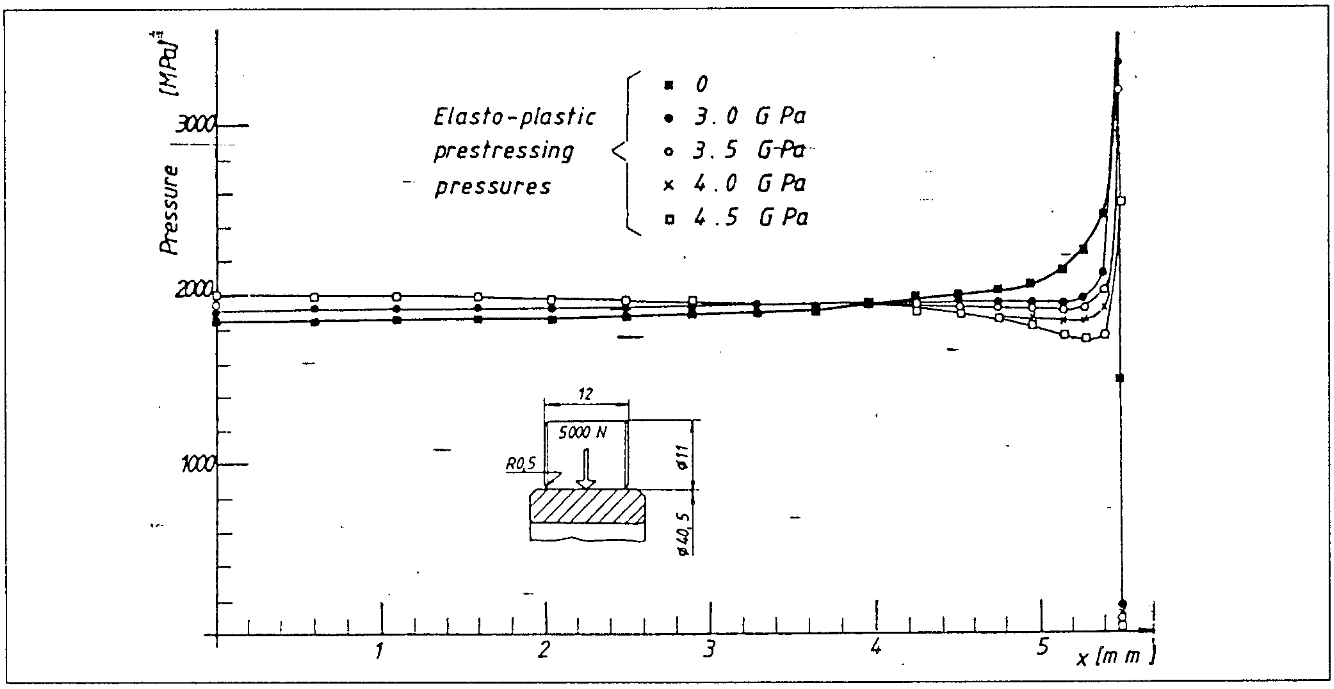

Fig. 3.

5. CONCLUSIONS

A modified profile, very close to the optimum one may be obtained for cylindrical roller using a suitable prestressing operation in elastic - plastic domain. In comparison with the old crowned profile, the plastically deformed profile is able to utilize the entire length of the roller, even at light loads. Comparative to straigh - crowned profile that maintains a stress concentrator in the passing point of the generatrix, the plastically deformed profile leads to a quite uniform pressure distribution along a centerline without any concentrator, as long as the load does not overcome the value established in the design stage.

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RESEARCH

The Determination of Load Distribution and Friction Torque of Angular Contact Ball Bearings

In the present paper, a computer supported method to determine the load distribution between the rolling elements of ball bearings considering friction is introduced. The ball bearing is subjected to a five degrees of freedom load vector. The centrifugal forces and the gyroscopic moments are calculated and the friction forces and moments as well. The friction losses are due to the rolling resistance and the sliding friction at the contact areas between the balls and the raceways. The kinematic parameters which describe the relative motions of rolling elements and rings are determined using penalty function method. During optimisation procedure, the forces and moments acting on the rolling elements have to be in equilibrium. The calculation of load distribution with friction is iterative considering friction forces and moments to be external loads. This procedure finishes when the effect of the friction forces and moments is insignificant to the load distribution of the balls. A mathematical procedure for the determination of the bearing friction torque due to load is proposed. The validity of the developed method is evaluated with the aid of friction torque measurements.

Keywords: *Ball bearing, friction, load distribution.*

1. INTRODUCTION

Nowadays demands for high speed cutting with high efficiency and productivity, lead to the use of advanced angular contact ball bearings in high speed spindles of machine tools. The significant factor, for bearing life time and tribological behaviour, is the magnitude of the occurring stresses within the bearing components during the operation and especially at the contact positions between the rolling elements and the races. To determine the stress distribution the loads between the rolling elements of the bearing have to be defined.

Andreason [1] uses a vector method in order to describe the loads and displacements that are involved in the solution of the equilibrium of deep groove angular contact ball bearings and taper roller bearings. Liu [2] extends Andreason's analysis taking into account the effect of high rotational speed on the centrifugal force and gyroscopic moment in taper roller bearings. A matrix method for calculation of the equilibrium and the load distribution in all main types of rolling elements is described by de Mul [3]. The described mathematical model considers the effect of the rotational speed on the gyroscopic moment and allows the calculation of the bearing stiffness matrix, which is essential in rotor dynamic analysis of any elastic system comprising bearings. It is well known that high speed affects the rolling element inertial load, i.e. the centrifugal forces and gyroscopic moments as well as the firiction in ball bearings. This has an influence on internal load distribution of loads and stresses. The effect of friction, on internal load distribution including skidding in high speed ball bearing, is taken by Harris [4], where the bearing is only thrust loaded.

The present paper extends a step forward de Mul's method for the determination of load distribution in high speed ball bearings considering the effect of friction in bearings, loaded by five degrees of freedom load vector. The calculated friction forces and momems that are developed between balls and outer ring are used for determination of the bearing friction torque.

2. DYNAMIC ANALYSIS OF BEARING

The outer ring in the developed calculating method is considered fixed and the inner ring can be displaced due to the load of the bearing. The extemally applied loads are referred to a point on the inner ring rotational axis, as are the displacement of the bearing (see Fig. 1). The problem is to determine the equilibrium of the inner ring under the applied loads. The determination of the load distribution without friction is carried out according to de Mul's method [3].

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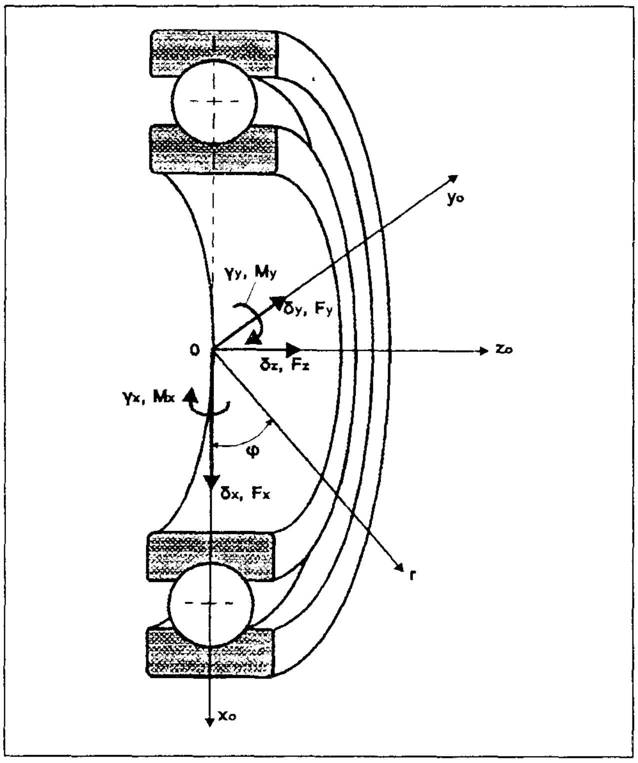

Fig. 1. Bearing Cartesian and cylindrical coordinate systems with loading and displacements

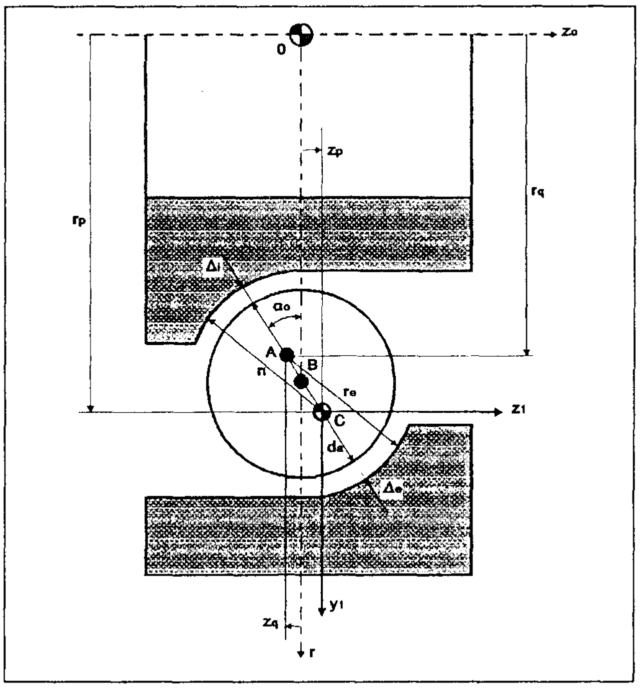

Fig. 2: Cylindrical and local Cartesian (xyz) coordinate systems in the cross-section of the ball bearing

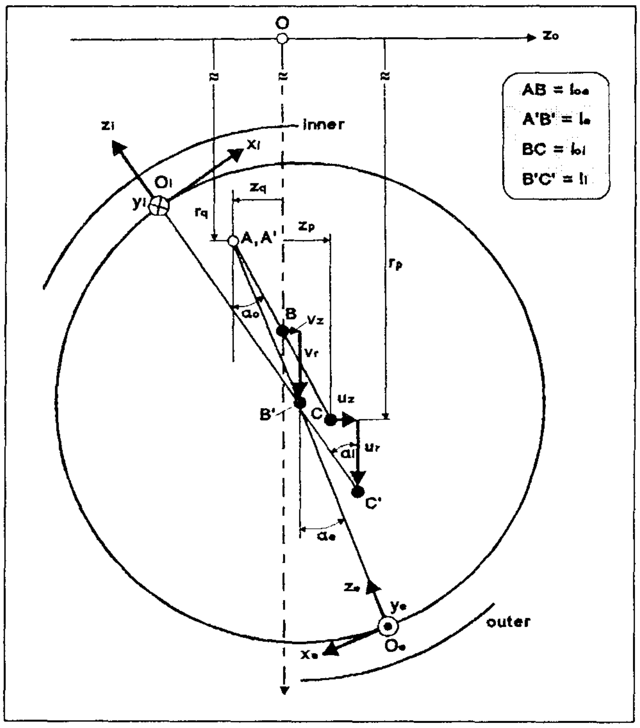

Fig. 3: Local Cartesian coordinate systems in the ball raceway contact area

2.1. Coordinate systems

For the analysis, the following coordinate systems are chosen:

- A right handed fixed Cartesian system $(xyz)_0$ with origin in the bearing (inner ring) reference point and z_0-axis along the bearing rotational axis (see Fig.1).
- A cylindrical system $(r\varphi z_0)$ with the same origin and z_0-axis, φ being the angle between r- axis and positive x_0-direction. The angle φ is chosen such that the r, z_0 plane passes through the center of the rolling element being considered (Fig. 1, 2).
- A right-handed Cartesian system $(xyz)_1$ with origin in the inner groove center, the z_1-axis parallel with z_0-axis and y_1-axis parallel with the r-axis (see Fig. 2).

In Fig. 2, the ball bearing is presented in the nominal position and the contact line that connects the centers of inner and outer ring grooves, forms the nominal contact angle α_0 with a radial axis. When the inner ring is displased, the cross-section generally moves and due to the ball centrifugal load, the ball center is no longer located on the contact line ABC, but is moved away from this line and reaches its new position B' (see Fig. 3).

In Fig. 3, two further local right-handed coordinate systems are presented. The systems $(xyz)_i$ and $(xyz)_e$ are located at the contact areas between balls and races, regarding the inner and the outer rings, with origins in the contact ellipse center and the axis x, y along the big and the small axes of the contact ellipses respectively.

Assuming that the displacements u_r, u_z (Fig. 3) are small, the homogeneous transformation matrix (4x4), that describes the relative position of system $(x,y,z)_1$ with respect to system $(x,y,z)_0$ is:

$$A_1^i = \begin{vmatrix} R_0^1 & p_0^1 \\ 0 & 1 \end{vmatrix} = \begin{vmatrix} \sin\varphi & \cos\varphi & 0 & r_p\cdot\cos\varphi \\ -\cos\varphi & \sin\varphi & 0 & r_p\cdot\sin\varphi \\ 0 & 0 & 0 & z_p \\ 0 & 0 & 1 & 1 \end{vmatrix}$$

in which the rotation matrix R_0^1 describes the orientation of system $(xyz)_1$ relative to system $(xyz)_0$. The vector p_0^i indicates the position of the origin of the system $(xyz)_1$ with respect to system $(xyz)_0$.

Furthermore the relative position of systems $(xyz)_i$ with respect to system $(xyz)_1$ is described by the matrix:

$$A_1^i = \begin{vmatrix} 0 & -1 & 0 & 0 \\ -\sin\alpha_i & 0 & -\cos\alpha_i & -(d_a/2 + l_i)\cos\alpha_i \\ \cos\alpha_i & 0 & -\sin\alpha_i & -(d_a/2 + l_i)\sin\alpha_i \\ 0 & 0 & 0 & 1 \end{vmatrix}$$

and the relative position of systems $(xyz)_e$ with respect to system $(xyz)_1$ is:

$$A_1^i = \begin{vmatrix} 0 & -1 & 0 & 0 \\ \sin\alpha_e & 0 & -\cos\alpha_e & -(d_a/2 + l_i)\cos\alpha_e - l_i\cos\alpha_i \\ -\cos\alpha_e & 0 & -\sin\alpha_e & -(d_a/2 + l_i)\sin\alpha_e - l_i\cos\alpha_i \\ 0 & 0 & 0 & 1 \end{vmatrix}$$

The relative position and the orientation of system $(xyz)_j$ with respect to system $(xyz)_0$ can be calculated by means of the following equation:

$$A_0^j = A_0^1 \cdot A_1^j = \begin{vmatrix} R_0^1 & p_0^1 \\ 0 & 1 \end{vmatrix} \qquad j = i,e$$

The matrices A_0^i and A_0^e are used to transform the friction forces and moments, occurring between the rolling elements and the inner and outer raceway respectively.

2.2. Forces and moments acting on the rolling elements

Fig. 4. illustrates the forces and moments acting on a rolling element. Friction forces and moments due to the rolling and sliding within the ring-ball contact areas, are determined after Harns [4] and are calculated with numerical double integration. The friction coefficient is selected as a function of maximum contact pressure, sliding velocity and oil viscosity [4, 5]. In the calculation of friction forces and moments besides the loads Q_i and Q_e, kinematic magnitudes as sliding and spinning velocities etc. are considered.

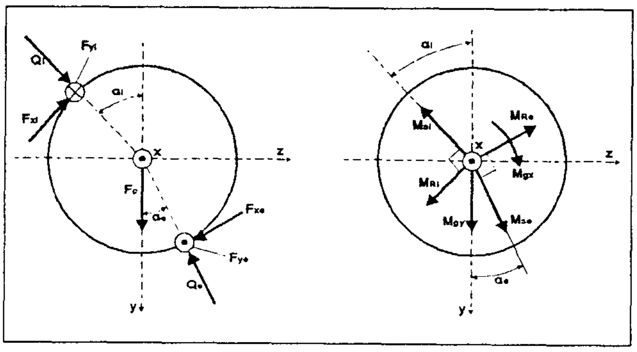

Fig. 4: Forces and moments acting on a ball

The used kinematic magnitudes depend on the rolling radii and on the angles β and β' which describe the relative motions of rolling elements and rings. These parameters must be appropriate ones in order to fulfill the equilibrium equations of forces and moments acting on the rolling element. The equilibrium equations are established in Cartesian coordinate system, parallel to system $(xyz)_1$, whose origin is the center of the rolling element. The solution of the nonlinear system of equilibrium equations can be obtained using the Newton-Raphson method. In order to prevent negative values for the rolling radii, a nonlinear programming method with restrictions is applied.

The flow chart diagram of the program that determines the kinematic parameters, used for the calculation of the centrifugal force, the gyroscopic moments and the friction moments and forces, is presented in Fig. 5. This calculation is included in an iterative procedure, which will be described in the following paragraph, and is carried out for each rolling element. The input data for this calculation are the values for the loads Q_i, Q_e and the contact angles α_i, α_e that have been calculated during the previous step of the current iteration. The initial values for these parameters, that compose the variable vector {x} are carried out by the "outer - raceway control"

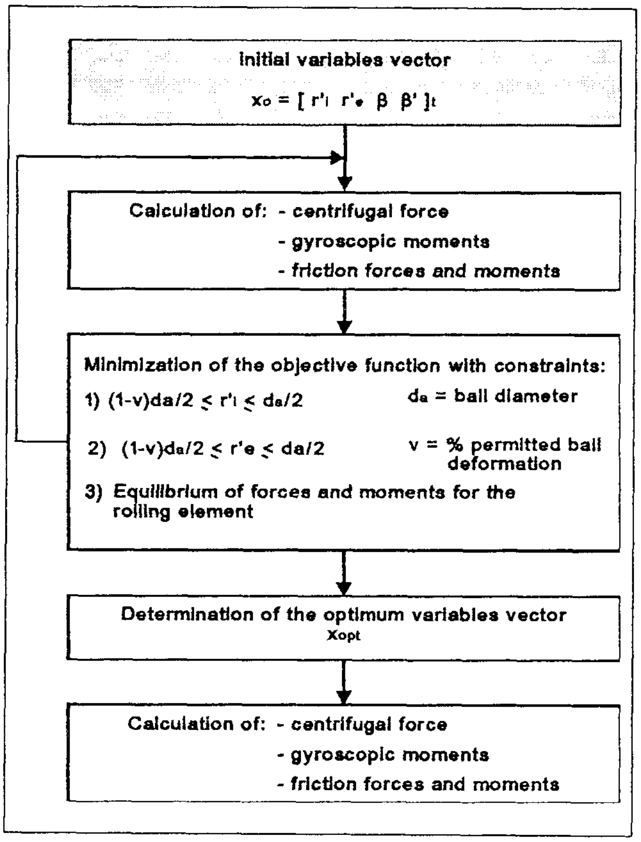

Fig. 5: Determination of the friction forces and moments using optimization method

condition. The objective function to be minimized is the square of the equilibrium equation along the x axis :

$$F(\{x\}) = [F_{yi} - F_{ye}]^2 \ min \qquad (5)$$

The restrictions that have to be fulfilled during the minimization procedure are:

- The rolling radii must have the following values in the interval:

$$(1 - \nu)\cdot d_a/2 \leq r'_j \leq d_a/2 \qquad j = i, e$$

  whereas ν is the percent permitted ball radius deformation
- Forces and moments acting on the rolling elements have to be in equilibrium

The determination of kinematic parameters is conducted by the penalty function method using the SUMT algorithm [6].

Furthermore, using the calculated kinematic parameters, it is possible to define the centrifugal force, the gyroscopic moments and the friction forces and momems, which are necessary for the determination of the load distribution with friction.

2.3. Determination of the load distribution with friction

The determination of the load distribution with friction among the rolling elements is performed, using an iterative procedure. The flow chart diagram to the program of this procedure is illustrated in Fig. 6. The initial load values are calculated considering the load distribution

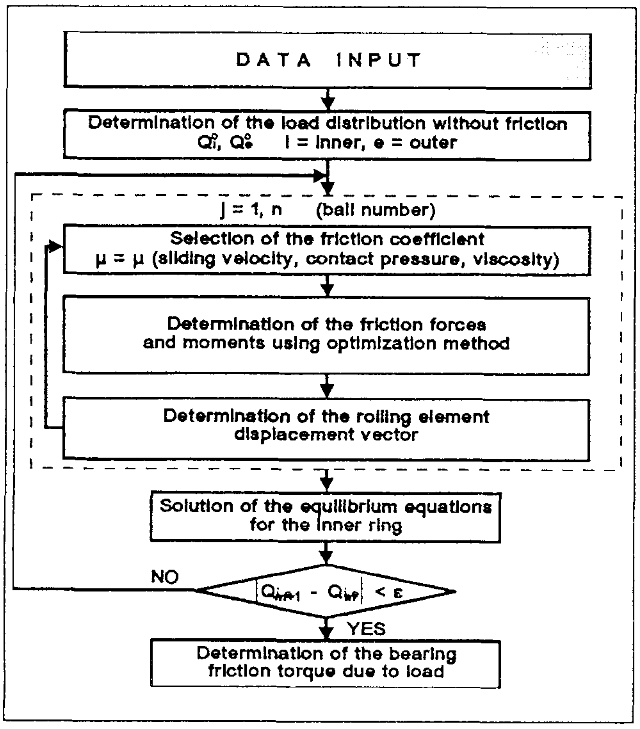

Fig. 6. Flow chart diagram of the program for the determination of contact loads Qi, Qe, contact angles α_i, α_e, and the bearing friction torque

without friction, according to de Mul's method. At the solution step $m+1$ of the iterative procedure, friction loads are assumed to be external loads, that have been defmed during the determination of the load distribution at the previous iteration step m. The iterative procedure converges when the difference between two successive values of the obtained loads is smaller, than a prescribed criterion, that indicates the desired accuracy of the calculation. The solution converges rapidly due to the low values of the friction forces and moments. Results ofthe described iterative procedure are used for the calculation of the bearing friction torque.

The determination of the load distribution with friction is based on the solution of the equilibrium equations of the inner ring. The establishment of these equations is conducted by reducing of the loads from the rolling elements to the reference point on the inner ring and then by adding them to the extemal loads acting on the bearing.

The bearing is external by loaded on it's reference point, by the load vector {F} shown in Fig. 1:

$$\{F\} = [F_x \ F_y \ F_z \ M_x \ M_y]^t \qquad (6)$$

and the corresponding displacement vector is:

$$\{\delta\} = (\delta_x \ \delta_y \ \delta_z \ Y_x \ Y_y)^t \qquad (7)$$

Friction forces and moments are considered to be external loads, calculated in the previous step of the iterative procedure. The reduction of the ball-inner ring friction forces vector, to an equivalent force vector at the reference point of the inner ring is performed by the equation:

$$\{F_1\} = [F_{1x} \ F_{1y} \ F_{1z} \ M_{1x} \ M_{1y}]^t \qquad (8)$$

where:

$$(F_{1x} F_{1y} \ F_{1z})^t = [R_0^i] \ \{F_i\} \qquad (9)$$

R_0^i is the rotation submatrix which yields by equation (4) for $j=i$,

$$\{F_i\} = (-F_{xi} \ -F_{yi} \ 0)^t \qquad (10)j$$

M_{1x}, M_{1y} are the components of moment M_1 that is determined by the equation:

$$\{M_1\} = \{p_0^i\} \, x \, ([R_0^i] \ \{F_i\}) \qquad (11)$$

p_0^i is the position vector, which yields by the equation (4) for $j=i$.

The friction moments M_{Ri}, M_{si} (Fig. 4) between ball and inner ring are transformed to the reference point ofthe inner ring, using the equation:

$$\{F_2\} = (0 \ 0 \ 0 \ M_{2x} \ M_{2y})^t \qquad (12)$$

where M_{2x}, M_{2y} are components of moment M_2, that determined by the equation:

$$\{M_2\} = [R_0^i] \ \{M_i\} \qquad (13)$$

$$\{M_i\} = (M_{Ri}\ 0\ -M_{si})^t \tag{14}$$

The load vector {Q} is transformed to an equivalent load vector {f} at the reference point of the inner ring by means of the equation:

$$\{f\} = [R\varphi]^t \{Q\} \tag{15}$$

where:

$$\{f\} = (f_x\ f_y\ f_z\ m_x\ m_y)^t \tag{15a}$$

[Rφ] is the transformation matrix (3x5) [3], and :

$$\{Q\} = (Q_r\ Q_z\ T)^t \tag{16}$$

The rolling elements forces depend on the inner ring cross section displacements according to the following:

$$\{Q\} = \{Q(\{u\})\} \tag{17}$$

where:

$$\{u\} = (u_r\ u_z\ \theta)^t \tag{18}$$

and

$$\{u\} = [R\varphi]\ \{\delta\}. \tag{19}$$

All the transformed forces at the reference point of the inner ring are summed and finally the equilibrium equations are established:

$$\{F\} + \sum_{j=1}^{n} (\{F_1\}_j + \{F_2\}_j + \{f\}_j = \{0\} \tag{20}$$

where n is the number of the balls. Due to the fact that the equations for the contact forces are nonlinear, the set of equations (20) is nonlinear as well. The unknown displacement vector [d] has to be solved using the Newton-Raphson method.

The cross section displacemsrt vector [d] yields from the inner ring displacement vector (equation 19). The displacement vector [u] and the displacement vector [v] of the ball center (see Fig. 3) are used in order to calculate the contact compression between the inner and the outer ring, which is necessary for the calculation of the contact load according to the Hertzian theory.

2.4. Determination of the bearing friction torque

The friction torque of rolling bearings is usually calculated with a two term equations, that include the torque due to the applied load and the torque due to viscous drag [4, 7].

The previously described method allows the calculation of the friction torque of a bearing, due to load. The friction torque is determined as the torque applied to the outer ring, due to friction forces and moments between ball and the outer ring. The friction torque is calculated out by the equation:

$$M_f = \sum_{j=1}^{n} (M_{f1z} + M_{f2z})_j \tag{21}$$

where M_{f1z} is the component along the axis z_0 of moment M_{f1} which is defined by the equation:

$$\{M_{f1}\} = \{p_0^e\} x([R_0^e]\ \{Fe\}) \tag{22}$$

The vector p_0^e and the rotation matrix R_0^e are determined using the equation (4) for j=e,

$$\{Fe\} = (-F_{xe}\ -F_{ye}\ 0)^t \tag{23}$$

M_{f2z} is the component along the axis z_0 of the moment M_{f2}, calculated using the equation:

$$\{M_{f2}\} = [R_0^e]\ \{M_e\} \tag{24}$$

$$\{M_e\} = (M_{Re}\ 0\ M_{se})t \tag{25}$$

3. APPLICATION OF THE PROPOSED METHOD

The proposed method is applied for the dynamic and tribological behaviour of a ball bearing that is axially preloaded [8]. The main geometrical features of this bearing are inserted in Table 1. The calculation of the load distribution with friction and the bearing friction torque is carried out for axial preload *600* and *1200 [daN]* and rotational speed of the inner ring from *5000* to *7000 [rpm]*, considering steel and ceramic rolling elements respectively. The load symmetry has as consequence an equal distribution of load among the rolling elements.

Table 1.

Pitch diameter	123.8 mm
Ball diameter	12.7 mm
Number of balls	26
Nominal contact angle	20.6 degrees

The outer contact load for one rolling element, considering friction, is presented in Fig. 7, whereas the outer contact angle is illustrated in Fig. 8 for various rotational speeds. The values for the contact load are lower for the ceramic balls, in comparison with the steel ones, due to the significantly smaller density that ceramic material has. Due to the low level of frictional forces, the load distribution is not strongly affected by the frictional losses.

Using the optimisation procedure, the rolling radii and the angles β, β' are obtained. The rolling radii are almost similar to the ball radius. The variation of the angles β, β' versus rotational speed is illustrated in Fig. 9 and 10, respectively. The angle β for ceramic ball is higher than the corresponding one for steel ball, whereas the change of the angle β' versus the rotational speed for ceramic ball is lower than the corresponding for ball bearings.

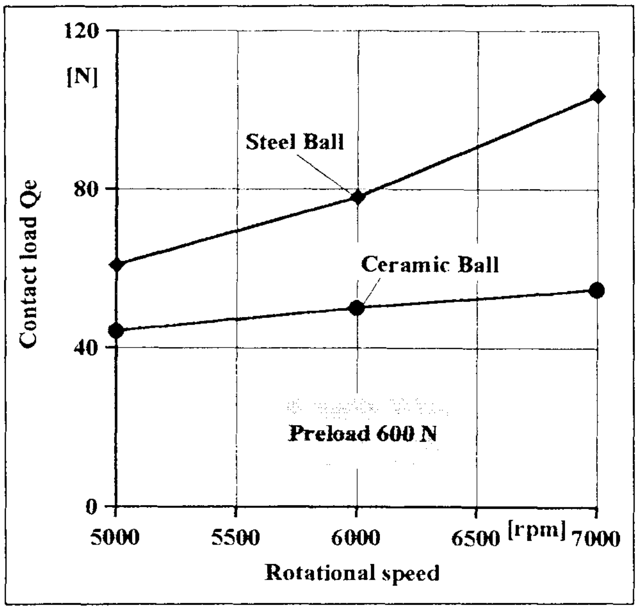

Fig. 7. Contact load developed between ball and outer ring versus rotational speed

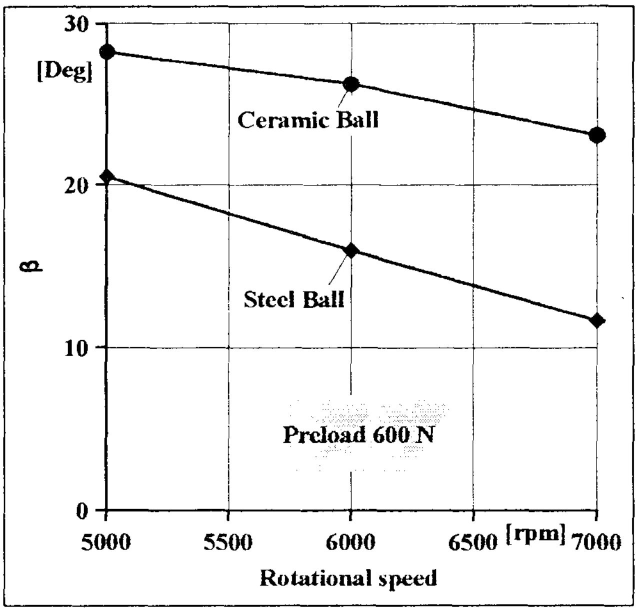

Fig. 9. Variation of the angle β versus rotational speed

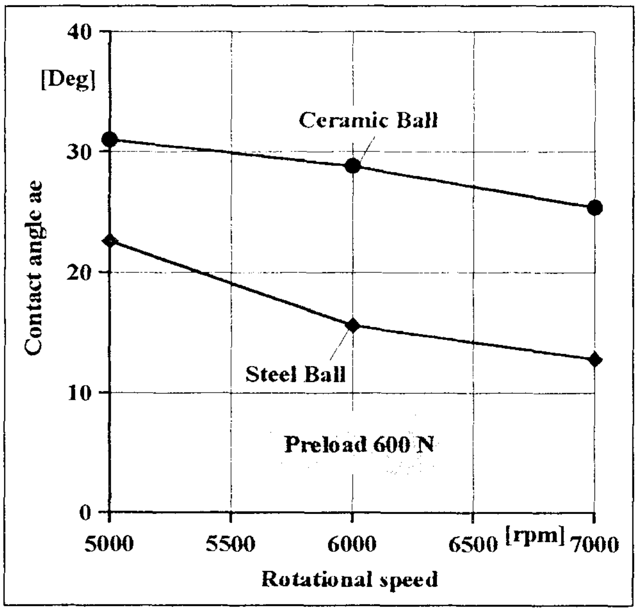

Fig. 8. Variation of the outer contact angle versus rotational speed

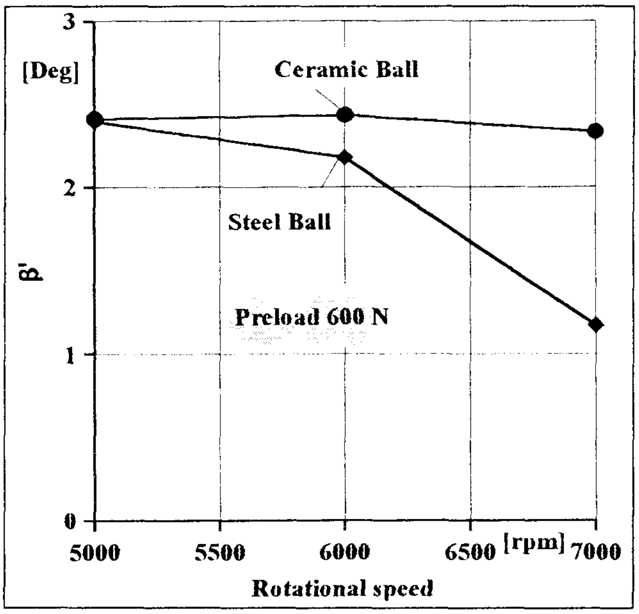

Fig. 10. Variation of the angle β' versus rotational speed

The influence of the preload to the bearing friction torque for various rotational speeds is illustrated in Fig. 11. The bearing friction torque increases while the rotational speed rises. The effect of ball material properties on the bearing friction torque is shown in Fig. 12. Hybrid bearings perform lower friction torque than steel ones.

In order to verify the validity of the developed numerical procedure, calculated values for bearing friction torque, due to load, have been compared with measured ones [8, 9] as it is shown in Fig. 13. The measured values are higher than the corresponding calculateri, because the developed algorithm does not take into account the friction due to viscous drag as well as the friction between the ball and the cage.

4. CONCLUSIONS

In the present paper a matrix method for the calculation of the load distribution among the rolling elements of angular contact ball bearings considering friction is introduced. The mathematical model takes into account the centrifugal forces as well as the gyroscopic moments and enables the calculation of the friction forces and moments at the contact surfaces of the ball and the inner and outer ring respectively. Furthermore, in this paper, a mathematical procedure is proposed for the determination of the bearing friction torque, due to the load. The validity of the developed method is evaluated with the aid of friction torque measurements.

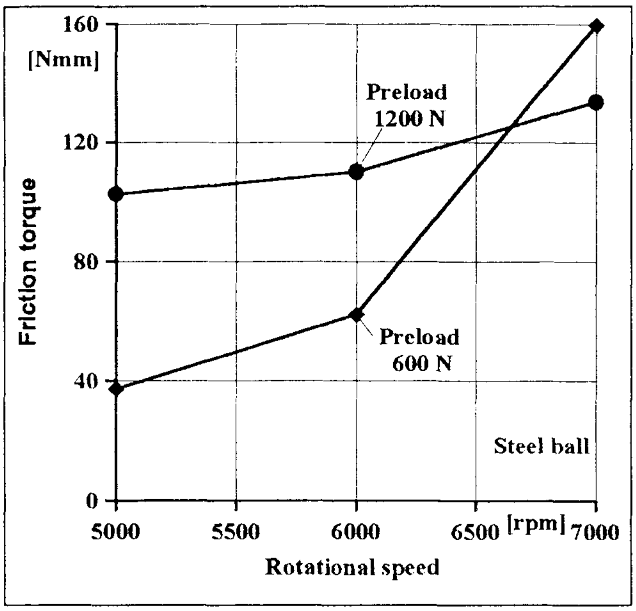

Fig.11. The effect of preload to the bearing friction torque

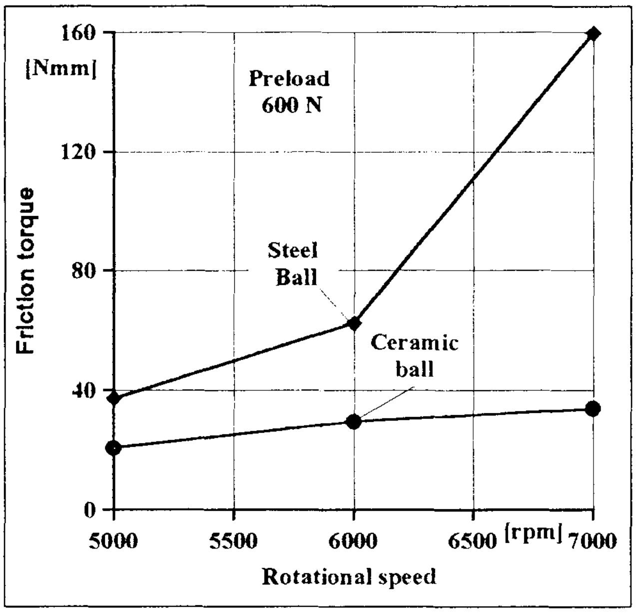

Fig. 12. The effect of rolling element material on the bearing friction torque

This method can be applied for all main types of rolling elements bearings. The mathematical model can be extended for the calculation of the load distribution considering the interaction between rolling elemectts and cage.

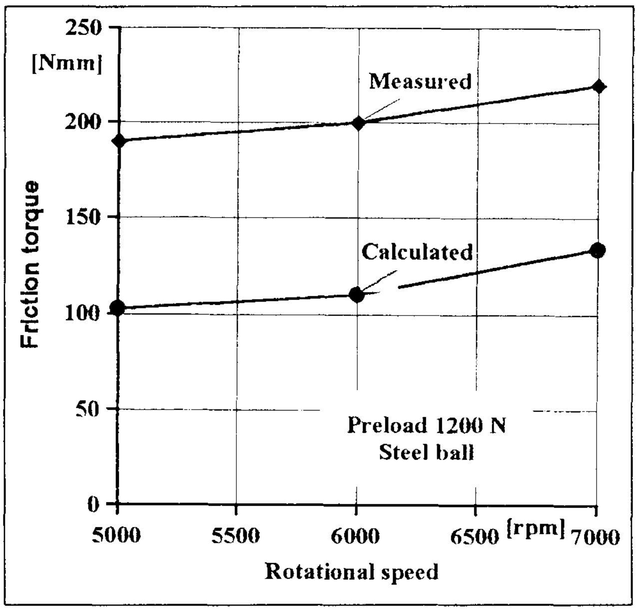

Fig. 13. Comparison between measured and calculated bearing friction torque

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RESEARCH

UDK 621.835.2.004.62

V. F. BEZYAZICHNY, A. N. SEMYONOV, I. N. AVERIANOV

Fretting Wear of Machine Parts

This article represents the results of investigating fretting wear mechanism of the main construction materials, the equipment and technigue of carrying out experiments. Experimental dependence of the main loading characteristics effects on fretting wear rate was produced, fretting wear model describing general regularities of the process proceeding was suggested.

Keywords: *Fretting, bearing, gear, surface layer.*

At present the quality problems of commercial products: machines, mechanisms, transportation facilities in particular have not only technical but great economic and social importance. When functioning, the machines are subjected to the loads corresponding to the product purpose and to the loads caused by service conditions as well. During the normal service the main reason of lowering working characteristics of machines. upon which purpose factors depend is the wear caused by components friction.

One of specific and the most dangerous types of material destruction in contact interaction zones is fretting wear - corrosive-mechanical wear of cont,acting bodies during short oscillating relative motions. In areas damaged by fretting wear the processes of setting, abrasive failure fatigue-corrosive failure are proceeding. These processes are observed as a rule in the point of contact of tightly contracted components of as a result of vibration between surfaces microscopic shear displacement appears. Bolted and riveted joints of frames, bearing fits, hinged joints, clutches, pipe couplings and others are subjected to wearing during fretting wear (corrosion).

One of the principal defects - fretting consequences is also the wear of structural member fits for bearing seats. Wear of fits results in the increase of vibration and dynamic loads. As a result long life of bearings, shafts, gears and other conjugate parts is decreased. Thus when fitting the bearing 307 with the clearance of *0.1 mm* its durability decreases 1.5 times and with the clearance of *0.2 mm* it decreases twice as compared with the durability by fit with zero clearance.

Among components and units determining reliability of machines an important place is occupied by rolling contact bearings. Durability of bearing assemblies is determined to a great extend by service conditions (especially by loads on seats) and the degree of their dynamics. One of the consequences of these working conditions and design features of bearing assemblies is the wear of structural member fits. Cylinder blocks, gear cases and rear-axle housings are referred to these parts. The increase of clearance between the fitting hole and the outer race of a bearing results in the change of almost all parameters of an assembly unit work, disaligment of shaft axes, the change of optimum gearing conditions, teeth crumbing and breaking, selfdisengagement of transmission, the increase of dynamic loads and vibrations and so on.

Wearing of fitting holes results from many complex processes beginning with bearing fit conditions, but the main reason of clearance increase during the service is fretting wear, while propping off the outer races of bearings.

Experimental equipment for fretting wear investigations of erecting joints was worked out on the base of a plant "JMACH-20-75" for metallographical investigations.

The contacting scheme determining the wear of a given conjugation is practically close to a plane scheme and can be modelled by the contact "plane-to-plane" under laboratory conditions.

The principal material of antifriction bearings is steel IIIX15; of structural parts of agricultural machines are low carbon steel CT3, CT5, possessing good casting properties, speed- torque characteristics and machinabi-

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lity. Hence as tested samples the materials mentioned above were used.

According to possibilities of testing equipment one of the samples was produced in the form of a cylinder with the diameter of *10 mm* and the length of *50 mm* from steel CT3, contrasample - a roller of a needle bearing was made of material ШХ15 with the diameter of *1.5 mm* and length of *10 mm.*

Face planes of samples were machined by lapping on a cast iron disk for obtaining the corresponding roughness $R = 0.8...1.32\ \mu m$.

Reference test conditions were defined from the adequacy condition of a fretting wear driving mechanism and the possibilities of a test unit and constitute standard pressure at the point of contact $P = 1...10\ MP$, vibromotions amplitude $A = 0.1...0.5\ mc$ vibromotion frequency $f = 15...30\ Hz$, the number of vibrocontact loading cycles $N = 4 \cdot 10$ *cycles.*

Fretting wear control carried out in the process of friclion showed that at the initial stage of the first cycles the samples parting - "negative wear" and a sudden inerease of friction coefficient occur. Then wear rate stabilizes and the curve takes the usual form (fig. 1).

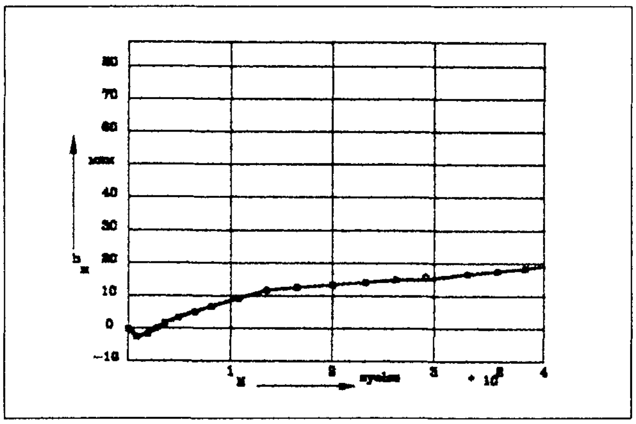

Fig. 1. Dependence of Wear on the Number of Cycles

The absolute load value affects the wear rate by external friction of any kind because it determines the area of actual contact.

As the carried out investigation showed, under fretting wear conditions the particles size remains practically constant irrespective of the contact pressure value. Only a number of wear products increases. The data obtained (fig. 2) showed that wear rate grew with the increase of contact pressure, linear variation of wear rate being observed in the investigated range after established wear had begun. With the increase of contact pressure the "negative wear" value changed either.

The effect of vibromotion amplitude under the chosen loading conditions is represented in fig. 3 and is well

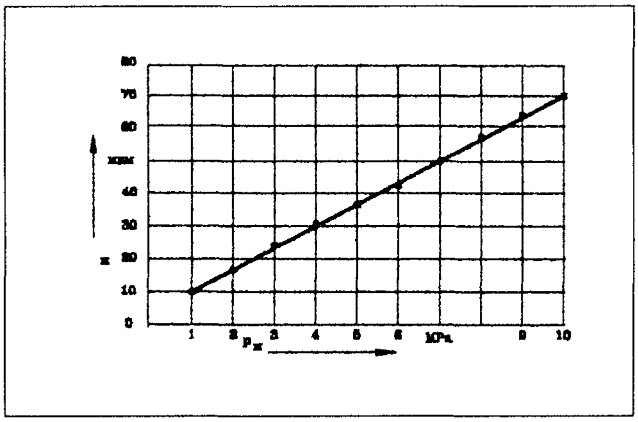

Fig. 2. Dependence of Fretting Wear on the Unit Load

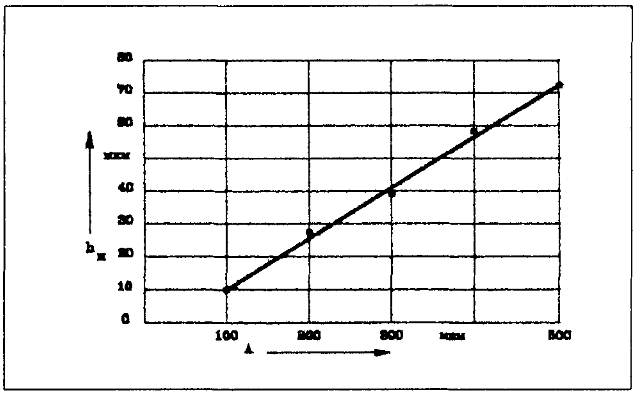

Fig. 3. Dependence of Fretting Wear on Slipping Amplitude

approximated to a straight line. Under the conditions of the small amplitude fretting the surfaces have a more uniform character and are covered with a dense layer of wear products. With the increase of the vibromotion amplitude the non-uniformity of surfaces increases, the processes of setting and plastic deformation are intensified. With the increase of slipping amplitude the number and value of the applied microvolumes of a surface layer increases, the removal of wear products from the friction area is facilitated, their damping ability decreases.

The study of fretting wear and friction surface dependences by means of an electronic microscope allowed to distinguish three characteristic stages for the part of driving wear mechanism and the proceeding regularities.

The first stage is the stage of adhesive interaction which is developed in intensive plastic deformation setting, depth tearing out, mutual metal transfer on contacting surfaces (fig. 4). The great intensity of setting during fretting results from the dynamic character of loading during which oxides covering the metal and adsorbtion films are quickly destroyed and the interaction of a juvenile metal occurs. Small fretting amplitudes and the closed type of contact prevent from penetrating oxidizing medium reproducing protective oxide films into actual contact areas.

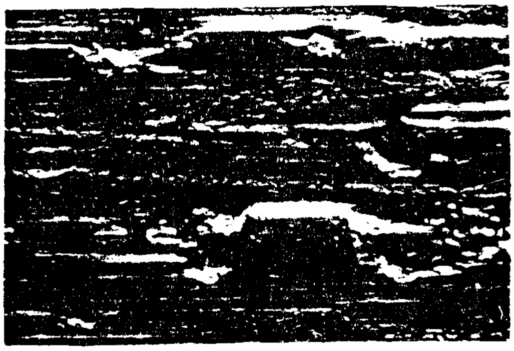

Fig.4. Plastic Pressing Back of Material in Places of Actual Contact at the Stage of Adhesive Interaction.

As a result of plastic deformation the surface layers are heavily packed, the arising adhesive bonds become stronger than the basic material and it results in breaking the rule of a positive gradient of mechanical properties, that is, adhesive bond becomes stronger than the layers lying below.

For fretting wear conditions with its low relative velocity and slipping amplitudes setting is practically inevitable and is stipulated by non- recoverable destruction of protective oxide films during anormally low necessary loads as compared with unidirectional friction.

The second transitional stage of fretting wear, the stage of running-in consists in wearing adhesive layers of transferred and redeformed material and the formation of nominal contact area.

The period of this stage is determined by features mentioned above. The approach of contacting surfaces as a result of a part of the scab wear to the original level results in the formation of the new areas of actual contact

Fig. 5. Fragment of Characteristic Friction Surface at the Stage of Running-in

where originally elastic and then plastic contact is realized. The greatest part of a nominal load is taken by the central area of contact and its lesser part is transmitted to the peripheral area of friction, therefore the progressive growth of scores in places of an actual contact is absent (fig. 5).

The formation of a nominal contact area, the stabilization of temperature-force processes, determined by the set up level of force, and the coefficient of friction characterizes the transition to the third stage, the stage of a setup wear. Accordingly the processes of a friction area destruction are stabilized: the fatigue mechamism becomes dominant. Abnormally high friction, strengthening of a surface layer and the increase of the yield limit result in the appearance of maximum stresses at some depth from the friction surface, and during the further deformation rudimentary cracks, which are merged in microcracks. are formed and grow. The inerease of the structural microdefects number and sizes, their merging in the closed contour form the peeling off material lobe (fig. 6).

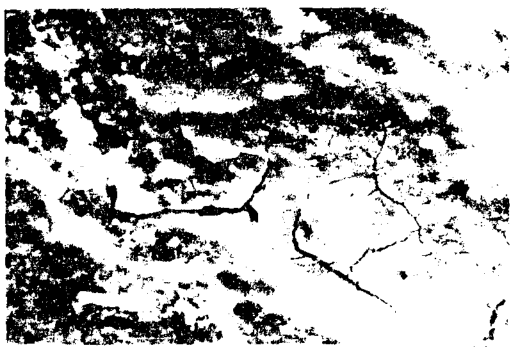

Fig. 6. Peeling off Plastic Particles of a Surface Layer as a Result of Fretting at the Stage of the Established Wear

Henee at the stage of the established wear the fatigue destruction of a surface layer by means of peeling off plastic surface fragments is prevailing and it stipulates the decrease and stabilization of wear rate as compared with earlier stages.

The suggested fretting wear model describes general regularities of all stages proceedings on the base of studying driving mechanisms of a surface layer destruction. The model reliability was confirmed by additional fretting wear study of titanium and nickel alloys and by fretting resistance investigation of different methods of friction surface strengthening.

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Noise and Vibrations Monitoring of Gearing Tribological Processes

RESEARCH

Vibrations monitoring and diagnosis of gears tribological processes was an efficient and modern method to determine gears failures.

Vibration signal analyzed in frequency or time domains, offered information about incipient failures and causes which produced them. Also, information can be obtained about gears running conditions.

In this paper are presented experimental test conclusions realized by authors, to establish diagnosis and monitoring possibilities of tribological processes.

Keywords: frequency spectrum, CEPSTRUM analysis, frequency lateral stripes.

1. INTRODUCTION

For gearing, noise and vibrations of engaged gears are modulated in magnitude or frequency because of manufacturing and running defects. Therefore, frequency lateral stripes in frequency spectrum are generated, placed around the engaging frequencies and their harmonics. The establishment of these modulated frequencies represented an important and supplementary sign for gearing defects diagnosis.

To this reason, the CEPSTRUM analysis is used. To establish noise and vibrations diagnostics and monitoring of gears tribological process, some authors' researches were done.

2. THEORETICAL ASPECTS

2.1 Theoretical consideration about gears failures vibration diagnosis

Generally, noise and vibrations frequency spectrum emitted by gears was complicated, but usually it had peaks corresponding to following frequencies:

- engaging frequency and its harmonics;
- secondary components (appeared accidentally);
- lateral stripes;
- low harmonics of rotational frequency.

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The peaks from frequency spectrum corresponding to engaging frequency and its harmonics, were created by periodical shocks. These shocks appeared in engaging process and they are generated by deviations from ideal shape appearing in manufacturing process. Also, they are generated by nonuniform flanks wear in running process. On the other hand, shocks are produced by periodical variation of engaged teeth deformation.

Secondary components appeared likewise at engaged frequency components, but they corresponded to a different teeth number and they are generated by kinematic errors of machine tools. These components had tendency to be attenuated by gears running in.

Frequency lateral stripes placed around engaging frequency and their harmonics, can be explained by gears uniform vibrations, in amplitude or frequency modulated, emitted by local failures.

For example, every tooth load fluctuation had tendency to generate a vibration amplitude changing, thus appeared an amplitude modulation.

Consequently, these tooth loads fluctuations, determined angular rotational speed fluctuations and a frequency modulation appeared.

These two modulations (in amplitude and frequency) determined the lateral stripes amplitude increase, proportional to modulating frequency. This frequency contained important information about modulated source.

The low harmonics of rotational frequency are generated by cumulated impulses, which are repeated at every gear rotation.

Except mentioned frequencies, in frequency spectrum appeared other important components, depending on rotational frequency suitable for gears vibration modes.

Therefore, in frequency spectrum appeared a lot of important components so that was very difficult to correctly identify.

2.2 Theoretical aspects concerning teeth flanks wear according to frequency spectrum and CEPSTRUM analysis

Referring to teeth flanks wear, it can be shown that it was bigger on rolling circle on both sides and produced an engaging frequency distortion. To this reason, over high engaging frequency ($f_a = Zf_1$) the low gear rotational frequency was overlapped. Thus, obtained an amplitude modulation.

In extreme case of 100% modulating, f_1 component disappeared, remained only lateral frequencies, defined by:

$$f_{1i} = f_a - f_1$$
$$f_{1s} = f_a + f_1 \qquad (1)$$

Because modulating process was distorted by manufacturing failures and modulation was not realized 100%, in frequency spectrum appeared new peaks at following frequencies:

$$f_{ji} = f_a - j \cdot f_i$$
$$f_{js} = f_a + j \cdot f_i \qquad (2)$$
$$j = 2, 3, 4, ...$$

To establish vibration monitoring and diagnosis possibilities of gears wear processes, experimental tests were realized. These tests were done for gears which were followed by teeth flanks wear degree, frequency spectrum and CEPSTRUM analysis.

In figure 1 are shown frequency spectrum and CEPSTRUM analysis for an initial gear (without teeth wear).

Also, in figure 2 are presented frequency spectrum and CEPSTRUM analysis for same worn gear.

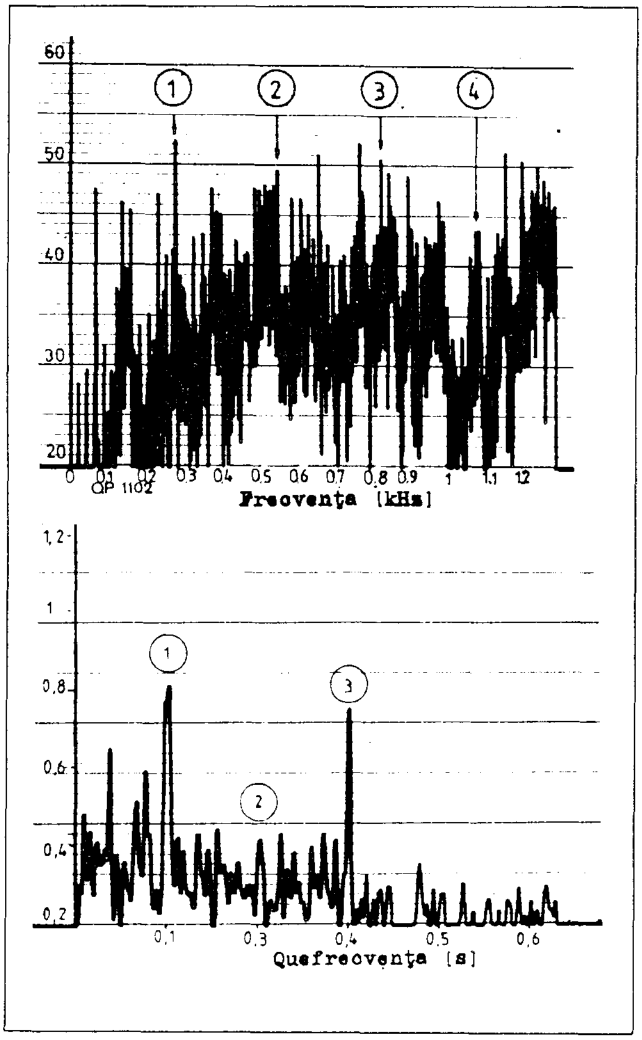

Figure 1

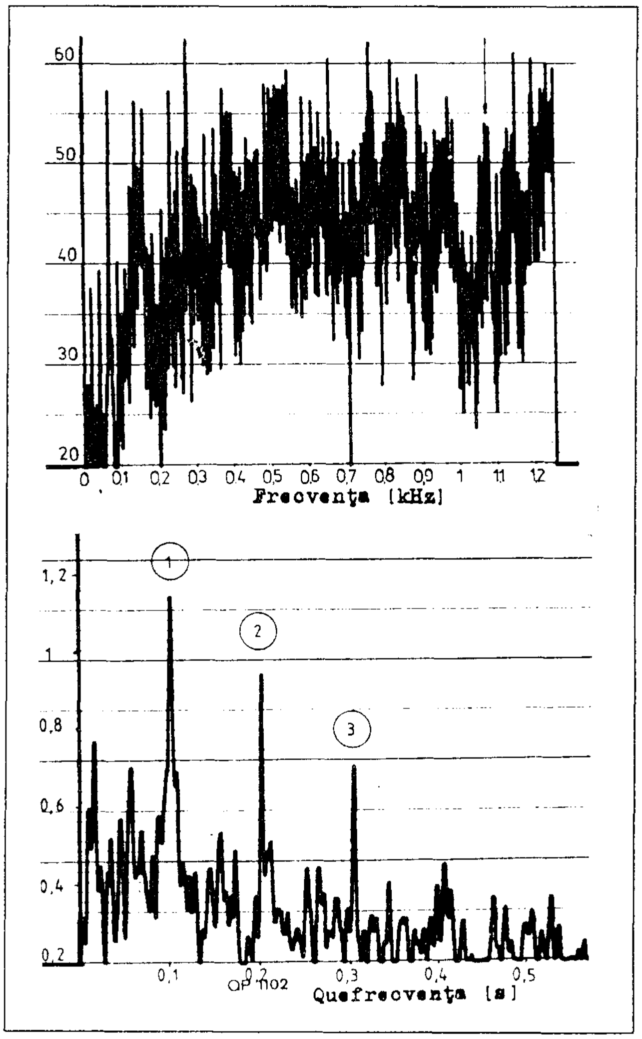

Figure 2

3. CONCLUSIONS

Analyzed experimental results, can be highlighted by following conclusions:

- gears frequency spectrum included frequency lateral stripes, placed around engaging frequency and their harmonics. The number of frequency lateral stripes increase denoted deteriorating conditions. Distance between frequency lateral stripes gave information about modulating source.
- wear teeth flanks produced an important frequency spectrum chenge, especially in high frequency domain,
- to detect frequency periodicity, the CEPSTRUM analysis was used which gave information about modulating frequencies. These frequencies can be related to mechanical failures. Consequently, CEPSTRUM analysis simplified experimental results interpretation, because in CEPSTRUM periodical peaks from frequency spectrum were cumulated.

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