tribology in industry

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World Tribology Congress Opening Address

by
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A World Congress, such as this WTC'97 has three principal objectives:

Firstly, there should be a Presentation of the latest state of Research and Developments in tribology.

This objective, and especially the demonstration of developments in research, traditional and new, and of a high level is likely to be outstandingly successful. Indeed the whole programme of this Congress is an excellent display of the state-of the-art of the science of tribology and - as such probably unique. The Programme Committee must therefore be congratulated on this achievement.

Secondly, it should present an opportunity for tribologists from all over the world to meet, to establish personal relations, to exchange views, and to up-date themselves of the state of the art and to fnd new friends, and this Congress has all the potentials of this objective being successfully met.

Making contacts between individuals, cross fertilisation of their ideas, and not forgetting contacts and arrangements by Tribology Societies or Groups with each other, is as important to tribology, as is the scientifc and technical content of the many papers, posters and lectures.

In spite of being members of the same umbrella organisation, namely the Intemational Tribology Council, there is too little contact between most of the world's Tribology Societies, too much inward looking, too much introspection.

In whatever country we operate, limited resources do not permit us t do as much as we wish. Therefore joint international activities - not necessarily on the scale of the World Congress - should be much in the minds of the Committees of the ITC Member Societies. There are already good examples, such as Nortrib, Balcantrib, ASME/STLE conferences which re regular joint endeavours that have been successful, and that do not put too much strain on individual societies.

I therefore put it to you formally that representatives of ITC's Member Societies at this Congress and those wishing to form Tribology Groups or Societies should get together during this Congress to establish closer links and more regular contacts, even if only by a regular exchange of letters and-above all - by sending information of their individual and joint activities and all happenings in the feld of tribology for inclusion in the Tribology ITC Information Sheets.

That brings us to the third and probably most important objective, viz the Economic and Coinpetitiveness aspects of Tribology - The link with the User.

Tribology is not a cocooned subject, nor can it exist on its own, it is a means to an end and that end is improved quality and performance, meeting new environmental requirements, lower costs, better energy use, in other words, increased industrial competitiveness.

The best and most advanced results of Research and Development in tribology, stored in a cupboard or left unused, will not be of much use. They will only justify the efforts and the resources spent on them only if they are applied. Tribology must not only have a USER, but that USER must also be part of the Tribology scenario. He must broadly know what tribology is and above all what it can do for him, not only by user directed R&D but also by use.r orientated Design and Application. It is a eneral fact that most Tribology Societies and Groups do not sufficiently identify themselves with the end-user of tribology, and often even less with the Funding Bodies, be these Government Departments, Educational Establishments or Industrial Undertakings. Not unexpectedly manufacturers of tribology materials, whether lubricants, bearings, friction materials and the like are doing their best to promote their wares, and there is no shortage of competitive advantages stated in their literature and publications and this is good.

Also, where tribology is a vital component e.g. in space, nuclear environment etc. the user exerts the market pull. But outside these,groups, comprising only about a quarter of tribology, there is a wide gap and there is lack of education in and promotion of tribology.

From this it follows that if in the other 75% or so areas of tribology awareness and understanding including education are not promoted, tribology's economic and competitive role cannot be fully recognised, and consequently the subject will suffer, and ultimately interest and funding will diminish and industry and a country's economy will be the loser.

In this context it might help you to be reminded how the concept of Tribology "the science and technology of interacting surfaces in relative motion" came into being and especially why it spread world-wide extremely speedily. At the joint IMechE and Iron and Steel Institution conference in Cardiff in 1964, one session had the title "Damages and Breakdowns, believed to have been due to Lubrication and their real Causes". During this session engineers from many countries showed slides of cemeteries of broken steel plant. It was quite obvious that most of these did not have their causes in what was conventionally known as lubrication.

These fndings were brought to the notice of the late Lord Bowden, then Minister for Science, who set up a Committee of investigation into Research, Education and - above all - Industry's needs.

The report by the group of industrialists, academics and educational experts was completed within 6 months and the now well known findings were obtained. However before finalisation the Senior Official in charge of the Ministry's Science side, who was much. impressed by the findings, expressed the need for the quantifications of the obtainable savings. "Unless you tell government or industry in terms of pounds, shillings and pence - the only language they understand - how much can be saved by the application of this new concept (i.e. tribology), your report will gather dust amongst many others".

This was excellent advice and work started in quantifying the savings easily obtainable through the application of tribological principles. They were vast, amounting for the UK to £ 515 million p.a. at the time, which in today's terms would be over £ 1.5 billion. No Government, no industry could lightly ignore such fgures which amount to just over 1% of GNP. Even rore thorough investigations in Germany, USA, Canada and China confirmed the estimated realistic savings to be between 1% to 1.4% of GNP. Therefore it was not merely the great truth but recognition of the economic advantages that led to the rapid world-wide recognition of Tribology.

Yet over the years this link between applied tribology and its benefits has weakened in many countries. Instead there has been a trend towards academic research and away from application and the financial benefts arising from it. Why is this so?

Let us admit, whilst Tribology research is intellectually very stimulating, this is not always so to the same extent in the educational and awareness promotional fields, but they are essential to retain public interest and public funding, and they too can become very exciting.

The current trend has already led to a loss of interest and understanding by many of the ultimate users of tribology and thus by Govemmental and other funding bodies. In many cases they are now insuffciently aware of the large fnancial and competitive benefits that can be obtained by tribology and of the methods required to achieve these benefits, nor of the sometimes enormous costs that ignorance of tribology can lead to.

This trend must be reversed. Academic research is vital, and must be supported, but it must not be in place of industrial directed developments, education in and application of tribology, nor of its constant promotion, especially directed at the next generation. We must regain the lost user participation, and - where a Tribology Society is part of an Engineering Body - regain the active support of that parent body. Otherwise there will inevitably be reduced recognition of tribology - this in spite of its increasing importance in our fast developing technological world, inevitably leading to reduced funding by ovemment and by industry - except probably only in those few areas in which tribology is recognised to be vital, e.g. space.

It is the task of all ITC Tribology Societies to ensure that their activities embrace education and the end-user i.e. the steelworks, the ship builder, the cement plant, the railroads, the computer and magnetic storage departments etc. but ešpecially Engineering Design offices and that - above all - the economic and competitive advantages of tribology, and the disadvantages and losses that can be suffered if tribological considerations are ignored - for whatever reason - be driven home, and loudly so. Ultimately,governments should be made and especially kept aware of the national economic effect of the application of tribology and these - as we know - are vast.

HOW CAN IT BE DONE? A simple guide might be given by an old American ditty that goes:

"He who whispers down a well
about the goods he has to sell
will not make as many dollars
as he, who climbs a tree and hollers"

Let us therefore holler to all those who can benefit from tribology and let us do so with solid facts and figures - and they must be solid - the significance of which can easily be understood by non-tribologists, which includes Engineers, Administrators and Politicians.

UDK 621.9.01.0031

RESEARCH

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Reliability of Aluminum Oxide And Graphite Modified Polytetrafluor-ethylene Sliding Contact Joints in A Compressor with Reduced Lubrication

Application of an automatic W-4 grease feeder in reliability investigations served to work out a totally new concept of investigation of a set of sliding surfaces of a compressor with significantly reduced lubrication. It was assumed that each injection of an oil dose in the moment of reaching the boundary state by the sliding surfaces set is its "renovation". This type of renovation, carried out in an appropriate moment, so as to prevent damaging the object, is called a preventive renovation. The results of an aluminum oxide - graphite modified PTFE sliding joint have confirmed applicability of this method for reliability investigations of a set of sliding surfaces: cylinder bearing surface - piston rings of a compressor with substantially reduced lubrication.

A set of sliding surfaces: a cylinder bearing surface - piston rings of compressors with significantly reduced lubrication according to PN-77/N-04010 is an unreperable object, working in a continuous way or with accidental breaks [1]. This object may continue its task after replacement of damaged set of sliding surfaces with a new one or after removal of a boundary state.

The knowledge of tribological characteristics of a sliding contact joint: aluminium oxide - graphite modified polytetrafluorethylene (PTFE) applied to a set of sliding surfaces of a compressor with reduced lubrication allows for a statement that this object will work on the basis of these materials in a continuous way until reaching a boundary state.

A model compressor KL-2 makes possible carrying out reliability investigations in which a parameter has been used - characterising the boundary state of a sliding contact joint, i.e. a threshold temperature difference between the average cylinder's temperature on the head side and on the crankshaft side as well as the average temperature of the central part of a cylinder. This is a parameter fully reflecting the effect of friction and wear on the course of sliding mating of contacted materials [2]. Application of an automatic W-4 grease feeder was considered in reliability investigations of a set of sliding surfaces of compressors with significantly reduced lubrication [3].

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And so, if we determine the value of Δt at a boundary level (this means that exceeding this level is tantamount with reaching by the investigated sliding contact joint of very unfavourable lubricating conditions - a boundary state) then the time necessary to reach this state is the time till "the first breakdown" occurence. Breakdown removal is the forcing, i.e. automatic feeding to the friction zone of an oil dose restoring lubrication to the stated conditions.

Investigation of reliability of a set of sliding surfaces of KL-2 compressor is a special case. It can be assumed that each injection of an oil dose in the moment of reaching the boundary state by the object is its "renovation". The object remains the same, it is not replaced. This type of renovation carried out in a correct time instant, so as to possibly avoid damage of the object, is called a prophylactic, preventive, ex ante type renovation. This is a kind of conditional renovation, in which the basis is the decision model, using a set of various current technical information in the object. In our case the temperature difference of contact joint Δt is this information.

Investigating a sliding contact joints: aluminium oxide - graphite modified PTFE in conditions of reduced lubrication, we deal with damages resulting from gradually occurring irreversible changes of ageing type and of wear of materials' outer layers. A normal distribution is the appropriate model of up time for objects of this type.

As reliability coefficients for the investigated object, i.e. for the set of sliding surfaces: cylinder bearing surface - piston rings of a compressor with reduced lubrication the following has been assumed: the average time of proper work till reaching, by the object of boundary state θ, the function of the damage intensity $\lambda(t)$ and the function $\Lambda(t)$ called the cumulative damage intensity or the leading function.

Investigations of reliability of the set of sliding surfaces: cylinder bearing surface - piston rings of a compressor with reduced lubrication have been carried out on a model compressor KL-2 using automatic W-4 grease feeder. The following parameters of compressor's work have been assumed: average sliding rate $V_{av} = 1\ m/s$, pumping pressure $P_t = 0.1\ MPa$, total time of compressor operation $\tau = 1000\ h$. Cylinder bearing surfaces made of PA2 aluminum alloy covered in an electrolytic way with a $60\ \mu m$ thick aluminium oxide and graphite modified piston rings made of PTFE have been used as the set of compressor's sliding surfaces. The reduced lubrication of the contact joint has been carried out using a method allowing for, by means of properly directed nozzles, single injection to the friction zone of *4.1 mg* of lubricant in the form of an oil mist. The boundary state of the investigated object has been determined at the level $\Delta t = 1.6\ ^oC$. During compressor's work injections boundary states have been consecutively recorded.

The investigations carried out resulted in determination that during *1000 hours* of compressor's work 94 boundary stated occured. This number of boundary states effected in injection of total of *290 mg* of oil that corresponds with oil consumption of $0.5\ mg/m^3$. We could say that this is the cost of running a compressor with substantially reduced lubrication in extreme conditions. The starting point to determine reliability characteristics is the distribution of boundary states in consecutive *100 hours*; time intervals encompassing *1000 hours* of compressor's work. The average time of proper work till reaching the boundary state has been calculated. For the studied object it amounts to $\theta = 10.6\ h$. The function of damage intensity $\lambda(t)$ and the leading function $\Lambda(t)$ have been presented in Fig. 1.

The mathematical model of the object's reliability has been verified on the basis of experimental data using the Kolmogorov - Smirnow test. At the significance level $\alpha = 0.005$ the obtained characteristics $\lambda(t)$ and $\Lambda(t)$ have a normal distribution what means that the assumed mathematical model is correct.

Analysing empirical reliability functions $\lambda(t)$ and $\Lambda(t)$ the following conclusions can be drawn on the investigated set of sliding surfaces: cylinder bearing surface - piston rings of a compressor with substantially reduced lubrication:

- gradual increase of both functions points out that the usable potential of sliding contact joint gradually decreases,
- this means that irreversible changes occurred in the tribological area, gradual wear of the outer layers of

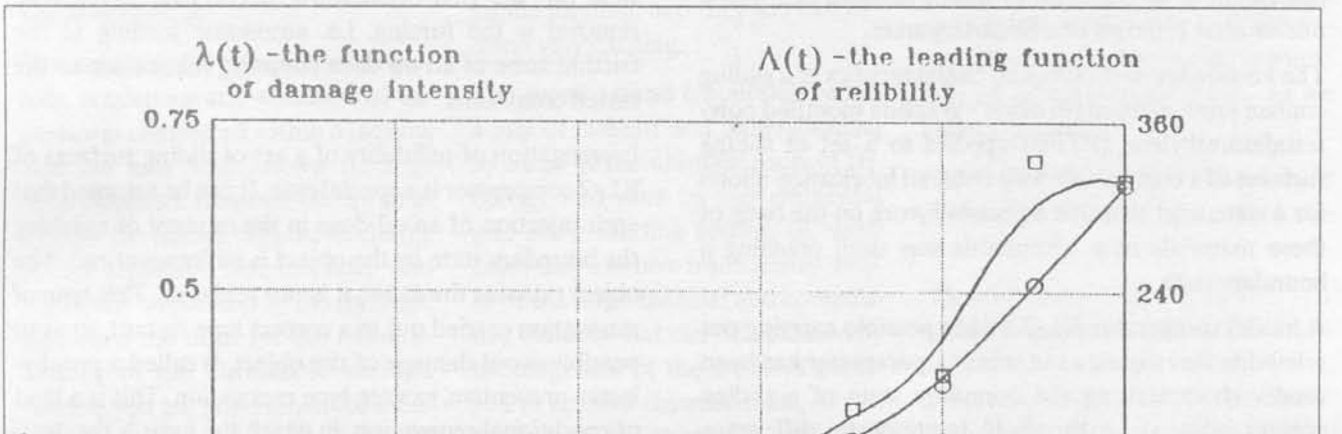

Fig 1. Reliability characteristics of a set of sliding surfaces cylinder bearing surface - piston rings of a compressor with subsantially reduced lubrication

aluminum oxide and PTFE resulting in a decrease of compressor's capacity,

- the course of the leading function ((t) points out at depleting the reserve of the object capability to perform the task and proceed according to the principle of increasing entropy in physical systems.

Keeping in mind the fact that the usable potential of the sliding contact joint: aluminium oxide - graphite modified PTFE is higher compared with numerous other sliding contact joints, depleting of its capability reserve occurs slower. Reliability characteristics for various materials sliding contact joints should be determined to find it out. This will allow to find the optimum sliding contact joints of a high aptitude, that indispensable in operating machines and installations.

CONCLUSIONS

Obtained results of investigations explain the structure of tribologically useful oxide layer. It has been shown that the oxide layer obtained on aluminium alloys can mate sliding with modified polytetrafluorethylene in conditions of technically dry friction and very significant reduction of lubrication. Advantageous values of friction and wear coefficients allow for application of contact joints in piston compressors. This has been used in the construction of lubricant-free compressors of air and freon. Results of operational and stand investigations revealed high usability in friction points of these compressors.

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To all our Reader,

we wish happy

and successful

New Year 1998.

Editorial Board

RESEARCH

UDK 621.892:621.9.027

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Influence of Contact Surface Modification on Tribological Behavior of Tools for Metal Powder Pressing

In the paper are presented and analyzed some tribological effects of different surface treatments (plasma nitriding, TiN reactive magnetron sputter ion plating and duplex diffusion-coating deposition treatment) applied to the set of materials for manufacturing the metal powder pressing tool elements. Tribological investigations are of the model type and they were conducted on the tribometer with the pin-on-disc contact pair geometry. The contact conditions were modeled based on data obtained by measurements of radial and residual sintering pressure in real machining conditions. As the simulation criterion were used the dominant mechanisms of the wear in the real system and on the model.

In all the tested cases the contact surfaces modification procedures have exhibit great potentials with respect to improvement of tribological properties of base materials, and especially from the aspect of wear resistance. In that, different materials are characterized by different degrees of improvement. Established effects of the contact surface modification, besides in view of contribution to savings of high quality materials, through increase of tool working life, are especially important from the aspect of their substitution with materials of lower tribological quality and price.

Application of metal powder metallurgy based products is today practically unlimited, and in many cases the only possible one as well. In that, the most present are products based on the sintered metals and their alloys. Fulfillment of strict requirements with regards to dimensions, shapes and density of the sintered parts, as well as the economic indicators of the machining process, is directly caused by decrease of numerous, mutually superimposed negative consequences of friction and wear in the contact of the working elements of tool and powder, during the pressing process and ejecting of the pressed parts.

Metal powder forming by pressing in molds is being realized within the specific tribomechanical machining system. In it, during the phases of the powder compression and ejecting of the pressed piece, the tribological interaction is unfolding of the contact surfaces of tools (mold and ejector) and abrasive material of the increasing density during the process, in conditions of very high contact pressures and elevated temperatures

As opposite to some metal forming processes, where friction can play the positive role in the process of metal forming, the friction in the powder pressing process affects extremely negatively the pressure distribution over the pressed piece volume, and with that also the density distribution. On the mentioned state of the density distribution depend the largest number of pressed piece characteristics, namely the characteristics of the sintered product. Direct negative consequences of tool working elements surfaces wear on the machining process represent the phenomena of increased clearances between the mold and compressor, and falling in of the smallest particles of the poured powder, unacceptably high roughness of the pressed piece surfaces, and exceeding of the prescribed tolerances of form and dimensions of the pressed piece.

In consideration of this, it is clear that special importance has investigation of the tribological aspect of the powder pressing, and improvement of the tribomechanical system in which the process is realized. Concerning the metal machining by cutting and forming, the machining of the metal powder pressing is much seldom the subject of tribological investigations, and especially in the very actual area of tribological improvement of tool by application of some of the modern methods of contact surfaces modification.

The published papers are aimed to measurement of friction and wear parameters, and especially from the aspect of lubricant and material of the tool elements

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[1-5]. In that, two approaches are present: (a) measurements on production equipment [1, 2, 3], and (b) measurement of the friction and wear parameters on the special model [3] or on the pin - on - disc tribometer [4, 5]. The experimental investigation, presented in this paper, was devoted to the tribological improvement of powder pressing tribomechanical system by application of different treatments for the tool contact surfaces modifying.

Three types of surface treatments (plasma nitriding, TiN deposition by reactive magnetron sputtering, duplex treatment by diffusion and coating deposition) and four tool materials (tool steel C 4150, high cutting speed steel C 7680, tool steel obtained by powder metallurgy ASP-23 and hard metal) were varied.

In all the tested cases the contact surfaces modification procedures have exhibit great potentials with respect to improvement of tribological properties of base materials, and especially from the aspect of wear resistance.

The established effects of the contact surface modification can bring to the decreasing of tribological dissipative effects in the powder pressing tribomechanical system by the optimal tool material and surface treatment selection.

EXPERIMENTAL

Tribological test method. From the standpoint of the tribological interaction of the tool and the pressed piece, regardless of the degree of complexity, one or two simplified models can be used, shown in Figure 1, that are related to pressing the cylindrical bushing and small cylinders.

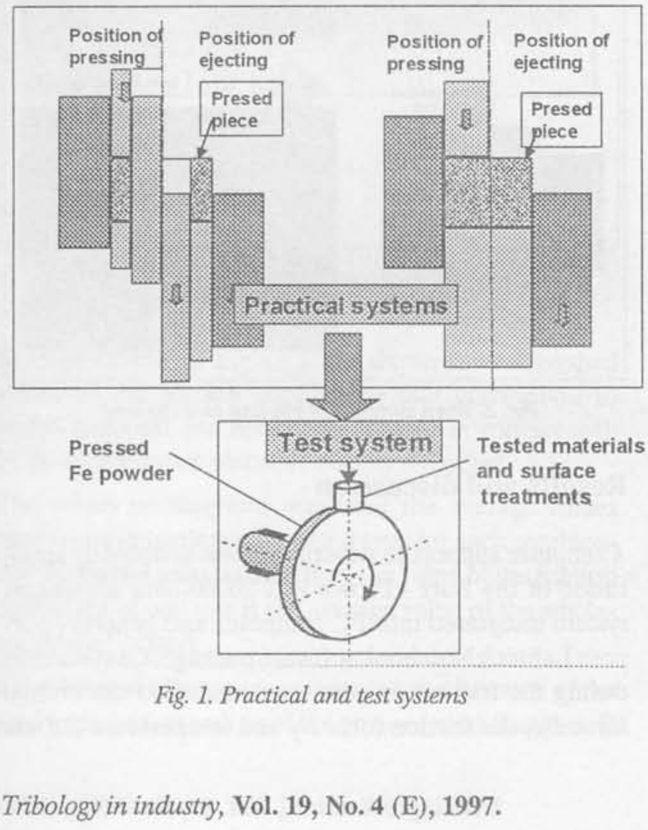

Fig. 1. Practical and test systems

As tribologically critical are treated two contact pairs: pressed piece/mold and pressed piece/mandrel. In them the sliding occurs over the working surface of the tool elements of the abrasive particles of the increasing density during the compression process, namely of the pressed piece of certain density during the pressed piece ejection process.

Due to the series of advantages in regard with possibility of controlling the numerous working conditions parameters, structure of tribo-system, and parameters of the tribological processes, high costs and short time needed for obtaining the results, advantage is given to the accelerated model investigations on tribometers. The pin - on - disc contact scheme was chosen with linear type of contact. As opposite to approach of Mallender and Coleman [4], and due to series of obvious advantages, pins as tribologically more endangered elements, were made of various tool materials, and discs were made by the powder sintering up to the prescribed density.

Tool materials. Wear pins, 10 mm in diameter and 15 mm in length, were machined of several tool materials: tool steel C 4150, high cutting speed steel C 7680, tool steel obtained by powder metallurgy ASP-23 and hard metal WC-Co. Their chemical composition is given in Table 1.

Table 1. Chemical composition of tool materials

	Material	Chemical composition
1	Tool steel C 4150	2.06C, 0.3Si, 0.3Mn, 11.8Cr, 0.1V (+ Fe)
2	High cutting speed steel C 7680	0.9C, 4Cr, 5Mo, 6.5W, 1.9 V (+Fe)
3	PM high cutting speed steel ASP-23	1.7C, 0.3Si, 0.3Mn, 4.2Cr, 5Mo,6.5V, 1.9V (+Fe)
4	Hard metal WC-Co	84.5WC, 15Co, 0.5(TaNb)C

Table 2. Conditions of surface treatments

Substrat	Plasma nitriding	TiN deposition	Duplex tretatment
C 4150	P=4.5 mbar 8%N T=370°C t = 60 min	Ub=105 V Ua=20 V Pc=5.1 kW p=0.6 Pa t=60 min T=450°C	Nitriding: P=4.5 Pa t=20 min TiN deposition: Ub=105 V Ua=20 V Pc=5.1 kW p=0.6 Pa t=60 min T=450°C
C 7680 and ASP 23	P=4 mbar 6% N T=430°C t=90 min		
WC+15Co	P=4 mbar 6% N T=430°C t=90 min		

Modification of the contact surfaces was done by application of three procedures (plasma nitriding TiN deposition by magnetron sputtering and the duplex treatment-diffusion and TiNdeposition), in the Institute in

CPT of Faculty of Electrotechnics and IMT from Belgrade, under conditions shown in Table 2.

Duplex technology consists of previous surface treatment of substrate by diffusion of nitrogen, carbon or boron in order to obtain a diffusion zone up to 1 mm tick with enhanced hardness, and subsequent deposition of a tribological wear resistant coating [8]. Diffusion surface treatment provides enhanced load bearing capacity and enhanced adhesion of the substrate material and enhanced adhesion at the interface.

The complete review of tool materials combinations with relevant parameters of hardness, roughness and coatings is given in Table 3.

Table 3. Tested samples characteristics

Materijal	Characteristics				
	Roughnes Ra (mm)	Hardness			Coating thickness (mm)
		HRC	HV0.05	HV0.03	
C 4150	0.05-0.10	61			
+ Nitriding	0.13-0.19	65	1272		
+ TiN	0.11-0.21	56		2020	3.1
+ Duplex	0.12-0.23	57		2680	3.3
AS-23	0.07-0.10	62			
+ Nitriding	0.10-0.17	63	1207		
+ TiN	0.18-0.31	64		2100	2.8
+ Duplex	0.19-0.29	65		2450	3.0
C 7680	0.06-0.11	62			
+ Nitriding	0.12-0.23	65	1362		
+ TiN	0.12-0.27	64		2380	2.9
+ Duplex	0.16-0.28	66		2840	3.1
Hard met.	0.03-0.06	70			
+ Nitriding	0.09-0.15	72	1577		
+ TiN	0.13-0.26	72		3140	3.2
+ Duplex	0.16-0.29	73		2690	3.1

Powder and pressed pieces. For experiment realization the corresponding mixtures were prepared of the Fe powder and lubricant powder. The pure mechanically reduced powder NC 100.24 of the Swedish company Hoganas was used, of the sponge-like structure and purity *99.76 %*, pouring density *2.45 g/cm*^{3} and leaking time 29s / 50g. Fe powder was mixed with the Kenalub lubricant for preparation of the testing pressed pieces in the form of discs of diameter *60 mm*, of thickness *9 mm*, with *6.8 g/cm*^{3} densities. Kenalub is more recent lubricant, basically the amide wax with addition of the zinc stearate. It is convenient for high densities of the pressed pieces.

Contact conditions and simulation criterion: All the tribological tests were realized in constant contact conditions: normal load $F_N=30\ N$ and the sliding speed $v=0.35\ m/s$. The chosen value of the normal force F_N corresponds to real radial pressure during the pressed piece ejection, that was experimentally determined in real conditions. The sliding speed value corresponds to the level of maximum speed of pressing, and it is larger than the speed of pressed piece ejection in real conditions.

Such contact conditions parameters and the very small degree of pin covering (due to linear contact) have provided for the desired accelerated unfolding of the wear process, namely the accelerated tests.

The basic problems of such accelerated tests on tribometers and using of the obtained results come from the fact that neither the friction indicators, nor the wear indicators represent the interior material properties, but they are results of complex and stochastic tribological interaction of surfaces in the given contact conditions. In order of model tests results to be applicable, to a certain extent, on the real process of powder pressing, in experiment preparation the elementary simulation algorithm was obeyed [6, 7]. The contact conditions were modeled based on data obtained by measurement of the radial and residual compression pressure in real conditions. The similarity of the contact surfaces wear forms that were determined in real and test conditions was used as the simulation criterion (Figure 2).

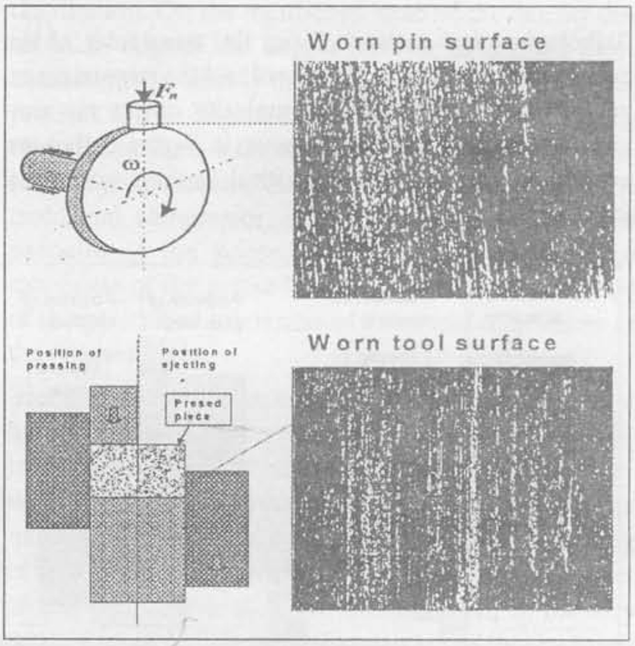

Fig. 2. Worn surfaces of pin and tool element

Results and discussion

Computer support to experiment was enabled by application of the Burr - Brown PCI 20000 data acquisition system integrated into PC computer and general - purpose Labtech Notebook software package. Continuously during the friction process were measured the normal force F_N, the friction force F_T and temperature *T*. Data was gathered with *10 Hz* sampling rate with forming of the numerical channel with calculated values of the friction coefficient. The example of the output signal is given in Figure 3.

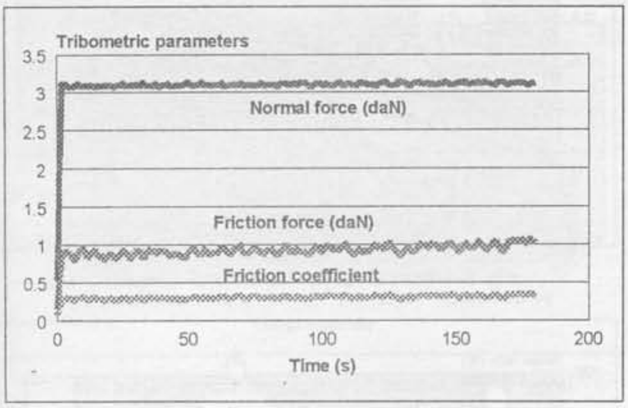

Fig. 3. Example of signal graphs

In regard with initial linear nominal contact, the development of the wear process on pins was monitored as growing of the wear scar average width, that was measured by optical microscope (Figure 4). Considering to the obtained results the corresponding wear curves were formed.

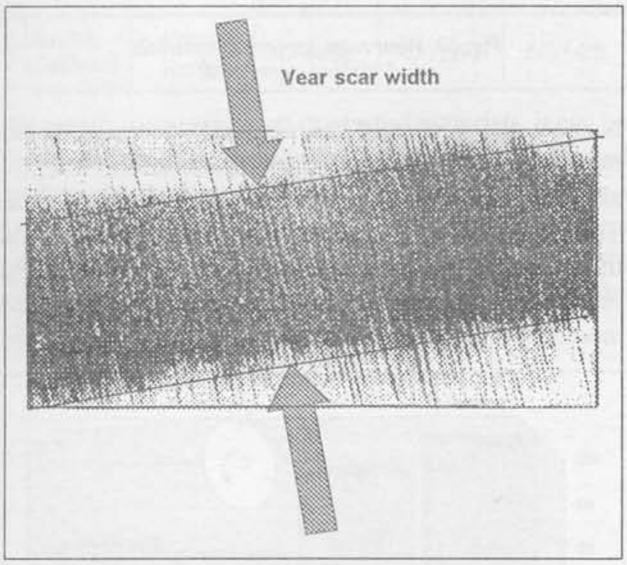

Fig. 4. Wear scar appearance

Friction effects. In Figure 5 are shown the established values of the friction coefficients that correspond to tested materials and surface treatments in contact with Fe powder pressed pieces of density *6.8* g/cm^3.

The values on diagrams represent the average values based on 10 completely repeated tests for each combination of contact materials. In that, the value of the friction coefficient of one test is the average value of the whole-considered period.

The frictional effects are represented as a function of the base materials, as well as of the types of modifications. It

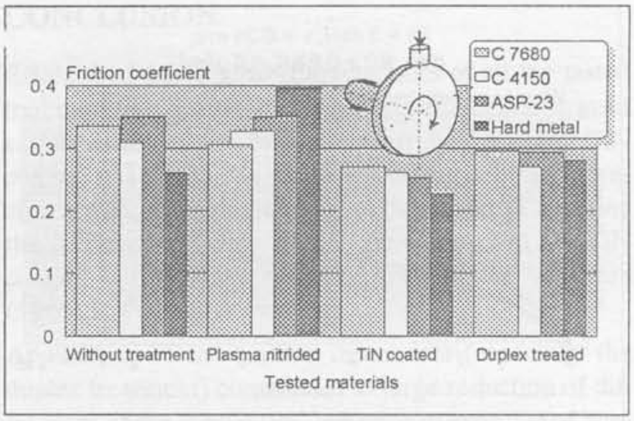

Fig. 5.Friction coefficient for tested materials and surface treatments

is visible that differences in frictional behavior of tested base materials are significantly expressed. The friction coefficients vary in the relatively wide range, from *0.26* up to *0.35*. For this case the smallest friction coefficient corresponds to hard metal, and the biggest to ASP-23 tool steel.

The nitriding of contact surfaces in described conditions has no effect for tool steel C 4150 and PM high cutting speeds steel ASP-23. It has positive effects for the case of the high cutting speeds steel C 7680, and for the hard metal it has very negative effect.

The TiN coating for the tested base materials shows very positive frictional effects, by lowering the friction coefficient for about *20 %*.

Somewhat more moderate positive effects (lowering of the friction coefficient for about *10 %*) are established for the application of the duplex treatment. Besides the lowering the friction coefficient levels, the deposited TiN coating and the duplex procedure of modification, obviously contribute to narrowing the dissipation of the materials friction coefficients, i.e., decreasing their frictional differences. In that, the ranking order, valid for the base material, remains unchanged.

Wear effects. The development of the wear process as a function of the varied materials is illustrated by the wear curves for the varied materials and surface treatments (Figures 6 - 9). They are based on ten repeated tests.

The positions of the wear curves that correspond to base (unmodified) materials, with respect to wear curves that correspond to modified materials, very convincingly show the contribution of all tested types of modifications to the wear resistance increase.

In order for these effects to be more clearly and comparatively presented, the diagram was formed in Figure 10, with calculated values of wear rate for all tested combinations of the materials and the surface treatments.

It is obvious that varying the pin materials and the surface treatments expresses more prominent effects on the wear resistance, than on the friction coefficient.

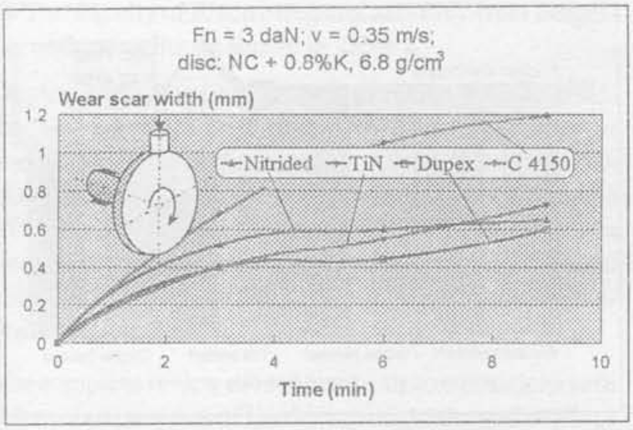

Fig. 6. Wear curves for C 4150 steel samples

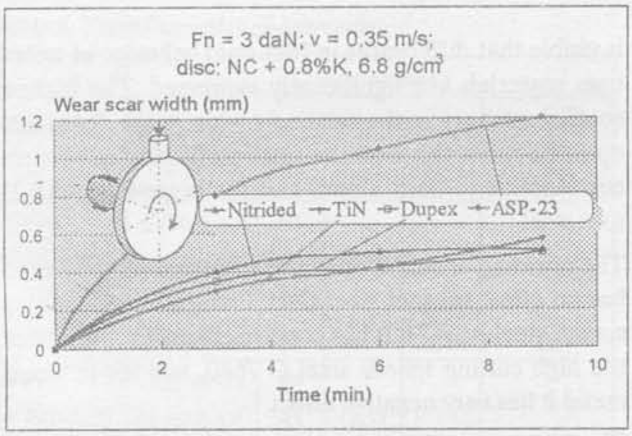

Fig. 7. Wear curves for ASP-23 steel samples

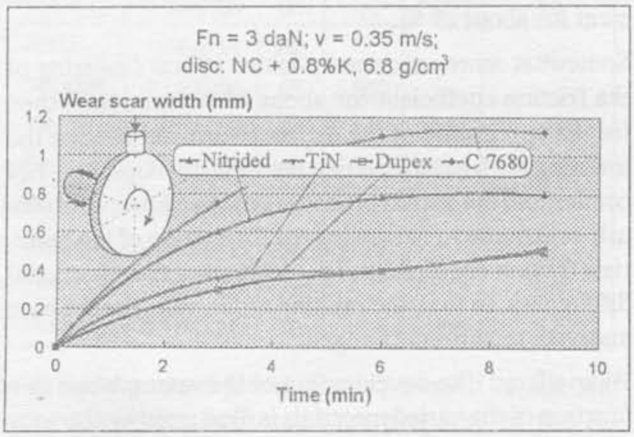

Fig. 8. Wear curves for C 7680 steel samples

Namely, the established differences show that, by change of material, the wear resistance can be increased twice. The convincingly best test results correspond to hard metal samples, and then the high cutting speeds steel C 7680. Wear rates of ASP-23 and C 4150 are very close.

The positive effects of modifications are expressed through the decrease of the wear rate of the base materials on the average of: *41 %* for ion nitriding, *49 %* for TiN deposition, and *52 %* for the duplex procedure. Those average values, considering all the tested materials, can be used only conditionally.

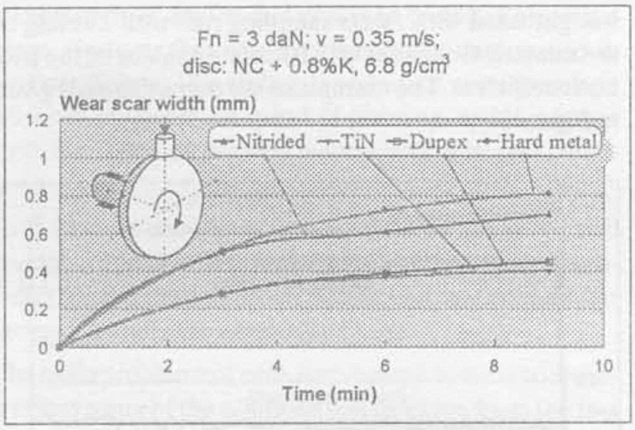

Fig. 9. Wear curves for hard metal samples

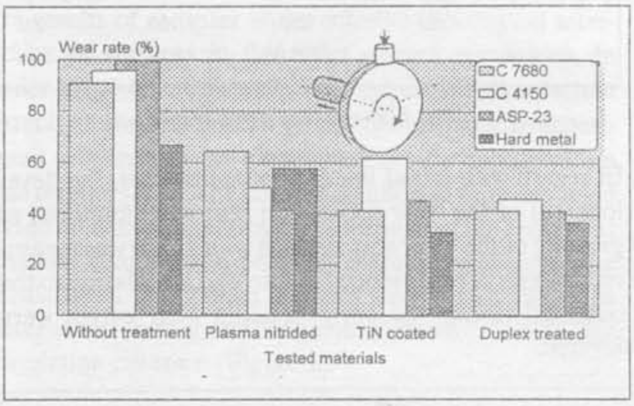

Fig. 10. Wear rates for tested materials and surface treatments

Namely, the degree of the wear rate decrease, at the expense of modification is different for different materials - it is higher for materials with lower wear resistance. The biggest improvement of a wear resistance is obtained in the cases of ASP-23 steel (Figure 11) and C 4150 steel modifications, and the smallest in the cases of hard metal modifications (Figure 12.).

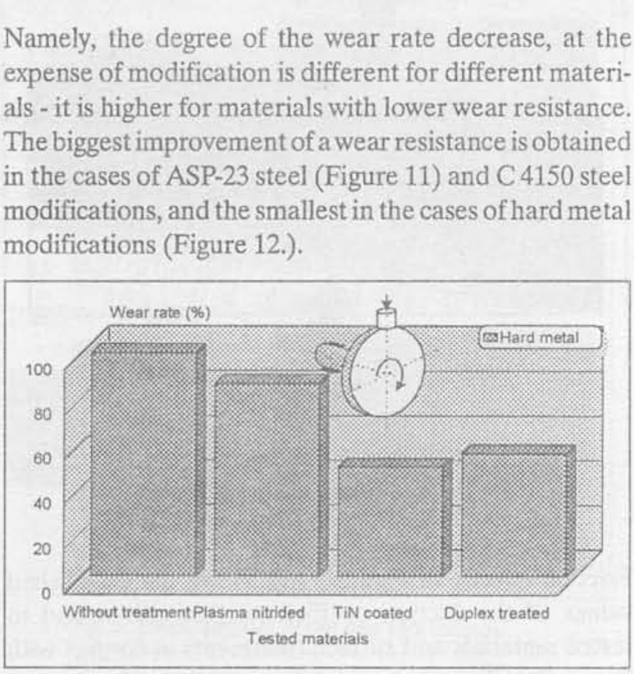

Fig. 11. Wear rate decreasing of treated hart metal

This contributes to significant elimination of differences between materials, in tribological sense, especially in the case of application of the duplex procedure. Thus, for instance, the difference in the wear degree of the PM steel and hard metal from the level of *34 %* is reduced down to *10 %*. Also, it is clear that selected surface treatments have not the same tribological improvement degree for the tested materials. It is illustrated in Table 4, with the ranking of materials depending on surface treatments antiwear effects.

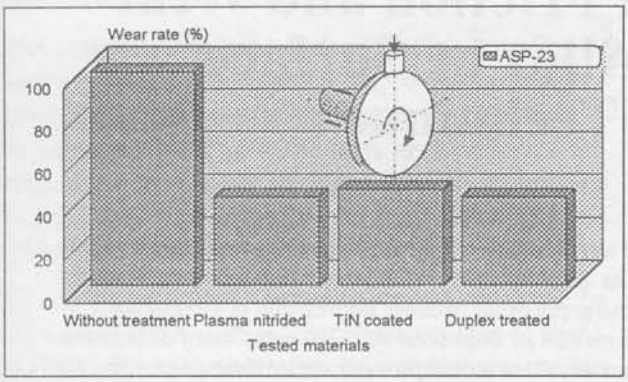

Fig. 12. Wear rate decreasing of treated ASP-23 steel

Table 4. Surface treatment influences on increasing of antiwear effects

	Antiwear effects increasing ⟶			
Nitriding	Hard metal	C 7680	C 4150	ASP-23
TiN coating	C 4150	Hard metal	ASP-23	C 7680
Duplex treatment	Hard metal	C 4150	C 7680	ASP-23

Generally, considering all the tested materials, it can be concluded that the best tribological effects are connected with duplex surface treatments. On the other hand, it is very important to underline the especially positive effects of ion nitriding of C 4150 and ASP-23 samples that are not far behind the effects of the much more expensive and complex duplex procedure.

CONCLUSION

Established tribological characteristics of all the tested tool materials are significantly improved by the contact surface modification by the ion nitriding, PVD of TiN coating and the duplex diffusion-coating deposition treatment. The largest effects, from the aspect of lowering the friction coefficient (*20 %*) correspond to the TiN coating, and from the aspect of lowering the wear rate (*52 %*) to the duplex treatment.

Application of the surface treatments (especially the duplex treatment) contributes to large reduction of differences of the tribological behavior parameters of base materials. In that way, at the expense of modification the equalizing of the tribological quality occurs of tools made of materials of very different tribological properties and prices. It gives a chance to replace expensive tool materials with the less expensive ones.

The obtained results indicate that the improving of tribomechanical system in metal powder pressing process must be based on optimal tool material and surface treatment combination selection.

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RESEARCH

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Decreasing of Friction and Wear of Ti-6Al-4V Alloy by Surface Modification Technique

Titanium alloys, such as Ti-6Al-4V, have high strength to weight ratio, as well as possessing excellent corrosion resistance, and high temperature mechanical properties, as demonstrated in aerospace applications. They are not used as widely as they could be despite their weight saving potential because they exhibit poor resistance to sliding wear. The wear resistance of titanium and its alloys can be improved by the formation of TiN in the alloy by the use of surface modification techniques. However, these techniques are either time consuming or higly specialised and hence other materials are used instead of titanium alloys for certain applications.

In this study, a new process that produces titanium nitrides in the Ti-6Al-4V alloy, quickly and simplistically, by the use of an electric arc to melt the alloy locally in the presence of nitrogen, with the objective of improving its tribological performance was investigated.

Two different treatments were performed on the titanium alloy. One involved melting the surface of the alloy with tungsten inert gas welder, in a shielding atmosphere of pure nitrogen. The other involved use of a shield of argon (80% by volume) and nitrogen(20% by volume) mixture.

The surface treatment under the shield of pure nitrogen resulted in the formation of high quantity of the TiN in the resolidified region. This compound, by its presence, increased the surface hardness to over 1000 VHN compared to 360 VHN for the untreated titanium alloy. The increased hardness value resulted in far superior wear resistance properties compared to the untreated titanium alloy.

The surface treatment under the shield of argon/nitrogen mixture also resulted in the formation of TiN, but in a smaller quantity in comparison with specimens treated in the presence of pure nitrogen. However, the wear resistant properties were once again an improvement over the untreated alloy, but were slightly inferior to the results obtained from specimens produced under a pure nitrogen gas shield.

On ground of the above stated, the application of this technique would result in more engineering applications being performed with less component replacement costs, less maintenance, and greater efficiency.

Keywords: Surface modification, shielding atmosphere, tungsten metal arc, micro-hardness, friction, wear.

1. INTRODUCTION

Titanium alloys, such as Ti-6Al-4V, are extensively used in the aerospace industry, where their high strength to weight ratio is of prime consideration. These alloys also have excellent corrosion resistance and high temperature mechanical properties. However, they suffer from poor surface wear resistance, which limits further applications in tribological systems [1].

A number of surface modification techniques have been used to improve wear properties of titanium alloys by modifying either the surface composition or microstructure.

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Conventional nitriding techniques, such as ion-nitriding, gaseous nitriding, salt-bath nitriding , and others, utilize the formation of TiN in the surface layer as a result of nitrogen diffusion [2]. The thickness of the layer is a function of the process temperature, time, and nitrogen partial pressure. However, these processes have disadvantage of requiring high temperatures and extensive hours of processing.

Ion implatation has recently emerged as a new technique to improve surface hardness of titanium alloys.When ion species such as nitrogen, carbon, and boron are implated into the surface layer of titanium alloys, they form hard titanium-base metallic compounds such as TiN, TiC, and TiB, giving a hardened surface layer of only a few micrometers. However, disadvantages include long processing time and the technique is limited by the size of specimens that can be treated [2].

Wear resistance of titanium can also be improved through chemical/physical vapour phase coating processes, but research has shown that the success of the hard surface coat is limited by the adhesive strength of the coat to the titanium substrate [3].

Laser processes involving surface melting, nitriding or alloying have been successful in improving surface properties, but are an expensive alternative [4]. Research has also identified cracking in the laser nitrided layers, although crack free layers were reported to be possible when the volume fraction of the hard phase TiN was kept low and by reducing the hard layer thickness [4] .

This study investigates a versatile process which can be used to modify the surface wear characteristics of titanium alloys. A tungsten metal arc heat source was used to provide surface melting of Ti-6Al-4V alloy in a shielding atmosphere of pure nitrogen, and nitrogen/argon mixture. The changes in the surface wear properties were assesed using metallographic, X-ray diffraction, microhardness and dry sliding wear testing techniques.

2. EXPERIMENT

The commercial alloy Ti-6Al-4V was cut into rectangular plates (*50 mm x 20 mm x 10 mm*) and the surface prepared to give a flat, polished finish followed by degreasing treatment in acetone before surface melting. A *3 mm* diameter tungsten electrode was used to create a metal arc betwen the tip of the electrode and the titanium alloy surface. This was achieved by holding the electrode stationary and at an angle of about 45^0 to the titanium alloy surface. A metal arc was produced by adjusting the distance betwen the tip of the electrode and alloy surface, and careful control of parameters such as current and voltage supply to the electrode. Shielding gases were channelled through the electrode gun, and flow regulators used to control the flow rate to give either a pure nitrogen shielding gas or a mixture of argon (80% by volume) and nitrogen (*20%* by volume).

Characterization of the resolidified region was undertaken using light microscopy and X-ray diffraction performed to identify the formation of nitride phases. Changes in the surface wear properties were investigated using a Leitz micro-hardness tester with the indentation load set at *500 g*. The wear resistance of modified surfaces was assessed using a reciprocating diamond pin on plate test carried out in air and changes in properties compared with the untreated (reference alloy). The wear test was performed using a constant sliding distance of 72 mm with a range of applied loads *(2.0 kg, 0.8 kg, 0.4 kg, 0.2 kg)*. The effect of surface treatments on wear behaviour were monitored by measuring changes in the wear rate, frictional force and the coefficient of friction.

3. RESULTS AND DISCUSSION

Surface melting to a depth of 2mm in a shielding atmosphere of pure nitrogen or an argon/nitrogen mixture was possible and resulted in the formation of dendrites in the resolidified surface as shown in fig.1.The X-ray diffraction analysis taken from the surfaces is given in figs.2-4. The results show that prominent peaks for TiN are present for surfaces treated with pure nitrogen and weaker intensities for surfaces treated by the argon/nitrogen mixture. It is reasonable then to say that the lower concentracion of nitrogen (only *20%* by vol.) in the argon/nitrogen mixture corresponds to a lower level of nitrides formation in the resolidified surface. The shielding gases were also successful in preventing the formation of titanium oxides in the surface melted zone, presence of wich could have been detrimental to surface wear properties.

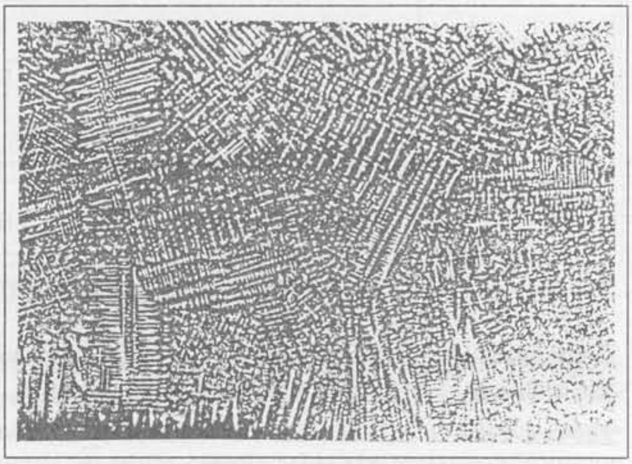

Figure 1. Melted zone produced under a gas shield of nitrogen showing dendritic structure

The formation of hard nitride phases on surface was further assessed using micro-hardness measurements which are shown in fig.5. The surface melted in the presence of nitrogen gave the highest Vickers hardness number (VHN) between *960* to *1014*, and a decrease in hardness was recorded outside the heat affected zone at a depth of *1.4 mm* from the surface. As expected, the surface treated by a mixture of argon/nitrogen gave lower VHN values of *560-607* which would correspond to a lower concentracion of nitride phases in the surface. Some increase in micro-hardness, particularly in the heat affected zone, has been attributed to the interstitial solid-solution strengthening of the α phase of the titanium alloy [5]. For comparison, the results showing variation in micro-hardness with depth for a laser nitrided surface have been superimposed in fig.5. For laser nitriding although high VHN values can be achieved at the surface, the hardness begins to drop rapidly with increasing distance from the surface. In comparison, the tungsten metal arc produces a much wider melted zone (which

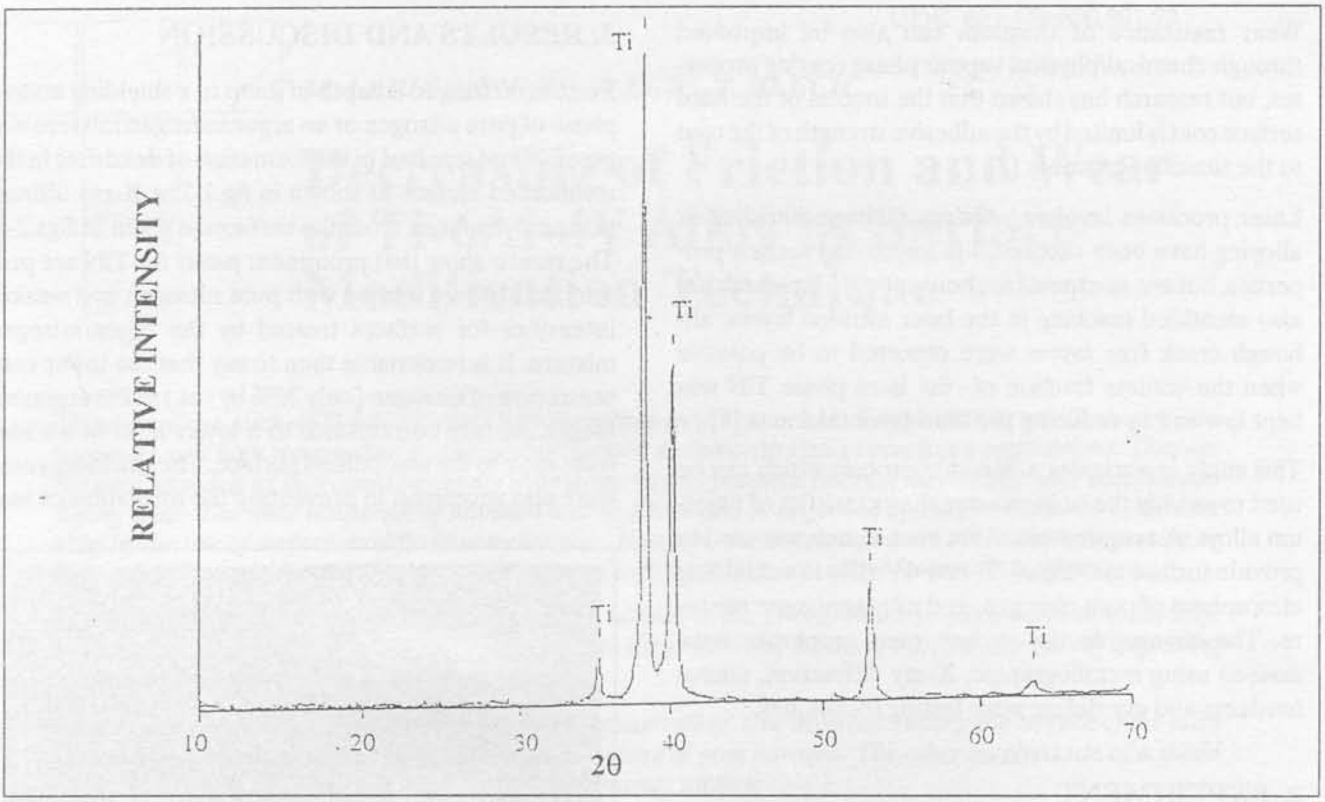

Figure 2. X-ray diffraction spectrum taken from untreated Ti-6Al-4V alloy surface

is determined by the diameter of the electrode used), but also a deeper, more uniform hardened surface region.

The resistance to surface wear was assessed by monitoring variations in the wear rate with increasing applied load, and making a direct comparison with the untreated (reference) titanium alloy. Earlier work by Lebedeva and Presnyakova [6] has shown that the wear behaviour of titanium alloys in air can be classified as severe over a wide range of sliding speeds. The results in this study also show that the untreated surface suffers high wear wich

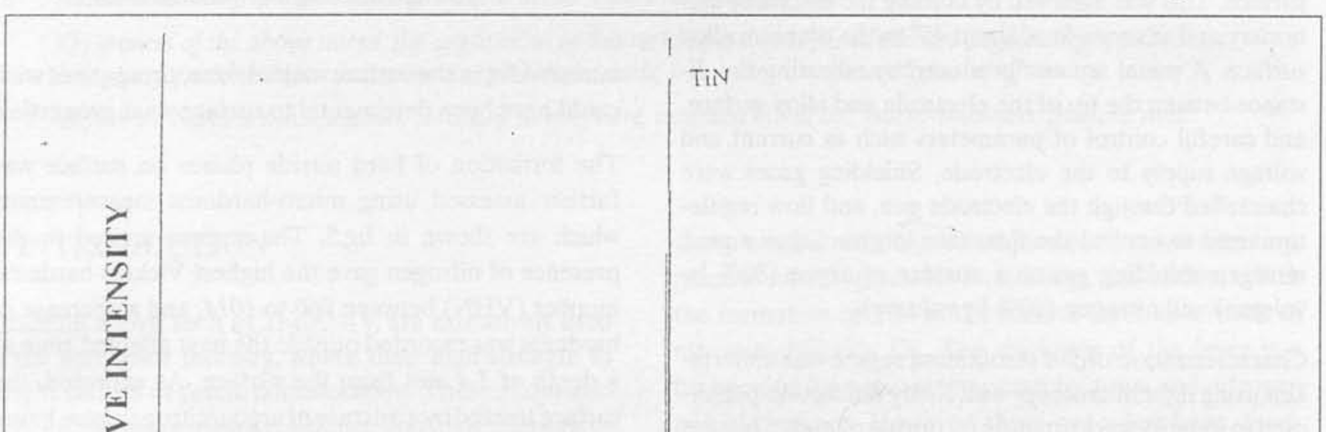

Figure 3. X-ray diffraction spectrum taken from surface melted under a nitrogen gas shield

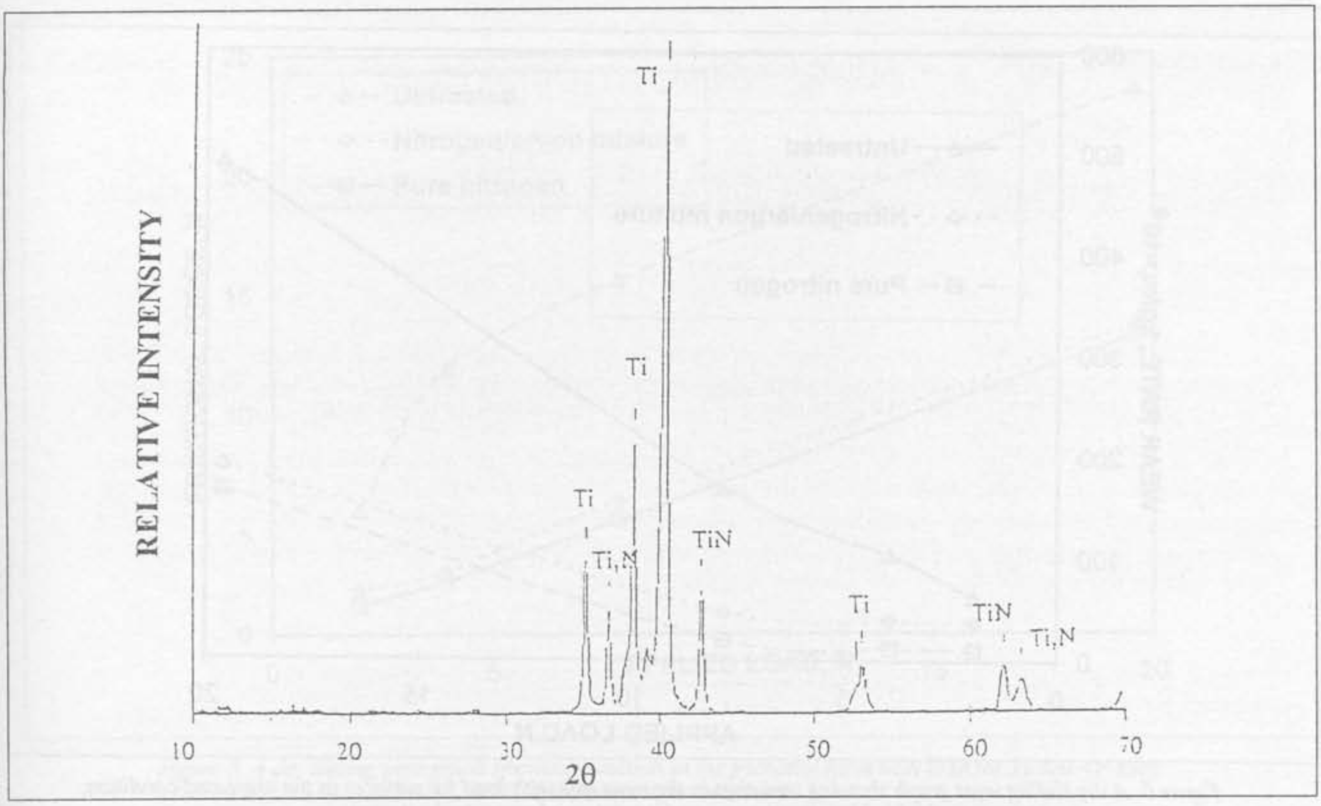

Figure 4. X-ray diffraction spectrum taken from surface melted under an argon/ nitrogen mixture

increases in severity as the applied load is increased (fig.6.).The scanning electron micrograph (SEM) in fig.9. shows adhesive wear dominated by plastic deformation involving the ploughing of materials and surface delamination.

Surfaces treated by melting in either pure nitrogen or argon/nitrogen mixture show better wear resistance than the untreated titanium surface. However, the least change in wear rate was recorded for surfaces shielded by nitrogen. These results are consistent with the microhardness values, showing that the grater concentration of nitrides formed in the surface improve the surface wear properties of the titanium alloy. Both Figs.7. and 8. show the reduction in the average frictional force and

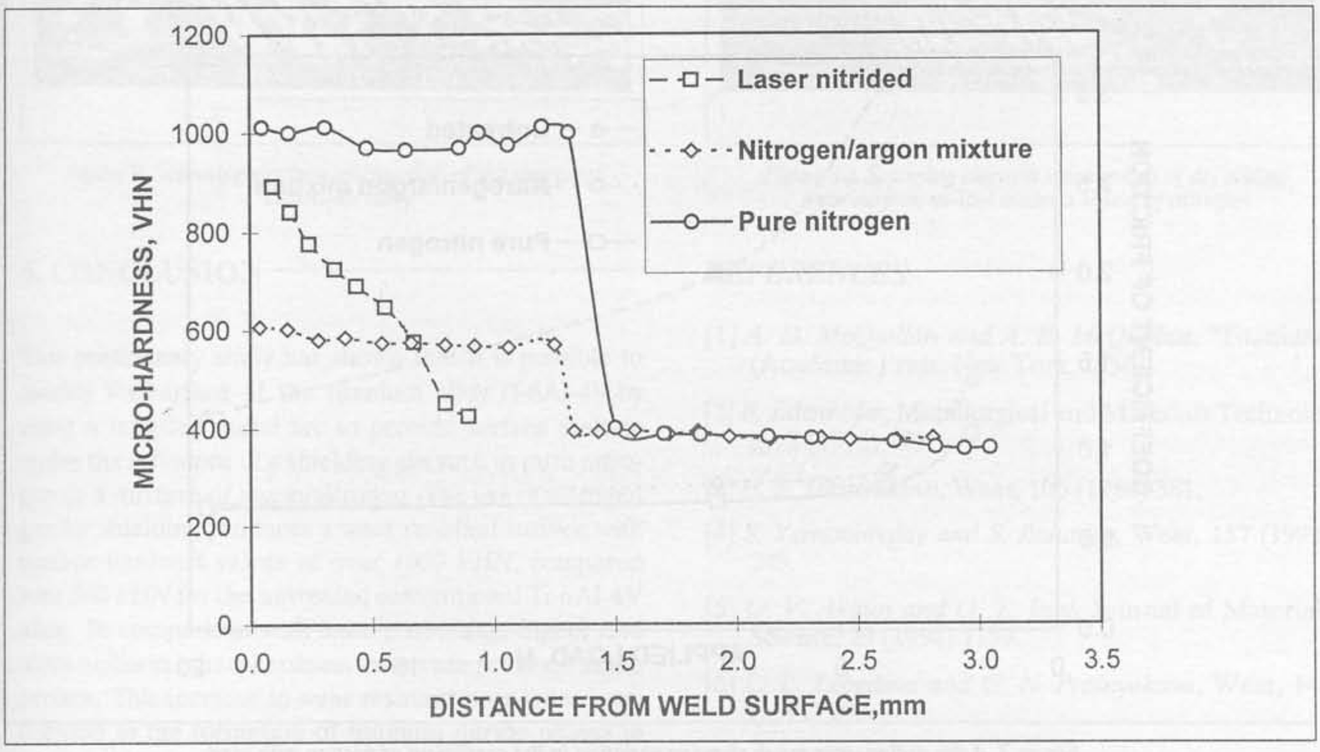

Figure 5. A comparison of micro-hardness depth profiles for surface treated Ti-6Al-4V alloy

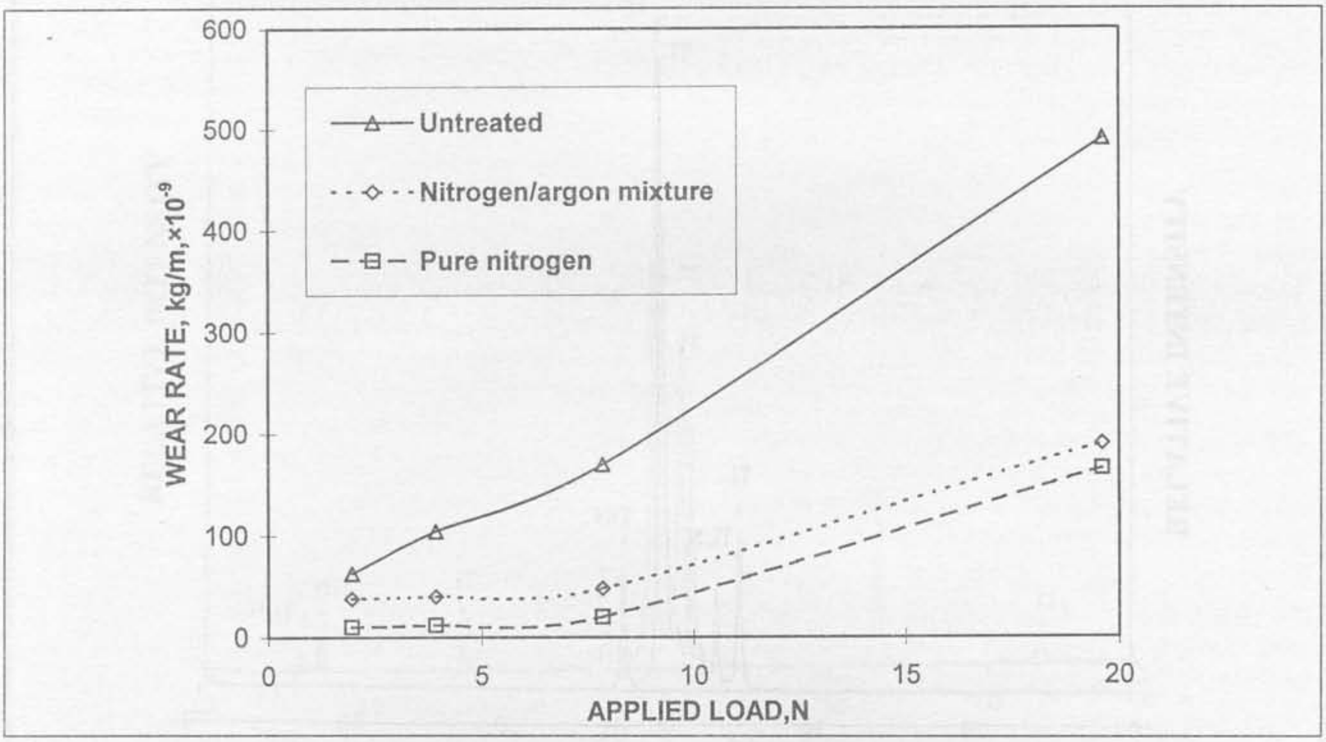

Figure 6. A dry sliding wear graph showing variation in the wear rate with load for surfaces in the untreated condition, shielded by nitrogen gas, and a mixture of argon and nitrogen

coefficient of friction for the treated surfaces and these properties corelate with the hardness changes mentioned above. The micrograph in fig.10. shows a distinct change in surface wear mechanisms. In both cases, in which the surface was shielded by nitrogen or argon/nitrogen mixture, severe adhesive wear was absent, and smooth regions were visible within the wear scar. These regions appear to have worn farr less than other surface areas. The dry sliding wear test was carried out in air, and these regions could be oxidised zones. However, because titanium oxides were not detected in the wear debris, these hardened regions could be then attributed to the formation of nitride phases.

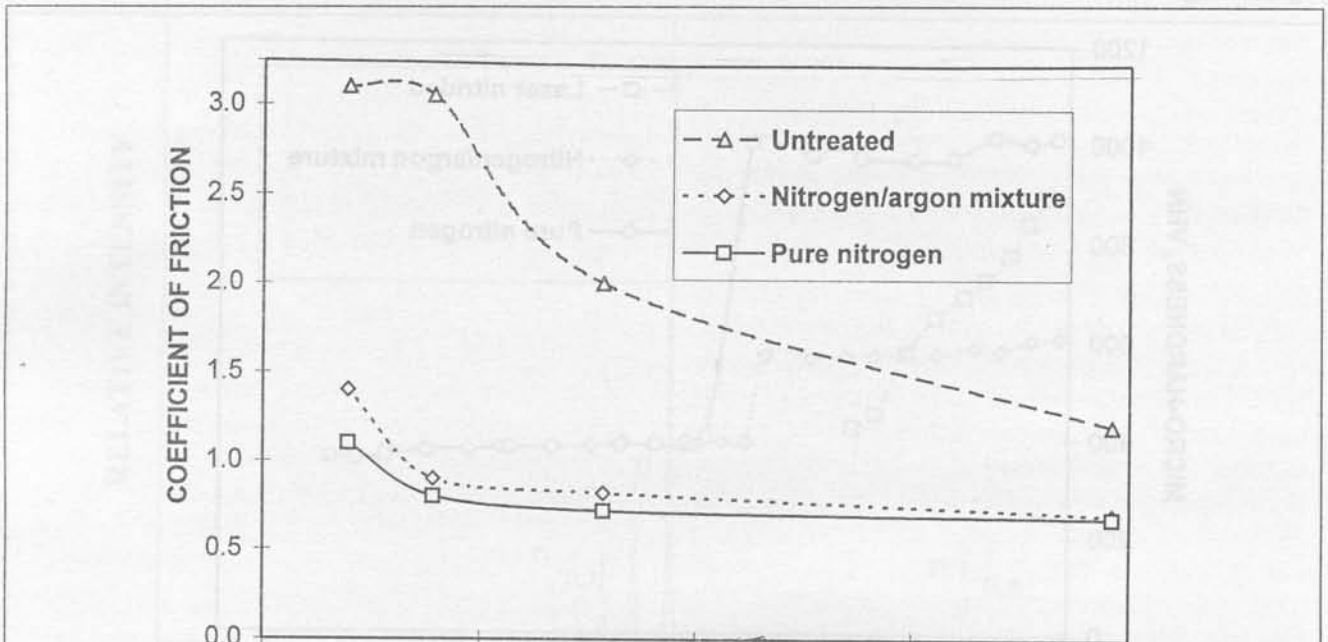

Figure 7. A dry sliding wear graph showing variation in the coefficient of friction with load for Ti-6Al-4V alloy

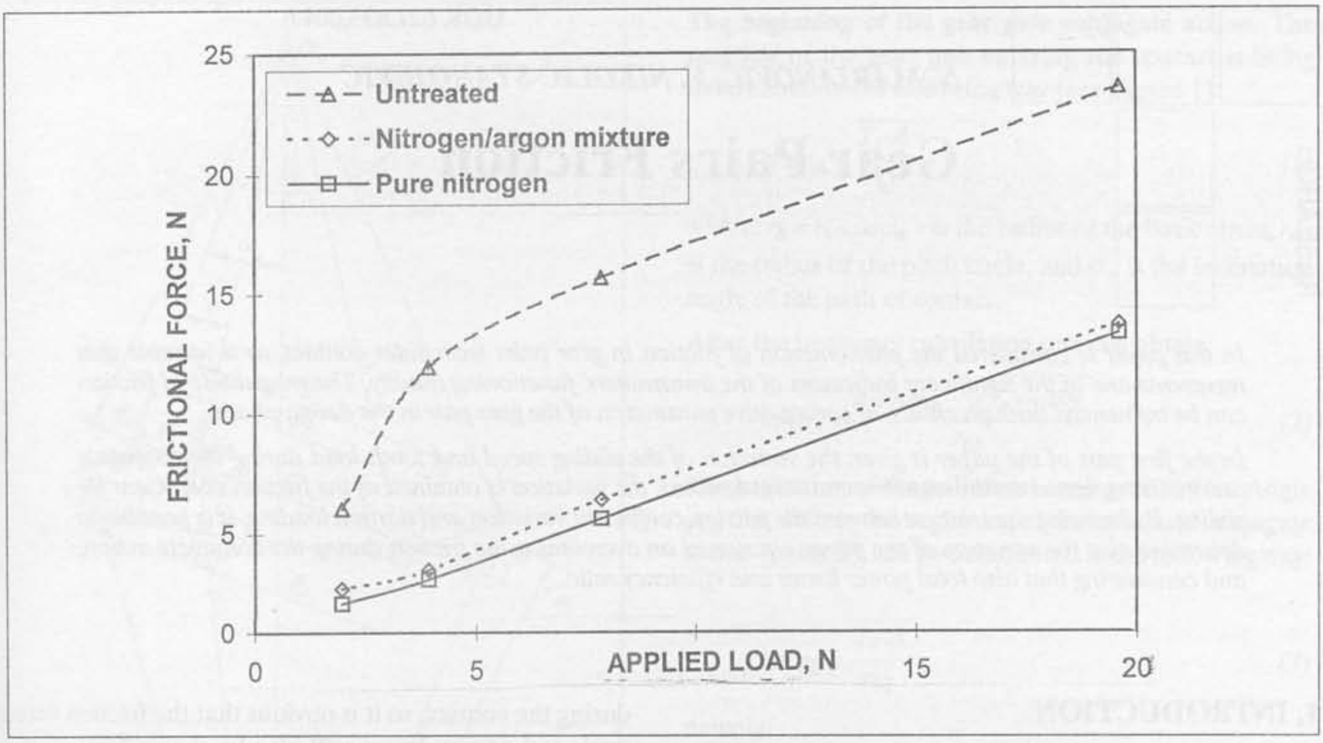

Figure 8. A dry sliding wear graph showing variation in the frictional force with load for Ti-6Al-4V alloy before and after treatment

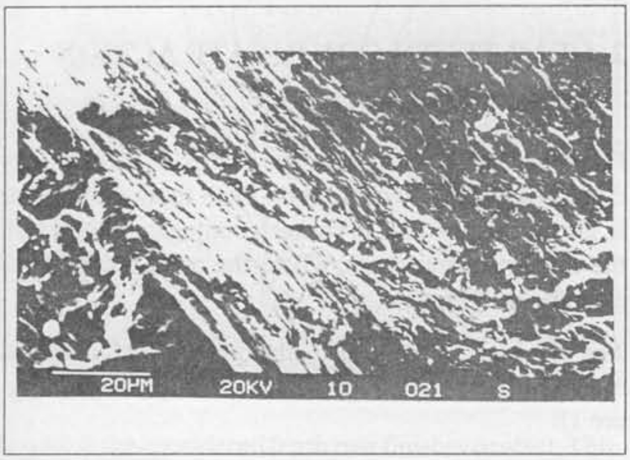

Figure 9. Scanning electron micrograph of the untreated Ti-6Al-4V alloy

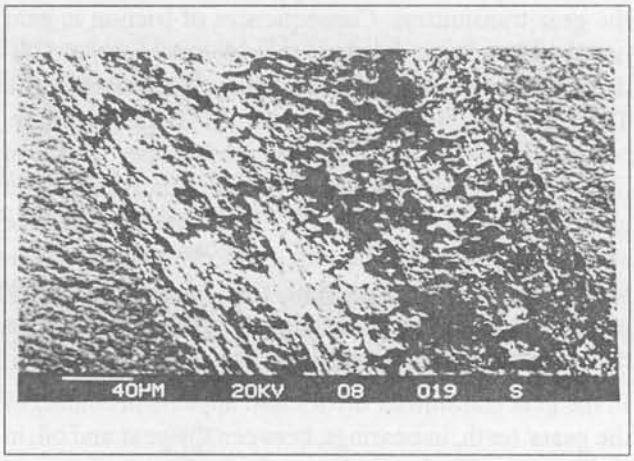

Figure 10. Scanning electron micrograph of dry sliding wear surface melted under a shield of nitrogen

4. CONCLUSION

This preliminary study has shown that it is possible to modify the surface of the titanium alloy Ti-6Al-4V by using a tungsten metal arc to provide surface melting under the influence of a shielding gas such as pure nitrogen or a mixture of argon/nitrogen. The use of nitrogen gas for shielding produces a wear resistant surface with surface hardness values of over *1000 VHN*, compared with *360 VHN* for the untreated conventional Ti-6Al-4V alloy. In comparison with laser processing, higher and more uniform micro-hardness values are achieved at the surface. This increase in wear resistant properties is attributed to the formation of titanium nitride phases in the resolidified surface microstructure.

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RESEARCH

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Gear Pairs Friction

In this paper is considered the phenomenon of friction in gear pairs teeth sides contact, as a variable that represents one of the significant indicators of the transmitters' functioning quality. The magnitude of friction can be influenced through choice of constructive parameters of the gear pair in the design phase.

In the first part of the paper is given the variation of the sliding speed and tooth load during the conjugate action. Then, based on the experimental investigations, the variation is obtained of the friction coefficient for sliding. By knowing the relation between the friction coefficient variation and normal loading, it is possible to determine also the variation of the power consumed on overcoming the friction during the conjugate action, and considering that also total power losses and efficiency ratio.

1. INTRODUCTION

Friction, as the resistance that appears during relative motion of two bodies, also accompanies the operation of the gear transmitters. Consequences of friction in gear power transmitters are the energy losses, wear and damage of transmitters' parts, heating of transmitters, ... These undesirable phenomena can, in the design phase, be reduced to the smallest extent by proper choice of influential variables. Studying of the friction process in gear transmitters is a complex problem due to large number of influential factors, their complex interrelations, and variation with time. The friction process is usually monitored by the friction coefficient, energy loss due to friction, and heating of the transmitter.

In the gear transmitter the friction appears in contact of the gears' teeth, in bearings, between the gear and oil, in sealings, etc. The largest losses due to friction appear in contact between the gear teeth, and their magnitude can be significantly influenced by the choice of the variables that define the gear pairs. Losses in other transmitters' elements can also be decreased, but constructive variables do not significantly influence their magnitude.

The gear pairs friction is usually monitored by the average values of friction [1, 2, ...]. Through that one obtains only approximate picture of this process, and it is hard to clearly notice the influence of individual transmitters' parameters on the friction magnitude.

During the gears conjugate action the relative motion occurs of the gears' sides, where the sliding and rolling speeds are variable. Gear teeth load is also variable during the contact, so it is obvious that the friction force work and energy losses will also be dependent on the instantaneous position of the point of contact.

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2. GEAR TEETH CONJUGATE ACTION REALIZATION

Power transmission by aid of the gear pairs is realized by direct contact of the gear teeth. Each tooth of one gear is, for the certain time, in contact (conjugate action) with the conjugate gear tooth, where the contact conditions are variable with time.

The unfolding of the gear teeth conjugate action can be monitored through several characteristic points (see Figure 1):

Point A represents the entering of the observed pair of teeth into the contact, and it lies on the crossing of the addendum circle of the driven gear (larger) and the path of contact.

When the contact of the observed teeth pair reaches the ***point B*** the previous teeth pair had left the contact, so the point B represents the beginning of the unilateral conjugate action.

Point C is the pitch point and in it are the projections of the both teeth speeds onto the common tangential line equal to each other, which means that in point C there is no sliding, namely only the pure rolling occurs of one over the other gear tooth sides.

At the moment when the contact reaches ***point D*** the new teeth pair enters the contact, which means that at point D the unilateral conjugate action of the observed teeth pair ends.

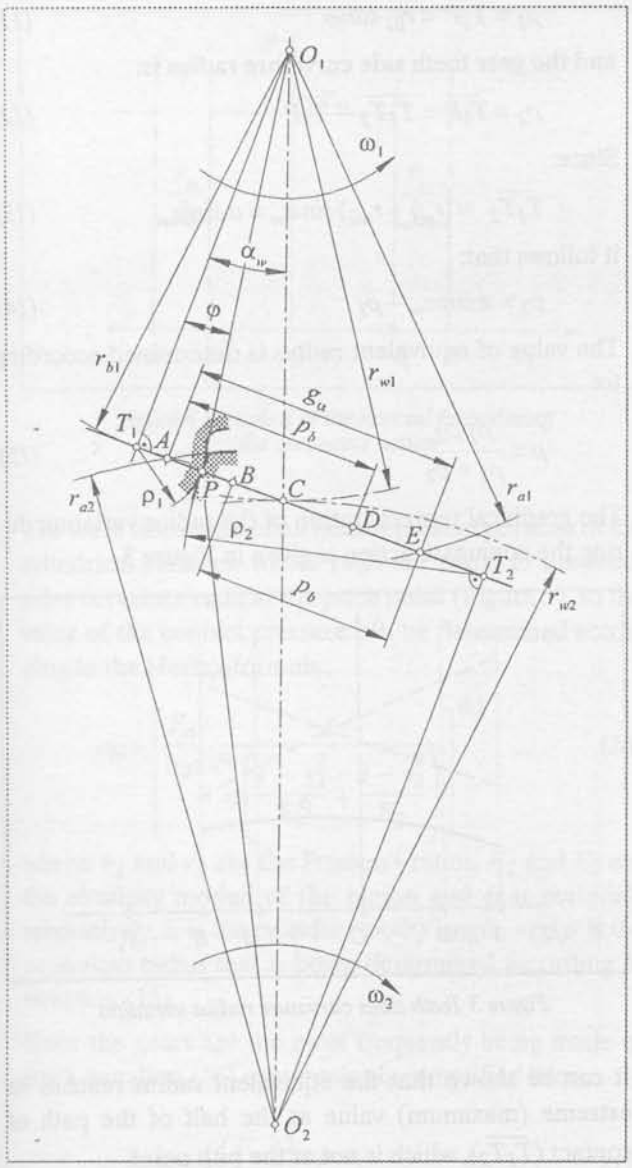


Figure 1 Kinematics of the gear pair conjugate action

At *point E* the considered teeth pair finishes contact. This point lies in crossing of the pinion addendum circle and the path of contact.

The conjugate action process is convenient to be monitored by the variation of the angle φ which defines the position of the instantaneous point of contact P with respect to the interference point T_1. If we assume that

$$\omega = \frac{d\varphi}{dt} = const.$$

it follows that $d\varphi = d\varphi{\cdot}dt$, namely $\varphi = \omega{\cdot}t$. Thus, by monitoring the variation of variables as function of angle φ, we obtain also their variation with time. The assumption $\omega = const.$ means that we are considering the stationary motion of gears, which is usual. The following consideration will refer to the spur gears, but it can also be extended to other gears.

The beginning of the gear pair conjugate action. The position of the gear pair entering the contact is being determined in the following way (see Figure 1):

$$tan\varphi_A = \frac{\overline{T_1 A}}{r_{b1}} \tag{1}$$

where: $r_b = r_w{\cdot}cos\,\alpha_w$ - is the radius of the basic circle, r_{w1} is the radius of the pitch circle, and α_w is the inclination angle of the path of contact.

After the necessary calculation one can obtain:

$$tan\varphi_A = \frac{a{\cdot}sin\alpha_w - \sqrt{r_{a2}^2 - r_{b2}^2}}{r_{b1}} \tag{2}$$

The beginning of the unilateral conjugate action. Angle φ_B, that defines the beginning of the unilateral conjugate action (point B), can be determined in the following way:

$$tan\varphi_B = \frac{\overline{T_1 B}}{r_{b1}} \tag{3}$$

namely:

$$tan\varphi_B = \frac{\sqrt{r_{a1}^2 - r_{b1}^2} - m{\cdot}\pi{\cdot}cos\,\alpha}{r_{b1}} \tag{4}$$

Pitch point. The point C is the pitch point and its position, with respect to point T_1 is defined by the angle:

$$\varphi_C = \alpha_w \tag{5}$$

From point A to point C the dedendum of the pinion is in contact with the addendum of the gear, and from point C to point E is opposite.

The end of the unilateral conjugate action. The position of the point D, that defines the end of the unilateral conjugate action, is determined by the angle that can be calculated in the following way:

$$tan\varphi_D = \frac{\overline{T_1 D}}{r_{b1}} \tag{6}$$

namely:

$$tan\varphi_B = \frac{a{\cdot}sin\varphi_w - \sqrt{r_{a1}^2 - r_{b1}^2} - m{\cdot}\pi{\cdot}cos\,\alpha}{r_{b1}} \tag{7}$$

The end of the gear pair conjugate action. The exit of the considered teeth pair from the contact is defined by the angle φ_E that can be determined based on the expression:

$$cos\varphi_E = \frac{r_{b1}}{r_{a1}} \tag{8}$$

In this way positions are defined of the conjugate action characteristic points with respect to the point T_1, what enables monitoring of the changes of characteristic variables during contact.

Teeth sides sliding speed

Sliding speed varies during the gear pair teeth conjugate action, and its value can, according to the law of conjugate action, be expressed in the following way [3]:

$$v_{kl} = \overline{CP} \cdot (\omega_1 + \omega_2) \tag{9}$$

where the point P is the instantaneous conjugate point. After necessary rearrangements, the sliding speed can be determined from:

$$v_{kl} = \omega_1 \cdot r_{w1} \cdot (\sin\alpha_w - \cos\alpha_w \cdot \tan\varphi) \cdot \left(1 + \frac{1}{u}\right) \tag{10}$$

where: $u = \dfrac{\omega_2}{\omega_1}$ - is the transmission ratio of the considerer gear pair.

Equation (1) gives the dependence of the sliding speed on angle φ, namely on time, and the graphical representation of this relation is shown in Figure 2. On the abscissa axis is, for the sake of clearer presentation, instead of angle φ, is given the path of contact with the characteristic points. The dashed lines represent the sliding speeds of the teeth pairs that are in the conjugate action before and after the considered pair.

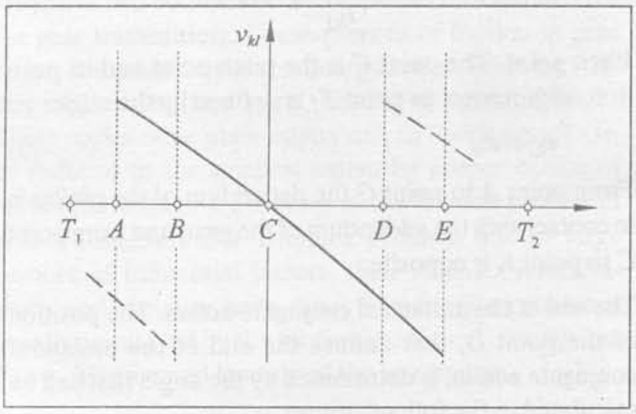


Figure 2 Sliding speed variation

If in the equation (10) the angle φ is substituted with the value $\varphi = \alpha_w$, one obtains that the sliding speed is equal to zero, what proves the statement that in pitch point (point C, $\varphi = \alpha_w$) there is no sliding.

Loading during the conjugate action

During the gear pair teeth conjugate action the point of contact moves on pinion from dedendum towards addendum, and on gear from addendum towards dedendum. This means that the curvature radii of teeth sides in the contact point change with time. The stresses (Hertz contact pressure) depend on the equivalent radius, so their change can be monitored if one knows the variation of loads and equivalent radius.

The curvature radius of the pinion tooth side in the instantaneous conjugate point P (see Figure 1) amounts to:

$$\rho_1 = \overline{T_1P} = r_{b1} \cdot \tan\varphi \tag{11}$$

and the gear teeth side curvature radius is:

$$\rho_2 = \overline{T_2P} = \overline{T_1T_2} - \overline{T_1P} \tag{12}$$

Since:

$$\overline{T_1T_2} = (r_{w1} + r_{w2}) \cdot \sin\alpha_w = a \cdot \sin\alpha_w \tag{13}$$

it follows that:

$$\rho_2 = a \cdot \sin\alpha_w - \rho_1 \tag{14}$$

The value of equivalent radius is determined according to:

$$\rho = \frac{\rho_1 \cdot \rho_2}{\rho_1 + \rho_2} \tag{15}$$

The graphical representation of the radius variation during the conjugate action is given in Figure 3.

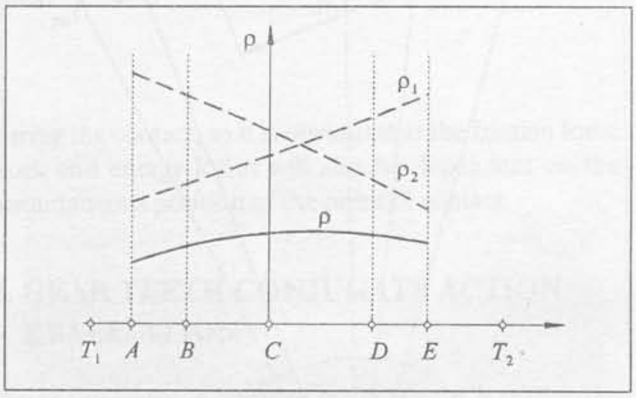


Figure 3 Teeth sides curvature radius variation

It can be shown that the equivalent radius reaches its extreme (maximum) value at the half of the path of contact ($\overline{T_1T_2}$), which is not at the pith point.

During the conjugate gear motion in conjugate action are one or two pairs of teeth alternatively, what was explained earlier. During unilateral conjugate action the total normal force F_{bn} is being transmitted by one teeth pair, while during the double conjugate action it is divided between the two teeth pairs. For analysis we shall adopt that during the double conjugate action the normal force is uniformly distributed to both teeth pairs, namely that each pair is loaded with the normal force $F_{bn}/2$. In practice, the distribution of load on several teeth pairs in conjugate action is not uniform. The largest influence on non-unifom distribution has the deviation of the tooth side profile pitch and the unequal stiffness of the teeth pairs. Non-uniform load distribution is, according to the ISO recommendations, in calculation of the gear carrying capacity, being taken into account by the factors $K_{H\alpha}$ and $K_{F\alpha}$.

In Figure 4 is presented the adopted variation of the normal force during the conjugate action. The dashed lines presents the variations of the normal forces of the preceding and following gear pairs.

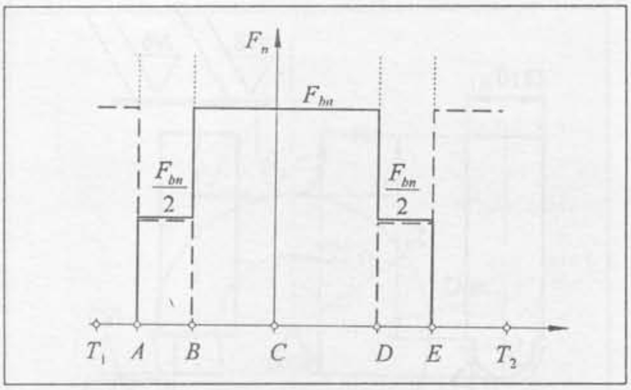


Figure 4 Variation of the normal force during the conjugate action

The teeth sides contact surfaces represent portions of the cylindrical surfaces, whose radii are equal to the teeth sides curvature radii at the pitch point (Figure 5), so the value of the contact pressure can be determined according to the Hertz's formula:

$$\sigma_H = \left[\frac{F_n}{b \cdot \rho} \cdot \frac{1}{\pi \cdot \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)} \right]^{0.5} \qquad (16)$$

where: ν_1 and ν_2 are the Poisson's ratios, E_1 and E_2 are the elasticity moduli of the pinion and gear materials respectively, b is the cylinder (tooth) length, and ρ is the equivalent radius that is being determined according to equation (15).

Since the gears are the most frequently being made of steel, equation (16) can obtain the simplified form:

$$\sigma_H = 189.8 \cdot \left(\frac{F_n}{b \cdot \rho} \right)^{\frac{1}{2}} \qquad (17)$$

It is obvious that the contact pressure on the teeth sides can change during the conjugate action due to variation of the normal force value on the side (Figure 4) and variation of the equivalent radius (Figure 3). In Figure 6 is presented the variation of the contact pressure on the teeth sides during the conjugate action, where here the dashed lines also refer to those variations for the preceding and the following tooth pair.

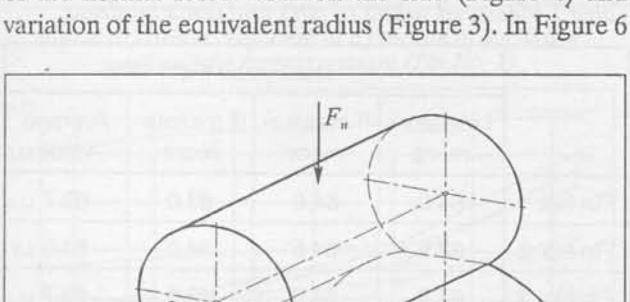


Figure 5 Contact pressure during two cylinders contact

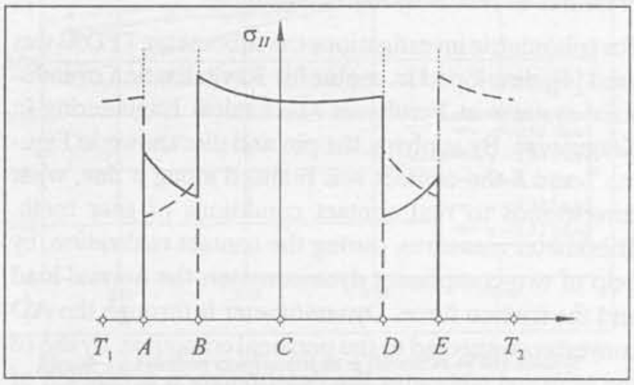


Figure 6 Variation of the contact pressure during the conjugate action

FRICTION COEFFICIENT

The friction coefficient represents the ratio of the friction force and the normal load, namely:

$$\mu = \frac{F_{tr}}{F_n} \qquad (18)$$

To define the value of the values of the coefficient at each contact point during the conjugate action of the gears' teeth sides is a very complex task, due to large number of the influential factors that are variable with time. The most important factors that influence the friction coefficient are: the sliding speed, the contact pressure, the gears' materials, technological inheritance, etc. Here shall be considered only the influences of the first two factors, since their values can be influenced by choice of the gear transmitter parameters (module, teeth number, gear width, ...) during the design process.

During the conjugate action of gear teeth sides the great change occurs of the sliding speed and the contact pressure, what was analyzed earlier. The friction coefficient, that depends on these variables, will also have variable values during the conjugate action. To establish the dependence of the friction coefficient on the sliding speed and the contact pressure, the tribometric investigations were performed.

Gears usually operate in conditions of the high pressure on the teeth sides. In design of gear pairs one of the criteria is also the teeth sides strength. This means that, if the gear is correctly designed and loaded with nominal load, the stress on sides will be close to dynamic strength on sides. Since the power transmitter's gears are usually made of high quality, thermally treated steels, with high dynamic strength of sides ($1000 \div 1500\ MPa$), this means that also the contact pressures on the teeth sides will be of the same order of magnitude. It is thus necessary to consider the case of relative motion of two cylindrical bodies at different sliding speeds and different contact pressures of relatively high magnitudes.

For tribometric investigations the tribometer TPD93 was used [4], developed in center for Revitalization of industrial systems at Faculty of Mechanical Engineering in Kragujevac. By applying the pin and disc shown in Figures 7 and 8 the contact was realized along a line, what corresponds to real contact conditions of gear teeth. Tribometer measures, during the contact realization, by help of two-component dynamometer, the normal load and the friction force. Dynamometer is through the AD converter connected to the personal computer. By use of the personal computer the determination is enabled of the friction coefficient, as well as the graphical representation of variations of all the variables during contact. Measured and calculated values are then recorded and

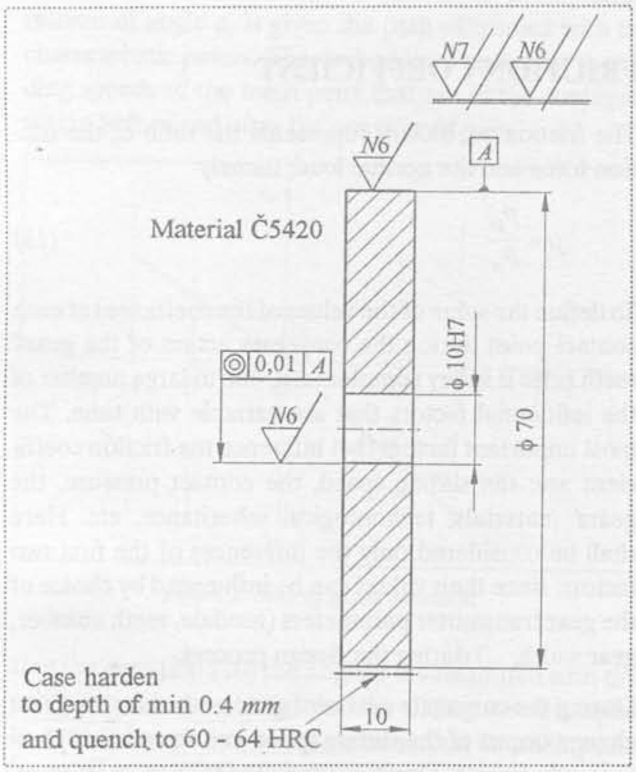
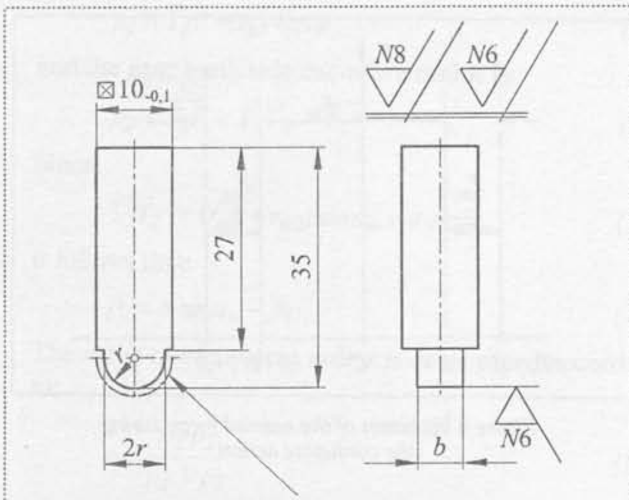


Figure 7 Disc used for investigations

kept in the form of data files (ASCII), what enables further processing.

The pin and disc material was the cemented and thermally treated steel 5420. The hardness monitoring gave values for hardness presented in Table 1. Measured average value of the roughness height, at three referent lengths for pins was $R_z \approx 1.4\ \mu m$, and for disc $R_z \approx 0.92\ \mu m$. Lubrication was done with oil for gear trains with viscosity code SAE 90 (GALAX HIPOL B).

In Figure 9 is presented the schematic representation of the tribometric investigations.



Case harden to depth of min 0.4 *mm* and quench to 60 - 64 HRC

Number	$r \pm 0{,}05$ *mm*	$b \pm 0{,}05$ *mm*	Pieces
1	∞	9	1
2	3	4	1
3	4	4	1
4	5	4	1
5	4	3	1
6	5	3	1
Material: Č 5420			

Figure 8 Pins used for investigations

During tests five sliding speeds were varied: $v_{kl} = 0.56;\ 1.09;\ 1.71;\ 2.18$ and $2.72\ m/s$, and five different normal loads of approximately *100* to *450 N*.

Table 1 Measured values of pin's and disc's hardness

	Hardness HRC			
	I measurment	II measurment	III measurment	Average value
Pin No. 1	64.0	64.0	63.0	63.7
Pin No. 2	63.5	64.5	64.0	64.0
Pin No. 3	64.0	64.0	63.0	63.7
Pin No. 4	63.0	63.0	62.5	62.8
Pin No. 5	64.0	64.0	64.0	64.0
Pin No. 6	64.0	64.0	64.0	64.0
Disc	63.0	64.0	63.0	62.3

The shape and dimensions of pins and disc enabled realization of high values of contact pressures.

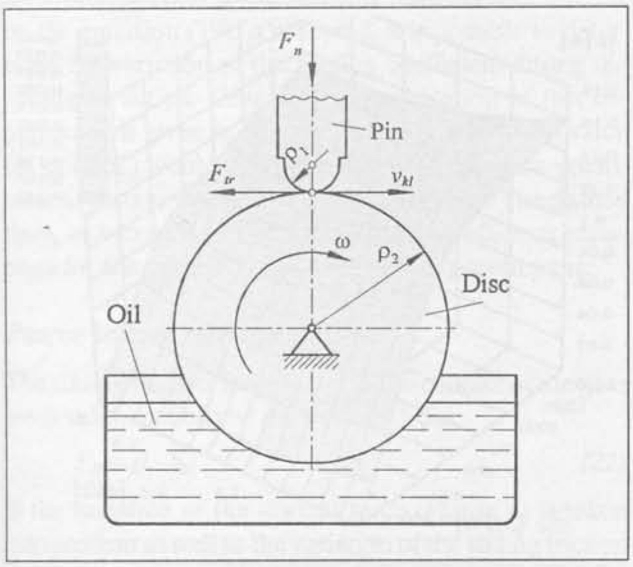


Figure 9 Schematic representation of the tribometric investigations

In Figure 10 to 15 are shown the graphical representations of obtained results in the form of diagrams, and in Figure 16 is given the tri-dimensional view of the friction coefficient dependence on the sliding speed and the contact pressure.

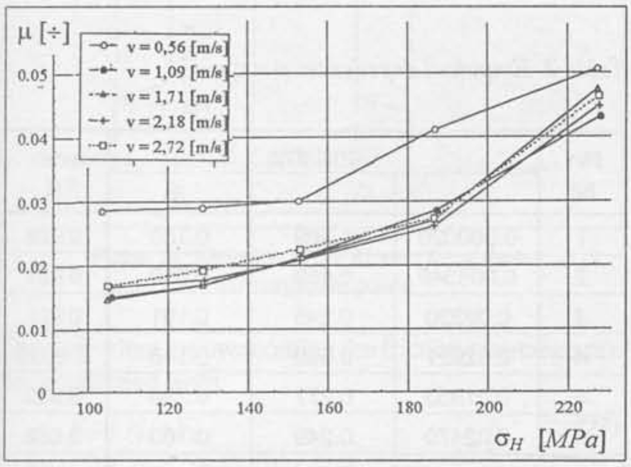


Figure 10 Friction coefficient as a function of the sliding speed and the contact pressure (Pin No. 1)

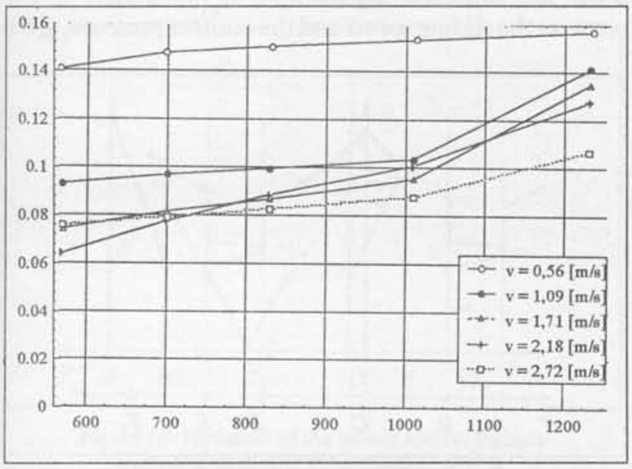


Figure 11 Friction coefficient as a function of the sliding speed and the contact pressure (Pin No. 2)

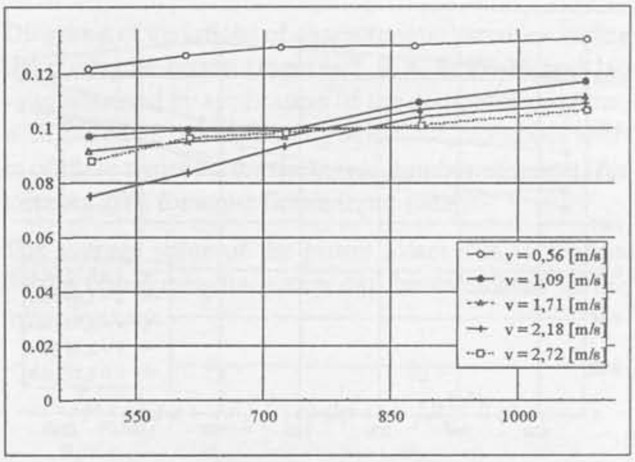


Figure 12 Friction coefficient as a function of the sliding speed and the contact pressure (Pin No. 3)

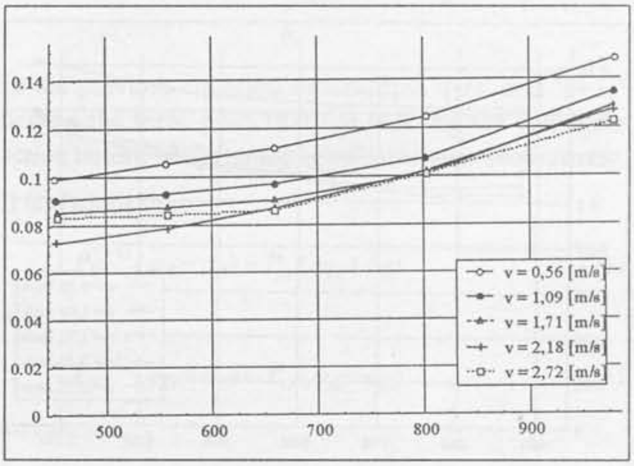


Figure 13 Friction coefficient as a function of the sliding speed and the contact pressure (Pin No. 4)

It is obvious that the friction coefficient increases with the increase of the contact pressure, and the with the increase of the sliding speed the friction coefficient ate first decreases rapidly, and then at higher speeds it depends only slightly on the speed variation. This can be explained by the change of the contact and lubrication conditions with change of the sliding speed and contact pressure.

Based on data obtained by measurement the dependencies were predicted of the friction coefficient on the sliding speed and the contact pressure. The three forms of the empirical relations were applied:

$$\mu = c \cdot \frac{\sigma_H^{\alpha}}{v_{kl}^{\beta}} \tag{19}$$

$$\mu = c_0 + c_1 \cdot \frac{\sigma_H^{\alpha}}{v_{kl}^{\beta}} \tag{20}$$

and:

$$\mu = c_0 + c_1 \cdot v_{kl} + c_2 \cdot \sigma H + c_{11} \cdot v_{kl}^2 + \\ + c_{12} \cdot v_{kl} \cdot \sigma_H + c_{22} \cdot \sigma_H^2 \tag{21}$$

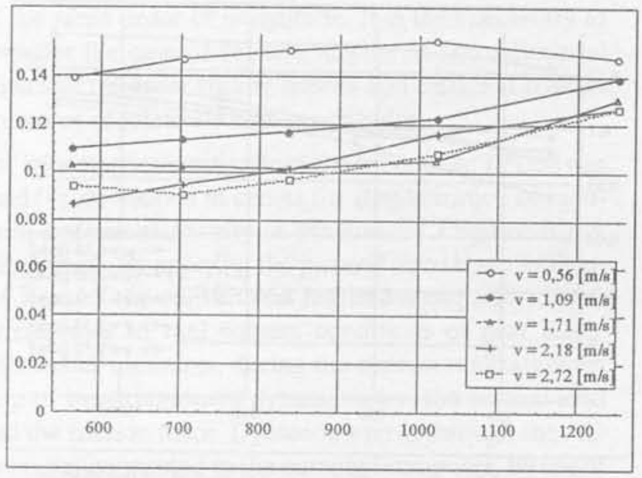


Figure 14 Friction coefficient as a function of the sliding speed and the contact pressure (Pin No. 5)

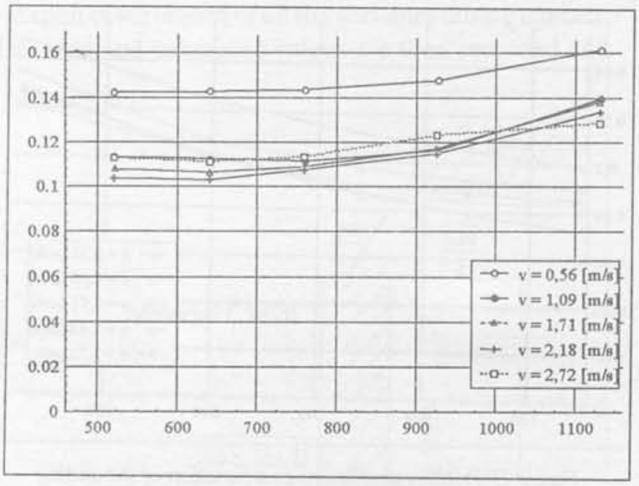


Figure 15 Friction coefficient as a function of the sliding speed and the contact pressure (Pin No. 6)

The constants in these expressions were determined by the least squares method. When al the data are taken into account, the indices of the curvilinear correlation are obtained that differ only slightly, where they all have values greater than *0.9 (0.937, 0.955, 0.954*, respectively). Considering these values one can conclude that each of the proposed distributions gives good agreement with the experimental data.

In all the cases the high value is obtained for the index of the curvilinear correlation (*R>0.9*), what points to the good agreement of experimental data with the proposed empirical distribution. The highest value of index *R* is obtained for the second order polynomial distribution (21), due to large number of constants and ability of this polynomial to adapt itself to experimental data.

For the further work the distribution was adopted given by the equation (19) due to its simplicity and clear physical meaning. The constants of this distribution are given in Table 2. By analysis of this distribution it is obvious that the friction coefficient increases with increase of the contact pressure, and that it decrease with the increase of the sliding speed up to a certain value. The values of

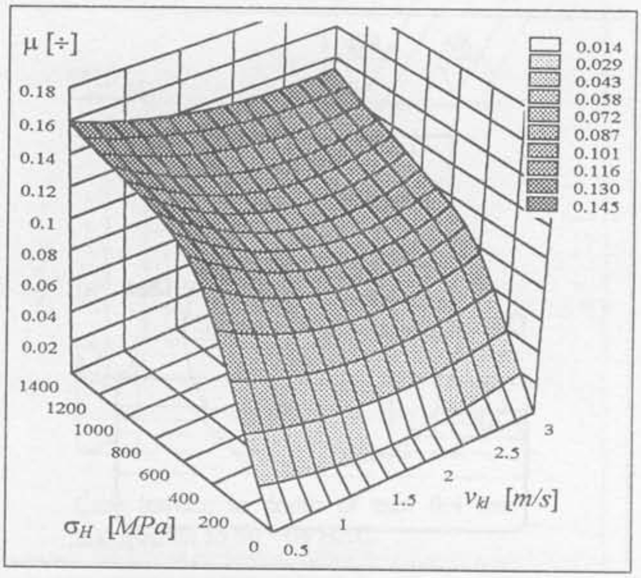


Figure 16 Friction coefficient as a function of the sliding speed and the contact pressure

coefficients c, α and β from Table 2 show that the influence of the sliding speed and the contact pressure on the friction coefficient is more distinguished at small speeds, what agrees with conclusions drawn based on experimental data.

Table 2 Empirical depedence: $\mu = c \cdot \frac{\sigma_H^{\alpha}}{v_{kl}^{\beta}}$

Pin N°	Constants c	α	β	Index R
1	0.000028	1.369	0.180	0.924
2	0.005349	0.458	0.350	0.921
3	0.02220	0.245	0.191	0.921
4	0.02921	0.589	0.145	0.9505
5	0.01953	0.277	0.236	0.930
6	0.02470	0.249	0.160	0.862
Σ	0.00234	0.584	0.231	0.937

Since the empirical dependence of the friction coefficient on the sliding speed and the contact pressure, given

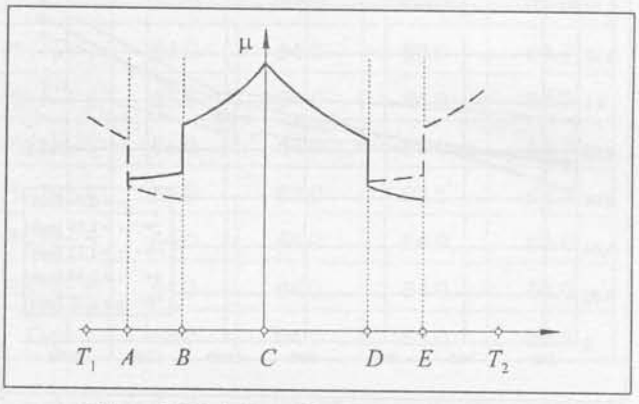


Figure 17 Variation of the friction coefficient during the conjugate action

by the equation (19), is accepted, it is possible to determine the variation of the friction coefficient during the conjugate action. Graphical representation of this dependence is given in Figure 17, and its maximum value (at point C) would correspond to static friction coefficient at certain value of the contact pressure. The dashed lines, as before, represent the friction coefficient variations for the preceding and the following teeth pairs.

Power losses due to friction

The sliding friction force arises in the contact of the gear teeth sides, and can be determined from:

$$F_{tr} = \mu \cdot F_n \tag{22}$$

If the variation of the normal force (Figure 4) is taken into account as well as the variation of the sliding friction coefficient (Figure 17) during the conjugate action, the dependence of the friction force during the conjugate action can be obtained (Figure 18).

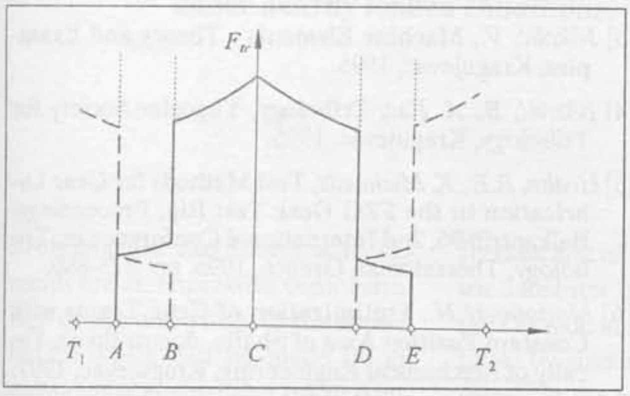


Figure 18. Variation of the friction force during the conjugate action

The power lost for overcoming the friction resistance can be determined from:

$$P_{tr} = v_{kl} \cdot F_{tr} \tag{23}$$

and its variation during the conjugate action is shown in Figure 19.

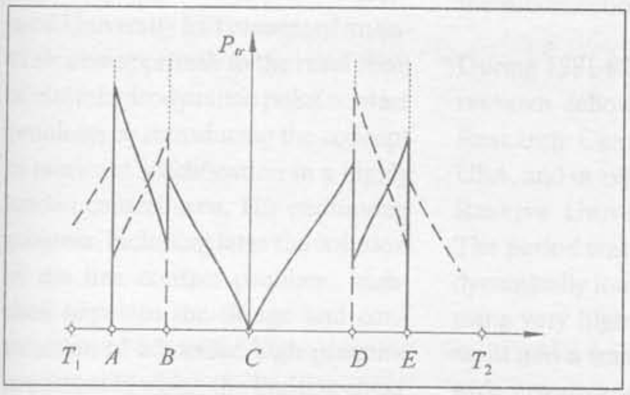


Figure 18. Variation of the power lost on friction overcoming during the conjugate action

Diagrams of variations of characteristic variables duting the conjugate action (Figures 2, 3, 4, 6, 17, 18 and 19) were obtained by application of the computer program, wich, based on corresponding equations, calculates values of these variables for the cerain number of points (for instance *100*) for some fictive input data.

The average value of the power losses due to friction during one conjugate action can be calculated in the following way:

$$\bar{P}_{tr} = \frac{\int_{\varphi_A}^{\varphi_E} P_{tr}(\varphi) \cdot d\varphi + \int_{\varphi_A}^{\varphi_B} P_{tr}^{(-1)}(\varphi) \cdot d\varphi + + \int_{\varphi_D}^{\varphi_E} P_{tr}^{(+1)}(\varphi) \cdot d\varphi}{\int_{\varphi_A}^{\varphi_E} d\varphi} \tag{24}$$

In the previous equation superscripts "*(-1)*" and "*(+1)*" denote the teeth pairs that are entering the conjugate action before and after the considered pair, respectively. It is obvious that:

$$P_{tr}^{(-1)} \cdot (\varphi_A \div \varphi_B) = P_{tr} \cdot (\varphi_D \div \varphi_E) \tag{25}$$

and:

$$P_{tr}^{(+1)} \cdot (\varphi_D \div \varphi_E) = P_{tr} \cdot (\varphi_A \div \varphi_B) \tag{26}$$

and since:

$$\int_{\varphi_A}^{\varphi_E} d\varphi = \varphi_E - \varphi_A \tag{27}$$

equation (25) gets the following form:

$$\bar{P}_{tr} = \frac{\int_{\varphi_B}^{\varphi_D} P_{tr}(\varphi) \cdot d\varphi + 2 \cdot \left(\int_{\varphi_A}^{\varphi_E} P_{tr}(\varphi) \cdot d\varphi + \int_{\varphi_D}^{\varphi_E} P_{tr}(\varphi) \cdot d\varphi \right)}{\int_{\varphi_A}^{\varphi_E} d\varphi} \tag{28}$$

The efficiency ratio of the gear pair, with respect to the friction losses due to sliding, can be determined based on the equation:

$$\eta = \frac{P - \bar{P}_{tr}}{P} \cdot 100\ [\%] \tag{29}$$

where: P - is the nominal transmitter's power, and $\bar{P}_{sr}$ is the power spent on overcomming the friction resistance, and wich is determined based on the equation (28)

In this way determined the efficiency ratios have values of about *96 - 99 %*, what corresponds to real values.

CONCLUSION

In this paper was investigated the influence of the sliding speed and the contact pressure on the teeth sides on the efficiency ratio (power losses) of the gear pair.

With decrease of the contact pressure the friction coefficient decreases also, and with the increase of the sliding speed over certain limit, it increases only slightly (see equation (19)). This means that if the outside nominal load of the gear pair remains unchanged, the power losses due to friction will decrease with increase of the teeth dimensions. This conclusion is logical, since here were considered only the power losses due to sliding friction on teeth sides. The maximum value of the efficiency ratio will be obtained for the combination of the constructive variables that gives as small as possible contact pressures, at sufficiently high sliding speeds.

For investigation of the gear pairs friction can be applied also other types of tribometers (for instance: disc/disc, four balls, ...), and for investigation of lubrication is frequently used the specific equipment and methods like: IEA in Great Britain, Ryder in USA, or FZG in Germany and other Western European countries [5].

The more realistic picture about the teeth pairs onjugate action can be obtained by application of the disc/disc type tribometers. Such a construction must provide for significantly higher normal loading forces, due to larger disc radii, and with that also the higher driving engine power. Besides that, the controlled motion must be ensured of both discs (where the percent of sliding is set in advance), etc. This kind of tribometers is used for testing the quality of the lubricating oils.

In multi-gear transmitters the total efficiency ratio based on the losses due to friction of the gear pairs, can be determined as a product of efficiency ratios of each individual gear pair.

In here conducted investigations of the friction coefficient the rolling friction was neglected, since it is negligible with respect to the sliding friction, and significantly complicates the investigation procedure.

The obtained results show good agreement with the real values with application of the relatively simple investigation procedure on the available equipment.

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