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Experimental study and modelling of mixed particulate lubrication with MoS_2 powder solid lubricant

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ABSTRACT

Purpose: The purpose of the paper is experimental study and modelling of mixed particulate lubrication with MoS_2 powder solid lubricant.
Design/methodology/approach: In the present investigation, ball-on-disc experiments were carried out to determine the lubrication performance of MoS_2 solid lubricant powder that could be used for hard PVD coatings applied for forging and stamping tools.
Findings: The proposed solid lubricant nano- and submicroparticles mixture demonstrates excellent potential for use in mixed lubrication regimes The quasi-hydrodynamic behaviour of solid lubricant and wear debris particles results in low friction coefficients of hard coating – steel ball friction pairs.
Research limitations/implications: The solid lubricant particle exfoliation and formation of tribofilms on micro-asperities allow to achieve the boundary lubrication effects which is found to more preferable for steel contacts rather than for hard coatings.
Originality/value: The model of mixed lubrication based on non Newtonian behaviour of powder solid lubricant was validated based on the experimental results. Results of calculation of Stribeck curves demonstrate the potential of modelling of friction process by sharing boundary and quasi-hydrodynamic processes.
Keywords: Materials design; Wear resistance; MoS_2 solid lubricant; PVD coatings; Microchannels

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1. Introduction

Solid lubricants such as MoS_2, WS_2 and other layered compounds have been widely used and the preference of one over the other has been the controversial subject of numerous papers [1]. To elucidate suitability of as powder lubricants for various sliding components, tools and bearings a deeper insight of lubrication mechanisms is needed. At first glance, nanoparticle MoS_2 powders chosen for lubrication may appear to be attractive solid lubricants because they exhibit low values of friction coefficients [2, 3]. For example, experimental results from a conventional pin-on disk apparatus at low speeds have suggested tungsten disulfide, WS_2, as superior lubricant to molybdenum disulfide, MoS_2, because it maintains a lower coefficient of friction for a higher temperature capability [2]. Irrespective of the value of coefficient of friction, high or low, it is essential that a lubricant possess the ability to diminish surface wear and to provide a high durability of the sliding counterparts.

Dry particulate materials have been proposed as viable candidates for lubrication in extreme environments (i.e., temperature and/or loads), where conventional lubricants cannot perform [1]. At the high contact stresses and temperatures greater than 500°C, conventional liquid lubricants are unable to sustain loads, hence, the advent of solid/particulate lubrication [2]. Nanoscale solid lubrication has also come into fruition, as nanopowder lubricants have recently demonstrated excellent lubrication capabilities at extreme temperatures [3-6]. Additionally, there are huge technological gains that can be achieved due to understanding the behaviour of dry particulates in sliding contacts under load in forging and stamping tools, high temperature bearings and other friction units.

Authors [1] propose two forms of dry particulate lubrication, powder and granular. Powder lubricants are classified as dry, “cohesive,” soft particles that radically deform under load and accommodate surface velocity differences mostly by adhering to surfaces and shearing in the bulk medium, similar to hydrodynamic fluids. Granular lubricants are classified as dry, “cohesionless,” hard spherical particles that accommodate surface velocity differences through sliding and rolling at low shear rates, and largely through collisions at high shear rates. Both powder and granular lubrication mechanisms have demonstrated that the particulates in the sliding contact can enhance lubrication and lower friction below boundary lubrication levels. At low speeds (i.e., low nominal shear rates), powder and granular lubricants may appear to take on similar velocity accommodation behaviour. One distinguishing phenomenological factor is the powder lubricant behaviour at the boundary, where powders usually adhere and coat surfaces (boundary lubrication), and granules slip, roll and/or collide with surfaces (quasi-hydrodynamic lubrication).

Based on theoretical work and on experiments conducted by H. Heshmat [7], a mechanism of particulate flow that possesses some of the features of hydrodynamic lubrication was conceived. The basic elements of the tribological system of an interface are shown to be tribomaterials, the intermediate film, wear debris, powder, and boundary layers (Fig. 1). The relative velocities between the surfaces are accommodated by different modes at the different sites.

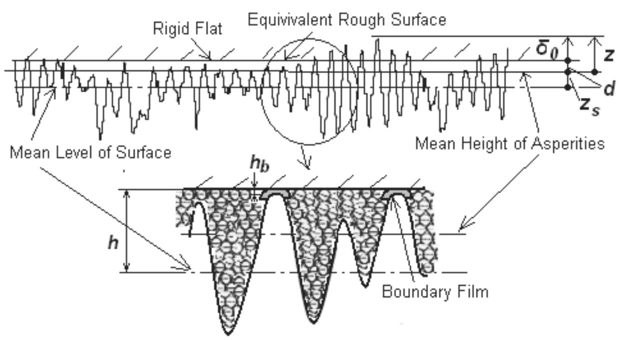


Fig. 1. A sliding contact model of mixed lubrication with powder lubricant

Effective lubrication of boundary lubricated systems such as the forging and stamping tools and high load sliding bearings is ensured by the formation of very thin films in the contact as a result of interaction between counterpart surface with the lubricant particles or oil. These films are known as tribofilms. There are different mechanisms by which the tribofilms ensure low wear and friction. In terms of tribofilms which protect component surfaces from excessive wear, the most common is the tribofilm formed from MoS_2 particles. Despite this not much is known about the tribofilms kinetics of formation and removal a tribofilm formation is a dynamic process which involves creation, removal and replenishment [8], and so in order to understand the formation of stable tribofilms it is very important to understand the kinetic of these processes.

Dry particulate lubrication schemes have been proposed for innovative bearing technologies: an oil-free journal bearing, a granular lubricated bearing by formulating a model that predicts the behaviour of colliding granules in a slider-bearing configuration [1, 7], forging tool lubrication [8]. In such applications the sliding contacts operate in a specific mixed lubrication regime. In order to optimize the interface behaviour with regard to friction on one hand and lifetime of friction pair on the other it is necessary to characterize and predict the lubrication regime in which such interface operates. Thus modelling and validation of the calculation methods of the mixed particulate lubrication parameters is an ultimate motivation for this work.

Numerous other related studies have been carried out over the past several decades on the modelling of the mixed lubrication regimes [10-14]. Some of these investigations [7, 12] primarily focused on the performance of MoS_2 powder in turbine bearing applications. Others, including M.A. Kabir et al. [13] and E.Y.A. Wornyoh et al [1], evaluated boric acid and numerous layered lattice powders or “powder lubricants.” Comprehensive studies aimed at determining the feasibility of using powder lubricants in general engineering systems were carried out by I. Jordanoff et al. [14] and R. Tenne et al [4-6]. Their research indicated that MoS_2 nanoparticle unique layered lattice structure made it a very promising solid lubricant material because of its high load carrying capacity and low steady state friction coefficient. However, there is still lack of data about solid lubricant particle behaviour at boundary and mixed lubrication of forging and stamping tool. It is the intent of this paper to provide physical based criteria through the rheodynamics of solid lubricant particles flow at the sliding interface, as a guide for the selection of solid lubricants, as well as elucidate behaviour of the solid lubricant particles with respect to tribosurfaces.

2. Experimental procedure

To study the influence of dry MoS_2 powder and oil - MoS_2 suspension with solid volume fraction of 5vol% on the friction performance, ball-on-disk experiments were conducted using a commercially available ball-on-disc T 10 of the PIB ITE in Radom, Poland. Various operating conditions (sliding speed, normal load, roughness) were investigated using spherical (9 mm diameter, hardness 62 HRC) steel ball with and steel disks and discs coated with PVD coatings (Table 1) Before each experiment, the disk and ball surfaces were cleaned with acetone. For the dry tests the MoS_2 powder was placed into microchannels made on the disc surface by laser engraving. A rubbing technique was applied for this purpose. Fig. 2 shows the microchannel image.

Table 1.
Geometry and properties parameters

No	Material used	Counterpart name	Coating Thickness, µm	Roughness, R_a, µm	Microhardness, MPa, and Nanohardness, GPa	Modulus of Elasticity, GPa	Friction Coefficient (Fretting)
			Geometry and Properties Parameters				
1	Garbon Steel (HS 6-5-2)	disc	-	0.025	980 MPa	210	0.3
2	TiAlN coating	disc	3.0	0.04	38 GPa	310	0.5
3	TiCN (CBC) coating	disc	3.0	0.06	37 GPa	350	0.2
4	TiC (Black top)	disc	1.0	0.047	24 GPa	410	0.3
5	DLC coating	disc	1.9	0.049	25 GPa	440	0.12
6	Garbon Steel (100Cr6)	ball	-	0.12	1010 MPa	210	0.3

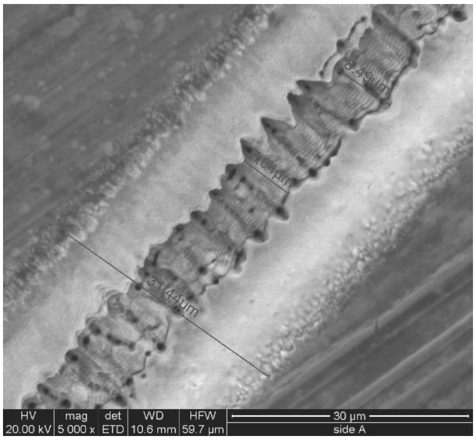
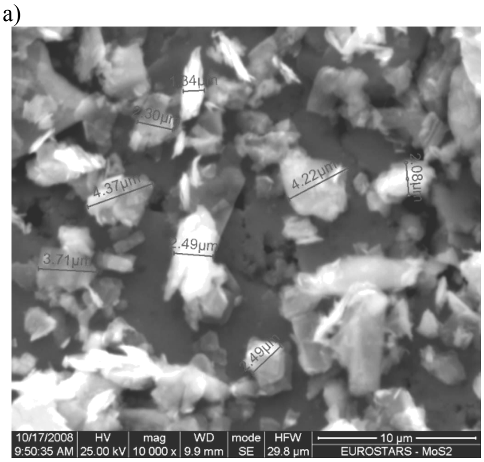


Fig. 2. SEM image of the microchannels

In the case of fluid lubrication the drops of oil-MoS_2 suspension were carefully placed on the disk so that the surface of the disk was in a fully flooded condition. For each test, a 6, 7, 8 N normal loads were applied with a constant sliding speed of the disc. The sliding speed varied in the range of 160-600 mm/s. A wear of the ball was calculated on the base of measurement of sliding contact spot diameter (d_{scar}) after end of the test. From other side the vertical displacement of a ball holder was registered to evaluate the current value of the ball wear. All experiments were carried out at room temperature (23°C). A scanning electron microscope (SEM) image of the MoS_2 powder made by rolling cleavage technique [9] is shown in Fig. 3. To obtain the suspension with MoS_2 solid volume fraction of 5vol%, the particles were mixed with NT oil using an ultrasonic mixer.

a)



b)

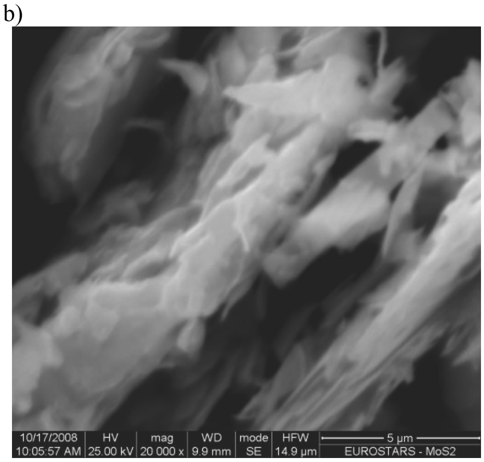


Fig. 3. SEM images of MoS_2 powder lubricant

3. Results

The triblological behavior of the samples depends on average normal contact pressure which may be calculated as

$$p = N/(\pi d_{scar}^2/4) \tag{1}$$

where d_{scar} is a diameter of sliding contact spot. It is defined on the base of Hertz formula at the beginning of test:

$$d_{scar} = 2\left(\frac{2}{3}\cdot\frac{NR}{E'}\right) \tag{2}$$

where: R is a radius of steel ball, E' – equivalent elastic modulus, N – normal load.

The results of friction tests at various normal contact pressures are depicted in Fig. 4 which provides the variation of friction coefficient with increase of average contact pressure for tool steel with dry MoS_2 powder lubricant and oil- MoS_2 suspension. The results reveal about strong dependence of the friction coefficient both on the normal pressure and sliding speed in the case of the dry MoS_2 powder application. An increase of the normal pressure leads to change of the lubrication regime to boundary lubrication which results in sharp raise of the friction coefficient. The increase of sliding speed allows to diminish the friction coefficient to minimal values (0.05-0.06). Thus the tribological behaviour of this sliding pair seems to be well described by Sribeck curves which allow differentiating the boundary, mixed and hydrodynamic lubrication regimes. The use of oil- MoS_2 suspension significantly changes the friction behaviour of sliding pair. The friction coefficient in this case is found to vary in the range of 0.13-0.15 that reveals about stable boundary lubrication regime in accordance to E. Gelinck and D. Schipper [10].

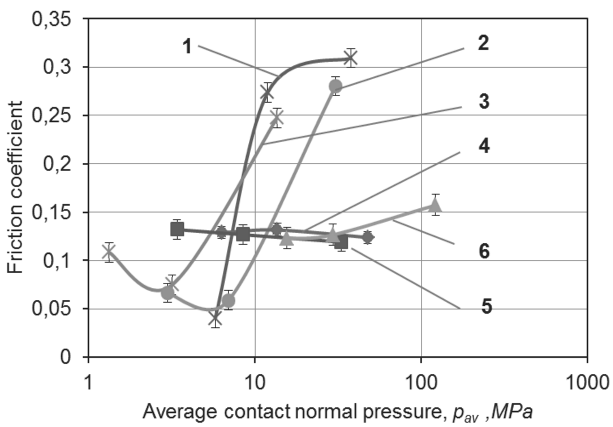


Fig. 4. Dependence of friction coaefficient on average normal contact pressure for tool steel. 1, 2, 3 – lubrication with dry MoS_2 powder, 4, 5, 6 – lubrication with oil- MoS_2 suspension. 1– sliding speed 0.194 m/s; 2 – sliding speed 0.381 m/s; 3 – sliding speed 0.571 m/s; 4 – sliding speed 0.162 m/s; 5 – sliding speed 0.320 m/s; 6 – sliding speed 0.482 m/s

Application of hard PVD coatings to enhance the performance of various tool is wide spread [15]. However, the mechanisms of lubrication with dry powder solid lubricant have not been studied yet. An analysis of friction behaviour of Fullerene like (IF) nano-particles in the oil suspension is presented in the recent work [16] for boundary lubrication conditions. It is shown that the main boundary lubrication mechanism is exfoliation of IF MoS_2 nanoparticles under compression and shear stress and formation of tribofilms. Whilst the mechanism of gradual exfoliation and transfer of molecular sheets onto asperities of sliding contacts [3] is being successfully realized for steel surface the MoS_2 particles don't adhere to hard ceramic coatings that results in absence of tribofilms at these surfaces, and, consequently, the friction coefficient in these cases is so high. We use the dry powder lubrication which allows to achieve the various lubrication regimes. For this reason examination of the hard coating lubrication with dry powder solid lubricant is believed to be important. The types of the PVD hard coatings made by Platit technology [18] are shown in the Table 1.

The results of friction tests of the hard coatings with powder solid lubricants are shown in Fig. 5 as a dependence of friction coefficient on average contact pressure. The low friction coefficients are achieved in the range of 10-300 MPa. An increase of the normal pressure up to 1-2 GPa leads to sharp raise of friction coefficient which becomes in the range of 0.3-0.45. Thus, the effect of dry solid lubrication is minimal at boundary lubrication regime. Such a behaviour is similar to those described in the paper [16]. The exfoliation process seems to be the main feature that controls the friction force at boundary lubrication. In our case the quasi-hydrodynamic behaviour of dry powder lubricant seems to be achieved, and low friction coefficients are observed for all studied coatings due to this effect. The experimental data are well approximated by exponent

$$f = Be^{mp_{av}} \tag{3}$$

where B and m are the constants of exponent approximation.

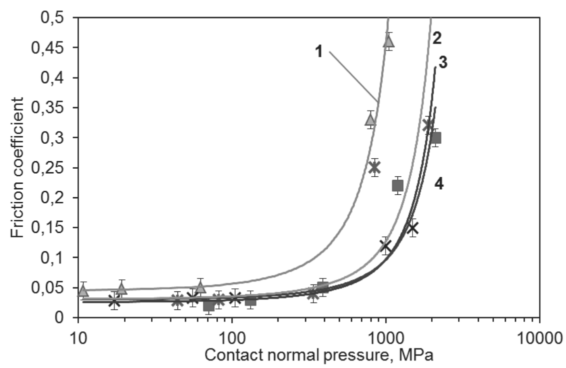


Fig. 5. Friction coefficient of PVD coatings with dry MoS_2 powder lubricant. 1 – TiCN(CBC), 2 – TiAlN, 3 – Black top, 4 – DLC

We tried to evaluate the exfoliation effect by SEM and EDS analysis. The average values of the content of main elements both in the disc surface and ball contact surface are presented in the Table 2.

Table 2.
EDS analysis results of the friction counterparts

Disc Material	Area of definition	Composition, wt%							
		Mo	S	O	Fe	C	Ti	Al	N
Steel	Microchannel	49.27	29.15	13.78	4.42	3.89	-	-	-
	Sliding contact	8.68	1.92	9.72	68.23	3.53	-	-	-
	Ball	2.84	0.52	8.03	85.9	3.7	-	-	-
TiAlN coating	Microchannel	19.98	13.76	4.94	5.82	4.58	21.17	14.67	14.69
	Sliding Contact	1.11	0.25	0.93	10.87	3.5	40.16	27.38	17.59
	Ball (steel)	0.69	-	10.2	82.17	4.51	-	-	-
TiCN (CBC) coating	Sliding Contact, 1	-	-	04.85	1.43	29.84	62.95	-	-
	Sliding Contact, 2	0.6	0.2	10.69	3.55	15.63	68.83	-	-
	Ball(steel)	1.19	0.28	4.54	86.60	3.83	1.28	-	-

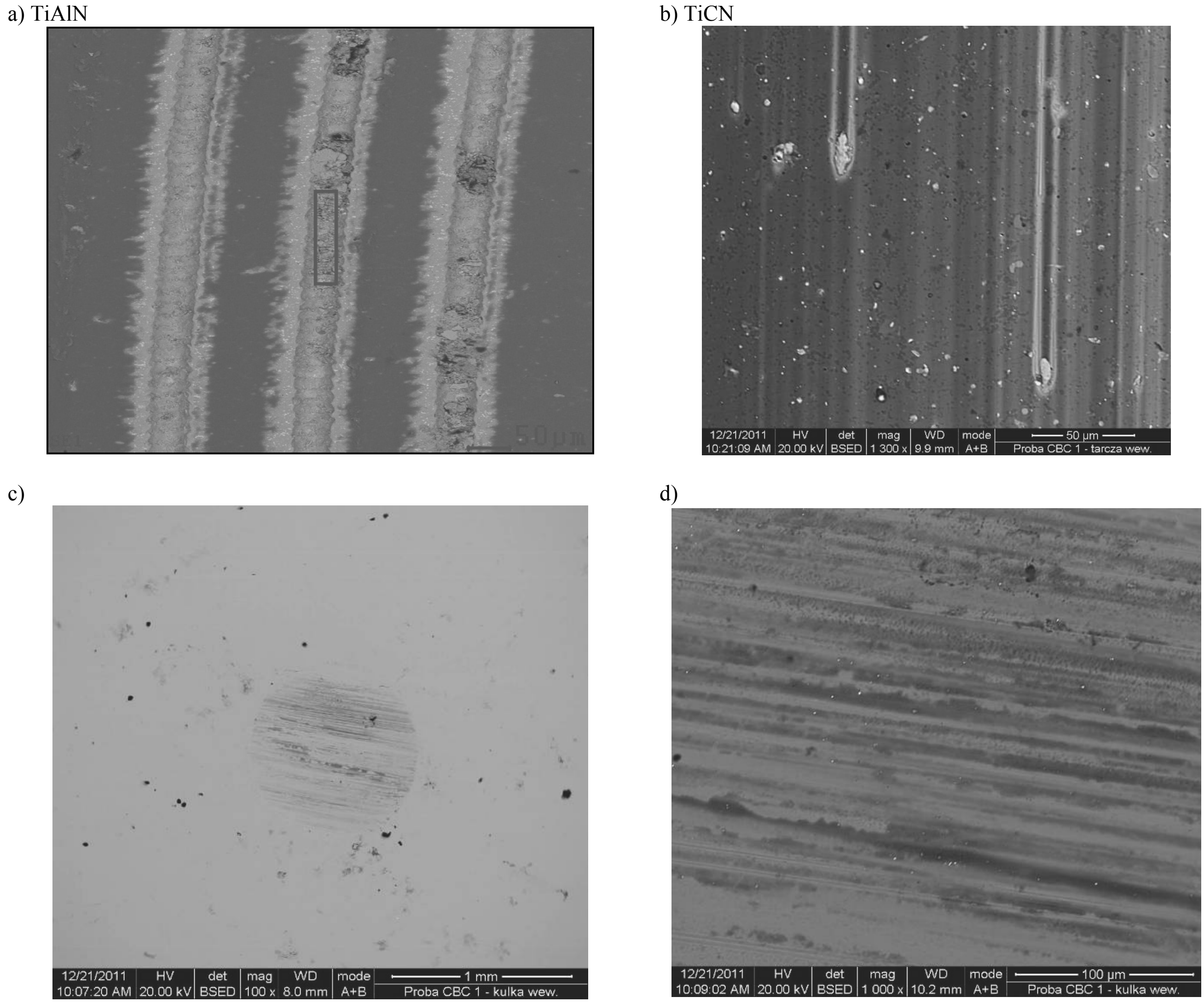


Fig. 6. SEM images of worn coatings and steel ball . (a) – TiAlN coating with microchannels filled with solid lubricant and wear debris particles; (b) – TiCN (CBC) coating with wear tracks; (c) and (d) – film on the sliding contact of the steel ball

Figure 6 shows the SEM images of the worn surfaces of the TiAlN, TiCN coatings and ball contact spot. The SEM images show the presence of MoS_2 particles in the microchannels and in worn tracks (Fig. 6b). Some thin oxide films are clearly seen on the ball contact spot surface. One can note there is quite small roughness of the worn surfaces (0.05-0.08 µm) due to high coating hardness (Table 1). The EDS analysis of the worn surfaces shows the following specific features:

- a content of MoS_2 particles in the microchannels is about of 20-50wt%;
- the steel disc contact surface contains more Mo and Sulfur then those of TiAlN and TiCN coatings, which have a small amount of Mo and Sulfur;
- the ball sliding contact surface of the steel disc-steel ball pair contains the maximal amount of Mo that reveals about presence of MoS_2 film;
- the content of Mo and S is smaller for ball surfaces tested with TiAlN and TiCN coatings.
- the Oxygen content in areas of examination is high in the most of cases that reveals about intensive oxidation of both solid lubricant particles and wear debris.

These results demonstrate that a permanent process of MoS_2 exfoliation and generation and fracture of MoS_2 solid lubricant film occur during sliding. The effects of MoS_2 exfoliation and surface film generation is more favourable for steel surfaces then for hard coatings. This result consents with the work [16]. However the exfoliation and tribofilm formation processes do take place at TiAlN and TiCN coatings probably due to presence of Ti. The features will be studied in more detail in the following work.

4. Discussion

The obtained experimental data exhibit that effective lubrication of boundary and mixed lubricated system such as the forging and stamping tool, sliding bearings is ensured by two processes: i) the formation of very thin films in the contact as a result of interaction between chemical components of solid lubricant with the lubricated surface and flow of the powder body at the interface (Fig. 1). Researchers [1, 14] have proposed to classify an innovative forms of dry particulate lubrication, namely powder and granular. Powder lubricants are classified as dry, "cohesive," soft particles that radically deform under load and accommodate surface velocity differences mostly by adhering to surfaces and shearing in the bulk medium, similar to hydrodynamic fluids. Granular lubricants are classified as dry, "cohesionless," hard particles that adequately maintain their spherical geometry under load and accommodate surface velocity differences through sliding and rolling at low shear rates, and largely through collisions at high shear rates. Both powder and granular lubrication mechanisms have demonstrated that the particulates in the sliding contact can enhance lubrication and lower friction below boundary lubrication levels. At low speeds (i.e. low nominal shear rates), powder and granular lubricants may appear to take on similar velocity accommodation behaviour. However, one distinguishing phenomenological factor is their behaviours at the boundary, where powders usually adhere and coat surfaces, and granules usually slip, roll and/or collide with surfaces [1]. Based on these features of dry powder lubrication there is need to exactly define the boundary and hydrodynamic lubrication modes in order to deeper understand the friction behaviour of sliding counterparts in mixed lubrication regime. It may be done on the base of advanced models of mixed lubrication developed by authors [12, 13] taking into account the recent achievements in the area of mathematical modelling of the tribological behaviour of boundary-lubricated systems [19-27] .

Constitutive equations

According to authors [10, 11] the Friction in the mixed lubrication regime may be adequately described with a model based on the fact that the total normal load N_T acting on a contact is shared between the hydrodynamic action and the interacting asperities of the interacting asperities of the surfaces:

$$N_T=N_C+N_H \tag{4}$$

where N_C is the load carried by the interacting asperities and N_H is the load carried by the hydrodynamic component. The total friction force F_f is the sum of the friction force between the interacting asperities and the shear force of the hydrodynamic component:

$$F_f=\sum_{i=1}^{n}\iint_{A_{C_i}}\tau_{C_i}\,dA_{C_i}+\iint_{A_H}\tau_H dA_H \tag{5}$$

Where n is the number of asperities in contact, A_C the area of contact of a single asperity i; τ_C the shear stress at the asperity contact i; A_H the contact area of the hydrodynamic component.

The coefficient of friction f_{C_i} of a single asperity can be written as:

$$f_{C_i}-\frac{\tau_{C_i}}{p_{C_i}} \tag{6}$$

with p_{C_i} the normal pressure of a single asperity. The coefficient of friction is assumed to be constant for all asperity contacts [26]. Thus, the first term of Eq. (5) can also be written as:

$$\sum_{i=1}^{n}\iint_{A_{C_i}} f_C\, p_{C_i} dA_{C_i}=f_C N_C \tag{7}$$

where N_C is the total load by all carried by all asperities. The value of f_C may be determined from friction tests at low sliding velocity and high normal load. The main difficulty is to define Nc, that can be done using an asperity-based mathematical model for boundary lubrication of nominally flat surfaces developed by H. Zhang and L. Chang [19, 20].

The model is made dimensionless for a more general representation. The asperity height and the surface separation are normalized by the rms roughness σ of the equivalent surface (indices of the equivalent surface are shown in Fig. 1),

$\bar{z} = z/\sigma \; and \; \bar{d} = d/\sigma$. Since the asperity contact force is a function of δ_0/R, where R − is the equivalent radius of collision asperity pair [20], and $\delta_0 = z - d$ (Fig. 1), δ_0/R may be expressed in terms of another dimensionless geometry parameter, σ/*R*, by

$$\frac{\sigma_0}{R} = \frac{\sigma}{R}\left(\bar{z} - \bar{d}\right) \tag{8}$$

Authors [19, 20] on the base of Greenwood and Williamson (CW) theory have shown that a GW plasticity index ψ for the two colliding surfaces is described by equation

$$\psi = \frac{E\prime}{H}\sqrt{\frac{\sigma_\alpha}{R}} \approx \frac{1}{3}\left(\frac{Y}{E\prime}\right)^{-1}\sqrt{\frac{\sigma_\alpha}{R}} \tag{9}$$

where *E*', *H,* and *Y* are the equivalent modulus of elasticity, hardness and yield strength of colliding surfaces, and σ_a is the standard deviation of the height distribution of the surface asperities.

The plasticity index is more meaningful to define the problem as a nominally flat contact system (with ψ less than 1) would elastically deform while those with ψ significantly above unity would substantially plastically deform [19]. We used the data of [19] and Fig. 6 to define the reduction in the load from the prediction of the classical GW model. The level of loading is defined by the mean separation of the two surfaces, $\bar{h}$=*h*/σ (Fig. 1). For a system of ψ=1.0, the contact is fairly elastic. The reduction in the load capacity associated with the collisions of contacting asperities is very small even with heavy loading of $\bar{h}$=1.0 at which the real area of contact reaches beyond 5% of the apparent area of contact. So, authors [20] suggest that reduction in the load capacity associated with the collisions of the asperities is not very sensitive to the applied load but rather more strongly linked to the plastic nature of the surfaces characterized by the plasticity index.

In the usual case of a Newtonian behaviour of the fluid lubricant the shear γ is given by equation (10):

$$\tau_H(\dot{\gamma}) = \frac{\eta u_{dif}}{h} = \eta\dot{\gamma} \tag{10}$$

where η is the dynamic viscosity, u_{dif} is the effective velocity difference between the contacting surfaces, *h* the local separation between the surfaces (Fig. 1), and $\dot{\gamma}$ is the shear rate ($\dot{\gamma} = u_{diff}/h$). Thus, the coefficient of friction *f* can be calculated by substituting Eqs. (7) and (10) in Eq. (5):

$$f = \frac{F_f}{N_T} = \frac{f_C N_C + \iint_{A_H} \tau_H(\dot{\gamma})\, dA_H}{N_T} \tag{11}$$

In the case of dry powder lubricant the flow models are based on rheological laws for the third body behaviour. These laws are then introduced into the continuity and momentum equations of fluid mechanics. When third body behaviour is close to the rheology of a solid, the equations of solid mechanics are used [14]. When the third body behaviour is closed to the rheology of a liquid, the equations of lubrication are adapted to a continuum flow that behaves like a non Newtonian fluid. Such a model has been developed by Heshmat [2]. According to [2] the role of solid lubricant and wear particles is to act as a form of lubricant between the initially dry surfaces. It has been shown that friction wear rate and other direct-contact phenomena decrease as wear particles accumulate between the surfaces. Moreover, the dynamics of the particles was shown to function not according to the prevailing theories of aggregate or compacted individual bodies, but rather more closely to a continuum with many of their velocity and shear characteristics akin to fluid films. Obviously, they also exhibit substantial differences, but on the whole they are closer to the nature of a lubricant than to an aggregate volume or discrete particles. As such powders can be treated here as a continuum and the powder flow can be called as quasi-hydrodynamic. Powders that are made to flow through a thin gap under the action of sliding or rolling exhibit a non Newtoman behaviour [1, 2, 10]. For the shear force in the powder lubricant film the Eyring model may be used [11]:

$$\tau_H(\dot{\gamma}) = \tau_0 \, arcsinh\left(\frac{\eta\dot{\gamma}}{\tau_0}\right) \tag{12}$$

where τ_0 is the Eyring shear stress.

So, Eq.(11) will be presented as:

$$f = \frac{F_f}{N_T} = \frac{f_C N_C + \tau_0 A_H arcsinh\left(\frac{\eta\dot{\gamma}}{\tau_0}\right)}{N_T} = \frac{f_C N_C + \tau_0 A_H arcsinh\left(\frac{\eta u_{diff}}{h\tau_0}\right)}{N_T} \tag{13}$$

The Eq.(13) represents in general the Stribeck curve which is the dependence of friction coefficient on a ratio $\lambda = \frac{effective\ film\ thickness}{surface\ roughness\ [CLA]}$ instead of classical Sommerfeld number $\eta u_{diff}/p_{av}\,\varepsilon$ (ε is clearance). We will use a lubrication number of Schipper [10] which is similar to. This number is defined as:

$$\mathcal{L} = \frac{\eta_0 u_s}{p_{av} R_a} \tag{14}$$

where p_{av} the average pressure in the contact, and R_a is the combined CLA (central line average) surface roughness, defined by

$$R_a = \sqrt{R_{a_1}^2 + R_{a_2}^2} \tag{15}$$

with R_{a_1} and R_{a_2} the CLA surface roughness of surface 1 and surface 2, respectively.

Stribeck curves

Figure 7 depicts the Stribeck curves $f = f(\mathcal{L})$ of the experimental data shown in Figs. 4 and 5. The fields of the boundary, mixed and hydrodynamic lubrication are clearly seen for dry powder lubrication that confirms Heshmat explanations [2] of quasi-hydrodynamic nature of the lubrication process at the relatively high sliding speeds and low pressures. The minimum of the Striebeck curve for dry powder lubricant is shifted towards lower values of Schipper parameter as compared to generalized curve [10]. The data of friction coefficient of oil- MoS_2 lubricated pairs (curve 2) fit only to boundary lubrication regime and correspond to generalized curve 1 (Fig. 7). Thus the thickness of lubricant film for oil lubrication case is smaller than for powder lubrication. However the boundary lubrication friction coefficient is higher for powder lubricant ($f \approx 0.25$-0.3) than for oil- MoS_2 case ($f \approx 0.12$-0.15).

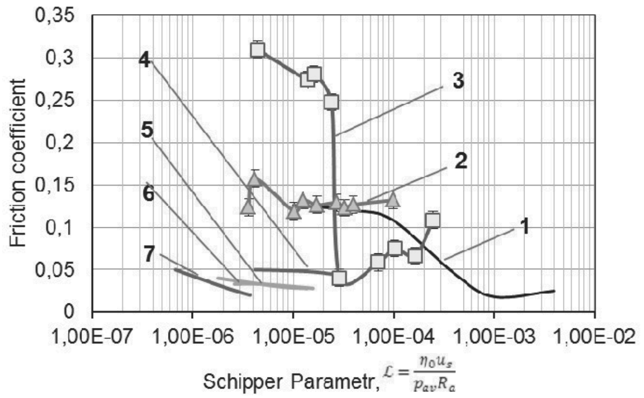


Fig. 7. Stribeck plot $f = f(\mathcal{L})$ of the experimental data shown in Fig. 4. 1 – generalized Stribeck curve for oil lubrication of line contact with parameters listed in [10]; 2 – experimental data of oil- MoS_2 suspension lubrication of steel disc – steel ball friction pair; 3 – experimental data of dry powder lubrication of steel disc – steel ball friction pair; 4 – experimental data of dry powder lubrication of TiCN coating-steel ball pair; 5 – experimental data of dry powder lubrication of TiAlN coating-steel ball pair; 6 – experimental data of dry powder lubrication of DLC coating-steel ball pair; 7 – experimental data of dry powder lubrication of Black top coating-steel ball pair

The data of friction coefficient of various coatings (curves 4-7) fit to minimum of the Stribeck curves that reveal about positive effect of the hard coatings in the dry powder lubrication at optimal hydrodynamic conditions. The stable lubricant hydrodynamic film of the hard coatings occur at higher contact pressures (low values of the Schipper number). The boundary lubrication friction coefficients are higher than for steel disc – steel ball pair due to lack of MoS_2 adhesion to coatings (Fig. 5). Nevertheless the lack of MoS_2 adhesion does not diminish the shear stress of solid lubricant and wear debris powder mixture which accommodates velocity at the interface in hydrodynamic conditions, and friction coefficient is small ($f \approx 0.02$-0.05). Therefore application of the hard coatings with dry MoS_2 powder lubrication is observed to be very promising.

The valuable friction parameters such as separation of the contact surfaces h, boundary lubricant film thickness, h_b (Fig. 1), the powder lubricant layer shear stress and others are difficult to define. We tried to evaluate some of them based on friction model described by Eq. (13). To validate the model the Eq.(13) is transformed to

$$ln(fN_T - f_C N_C) = \ln(\tau_0 A_H) + \ln[arcsinh\left(\frac{\eta u_{diff}}{h\tau_0}\right)] \quad (16)$$

Sribeck curve of steel disc – steel ball friction pair for powder lubrication conditions was calculated varying sliding speed and contact pressure while keeping other operational conditions constant. Values for parameters used for calculation were taken in the works [2, 7, 11].

The Experimental data of the ball-on-disc friction tests of steel disc – steel ball pair with dry powder lubrication calculated with the Eq.(13) are presented in coordinates $ln(fN_T - f_C N_C) = f\{\ln[arcsinh\left(\frac{\eta u_{diff}}{h\tau_0}\right)]\}$ in Fig. 8. It is seen the experimental data fit to linear relationship (15) with linear approximation inclination of about 45^o, that proves validity of the non Newtonian model (13). The condition $\ln(\tau_0 A_H) = -11.834$ allows to estimate the value of the contact area of hydrodynamic component A_H providing τ_o = 2.5 MPa [7]. The estimation results in the value of $A_H \approx 100$ µm^2. Thus, the modelling of mixed lubrication regime based on sharing of total normal load between asperities interaction and quasi-hydrodynamic component of powder lubricant layer is valid.

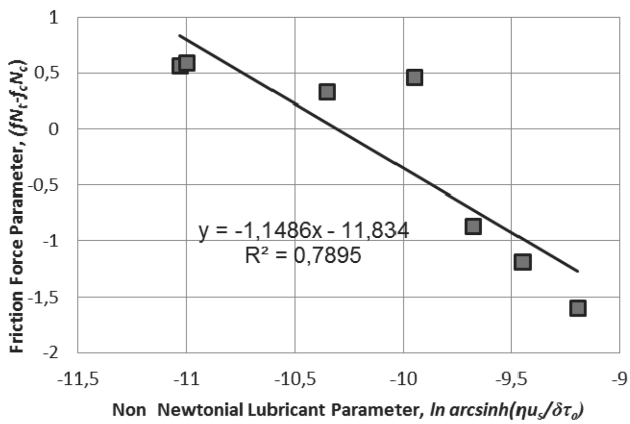


Fig. 8. Experimental data of the ball-on-disc friction tests of steel disc – steel ball pair with dry powder lubrication calculated with the Eq.(13) and presented in coordinates $ln(fN_T - f_C N_C) = f\{\ln[arcsinh\left(\frac{\eta u_{diff}}{h\tau_0}\right)]\}$

5. Conclusions

In the present investigation, ball-on-disc experiments were carried out to determine the lubrication performance of MoS_2 solid lubricant powder that could be used for hard PVD coatings applied for forging and stamping tools. The proposed solid lubricant nano- and submicroparticles mixture demonstrates excellent potential for use in mixed lubrication regimes The quasi-hydrodynamic behaviour of solid lubricant and wear debris particles results in low friction coefficients of hard coating – steel ball friction pairs.

The solid lubricant particle exfoliation and formation of tribofilms on micro-asperities allow to achieve the boundary lubrication effects which is found to more preferable for steel contacts rather than for hard coatings.

The model of mixed lubrication based on non Newtonian behaviour of powder solid lubricant was validated based on the experimental results. Results of calculation of Stribeck curves demonstrate the potential of modelling of friction process by sharing boundary and quasi-hydrodynamic processes.

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