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MODELING OF WEAR AND FRICTION UNDER BOUNDARY LUBRICATED CONTACT – ROLE OF PALM OIL METHYL ESTER (POME)

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Abstract This paper explores an approach of empirical model of wear and friction based-on experimental results from palm oil methyl ester (POME) added lubricant to contribute to the understanding of wear and friction process under boundary lubricating condition. In order to develop an empirical model, the specific wear rate against temperature, sliding distance and running time was statistically analyzed using MATLAB and an initial regression equation was obtained. The model reflects a general wear and friction equation of ball-plate configuration under 5% POME added lubricant. From the above findings it can be concluded that the sliding distance and time have very less influence on the wear of the mating surfaces. The friction equation shows that temperature has stronger influence on the friction coefficient compared to other two variables. It is reasonable to assume that wear and friction of mating materials up to a certain level are mostly controlled by boundary film of the POME even at higher temperature.

Keywords: Palm oil methyl ester, empirical model and boundary lubrication

LIST OF NOMENCLATURE

Mathematical symbols

A	contact area
K	specific wear rate; k, empirical constant
T	absolute temperature of the surface; t, time
V	wear volume; v, velocity
W	applied load; w, wear rate
a, b, c, m, n & q	empirical constant

Greek letters

Δ	an increment
α	probability of metal contact
β	constant
μ	friction coefficient; μ_f, friction coefficient in the conditions ensuring the absence of metal contact; μ_m, friction coefficient of metal-to-metal contact.

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INTRODUCTION

Modeling wear and friction is the ultimate goal of practical tribologists. There are many types of models (e.g., physical model, empirical model, mathematical or numerical model etc.) but the type tribologists and designers are more interested in are those that allow the prediction of wear and friction. The physical model predicts the performance of typical tribological contacts. Such models have the capability of being used in design and material selection, and are able to predict the values of friction coefficient, wear rate and maximum life of components. It is important to characterize geometrically, chemically and structurally the surface of each material in the system. The empirical approach of a model involves characterization of surfaces, collection of tribological data in laboratory tests, correlation of data with field tests, and sorting out of data to build an empirical model. Finally, the approaches that are mentioned above are not unique, and can be changed. This type of model is dynamic in the sense that one can update it easily with new parameters, correlation, etc.

For mathematical or numerical models, the works to be done include the derivation or formulation of different types of differential equations, solution of the equations by different numerical methods (e.g., Initial Value Problem, Boundary Value Problem, Finite Element Method, etc.) and development of tribological model. The theoretical models must be checked and validated by experimental data to ensure the general applicability of the models. It has been suggested recently that wear modelling must take new directions in order to succeed (Ludema, 1996). The need has been identified for involving many different disciplines in the development of practical and complete wear models.

Tribology modelling discussed above has always provided engineers with some information and understanding about the tribological phenomenon and given advantages to identify important factors which influence friction, wear and lubrication. Therefore, the importance of modelling for tribologists can be considered as a significant one for more effective approach of understanding the problems of tribological behaviour in tribo-systems. According to engineering practice, many models have been simplified to obtain practicable solutions. Frequently, the results of simpler models can be applied much more easily to practical problems than those of theoretically more precise but much more complicated, containing ill-defined or unknown factors. One area that is important in modelling friction and wear is the lubricating conditions. The next section explores how the results obtained from the experiment can be interpreted to contribute to the understanding of wear and friction process under boundary lubricating condition.

APPROACH TO MODELING OF WEAR AND FRICTION

Modeling of tribology is an effective tool to predict the tribological behaviour of mechanical components (e.g., piston ring, piston, liners, machine tools, cutting tools, gears etc.). Following a trend observed in all engineering fields, tribologist recently started applying various modeling methods at various levels of sophistication to solve the tribological (wear, friction and lubrication) mechanism, effect of different parameters on tribological behaviour including prevention of these problems in a more accurate and efficient way. Due to the complex nature of tribo-mechanism, a thorough understanding of the subject is still far off. The modeling approach can be a very useful tool as it provides engineers and scientists with a wide range of methods. Two approaches can be taken into account in developing models. One approach is based on empirical data which involves characterization of surfaces, collection of wear and friction data from laboratory tests, analysis and sorting of data. A second approach involves generation of models based on known physical principles, followed by confirmation through data collection.

Modeling of wear in metals has been conducted by many researchers (Jahanmir, S., 1986; Ling and Pan, 1988; Lancaster, 1990; Ludema and Bayer, 1991; Harnoy, A. and Fried L.B., 1994). Over the years, many models have been proposed for many different situations and most models are correlated in nature and therefore, system specific i.e., they only work for the particular material pairs, contact geometry, operating condition range and the particular environment and lubricant. It is useful to first examine the concept of empirical models. Engineering needs focus on one goal: guidance in the development or selection of an optimum design for a particular tribological function. A typical engineering problem involves four main issues (Hsu et al., 1997): (1) a geometric design (for example, a sliding bushing/shaft combination); (2) a set of usual materials of interest (e.g., 440C steel, various coatings, oil-filled bronze polymers etc); (3) an envelope of operating parameters (e.g., load, speed, temperature, environment, etc.); and (4) a set of relevant data to develop and verify the analytical expression for wear in the system (test results, properties data, etc.).

Empirical wear modeling means depending on experience or observation alone without due regard to science and theory. It has been more favorably called '*design oriented modeling*' (Bayer, 1991) or '*simulation modeling*' (Godet, 1988), and many who practice this approach, in fact, use scientifically guided empiricism. A few of the scientific principles of the complex process of wear is well enough known to formulate from fundamentals, e.g., the effect of increasing bulk temperature. But most of the science remains too poorly understood or is too multi-dimensioned for the use of simple fundamental equations to succeed. Therefore, an engineer, faced with an assignment to explain and predict wear performance in a design, must of necessity adopt an empirical approach.

EMPERICAL MODELS OF WEAR AND FRICTION UNDER POME ADDED LUBRICANT

Many equations have been derived using the methods of solid mechanics and most included material properties, thermodynamic quantities and other variable. Empirical equations are directly constructed with data taken from tests in which a few testing conditions are varied.

Barwell (1957-1958) suggested that wear rates may be explained by one of three equations:

$$V = \beta/\alpha\{1 - \exp(-\alpha t)\} \quad (1)$$

$$V = \alpha t \quad (2)$$

$$V = \beta \exp(\alpha t) \quad (3)$$

where V is the volume loss, α is a constant and t is time.

The parameter β is one of the mysterious terms, identified as 'some characteristic of the initial surfaces'. These equations simply describe the shape of a curve for V vs. t or V vs. β, the latter quantified in some way.
Rhee (1970) found that the total wear of a friction material (polymer-matrix) is a function of the applied

load (W), speed (v) and time (t) according to

$$\Delta w = KW^a v^b t^c \quad (4)$$

where Δw is the weight loss of the friction material and K, a, b and c are empirical constants.

In order to develop an empirical model, wear and friction test were performed with 5% POME added lubricant under boundary lubrication. Table-1 shows the general experimental conditions for wear and friction tests. The experimental results obtained, the specific wear rate against temperature, sliding distance and running time (or running cycle) was statistically analyzed using MATLAB and an initial regression equation was found to be:

$$\ln K = -13.613 + 0.515 \ln T - 0.022 \ln S + 0.0089 \ln t \quad (5)$$

where K = specific wear rate (mm^2/N), T = temperature (^0C), S = sliding distance (m) and t = time (min).
Equation (5) reflects a general wear equation of ball-plate configuration under 5% POME added lubricant of the type

$$K = k\,(T^a S^b t^c) \quad (6)$$

where, k, a, b, and c are empirical constants/coefficients, which can be derived from the constants in equation (5). The final form of equation (6) becomes

$$K = 1.229\, T^{0.515} S^{-0.022} t^{0.0089} \quad (7)$$

From the above equation it can be noticed that the sliding distance and time have very little influence on the wear of the mating surfaces.

The initial multiple regression equation for the coefficient of friction is,

$$\ln \mu = -10.8496 + 1.5287 \ln T + 0.2837 \ln S + 0.2055 \ln t \quad (8)$$

where, μ= friction coefficient.

Using similar considerations as in equation (6), equation (8) can be expressed as

$$\mu = s\,(T^m S^n t^q) \quad (9)$$

where, s, m, n, and q are empirical constants.
The expression for the friction coefficient of ball-plate configuration with 5% POME added lubricant was found to be

$$\mu = 1.94\, T^{1.5287} S^{0.2837} t^{0.2055} \quad (10)$$

The above equation shows that the temperature has stronger influence on the friction coefficient compared to other two variables. From the discussion presented above it is reasonable to assume that wear and friction of mating materials up to a certain level are mostly controlled by boundary film of the POME even at higher temperature.

PHYSICAL MODELS OF THE CONTACT WITH A CHEMICALLY ACTIVE POME ADDITIVE (MODEL OF BOUNDARY FRICTION)

The characteristic dependence of the coefficient of friction on the temperature under conditions of boundary lubrication and assumed physical models of the contact for the case of friction in lubricant with a chemically active POME are shown in Fig. 1. Up to the moment of reaching a certain temperature for the given materials combination, the contact is characterized by a rather low and stable coefficient of friction (~0.10-0.15) and moderate wear. In Fig. 1b, it is denoted by Zone-I (physical adsorption of the fatty acids of POME) corresponds to the model produced by Bowden and Tabor (1964). Here, the frictional resistance is due to interaction between the outer surfaces of the adsorbed mono and/or multiplayer, but where no metallic contact occurs. At this stage, POME reacts with the metal to form a metallic soap. Though this part of the contact is very small (at least, on the section OA), it exerts substantial influence on the friction processes. It could be observed over a very large range of loads as well.

After reaching the critical temperature T_{cr1} (point A on the curve) the first transition process begins – sharp growth of coefficient of friction, followed by the increase of wear (section AB on the curve). Then, after reaching the maximum value at the point B, (Fig. 1a, the corresponding temperature is $T_{f\ max}$), the friction decreases up to the moment, when at the point C (the second transition temperature or the temperature of chemical modification) a new value of the coefficient of friction is established for shorter period within narrow range of temperature. The corresponding contact geometry (model) of the mating surface is denoted by Zone-II breakdown of the lubricant film at small localized regions taken place. The metallic junctions so formed are partly responsible for the friction and almost entirely responsible for the wear and surface damage involved. The load is supported only over the area A (Fig. 1b), whilst metallic junctions are formed through the lubricant film over a much smaller area αA. The coefficient of friction μ in such a model of contact according to Bowden and Tabor (1964) may be calculated with the help of the following equation:

$$\mu = \alpha\mu_m + (1-\alpha)\mu_f \quad (11)$$

where α - is the probability of metal contact (friction film defects); μ_f - coefficient of friction in the conditions ensuring the absence of metal contact; μ_m - friction coefficient of metal-to-metal contact.

On the other hand, at temperature, T_{cr2} the third transition process begins which is characterized by a new sharp growth of the coefficient of friction, corresponding to the failure of the chemically modified layer.

At the point B of the curve OD maximum coefficient of friction is attained. Evidently, under the temperature corresponding $T_{f\ max}$, decomposition of the POME additive takes place; when this temperature is read, the chemical reaction of the additive active components with surfaces begins and zones of chemically modified layer in the contact might form. It is assumed that at the point C there is equilibrium between the formation and destruction of the chemically modified layer, and the contact takes place through the surface zones covered with such a layer. The physical model of the process on section CD is not quite clear; either the actual surface of the contact is covered with the modified layer, or the adsorption layer is partially preserved and its part continues to decrease with the increase of temperature.

On the basis of the above physical models of contact at different temperatures it can be said that on the curve section OB most important criteria is the adsorption properties of the chemically active POME additive; on the section BD most important criteria is its tribochemical properties. By using this type of model for surface-active substances, it is possible to predict the behaviour of vegetable-based fatty acid additive in the adsorption zone.

Table –1 Test conditions for wear and friction study

Test ball	Diameter	10 mm
	Material	Hardened bearing steel (AISI52100)
Plate material	Hardened grey cast iron	
Normal load	100 N	
Speed	0.34 m/s	
Mode of experiment	Ball-on-Plate	
Motor speed	250 rpm (0.34 m/s)	
Bath temperature	40 - 140 deg C	
Duration	1 hr	
Oil sample	5% POME added lubricant	

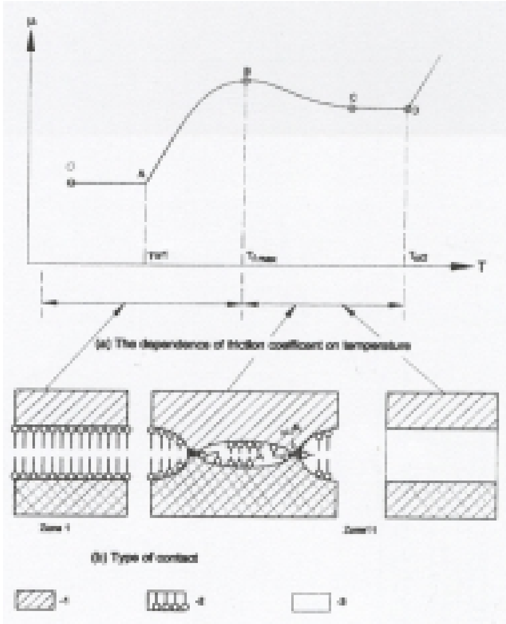

Fig.1 Physical models of the contact with a chemically active POME additive under boundary lubrication (model of boundary friction). Legend: 1 – mating surface; 2 - adsorption layer and 3 – chemisorption layer.

CONCLUSION

The following conclusions may be drawn based on the present investigation:

1. Empirical models showed that wear and friction of mating materials up to a certain level are mostly controlled by boundary film of the POME even at higher temperature.
2. On the basis of the physical models of contact at different temperatures, it can be concluded that up to certain temperature the most important criteria is the adsorption properties of the chemically active POME additive; beyond that the most important criteria is its tribochemical properties.
3. It could be concluded that wear and friction are more sensitive to higher temperature compared to sliding distance and time.

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REFERENCES

Barwell, F. T. (1957-1958), Wear of Metals, *Wear*, **1**, 317-332.

Bayer, R. G. (1991), Comments on Engineering Needs and Wear Models. In Tribological Modeling for Mechanical Designers, *ASTM STP 1105*, *(Eds. Ludema, K.C. and Bayer R. G.*) ASTM, Philadelphia, PA, 3.

Bowden, F. P. and Tabor, D. (1964), Friction and Lubrication of Solids, Part I, *Clarendon Press*, Oxford, U.K., 184-188, 200-230, 297.

Godet, M. (1988), Modeling of Friction and Wear Phenomena. In Approaches to Modeling of Friction and Wear, (*Eds. Ling, F. F. and Pan, C. H.*), *Springer-Verlag,* New York, NY, 12.

Harnoy, A. and Fried L.B. (1994), Dynamic Friction Model of Lubricated Surfaces for Precise Motion Control, *STLE Trib. Trans*., 37(3), 608-614.

Hsu, S. M., Shen, M. C. and Ruff, A. W. (1997), Wear Prediction for Metals, *Trib. Int*., 30(5), 377-383.

Jahanmir, S. (1986), Prediction Models for Sliding Wear, *Proc. Of Surface Deformation Models to Predict Tribological Behavior*, Columbia University, 17-29.

Lancaster, J. K. (1990), Material-Specific Wear Mechanisms: Relevance to Wear Modeling, *Wear*, 141, 159-183.

Ling, F. F. and Pan C.H.T (Eds.) (1988), Approaches to Modeling of Friction and Wear, *Springer-Verlag*, New York, NY, 1.

Ludema, K. C. and Bayer, R. G. (Eds.) (1991), Tribological Modeling for Mechanical Designers, *ASTM STP 1105*, *American Society for Testing and Materials*, Philadelphia.

Ludema, K. C. (1996), Mechanism-based Modeling of Friction and Wear, *Wear*, 1-7.

Rhee, S. K. (1970), Wear Equation for Polymers Sliding Against Metal Surfaces, *Wear*, 16, 431-445.