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TERMS OF PRESENCE OF LIQUID FRICTION CONDITIONS

The engineering practice shows that there are many bearings designed to operate in the field of hydrodynamic friction, but sometimes the oil film breaks and symptoms of wear appear on surfaces. We are interested in the possibility of determining these friction conditions under which the oil film effectively separates the two friction surfaces, and at the same time has a high load capacity and low friction. Parameters describing the selected range of hydrodynamic friction that interests us are as follows: pressure, sliding velocity, temperature and length of the slide at which the oil film not only is not interrupted, but still has a high load capacity and low friction. In this paper, we assume the boundary for the occurrence of hydrodynamic friction to be conditions under which a type of fluid friction changes into a kind of boundary friction. Establishing conditions in which one type of friction takes place helps to make conscious use of this fact to design a plain bearing with low friction and high strength non-interrupting film, and possible wear processes will occur only at moments of start-up and stop.

INTRODUCTION

There are still areas of engineering where knowledge is empirical in nature, commonly referred to as 'the art of engineering'. Parameters of human thought are arrived at 'by trial and error'. This 'art of engineering' characterizes a majority of solutions used in tribology. This conclusion can be drawn from concepts of the so-called 'tribological data banks' [1].

Such banks collect information concerning solutions of specific tribological pairs with regard to design, technology, materials, as well as conditions under which they operate. The issue of materials is addressed in two aspects: structural materials and lubrication materials. Computers are applied to building and employment of such banks, however, the information is still gathered by empirical means. It cannot be doubted that solutions produced by the method of trial and error continue to improve, yet the rate and cost of this development are incomparable to the development taking advantage of basic science. Basic science helps to understand phenomena and laws governing particular processes and then to put them into practice in an informed manner.

In view of the above, we have undertaken to develop test methods that would help to understand certain phenomena and make some contribution to a new perspective of problems in tribology, if not to solve all of them.

As far as issues of hydrodynamic friction are concerned, we would like to understand why the oil film loses its load capacity and is interrupted after a certain length of friction, as well as why the average unit pressure in plain bearings is so low compared to, for instance, wedge contacts, where elastohydrodynamic films withstand unit pressures which are several orders of magnitude greater without being interrupted.

Taking Reynolds and Navier-Stokes equations as the starting point, J. Kiciński [2] analyses models by Gumbel, Sommerfeld, Floberg et alia to conclude 'that phenomena occurring on the boundary of the continuous lubricant film as well as the mechanism itself of cavitation emergence and development have yet to be explored in detail'.

1. THE IDEA FOR EXPLICATING LOSS OF THE OIL FILM'S LOAD CAPACITY IN SLIDING FRICTION

Our approach to maintaining continuity of the oil film within the range of its minimum friction resistances or maximum durability (load capacity) is based on the following assumptions [4]:

- Boundary layers have considerable impact on phenomena taking place in the oil film as they provide thermal insulation when heat is conducted away from the zone in which the same heat is released [3],
- Heat accumulation along the oil film causes non-linear (exponential) variations (decline) of viscosity,
- Viscosity of a lubricant is the parameter determining load capacity and thickness of a hydrodynamic oil film until conditions arise in which triboelectric effects (dynamic boundary film) begin to prevail, friction resistances and film thickness increase,
- Friction surface should not be longer than the distance at which positive phenomena occur in the oil film (minimum friction resistances and maximum load capacity with an uninterrupted layer),
- The oil gap and its profile set by the design do not match viscosity variations along their friction surface, which is the gravest impediment to continuing maintenance of a high load capacity,
- The oil film can hold all along the designed profile if the rate of heat removed across friction surfaces is balanced with the heat released in the friction area.

Note: If the rate of heat removal across friction surfaces is lower than the rate of heat release produced by friction processes in the friction area, a lubricant in the oil profile accumulates the quantity of heat which has not been removed across the surfaces and lubricant temperature in the oil gap rises as a result. This dramatically reduces viscosity of the lubricant. The viscosity drop concurrent with the temperature growth is exponential. The following logical assumption should be made at this juncture:

The changing viscosity should be accompanied by modifications of the oil profile thickness reflecting the exponential viscosity fall concurrent with temperature growth in the friction area.

As progress of viscosity changes and temperature growth is not known, and the profile in actual friction pairs is set at the design stage and cannot adapt to viscosity variations and different parameters of friction (P, V, T), in the circumstances we would have to attempt forcing the lubricant to behave against its physical properties, which is impossible. The problem is illustrated in Figure 1.

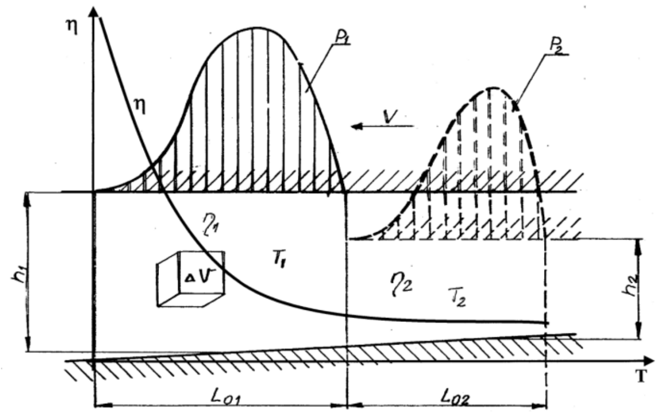

Fig.1. *Viscosity changes as a function of temperature and the associated adjustment of the oil gap thickness at which the layer's load capacity reflects the dwindling viscosity of lubricant*

The initial conditions correspond to the film's load capacity p_1, thus, the approximate film thickness h_1 can be determined. Hydrodynamic conditions described by η_1, T_1, p_1 and V_1 can be defined along the distance l_{01}, but viscosity falls dramatically to η_2 along a further section of the path as temperature rises to T_2, which implies that film thickness h_2

should change to adapt to the new conditions and pressure distribution across the film along the shorter distance l_{02} will change as a result too. As the designed profile cannot be changed, processes along the path l_{02} remain utterly unknown and are averages of the phenomena taking place beyond the initial section of the path. We do not know the film's load capacity beyond the initial stage of the path section (friction surface), therefore. In view of the foregoing discussion, a length of path needs to be determined where load capacity of the oil film is maximum. It is obvious that this length will depend on the sliding velocity V, friction pair temperature T, pressure P and friction pair (surface and lubricant) materials. Determining an optimum length of the friction path (with a maximum load capacity) requires development of an appropriate mathematical model of friction area phenomena. We tend to prefer a model based on optimisation employing both physical and mathematical modelling. The current stage of our research (discussion) should produce

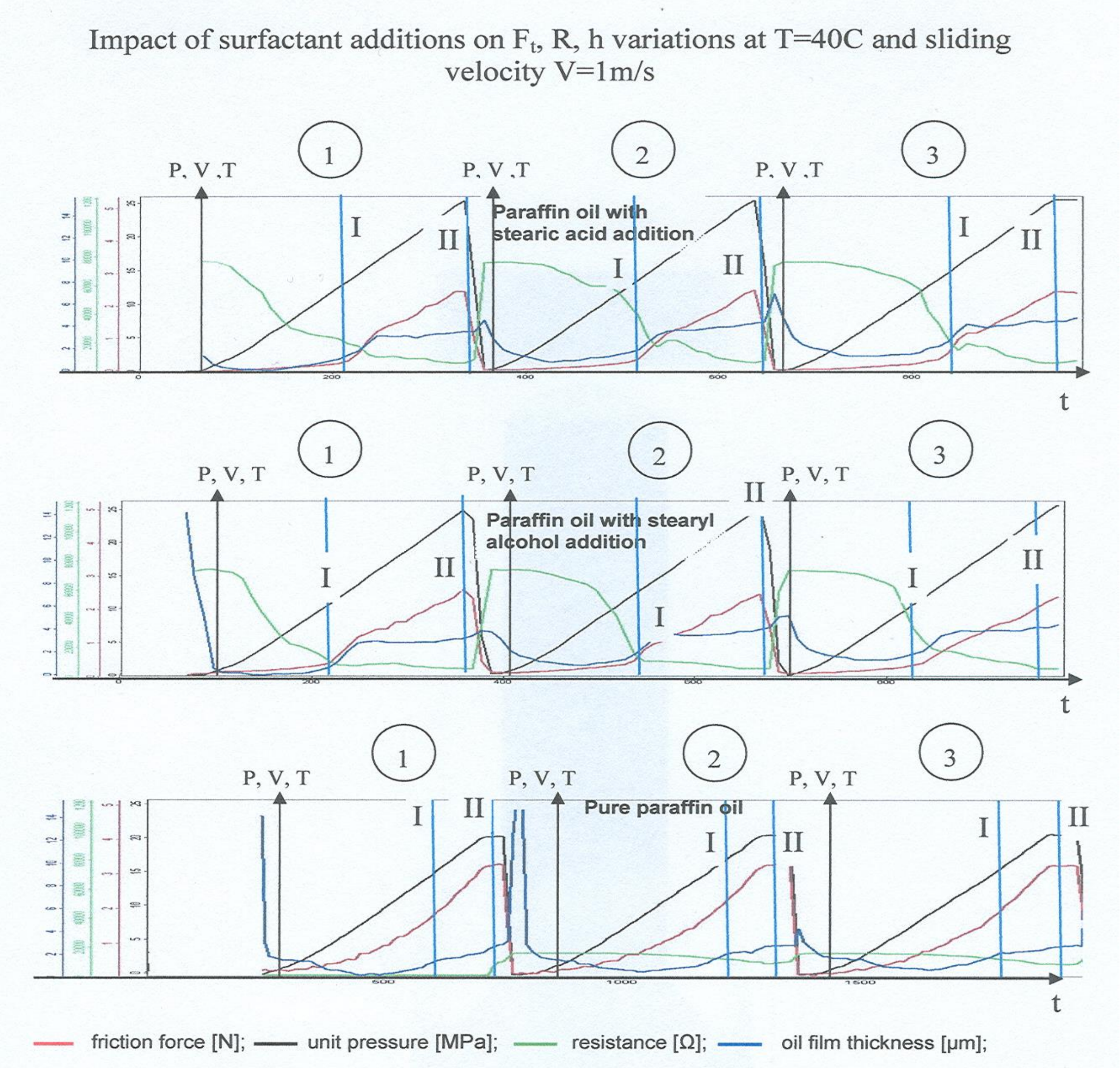

Fig.2. *Testing the boundary of hydrodynamic friction and the transition to intensive triboelectric phenomena in pure paraffin oil and paraffin oil with additions. The figure illustrates effect of the additions on the range of this friction and levels of friction; I – end of the hydrodynamic friction range, II – end of the experiment; the axes of ordinates were inserted at the start of experiments; (circled) 1, 2, 3 – repeated experiments (three runs were shown to illustrate recurrence and method of the experimentation; in fact, testing was carried out over 6 to 10 runs and averages were extracted to define boundaries of hydrodynamic friction for purposes of data processing).*

(via mathematical modelling) guidelines for physical experiments. Knowledge of an optimum length of the friction surface can be used to construct plain bearings of completely new operating parameters – friction with an uninterrupted hydrodynamic film, very high load capacity or minimum friction of the bearing, small dimensions, damped vibrations and application of cheaper materials and technologies, possibly to build clearance-free plain bearings that are easy to cool.

The issue of optimising the length of the friction distance surface can also be addressed with regard to minimising friction resistances. Such discussions are driven by results that indicate that, by using different lubricants, varied load capacity ranges of the hydrodynamic oil film and lower or greater friction resistances can be arrived at. The present testing was conducted with a 1mm friction surface. We are aware optimising the load capacity experimentally requires diverse lengths of the friction surface, but this is a further step in our research.

The figure 2 illustrates a possibility of determining a range of conditions where fluid friction occurs and the oil film is not interrupted. By varying the length of friction surface, one can define its optimum length in relation, for instance, to maximum unit load capacity.

When optimising the surface length in relation to maximum load capacity, such parameters as: temperature of friction pair T, sliding velocity V, as well as friction surface materials and lubricant, are taken into consideration.

CONCLUSIONS

- The possibility of distinguishing the range of hydrodynamic friction helps to define the length of friction surface over which the film remains continuous;
- The length of friction surface can be optimised with regard to a maximum load capacity of the oil film for a particular set of materials and range of friction conditions;
- The length of friction surface may also be optimised with regard to minimum friction resistances which depend not only on lubricants but also on the range of friction parameters P, V, T;
- The foregoing information can serve to construct sliding pairs of novel engineering capacities.

BIBLIOGRAFIA

1. Szczerek M.: Methodological problems of systematization of experimental studies of tribology. ITE-PIB, Radom 1997.
2. Kiciński J.: Theory and research hydrodynamic plain bearings transverse. Maszyny Przepływowe, Tom 15, PAN Instytut Maszyn Przepływowych, Wrocław 1994.
3. Krawczyk K.: Triboelectric phenomena in the film separating the surfaces of friction during sliding friction. Zeszyty Naukowe Instytutu Pojazdów PW nr 2/2010.
4. Żurowski W., Krawczyk K., Nowiński E., Wronka G.: Method of determining fluid friction conditions in TRIBOLOGY 2015 (ed. P. de Vaal). SAIT, Pretoria 2015.

WARUNKÓW WYSTĘPOWANIA TARCIA PŁYNNEGO

Praktyka pokazuje, że w technice istnieje wiele łożysk przeznaczonych do pracy w zakresie tarcia hydrodynamicznego, w których czasami ulega zerwaniu film olejowy i na powierzchni pojawiają się objawy zużycia. Ważna jest możliwość określenia warunków tarcia gdy film olejowy skutecznie oddziela dwie powierzchnie tarcia, a jednocześnie ma dużą nośność i niski współczynnik tarcia. Parametry opisujące wybrany zakres tarcia hydrodynamicznego są następujące: ciśnienie, prędkość przemieszczania, temperatura i długość prowadnicy, w których film olejowy nie tylko nie ulega przerwaniu, lecz wciąż ma wysoką obciążalność i niskie tarcie. W artykule założono granicę dla występowania tarcia hydrodynamicznego w celu ustalenia warunków w których rodzaj tarcia płynu zmienia się w rodzaj tarcia granicznego. Tworzenie warunków, w których występuje jeden rodzaj tarcia pozwala na dokonanie świadomego korzystania z tej wiedzy do projektowania łożysk ślizgowych z niskim współczynnikiem tarcia i wysokiej wytrzymałości bez przerywania filmu i występowaniu ewentualnych procesów zużywania tylko w momentach rozruchu i zatrzymania.

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