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EXPERIMENTAL INVESTIGATION OF THE LUBRICATION OF OPPOSITE RUNNING DISKS

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Abstract

Rolling elements in bearings without cage, suffer not only from the high pressure developed at the contact with the inner and outer ring, but also from the very high sliding at the contact between them. In the present study the lubrication conditions of such contacts are experimentally investigated, using a two-disk test rig developed recently by the laboratory of Machine Elements and Machine Design of the Aristotle University of Thessaloniki. The friction force, the temperature at several points of the disks as well as the asperities interactions are monitored during the experiments. From these measurements, the friction coefficient and the solid contact time percentage for each disk pair can be directly derived.

Keywords: *Opposite running disks, friction coefficient, high slide to roll ratio*

1. Introduction

Roller bearings without cage enable the designer to place more rollers in the bearing increasing thus the load carrying capacity. Especially in cases where an extreme space saving design is required, needle bearings without cage are used. For example, in planetary gear trains of heavy trucks just needles support the planets on the pins (see Fig. 1).

Due to the absence of a cage, the rolling elements contact each other. Although the pressure between them is not as high as that of the contact with the inner and outer ring, the lubrication conditions are very severe. The circumferential velocities of the rollers are in opposite directions and thus the sliding speed is very high. Therefore, increased wear, overheating or even scuffing of the rollers may occur.

Rolling elements are not the only machine elements where very high slide to roll ratio appears. Dyson and Wilson [1] have pointed out that in the cam and tappet system of internal combustion engines, twice in each valve-operating circle, the contact surfaces will have equal and opposite velocities. In their experiments they found that under such extreme operating conditions there still exists a quite thick lubricant film. They also pointed out that when load increases the central film thickness of such zero-entrainment conjunction does not decrease but increase!

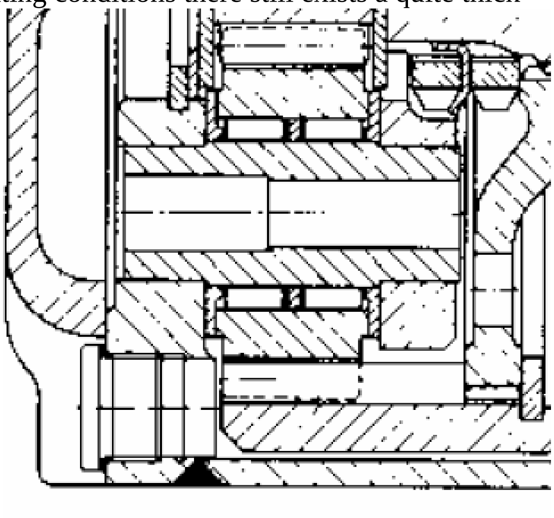

Fig. 1: *Needles used to support the planets of a planetary gear train mounted in the rear wheel hub of a heavy truck.*

According to the theory of isothermal EHL or of the low slide-roll ratio TEHL, the rolling velocity given by $v_R = \frac{1}{2} v_\Sigma = \frac{1}{2}(v_1 + v_2)$ and the lubricant viscosity have the most significant influence on the film thickness. In the case of opposite running disks, the circumferential velocities of both disks are the same, while their directions are opposite. This means that the slide to roll ratio defined as $S_R = v_g / v_R$ equals infinity. Therefore, the film thickness cannot be calculated by the classical EHL theory and several attempts have been made to develop new models. Hsiao and Hamrock [2] solved a non-Newtonian line contact problem with slide to roll ratio as high as 10, while Yang et. al. [3] solved a line contact problem with very high slide to roll ratio. They concluded that increasing the load, the oil film at the center of the contact gets thicker, while the minimum film thickness remains almost constant. They found that the friction coefficient must decrease with increasing load, since the shear rate decreases with the increasing film thickness.

In the present study the lubrication conditions of an elliptical contact that resembles the contact between two rolling elements are experimentally investigated using a two-disk test rig. The normal load is kept constant throughout the experiments, while the speed is varied from 0.25 to 25 m/s. Friction coefficient, disk surface bulk temperature as well as the contact time percentage are monitored and presented for two disk pairs that have different finish treatments and roughness profiles.

2. Two disks test rig

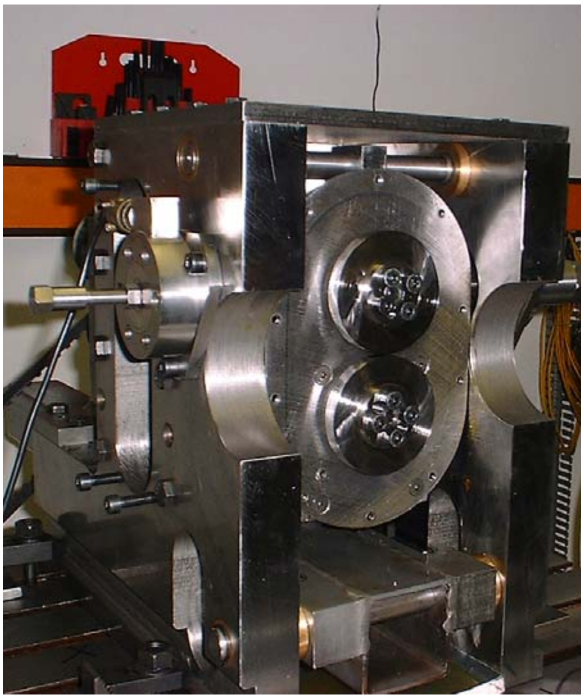

Fig. 2: *Two disks test rig with parallel shafts of the LME&MD*

The experiments have been carried out on the two-disk test rig with parallel shafts of the Laboratory of Machine Elements and Machine Design (*LME&MD*), shown in figure 2. It has been previously described in detail in [4]. Two inverter controlled AC motors drive the shafts independently. Each shaft can rotate from 60 to 6000 rpm (1 Hz –100 Hz) resulting in circumferential velocity from 0.25 m/sec to 25 m/sec for each disk.

In order to minimize the high stresses developed on the edges of the rolling elements in roller and needle bearings, they are not manufactured as cylinders. On the middle they are cylindrical but close to both edges they are crowned. In this way the contact of the rolling elements with the outer and inner ring as well as with each other is not a pure line contact but rather a very elongated ellipse. Therefore, a disk pair consisting of a cylindrical disk and a crowned one is used in the experiments. Both disks have a rolling diameter of 80 mm. The crown diameter of the latter is 990 mm. The disks are illustrated in figure 3. The above-mentioned disk pair gives a contact ellipse with ellipticity 12, which can safely be considered as a line contact.

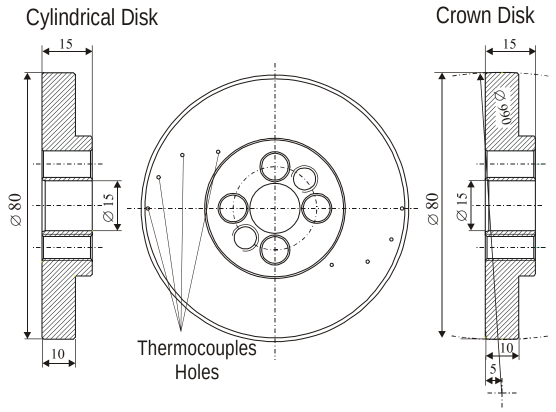

Fig. 3: *Experimental disks*

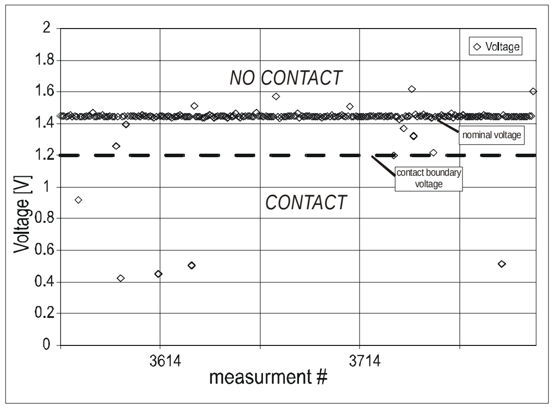

Fig. 4: *Asperity interactions of a fully flooded contact*

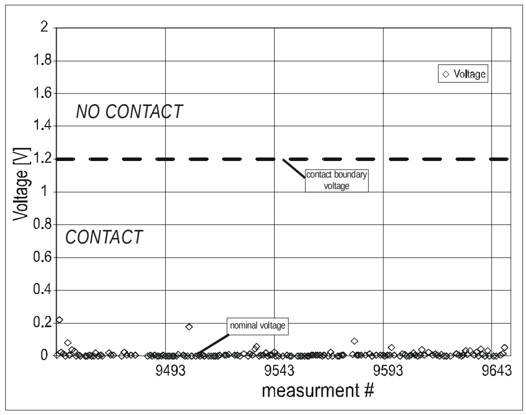

Fig. 5: *Asperities contact of a contact that operates in the mixed lubrication regime.*

Both shafts of the two-disk test rig are electrically insulated from each other. A battery of known voltage is connected to them in such a way that when the two disks come in contact the circuit closes. By monitoring the current voltage between the two disks, one can determine whether the oil film completely separates the disks or there is asperity contact, see figure 4.

Measured voltage equal to the voltage of the battery means that the two disks are completely separated by the oil film. When asperities come in contact, the circuit closes and the voltage measured should be 0 V. Whether this closed circuit voltage reaches 0 V or not, depends on the duration of the contact. The important fact is that a value significantly lower than the applied voltage of the battery means that at least one asperity pair is in contact. Certainly this measurement, cannot determine the solid contact area, since the contact of just one asperity pair would give theoretically exactly the same voltage read-out as the contact of the whole contact ellipse. Therefore, in order to evaluate if the contact operates in the full film lubrication regime, the contact time percentage λ_T is proposed as criterion:

$$\lambda_T = \frac{no.of\ asperity\ collisions}{no.of\ measurments} \cdot 100[\%]$$

$\lambda_T = 0\%$ means that during the measurement period no asperity contact took place, whereas $\lambda_T = 100\%$ means that throughout the same period at least one asperity pair was always in contact. Voltage measurements were taken at a constant rate of 6 measurements/sec. Therefore, λ_T represents the time percentage, at which at least one asperity pair of the disks is in contact .

The voltage measurements illustrated in figures 4 and 5, correspond to a fully flooded contact and to a contact that operates in the mixed lubrication regime respectively. Due to the fact, that certain electric and electronic devices produce noise, the boundary, i.e. the value under which solid contact occurs, was set to 1.2 V.

3. Experiments

The maximum circumferential velocity of the rolling elements of roller and needle bearings as calculated from the maximum rotational speed given in the catalogs, is 24 m/sec regardless the size of the bearing. The maximum Hertzian pressure developed between the rolling elements ranges between 200 and 400 MPa.

The experiments were conducted with 350 N/mm^2 maximum Hertzian pressure, while the maximum circumferential speed of the disks was set to 25 m/sec. The experiments were conducted using FVA 4 reference oil. An extensive data collection for this oil can be found in the literature [5], [6]. The oil is supplied directly on the two disks at a rate of 0.5 l/min at 45°C. The disks were made of 15CrNi6 steel case hardened to 710 HV. Two different surface finishing methods were used, in order to evaluate the influence of surface roughness. These are honing fine (HF) and turning (T). Typical rms roughness values for each disk pair and the resulting composite roughness R_q are shown in table 1.

Table 1: Rms roughness of the experimental disk pairs

Finish Treatment	Crowned Disk [μm]	Cylindrical Disk [μm]	R_q [μm]
HF	0.12	0.12	0.17
T	0.45	0.51	0.68

The experiment commences with the two disks rotating at 0.5 m/sec circumferential velocity. Then, the two disks are pressed together while rotating to ensure that their surfaces are lubricated. The circumferential speed is increased from 0.5 m/sec to 25 m/sec in 23 steps.

The disks operate at a given speed as long as required to reach a thermal equilibrium state; or in other words until the temperature field of the disk reaches a steady state. Steady state is considered when the temperature variation of each disk is less than 0.5 °C per minute. The experiment concludes when the speed reaches 25 m/sec or when the disk pair fails due to scuffing.

The maximum contact temperature ϑ_c can be modeled as the sum of two temperatures: (a) the bulk temperature, which is the weighed average of temperature of the disk surfaces just before contact, and (b) the flash temperature ϑ_{fla}, which is the maximum temperature rise during the contact:

$$\vartheta_c = \vartheta_m + \vartheta_{fla}$$

The bulk temperature of the disks is obtained by extrapolation of the temperatures measured by three type J (Fe-Ko) thermocouples positioned at a radial distance 2, 4 and 8 mm from the cylindrical or crowned surface of the disk, at a depth of 5 mm, equal to the half width of the disk. Figure 6 illustrates a disk with three thermocouples.

The flash temperature is calculated assuming one-dimensional heat transfer to the disks and a semi-ellipsoidal heat source [7] and [8]. These assumptions are justified due to the high Peclet number and the almost Hertzian pressure distribution of the elastohydrodynamic contact:

$$\vartheta_{fla} = 1.309 \cdot \frac{\dot{Q}_{1,2}}{A \cdot B_{m1,2}} \cdot \sqrt{\frac{2 \cdot b}{v_{1,2}}}$$

In the above equation A is the nominal contact area given by πab, $v_{1,2}$ is the circumferential velocity of the disk 1 or 2, a, b are the major and minor semi-diameters of the contact ellipse, and $B_{m1,2}$ the thermal contact coefficient of the materials of disk 1 or 2 defined as $\sqrt{\lambda_{1,2} c_{1,2} \rho_{1,2}}$.

The total generated friction heat flux $\dot{Q}_{tot}$ is obtained from the measured friction coefficient μ and the sliding velocity of the disks:

$$\dot{Q}_{tot} = \mu \cdot F_n \cdot v_g$$

The above total heat flux is equally distributed to the disks since they operate at the same speed and therefore the heat source is moving at the same velocity over their surfaces.

$$\dot{Q}_{1,2} = \frac{1}{2} \dot{Q}_{tot}$$

Fig. 6: *Experimental disk equipped with the mocouples.*

4. Results and discussion

The results of the experiments are presented in figures 7 and 8. Each figure consists of four separate charts. Charts "a" show the friction coefficient and the circumferential speed of the disks over the time. In charts "b", the friction coefficient and the bulk temperature of the upper disk are presented over the total time of the experiment. In charts "c" the friction coefficient and the contact time percentage are illustrated over the circumferential velocity. Finally, charts "d" show the bulk, flash and maximum contact temperature over the circumferential velocity.

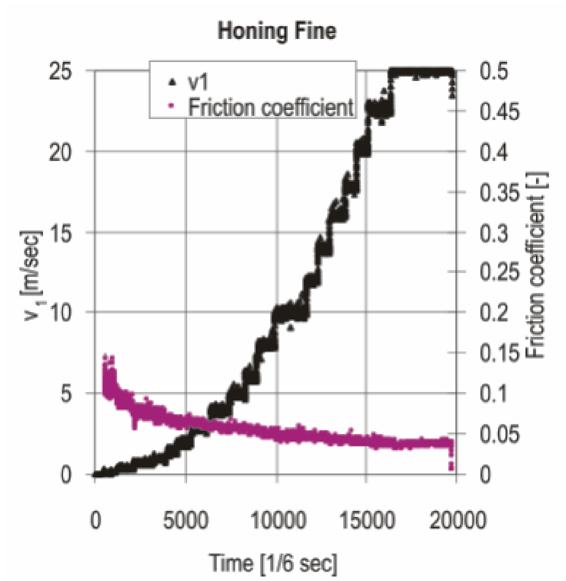

Fig. 7a: *Velocity and friction coefficient over time for the HF disk pair*

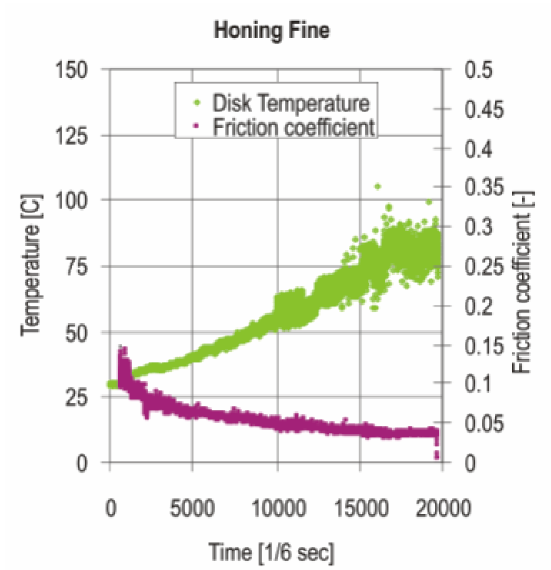

Fig. 7b: *Disk temperature and friction coefficient over time for the HF disk pair*

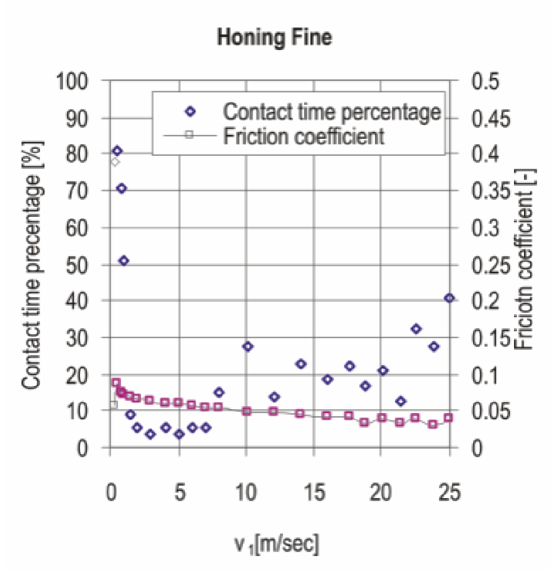

Fig. 7c: *Solid contact time percentage and friction coefficient over the velocity for the HF disk pair*

In figures 7a, b, c and d the results of the HF disk pair are illustrated. At the beginning of the experiment, the friction coefficient has a value of 0.15 and the contact time percentage is over 80%, which indicates that almost always some asperities are in contact. The circumferential speed is too low to support the development of an adequate oil film. When increasing the speed of the disks the friction coefficient decreases, while the bulk temperature of the disks increases. Obviously, the increase of speed causes a thicker film even though the disk surface temperature, and consequently the oil temperature, is also increasing. This can be concluded from the contact time percentage, which reaches a value of just 10-30%. The friction coefficient decreases to 0.04. The contact temperature increases gradually with speed.

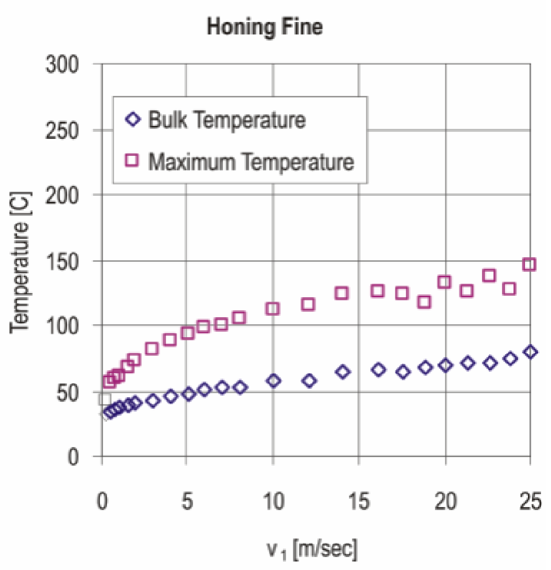

Fig. 7d: *Bulk and maximum temperature of upper disk over the velocity for the HF disk pair*

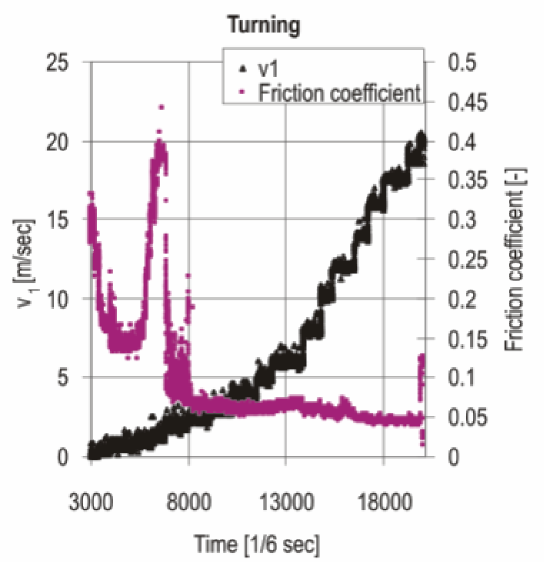

Fig. 8a: *Velocity and friction coefficient over time for the T disk pair*

Figures 8a, b, c, and d show the results of the T pair. At 1.5 m/sec scuffing occurred and the friction coefficient reached the very high value of 0.4, despite the low bulk temperature of about 52°C. However, the experiment was not aborted, but continued increasing the speed further. Both friction coefficient and bulk temperature dropped, a fact that indicates that the surface distortion caused by scuffing wasn't permanent, but a certain degree of self-curing has been made possible due to running-in effect at higher speed. Finally, at 20 m/sec, scuffing occurred for second time. It was much more severe since the friction coefficient and the bulk temperature reached the very high values of 180°C resulting in a contact temperature of over 200°C.

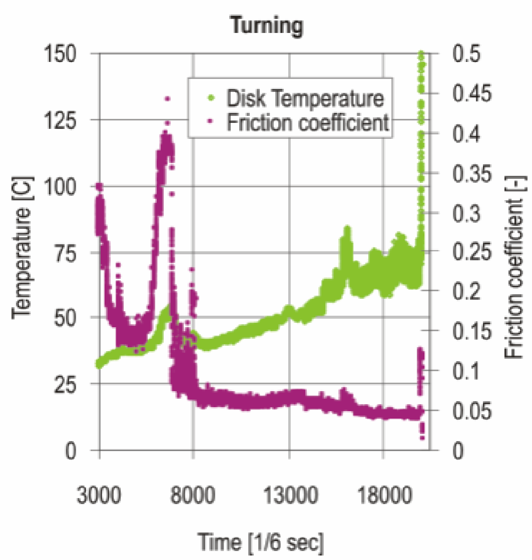

Fig. 8b: *Disk temperature and friction coefficient over time for the T disk pair*

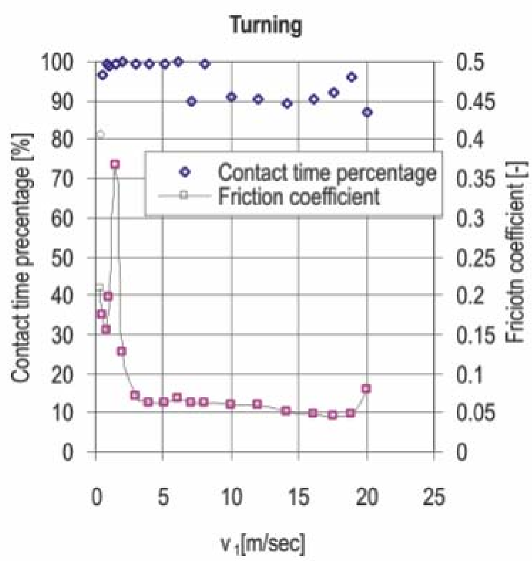

Fig. 8c: *Solid contact time percentage and friction coefficient over the velocity for the T disk pair*

Figure 9 presents the friction coefficient and the temperature of the disk before scuffing occurred for the T disk pair. At that moment the disks were rotating with circumferential velocity of 20 m/sec each. The friction coefficient was 0.048, while disk temperature was 80 °C. Then the friction coefficient started to increase rapidly and climbed to 0.13. This increase in the friction coefficient was followed by an increase of the disk temperature, since a higher value in the friction coefficient means that more heat is generated in the contact. The temperature finally rose to 180°C. The sudden rise of the friction coefficient from 0.048 to 0.13 lasted just 5 sec.

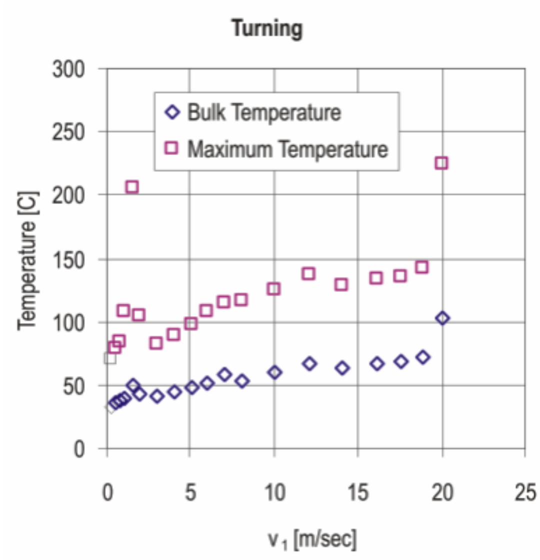

Fig. 8d: *Bulk and maximum temperature of upper disk over the velocity for the T disk pair*

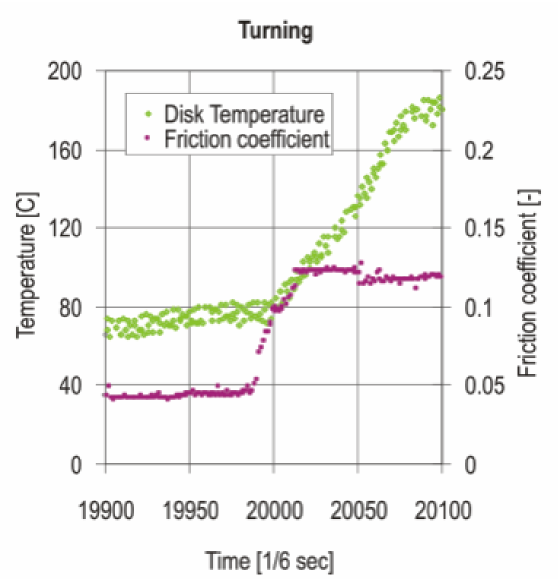

Fig. 9:*Friction coefficient and temperature rise before scuffing*

In some experiments, sparks were observed coming out from the contact, just prior to scuffing. Figure 10 illustrates a cylindrical disk after scuffing. The contact solid percentage was throughout the experiment more than 90 %.

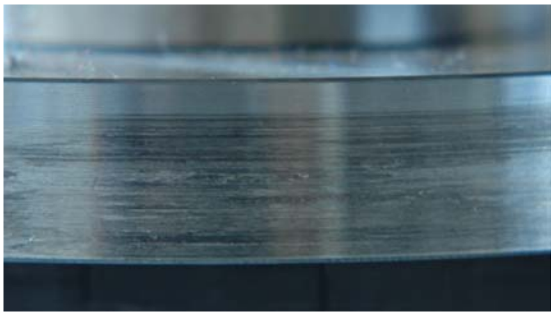

Fig. 10: *Disk surface after scuffing (cylinder of T disk pair)*

5. Conclusions

The friction behavior of opposite running disks is experimentally investigated using a two-disk test rig. The measurements are conducted with 350 N/mm^2 maximum Hertzian pressure, FVA 4 reference oil at 45°C oil supply temperature, at a rate of 0.5 lt/min. One disk is cylindrical and the other crowned resulting in an elongated contact ellipse (ellipticity 12). Several different disk pairs were used, manufactured with two different surface finish treatments resulting in composite roughness $R_q = 0.17\,\mu m$ or $R_q = 0.68\,\mu m$. The circumferential velocity of the disks varied from 0.5m/sec to 25 m/sec.

The experiments of this study lead to the following conclusions:

1. Even though the rolling velocity equals to zero, a lubricant film is built up. Solid contact time percentage shows a decrease with the increase of speed, meaning that the oil film is getting thicker. This happens until the influence of temperature on the development of the oil film prevails on the influence of the velocity, so that the contact time percentage starts to rise again. If the disks are smooth enough, it is possible to achieve fully flooded lubrication conditions.
2. The friction coefficient of opposite running disks at low speeds (<2m/s) can be very high and cause increased power losses in needle or roller bearings without cage. At higher speeds the friction coefficient decreases.
3. The lubrication conditions are very severe. Scuffing may occur even at low speeds, especially when the disks are rough.

6. References

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