EXPERIMENTAL SET-UP FOR THE STUDY OF SPEED EFFECT ON THE OCCURENCE OF SCUFFING

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Abstract: The present paper illustrates the construction and operation principles for an experimental set-up for the investigation of scuffing. Experimental results are presented with the highlight of various agents that can lead to scuffing.

Keywords: *scuffing, friction, speed, experimental investigations*

1. Introduction

As a typology of early adhesion wear that can develop from a moderate adhesion phase into a severe one, scuffing can practically appear in most friction couplings: bearings, dismountable assemblies, guides, piston-cylinder assembly, gears, variable speed devices, cutting tools etc. Although recent theoretical and experimental investigations clarified many of the triggering factors for scuffing, the phenomenon as a whole still remains a challenge for researchers. Scuffing phenomenology and its prevention depends on many variables, such as: materials, machines and equipment, surface preparation methods and procedures, lubricants etc. As a complex dynamic phenomenon, scuffing implies failure of the lubricant film. One of the parameters that influence scuffing occurrence and evolution is speed. Both rolling and sliding speeds can have important consequences with regards to scuffing appearance risk.

Nelias, [1], advanced a new theory on scuffing phenomena, taking into consideration speed, temperature, load and surface microgeometry as study parameters, for various materials and lubricants. From the experimental investigations conducted by aid of a two-disk machine, Nelias, [1], found a correlation between sliding speed and the appearance of micro-scuffing. Increasing progressively sliding speed while rotational speed and loads are maintained constant, an abrupt increase of friction was observed, which lead to machine failure. The used a lubricant specifically designed for aerospace demands, Tetraesther having a viscosity of 5CSt. Experimental investigations presented in [1] were conducted at speeds of 25m/s and 50 m/s, under working pressures of 0.5; 0.8 and 1.6 GPa and for lubricant temperatures of 40°C; 80°C and 120°C. The experimental results advanced in showed that scuffing failure appear at sliding speeds above 40 m/s under low pressure conditions.

In 1998, Bujoreanu, Olaru and Popinceanu, [2] determined the scuffing limits corresponding to elastohydrodynamic (EHD) contacts. Their tests were conducted for bearing balls subjected to speeds ranging from medium to very high.

Starting from the hypothesis that no matter what the scuffing mechanism, an energy unbalance must take place in contact, Bujoreanu, in [3], estimates scuffing limits by assessing the dissipated energy in rolling contacts. Also, distinct relations were advanced for each lubricating regime. The fact that scuffing limiting sliding speed decreases as contact pressures and lubricant working temperatures increase, is presented as well in [3].

The conducted investigations showed that sliding speed, scuffing limit and contact pressure coexist in an antagonistic relation and that the lubrication regime is very important for failure initiation, with significant influence on speed.

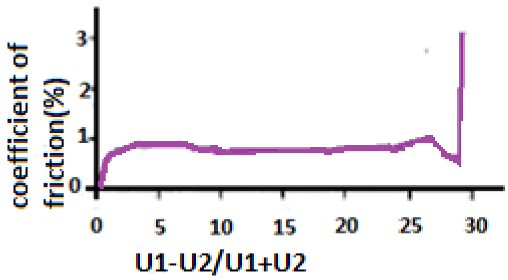

Figure 1: *Sketch of a traction curve after Nelias, [1]*

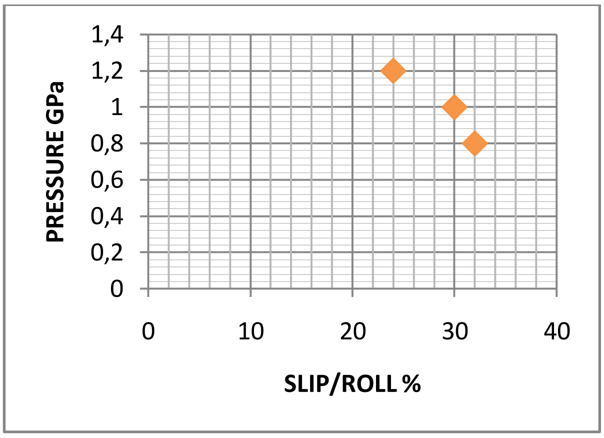

Figure 2: *Pressure-slip/roll correlation for scuffing occurrence, ELF 154 NS. Oil, after Bujoreanu.*[2]

The present study aims to investigate speed influence on the occurrence of scuffing. To that end, it was assumed that in the case of hydrodynamic lubrication (HL) bearing capacity increases proportionally to the product between lubricant viscosity, η, and the relative speed of the two contacting surfaces, *v*. As this product increases, shear stresses in the lubricant film also increase according to eq. (1):

$$\tau = \eta_e \dot{\gamma} = \eta_e \frac{\partial v}{\partial n} , \qquad (1)$$

where η_e is the effective viscosity of the lubricant film and $\dot{\gamma}$ represents the shearing speed, given as the speed gradient along normal direction to the flow plane. Once shearing speed increases, lubricant viscosity decreases, but this fact cannot avoid the shear stress evolution, which rise up to a critical value, [4]. Reaching of the critical value unavoidably leads to scuffing. In conclusion, scuffing phenomena are linked to shear speed of the lubricant film and more precisely to its plastic shear, determined by reaching of a critical shear threshold (shear limit), which in its turn is dependent on pressure and temperature.

2. Experimental setup description

Starting from the abovementioned hypothesis, an experimental setup was conceived and built that permits investigating the lubricated contact between a disk and a specimen, under variable conditions of loading and speed. The presented experimental setup operates on a similar principle to that of a two disk machine, but it is adapted to the contact of only one rotating disk, fixed on the shaft of a motor (which ensures rotation) and a specimen, fixed by means of prismatic device on a swing arm which allows measurement of contact frictions forces.

The friction force measuring principle consists of monitoring the correlation between the pushing force (load) and the friction coefficient, under the pre-established conditions. This setup permits determination of the contact operating parameters values throughout the entire experiment, until scuffing appears.

An overview of the experimental setup is illustrated in Fig. 3, with highlighting of its structural systems.

The main components of this constructively simple setup are: a drive system for the disk, a system that ensures loading of the investigated contact, a measuring system and a system that ensures lubricant feed by free fall.

2.1 The drive system

The driving system that ensures motion of the disk consists of a MCA 45/64-148/IRN variable speed motor. Its technical and operational parameters permit speeds of up to 12,500 rpm, variable in time, which allows easy tracking of the evolution for such parameters of interest as friction force and lubricant film presence. A speed transducer is also integrated to the motor. Speed variation is ensured by a variable DC power supply. The active disk, which constitutes one of the contacting bodies, is mounted on the end of the motor shaft.

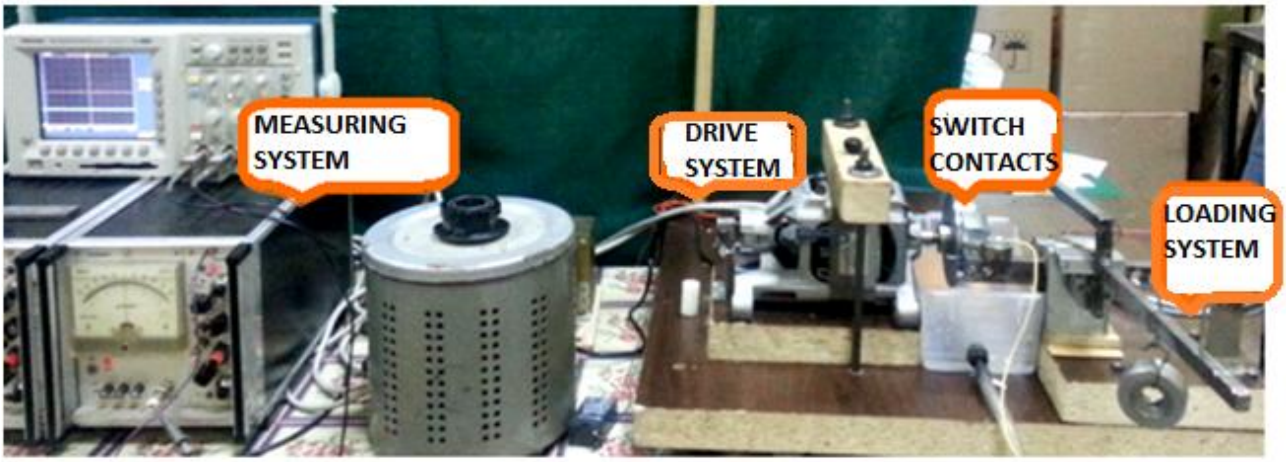

Figure 3: *Experimental setup for the study of scuffing*

2.2 Loading system

The deadweight G is placed on one side of a lever system, which makes the other side push, by means of an axial bearing, against the element that supports the investigated specimen, as shown in Fig. 4. The different length of the two sides of the lever system ensures a load multiplication that can be determined as:

$$F = \frac{c}{x} \cdot G \tag{2}$$

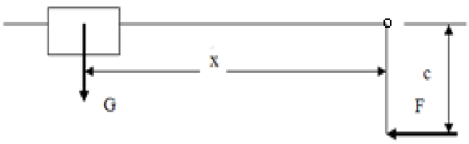

Figure 4: *Loading system principle*

Various deadweights can be attached to the lever, thus determining proportional loads applied to the investigated contact. This leads to a bending of the lamella attached to the lever system. The strain of the lamella is sensed by a strain gauge placed on the lamella and viewed on the oscilloscope screen, as units. Various deadweights with known mass were placed on controlled positions on the lever, thus applying controlled loads to the studied contact. The effective load value is determined by aid of a force transducer, placed behind the specimen fixing system and marked as 2 in the representation illustrated by Fig. 6.

2.3 Friction coupling

The monitored friction coupling is created by the disk (cylinder) – specimen (roller) assembly as shown in Fig. 5. The metallic disk has an outer diameter of 100 mm, it is 25 mm wide and it is made from bearing steel, with a polished active surface. The disk was mounted on the main motor shaft. The used specimens (one for each test) were either cylindrical or barrel shaped metallic rollers, adapted from radial or radial-axial roller bearings. The load applied to the contact ensures fixing of the rollers on their prismatic support. This system allows for fixed positioning, as well as fast mounting/dismounting of the specimens.

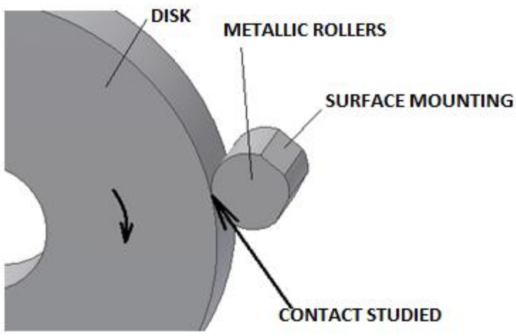

Figure 5: *Friction coupling*

2.4 Lubricating system

Contact lubrication is ensured by free fall from a lubricant reservoir mounted above the friction coupling. Flow regulation ensures abundant lubrication of the contact, which also ensures partial elimination of the heat generated by friction.

Industrial lubricants, destined for lubrication of contacts predisposed to scuffing, such as those between gear teeth or between bearing components, also contain additives for extreme pressures and anti-scuffing. Use of such lubricants for the proposed experimental investigations would not allow for an eloquent study regarding the influence of lubricant shear, as the results would be influenced by additive presence. Therefore, preliminary investigations, meant to experimentally demonstrate the envisaged phenomena were conducted using a pure substance (not a mixture) with lubrication properties and for which, shearing properties are known from previous works. The substance chosen to fit these requirements was glycerin.

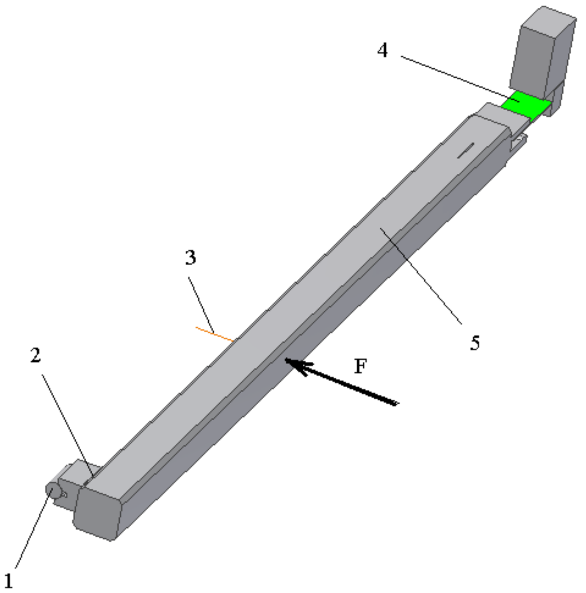

Figure 6: *Assembly of the loading and measuring system. (1 - specimen, 2 - force transducer for load measurement, 3 - rotation axis, 4 – friction force measuring system, 5 - swing arm).*

2.5 Measurement and control system

The friction force is measured by aid of strain gauges placed on an elastic lamella attached to the specimen fixing system, as illustrated by Fig. 6. The bending of the elastic lamella 4 induced by the friction force leads to variations of the attached strain gauge's electrical resistance, detectable by a strain gauge indicator.

In order to correlate the strain gauge indications to the actual friction force, the system was first calibrated by application of known loads. The obtained calibration diagram for the friction force transducer is presented in Fig. 8.

Lubricant film presence at contact interface was determined by resistive methods. It is known that contact electrical resistivity increases with the film thickness which allows the study of thin film regions. In order to evaluate the lubricant film thickness, the two contacting bodies (specimen and disk) were connected to a DC power supply, allowing current to pass through the contact. As the lubrication regime is not completely fluid, some direct metal on metal contacts take place between contact surface asperities. Contact resistance diminishes significantly in this case, which imposed integration of the registered electrical signal and then visualization using an oscilloscope. This allowed for a mean appreciation of lubricant film presence.

2.6 Electronic circuit for data monitoring

The data monitoring circuit consists of the two strain transducers, a strain indicator, an integration circuit and a using, as illustrated by Fig. 7.

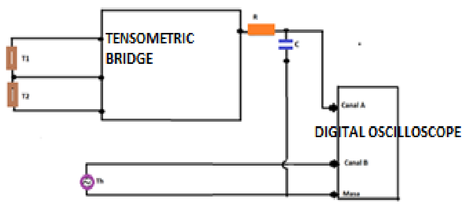

Figure 7: *Electronic circuit for data monitoring*

3. Experimental methodology

3.1 Calibration of the loading system

The force transducers were calibrated by repeatedly applying loads using various deadweights with known calibrated masses. The obtained measurements were then used to correlate the transducer signal with applied loads. An example of calibration curve, determined for the friction force measuring system is plotted in Fig. 8. Signals from the transducers are converted by the electronic system in units displayed by the oscilloscope.

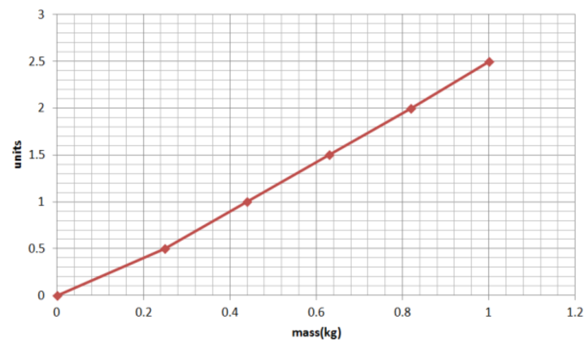

Figure 8: *Calibration curve determined experimentally for the friction force transducer*

3.2 Experimental methodology

Each experiment implies the following steps:

1. The roller specimen is placed on the prismatic support without making contact with the disk.
2. The corresponding electrical connections are made for the lubricant film measuring system.
3. The appropriate deadweight is positioned on the loading system lever so that the needed load is applied to the contact.
4. Contact lubrication is ensured by opening the lubricant feeding circuit.
5. The drive motor is started.
6. When the disk reaches speeds of at least 800-900 rpm, the specimen is pressed against the disk, and the contact is established.
7. Friction forces generated in contact are monitored while the speed is progressively increased until a rapid increase of friction is observed, which signals scuffing appearance.
8. The experimental setup is turned off.

As speed was increased, a decrease of the friction force was observed, which confirms the appearance of carrying capacity. Friction force values were determined using the classical linear equation: $F_r = \mu \cdot F_a$. As the low applied loads and the lubricant is in large quantity provide the framework for relatively low working temperatures, the effects of local heating were neglected. By further increasing speed, it was observed that, for all tests, a speed of 2500 rpm can be considered critical because reaching it lead to a sudden increase of friction force, which indicated appearance of scuffing.

4. Experimental results

In order to verify the proposed experimental setup, two scuffing experiments were conducted using glycerin as lubricant. Contacts between cylindrical rollers with 10mm diameters and the system's rolling disk were loaded by F=5.27 N forces.

The conditions for scuffing occurrence were determined by plotting the evolution of friction coefficient against speed corresponding to all conducted tests. The friction coefficient evolution determined experimentally is illustrated in Fig. 9. It can be clearly noticed that the plot has an initially slow evolution path, until speed reaches 2250 rpm, when friction suddenly increases, an obvious indicator for scuffing. As in this region, friction coefficient reaches values of 0.426, this also indicates that the lubricant film failed and contact takes place between surface asperities. The similarity of the experimental results is a solid argument for validation of the experimental setup and methodology.

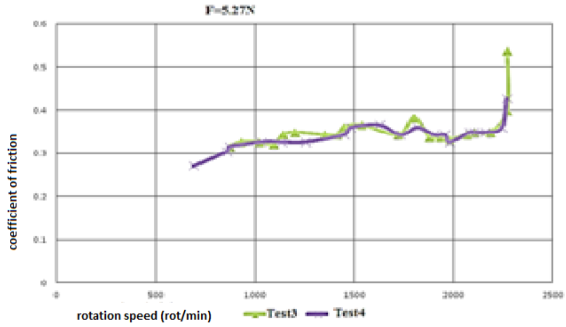

Figure 9: *Friction coefficient vs. speed for the contact between a cylindrical roller (d=10mm) and a rotating disk (D=100mm.), under normal loads of 5.27 N.*

5. Conclusions

The present paper presents an experimental setup conceived and built in order to investigate the effect of speed upon scuffing phenomena for various friction coupling elements geometries and under different applied loads to the lubricated contact.

The adopted constructive solutions are simple, secure and easy to use. The experimental setup allows permanent control over the following parameters:

- Rotating disk speed (variable);
- Electrical resistance of the lubricated contact – parameter that gives information regarding the lubricant film thickness and implicitly the lubricating regime for the studied contact;
- Normal load globally applied to the contact;
- Friction force between contact surfaces.

The conducted experimental investigations aimed to evaluate the friction coefficient evolution with speed, under constant load.

It was found that over the duration of a test (contact initiation takes place at roller-disk relative speeds corresponding to 800-900 rpm disk speeds, followed by a progressive increase of speed until scuffing occurs), the friction force has a relatively constant value, slightly increasing with speed and presents rapid increase corresponding, for the presented experimental conditions, to speeds around 2500 rpm. This threshold represents the moment when in the investigated contact scuffing occurs.

The results of the present investigations are similar in principle to results previously advanced in literature, [1-3], for tests conducted on two-disk machines. The shapes of obtained plots are very similar to those advanced in [1-2].

The conducted experimental investigations validate the proposed experimental setup. Load conditions, speed, temperature, lubricant parameters and surface roughness are essential elements that, together with the lubricating regime, can create the determining framework for scuffing phenomena.

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