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Study of Processes in Critical Operating Conditions of Sliding Pairs

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Preliminary note

The occurrence of working irregularities in journal bearing such as loss of geometry correctness, sudden failure of lubrication, overloading, etc., requires that serious damage of journal bearings have to be overcome in a short period of time. In such cases, it is important that the applied combinations of materials and lubricants do not have a tendency to seizure. Such critical conditions were simulated in this work for various combinations of materials and lubricants. Model research was carried out using the adapted Timken device and results were compared with the real loaded bearing. Data for nominal working conditions has shown relatively good agreement with calculated values whereas maximum frictional moment was significantly higher for the real bearing. This difference was explained as a consequence of different lubricating conditions. Nevertheless, a profile of the frictional moment was qualitatively similar compared to a model testing for equal combination of material and lubricant, thus confirming initial assumptions.

Istraživanje pojava pri kritičnim uvjetima rada kliznih parova

Prethodno priopćenje

Kod pojave nepravilnosti u radu kliznih parova kao što su gubljenje geometrijske točnosti, iznenadni otkaz podmazivanja, preopterećenost i slično, poželjno je da u nekom kratkom roku ne nastupi havarija kliznog para. U tom je slučaju nužno da primijenjene kombinacije materijala i maziva nemaju sklonost zaribavanju. U radu su simulirani takvi kritični slučajevi za različite kombinacije materijala i maziva. Istraživanje je vršeno modelno na prilagođenom Timken-uređaju te su dobiveni rezultati uspoređeni sa realnim dinamički opterećenim ležajem. Podaci za nominalno opterećenje pokazali su dosta dobro poklapanje, dok su za najveći moment trenja znatno veći kod realnog ležaja. Ovakva se razlika tumači kao posljedica različitih uvjeta podmazivanja kod modelnog i realnog ispitivanja. Ipak, dijagram promjene momenta trenja za istu kombinaciju materijala i maziva je kvalitativno sličan potvrđujući inicijalne pretpostavke za modelno ispitivanje.

1. Introduction

Sliding contact between two surfaces with less resistance calls for a multidisciplinary problem solving methods. Journal bearings and worm gears are the most common machine element representatives for such problems. It is known that steel, widely used in machine building, is harder compared to many other metals. Based on that and expected successful lubrication regime, one could assume an endless lifetime for journal bearing made of hard metal components. However, this combination of materials will inevitably result in the bearing breakdown in a very short operational time. Poor performance of

such combination of materials is commonly explained by interaction of influencing parameters (rotating speed, load, temperature, viscosity of lubricant etc.) and dynamics of change. Unfavourable combinations of some of those parameters affect lubrication, degrading it from full down to mixed or even boundary regime. There, in critical working conditions, a wear-out of components will occur.

Experience and results of scientific work have shown that hard/soft combination of materials is the best choice for loaded machine elements, where sliding movement

Symbols/Oznake

| | | | |
|-------|---|--------------------------|---|
| A_L | – cooling surface of bearing housing, m ² – rashladna površina kućišta ležaja | So | – Sommerfeld's number, - – Sommerfeldov broj |
| b | – with of bearing, mm – širina ležaja | ϑ_0 | – ambient temperature, °C – temperatura okoline |
| d | – nominal diameter, mm – nazivni promjer | ϑ_u | – temperature of lubricant film, °C – temperatura mazivog filma |
| F | – load carrying capacity, N – nosivost | v | – tangential speed, m/s – obodna brzina |
| h_0 | – thickness of lubricant film, mm – debljina mazivog filma | ε | – relative eccentricity, – relativni ekscentricitet |
| k_L | – coefficient of heat conduction, W/m ² K – koeficijent provođenja topline | η | – dynamic viscosity, Pa s – dinamički viskozitet |
| n | – rotating speed, 1/s – brzina vrtnje | μ | – friction coefficient, - – faktor trenja |
| p_s | – mean contact pressure, MPa – srednji dodirni pritisak | $\nu_{40^\circ\text{C}}$ | – kinematic viscosity at 40°C, mm ² /s – kinematski viskozitet pri 40°C |
| P_g | – power loss, W – gubitak snage | ω | – angular speed, rad/s – kutna brzina |
| R_a | – roughness average, m – srednje odstupanje profila | ψ | – relative clearance, % – relativna zračnost |
| R_m | – tensile strength, MPa – vlačna čvrstoća | | |

of contacting surfaces is significant compared to rolling. Investigation of such combinations of materials was performed in order to identify tendency to seizure and wear for few lubricating oils as the third component in the system. This work presents the results of the investigation.

2. Used component data

Common materials used for journal bearings shafts are steel for tempering or case hardening and for bearing bushes alloys with tin, copper and aluminium.

For shaft material steel DIN C45 was chosen, tempered to HB250 and ground to $R_a=0.4 \mu\text{m}$. Selection of materials for bearing bushes was based on literature dealing with similar problems [1], [2]. The properties of three selected materials are shown in Table 1.

Table 1. Characteristics of materials used in investigation

Tablica 1. Svojstva materijala korištenih u ispitivanju

| | R_m , MPa | E , MPa | ρ , kg/m ³ |
|-----------|-------------|-----------|----------------------------|
| CuSn12Ni2 | 330 | 88.300 | 8.800 |
| AlSn6 | 180 | 100.000 | 2.840 |
| PbSn80 | 80 | 56.000 | 7.500 |

Four different oils were used in experiments forming 12 results per oil and each load value. Since the goal of the investigation is to identify processes in critical operating conditions of sliding pairs, a poor lubrication regime should be applied. This is achieved by using a drop injection system that supplies oil in small amounts near the contact zone. In order to maintain constant temperature of the lubricant circular, a lubrication system with isolated reservoir and heating/cooling capability was established. Some basic properties of lubricating oils used in investigation are shown in Table 2.

Table 2. Properties of the used lubricants

Tablica 2. Svojstva korištenih maziva

| | v , mm ² /s | Flaming point, °C / Plamište | Viscosity Index / Indeks viskoznosti |
|--------------------|--------------------------|------------------------------|--------------------------------------|
| Mineral oil (MO1) | 19,3 / 100 °C | 225 | 100 |
| Mineral oil (MO2) | 18,2 / 100 °C | 210 | 95 |
| Synthetic oil (SO) | 24,5 / 100 °C | 277 | 100 |
| Diesel oil (D2) | 21,1 / 20 °C | 65 | - |

3. Description of the experiment

In order to speed up the investigation and cut expenses, a model approach to experiments is chosen while results would be tested and compared for real journal bearing. Each bearing bush material is represented by a specimen that was tested against a steel disk in the presence of a lubricant. All model experiments were performed on a device, developed according to Hunsicker and Kempf [3]. The acting principle of the device is shown in Figure 1.

A tendency to seizure is determined by monitoring the moment of friction against time as the quantitative ratio for material quality and applied lubricant. Run time of 1 minute is prescribed, based on experience in previous experiments [4]. The specimen is slanted for 5 degrees and loaded axially with force F against steel disk. Size and slant of the specimen are chosen to simulate a state that could arise in critical working conditions of journal bearing.

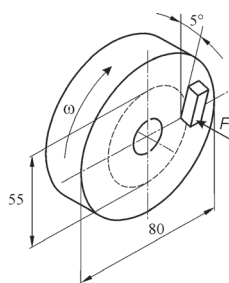


Figure 1. Loading the specimen (5x5x13) against rotating disk

Slika 1. Princip opterećivanja ispitne epruvete (5x5x13) na rotirajući disk

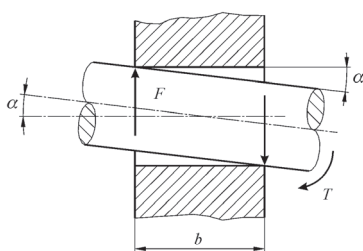


Figure 2. Determining slant angle for the model investigation

Slika 2. Određivanje kuta nagiba za modelno ispitivanje

For some unfavourable ratio b/d and large clearance in journal bearing, slant angle of up to 5 degrees could be set, which is adopted to be the slant angle between the specimen and the disk.

Due to slant of the loaded shaft, the edge pressure rises on both ends of the bearing inducing more traction and higher friction moment. A theoretical profile of such pressure is shown in Figure 3 [5]. It is obvious that pressure reaches its maximum value on the contacting edge between bearing bush and the shaft, hence a rupture of the lubrication film is more probable. An angular position of the maximum pressure depends on the rotating speed and load introduced to the bearing.

Low load frequencies were simulated by cam mechanism and set with appropriate transmission ratio to 3 Hz. In between those oscillations, the load changed from zero to a maximum defined value.

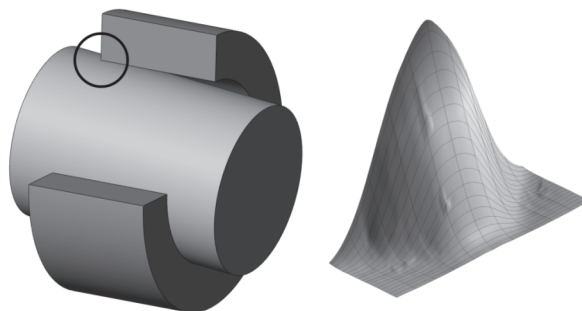


Figure 3. Edge pressure profile due to slant of the shaft
Slika 3. Profil rubnog pritiska uslijed nagiba vratila

For each presented result (Table 3), three specimens were tested. In the case of large data scattering, the experiment was repeated with five specimens.

During the experiment, sliding speed and lubricant temperature were kept constant for each test load.

In order to isolate sliding properties of the material itself, some specimens were lubricated with diesel oil (D2).

The monitoring of moment of friction was performed through a strain gauge sensor and measurement amplifier and the data was stored on a personal computer.

Theoretical contact between the disk and the specimen at the beginning of the experiment is line contact. As wear increases, the surface of contact increases as well. When seizure occurs, the moment of friction undergoes rapid changes (Figure 5). If the moment of friction decreases with increasing load, one could conclude that tested material has a low tendency to seizure.

Journal bearing testing

Based on the data gathered during model experiments performed on the adapted Timken device the best combination is selected for testing with real journal bearing. Basic parameters were calculated prior to testing by using computer program [6] based on standardized calculation procedure [7], [8]. Those parameters, involving geometry of bearing, properties of components and aimed working conditions, are as follows:

- | | |
|----------------------------|------------------------------|
| 1. Nominal diameter | $d = 37 \text{ mm}$ |
| 2. Diameter/with ratio | $d/b = 0,75 -$ |
| 3. With of bearing | $b = 27,75 \text{ mm}$ |
| 4. Relative clearance | $\psi = 1 \text{ ‰}$ |
| 5. Relative eccentricity | $\epsilon = 0,388 -$ |
| 6. Mean contact pressure | $p_s = 1 \text{ MPa}$ |
| 7. Angular speed | $\omega = 100 \text{ rad/s}$ |
| 8. Rotating speed | $n = 955 \text{ rps}$ |
| 9. Tangential speed | $v = 1,85 \text{ m/s}$ |
| 10. Load carrying capacity | $F = 1,026 \text{ kN}$ |

| | |
|--|---|
| 11. Cooling surface ratio | $A_L/(l \cdot d) = 25$ - |
| 12. Cooling surface of bearing housing | $A_L = 0,02567 \text{ m}^2$ |
| 13. Coefficient of heat conduction | $k_L = 20 \text{ W/m}^2\text{K}$ |
| 14. Ambient temperature | $\vartheta_0 = 30 \text{ }^\circ\text{C}$ |
| 15. Lubricant oil ISO VG 100 | $\nu_{40^\circ\text{C}} = 100 \text{ mm}^2/\text{s}$ $\eta = 0,0291 \text{ Pa}\cdot\text{s}$ |
| 16. Thickness of lubricant film | $h_0 = 11,3 \text{ }\mu\text{m}$ |
| 17. Temperature of lubricant film | $\vartheta_u = 64 \text{ }^\circ\text{C}$ |
| 18. Sommerfeld's number | $So = 0,344$ - |
| 19. Friction coefficient | $\mu = 0,009$ - |
| 20. Power loss | $P_g = 17 \text{ W}$ |

Testing of the journal bearing is performed on a device designed and produced [9] for that purpose (Figure 4). Its components are measured shaft, tested journal bearing, loaded part as well as measuring device and supporting instruments. A radial load is measured directly as the dynamometer is placed between the bearing and the load. Friction moment is measured on the angular shift of the bearing bush. Additional moment that induces edge pressure simulating shaft slant in the bearing is set with appropriate bar loaded via second dynamometer. The temperature is measured by use of contacting thermometer with thermocouple.

Data gathered for nominal working conditions has shown relatively good agreement with calculated values. For half of the nominal load measured temperature

was close to theoretical ($58 \text{ }^\circ\text{C}$) despite relatively poor lubrication regime. After a period of run-in (approximately three hours with 33 % and five hours with 50 % of the nominal load) of the device and the bearing, bearing bush control was performed with no damage of any kind found. Testing was continued with remounting of the bearing and introducing additional moment acting in the plane perpendicular to the one of friction moment. The amount of the moment was such as to induce specific load size at the contact zone (see input and calculated values). As for model testing, the duration of run was 1 minute but without circular lubrication. Based on three runs, the following results were gained:

- Maximum measured temperature on outer side of bearing bush was $85 \text{ }^\circ\text{C}$ and
- Maximum friction moment was 11.1 Nm.

The profile of maximum friction moment is of one streaming toward some constant value. Qualitatively, it looks similar to the result of the model testing for corresponding conditions (equal material and lubricant). Absolute value of the moment for real journal bearing testing is more over those of model testing (see table 3). It is surely a consequence of different lubricating conditions. However, the moment profile is such that confirms assumptions stated on the basis of model testing. Unfavourable combinations of materials and lubricants that were not tested and compared could perhaps fulfill a whole image of the investigation.

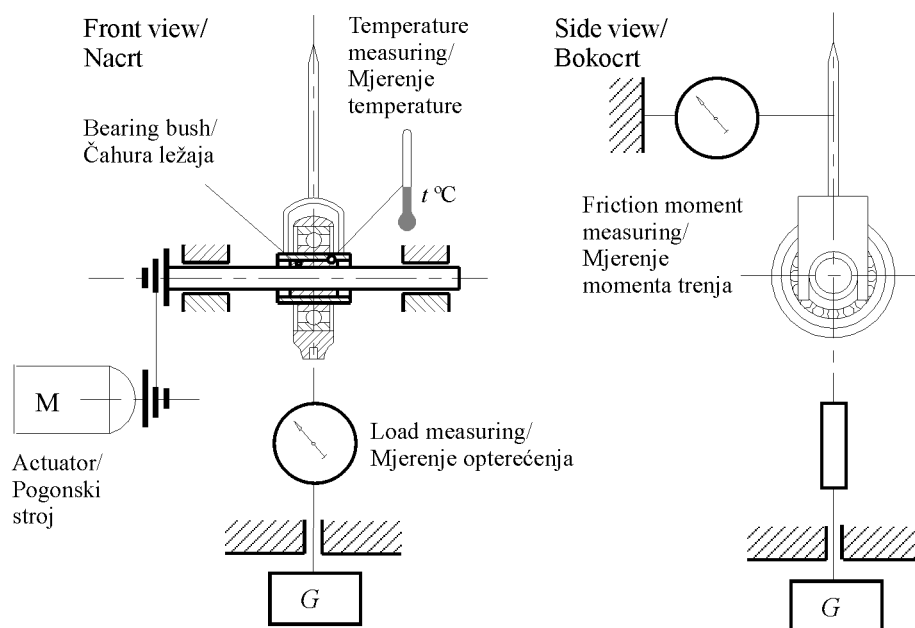


Figure 4. Schematic review of the device for investigating sliding bearings

Slika 4. Shematski prikaz ispitnog uređaja za klizne ležaje

4. Testing results

Testing was performed for every combination of bearing bush material, lubricant and load. Results shown in Table 3 represent mean value for three specimen of each bush material and both constant and dynamically changing load. Non-significant data as well as extreme values are not included.

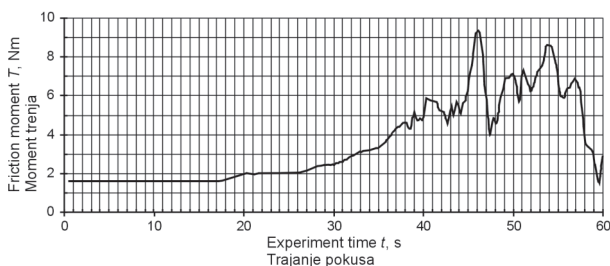


Figure 5. Aspect of the data registration with the appearance of seizure

Slika 5. Zapis promjene momenta trenja pri pojavi zaribanja

Profile of friction moment for one combination of bush material and lubricant is shown in figure 5. In the range of lubrication seizure, oscillations of the friction moment are obvious. Values of friction moment among all combinations of material, lubricant and load are expectedly lowest for the synthetic lubricant.

For lubricants with a mineral base, friction moment varies for higher loads depending on selection of bearing bush material.

5. Conclusion

Based on performed tests, the following conclusion can be made:

- As the load grows, the difference between friction moments is bigger on account of synthetic oil (SO) for all tested combinations,
- There is a small friction moment difference at higher loads between mineral base oils on account of oil 2 (MO2) for all tested combinations,
- Tendency to seizure is smaller for all combinations of material and load if lubricated with synthetic oil (SO)
- Tendency to seizure is smaller for all combinations of material and load if lubricated with mineral base oil 2 (MO2) compared to oil 1 (MO1),
- When lubricating with diesel oil (D2) friction moment is the highest for material CuSn12 compared to others,
- Wear of bearing bush is proportional to material hardness and friction moment.

The smallest difference in value of friction moment regarding bush material and load for synthetic lubricant against other tested oils could be explained through the lubricant properties. Since synthetic oils usually have extreme pressure additives (EP) their performance is overall better and boosting higher temperatures [10]. On the other hand, change of lubricant viscosity with temperature and pressure affects thickness of lubricating film where higher viscosity results in a thicker film [11]. Since the change of viscosity is commonly smaller for synthetic compared to mineral oils it is reasonable to have smaller variations of friction moment.

This work presents a comparison of common bearing bush materials and journal bearing lubricants. Continuance of this work includes more different loads as well as various sliding speeds. The aim of the work is to show that economically effective substitutions of materials and lubricants could be made.

Table 3. Friction moment values in Nm at the end of experiment

Tablica 3. Momenti trenja na kraju pokusa iskazani u Nm

| | AlSn6 | CuSn12 | PbSn80 |
|------------------|-------|--------|--------|
| <i>F</i> =400 N | | | |
| MO1 | 0,9 | 1,5 | 4,0 |
| MO2 | 0,9 | 1,2 | 1,5 |
| SO | 0,7 | 0,7 | 1,0 |
| D2 | 2,4 | 6,6 | 6,5 |
| <i>F</i> =1200 N | | | |
| MO1 | 2,4 | 2,7 | 6,0 |
| MO2 | 2,1 | 2,3 | 2,3 |
| SO | 1,2 | 1,7 | 1,7 |
| D2 | - | 11,9 | 8,5 |
| <i>F</i> =2000 N | | | |
| MO1 | 4,4 | 3,8 | 7,9 |
| MO2 | 3,6 | 3,2 | 3,9 |
| SO | 0,8 | 1,5 | 1,9 |
| D2 | 5,0 | 18 | 10 |
| <i>F</i> =2800 N | | | |
| MO1 | - | - | 8,5 |
| MO2 | - | - | 7,0 |
| SO | 1,5 | 1,2 | 1,8 |
| D2 | - | - | 16 |

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