Research of journal bearings for using in agricultural mobile machines

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Abstract

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The competitive environment forces producers in agricultural machine industry to decrease the costs. Producers as well as sub-suppliers need to find possible savings. The paper presents results of laboratory experiments with real journal bearings made of bimetallic alloy performed to find out possible replacement of a rolling bearing by a journal bearing. An important correlation between the results of laboratory experiments with a model of tribological system and the real journal node can be achieved by a maximum approach of simulation features by real running conditions. Thus, the given experiment conditions result from the chosen application, i.e. a steering servo unit in mobile machines. The experiments were performed on the Tribotestor M'06" testing machine.

Keywords: bearings; bimetallic alloy; tribological experiment; Tribotestor M'06"

Nowadays, the experimental determining of tribological features is performed via devices with different configurations. It is very common that experiment parameters are always chosen based on needs and demands. Each experiment is influenced by several factors, whereas the weight of factors is different, and each of them is determined to solve a partial tribological task (Mang, Dresel 2001; Bayer 2004; Ünlü, Atik 2007; Kučera, Pršan 2008a). The data obtained from experiments have an important influence on interpreting the results where friction and wear are measured. The development of microtribology and nanotribology influences the parameters of experimental testing devices. There is a trend of using devices with low surface speed and low load. In most situations, the real friction node is replaced by a line contact or a spot contact; however, the reached friction coefficient cannot be compared with values of real journal nodes (Žiačik et al. 1995; Kučera 2008). There also exist minimum experimental devices which are able to perform an experiment with real journal node during real running conditions as they are usually used to provide durability tests (Bayer 2004; Kučera, Pršan 2008b; Kadnár et al. 2011). According to the simple design of existing devices, the minimum possibility of changing the parameters during the experiment may be seen as their disadvantage.

MATERIAL AND METHODS

Test Rig. In experiments with real journal bearings, the Tribotestor M'06 testing machine (Slovak University of Agriculture in Nitra, Nitra, Slovak Republic) allows:

– to use the most modern measuring technology,
– to influence the range of running parameters,
– to provide modification easily,
– to provide several lubrication modes, etc.

When the journal node is loaded by normal force F_N, there is friction between the shaft and bearing, represented by friction force F_T effecting the rotation movement of the shaft. Friction also causes the transmission of torsion moment to the head of the testing machine (Fig. 1). The transmitted torsion moment is defined by friction force $F'_T = F_T$ and the radius (diameter) of the shaft r_H.

The friction coefficient is determined by the formula:

$$\mu = \frac{F_t a}{F_N r_H} \quad (1)$$

where:

F_t – force at the end of teasting head's arm (N)
a – length of testing head's arm (mm)
F_N – normal force (N)
r_H – radius (diameter) of the testing shaft (mm)

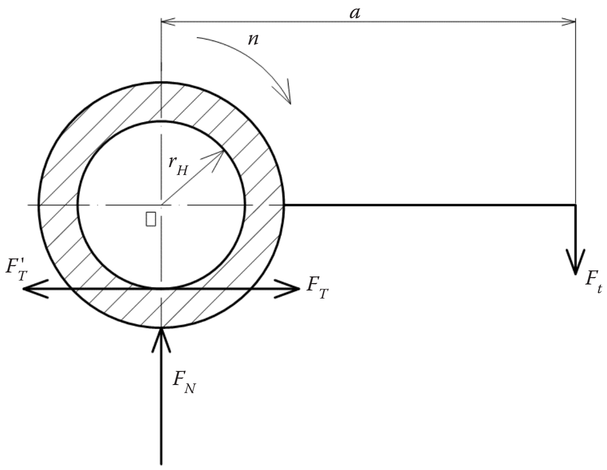

Fig. 1. Diagram of Tribotestor M'06 testing machine

F'_T – transmitted torsion moment force; F_N – normal load; F_T – friction force; F_t – force at the end of teasting head's arm; n – revolutions; a – length of testing head's arm, r_H – radius of the testing shaft

Material. For the producer of the steering servo unit, the experiments with different kinds of journal bearings were elaborated for the purpose of replacing the rolling bearing by the journal bearing. The reason of the purpose was an expected saving. Bimetallic bearings are made by curling bimetallic strips with different sliding materials. The active layer is represented by a sliding material which is coated on a steel base in the form of powder and is compressed by rolling. The smooth structure is considered to be the main advantage because the bearings of these materials are usable also within the critical friction. The structure of the materials is illustrated in Table 1.

The B10 material is a metal-polymer composite material with excellent friction features also without lubrication. The required shaft roughness cannot exceed 0.4 µm, the shaft hardness must be over 200 HB. B30 is a bimetallic material with bronze alloy (both Kajo Metal, Dolní Kubín, Slovak Republic). The required shaft features are the same as for the B10 material. The basic features of the B10 and B30 materials are illustrated in Table 2.

Experiments. The following conditions were followed during the experiments for friction measurement:

– shaft of Φ 10, cemented, hardened and edged – material ČSN 14 220/EN 16MnCr5 – common material for the selected application (each shaft used only for one measurement),
– bearing clearance of 0.02 mm,
– six tested samples of each materials,
– no lubrication.

Kadnár (2009) illustrates the selection of experiment parameters. Based on the parameters, the complex mode of the experiment for friction measurement was determined. The mode included several partial phases which are illustrated in Table 3. Before the experiment, each sliding node was exposed by a test run of 600 s with the revolutions of 2,000 min^{-1} and load of 150 N. The phase was considered to be a preparatory phase, and reached results are not taken into consideration further.

Table 1. Chemical structure of materials (% wt.)

Material	Cu	Pb	Sn	Zn	P	Fe	Ni	Sb	Other
B10 CuPb10Sn10	rest	9–11	9–11	≤ 0.5	≤ 0.1	≤ 0.7	≤ 0.5	≤ 0.2	≤ 0.5
B30 CuPb30	rest	26–33	≤ 0.5	≤ 0.5	≤ 0.1	≤ 0.7	≤ 0.5	≤ 0.2	≤ 0.5

Table 2. Basic features of B10 and B30 materials

Material	Chemical structure	Tensile strength (MPa)	Max. load in static stress (MPa)	Max. load in dynamic stress (MPa)	Max. operation temp. (°C)
B10	CuPb10Sn10	230–280	200	120	250
B30	CuPb30	90–107	120	40	160

Table 3. Phases of the experiment for friction measurement

Phase	Time (s)	Duration (s)	Load (N)	Revolutions (rpm)
Sliding node stabilisation	0	20	20	2,000
Measurement with constant speed	20	120	100	2,000
	140	120	150	2,000
	260	120	200	2,000
	380	120	250	2,000
Sliding node stabilisation	500	20	20	4,000
Measurement with constant speed	520	120	150	4,000
	640	120	150	500
Supporting measurement for checking of measuring device	760	30	100	0
	790	30	150	0
	820	30	200	0
	850	30	250	0

After the test run, each node was stabilised, i.e. loaded by 20 N with the revolutions of 2,000 min^{-1}. Consequently, the measurement with the revolutions of 2,000 min^{-1} was performed with the load of 100, 150, 200 and 250 N. Each measurement lasted 120 seconds. Before the measurement with constant load, each node undertook another stabilisation which lasted 20 s loaded by 20 N with the revolutions of 4,000 min^{-1}.

The measurement with constant load was undertaken with the load of 150 N and revolutions of 4,000 min^{-1} or 500 min^{-1}. Both measurements lasted 120 s. The diagram of load and rotational frequency depending on time is illustrated in Fig. 2.

There are many statistical interpretations of friction measurement. For an application in tribology, table data are used rather than additional information (Bhushan 2002; Bayer 2004; Ünlü, Atik 2009; Kadnár et al. 2011). The most important information is that in the unfiltered record during friction force measurement, the measured data generally reflect the reality in a tribological node. Further processed information in a table or figure is only its interpretation. Thus, we have decided for a compromise, i.e. for a figure interpretation with illustrating the average value of measurement and the statistical interval of 95%. The idea is supported by the fact that the total friction coefficient is not

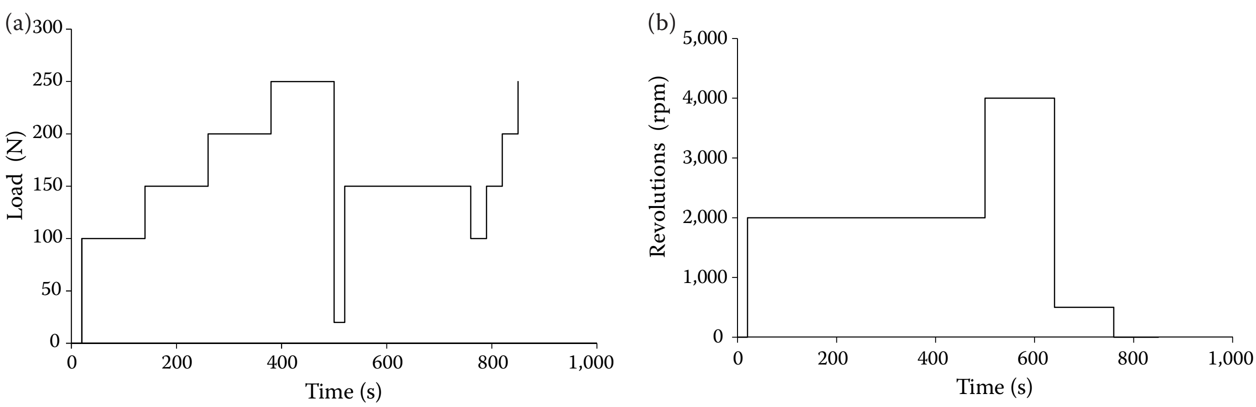

Fig. 2. Load (a) and rotational frequency (b) depending on time

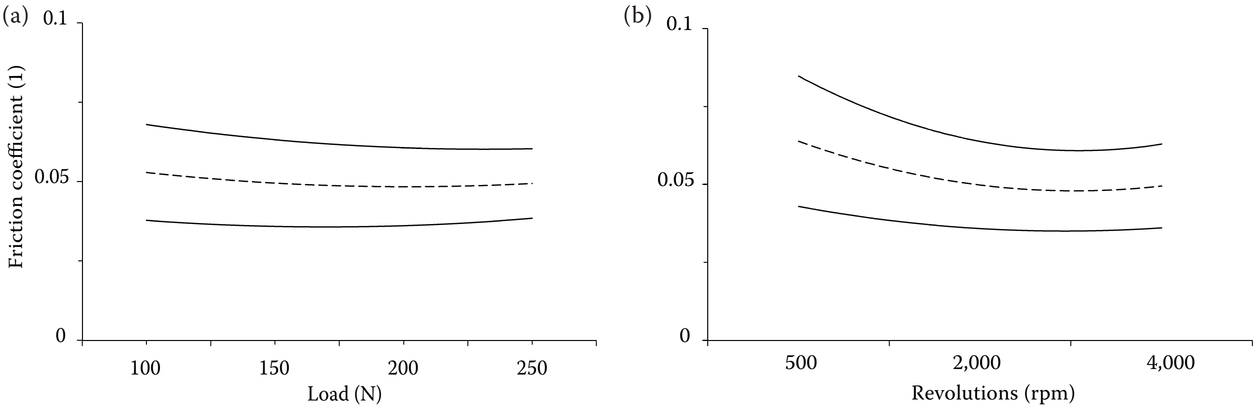

Fig. 3. B10 material – friction coefficient depending (a) on load and (b) on rotational frequency with 95% interval of confidence

a measured value but a calculated one. At the same time, rotational frequency was used rather than the surface speed concerning the features of the steering servo unit (Páltik et al. 2007; Poničan, Korenko 2008).

RESULTS AND DISCUSSION

Within the sliding node, the B10 material had stable features in connection with variable load or variable rotational frequency. No important vibrations were recorded during test run and experiments themselves. The diagram of friction coefficient within constant rotational frequency, i.e. depending on variable load, is illustrated in Fig. 3a.

At the load of 100 N, the friction coefficient was 0.05. When decreasing the load, there were only little differences in friction coefficient. The temperature was practically the same, i.e. 42°C. At the load of 250 N, the friction coefficient was 0.05. Thus, the sliding node of B10 material can be evaluated to be favourable. At the load of 150 N, there was only a little decrease in friction coefficient.

At the rotational frequency of 500 min^{-1}, the lubrication mode can be considered to be mixed. However, the friction coefficient did not exceed the limit of 0.1. During the experiment, the friction coefficient ranged from 0.06 to 0.07. The diagram of friction coefficient with constant load and variable rotational frequency is illustrated in Fig. 3b.

At the frequency of 4,000 min^{-1}, the friction coefficient decreased to 0.05. Thus, it is possible to conclude that the sliding node of B10 material is considered to have stable features. Despite the conclusion, the bearing surface had little wear, a local wear of thin film, i.e. a sliding layer on the surface of the material (Fig. 4).

Regarding the surface of the journal bearing of B10 material, the separation of surface was recorded. In some parts, there was a visible subsurface layer of sintered bronze. Based on the results, the wear resistance of B10 material towards the chosen application is not sufficient. The weight loss of B10 material ranged from 9 to 14 mg (Fig. 5).

The test run or experiments themselves when using the B30 material recorded more important vibrations. Fig. 6a illustrates the diagram of fric-

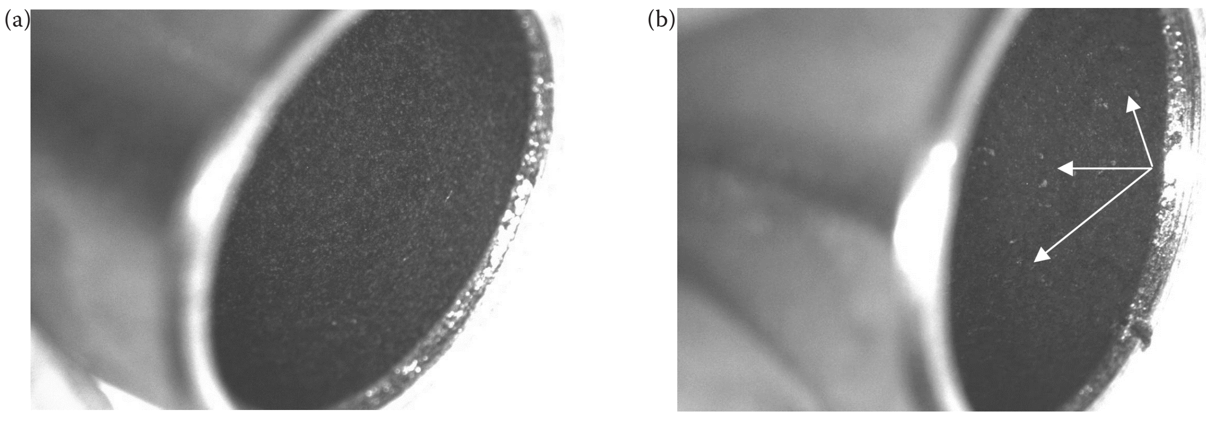

Fig. 4. B10 material – surface before (a) and after (b) the measurement

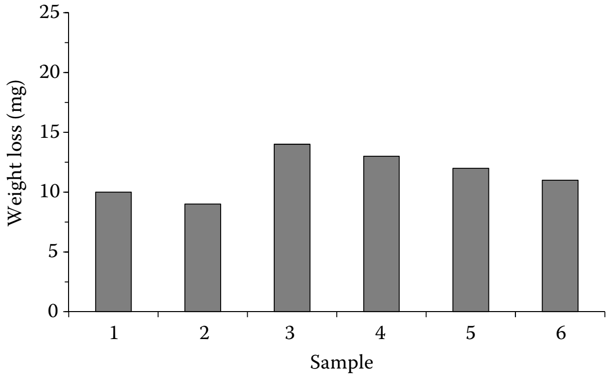

Fig. 5. B10 material – weight loss

tion coefficient depending on variable load. At the load from 100 to 200 N, the friction coefficient was 0.08. At the load of 250 N, the friction coefficient decreased to 0.07. The temperature was not more than 43°C.

At the load of 150 N and lower frequencies, the friction coefficient decreased to 0.10. It resulted from conditions in the sliding node which corresponds to the area of mixed friction.

The diagram of friction coefficient with constant load and variable frequency is illustrated in Fig. 6b. At the frequency of 4,000 min^{-1}, the friction coefficient decreased to 0.08.

According to the high value of friction coefficient, the B30 material is considered to be less favourable. The surface of sliding material had local wear (Fig. 7). A higher rate of noise and vibrations were also recorded. The sliding node had only average features regarding friction and wear, and therefore the bearings after the test can be considered to be damaged. Based on the experiment results, the wear resistance of B30 material is evaluated to be not sufficient.

The weight loss within the B30 material ranged from 11 to 16 mg (Fig. 8).

CONSLUSION

As the literature confirms, high values of friction coefficient were recorded within dry friction. The tested bearings were stable depending on load as

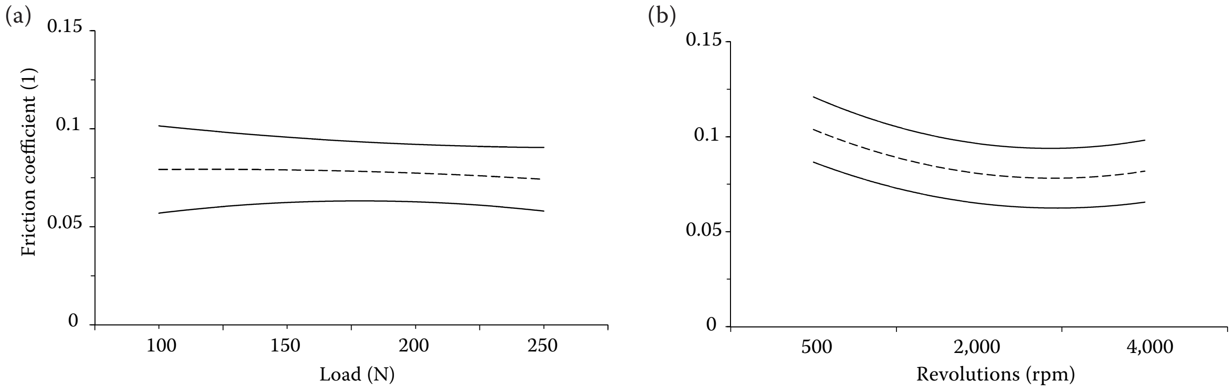

Fig. 6. B30 material – (a) friction coefficient depending on load and (b) on frequency with 95% interval of confidence

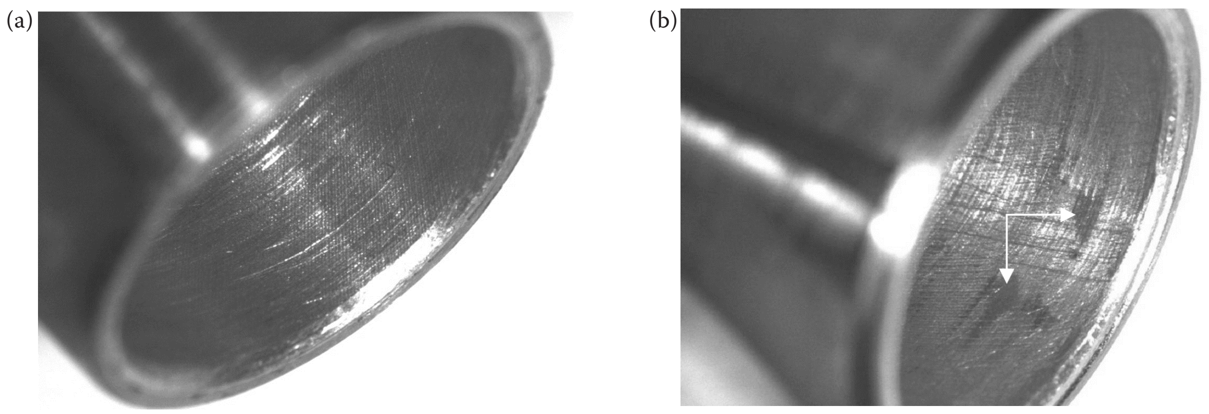

Fig. 7. B30 material – surface before (a) and after (b) the measurement

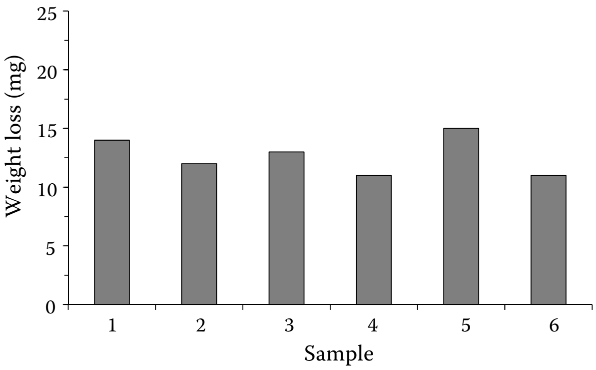

Fig. 8. B30 material – weight loss

well as frequency. For the chosen application, the tested bearings are considered not to be suitable. In the future, it is possible to verify tribological features of tested bearings also within hydrodynamic friction, whereas the structure of the testing head allows the circulation of lubricant and an additional influence of sliding node temperature, which helps to simulate real conditions better.

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