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TRIBOLOGICAL EXPERIMENTS IN AUTOMOBILE INDUSTRY

Конкурентоспроможне середовище змушує виробників автомобільної промисловості знижувати витрати. Виробникам, а також субпостачальникам необхідно знайти можливість для економії. У статті представлені результати лабораторних експериментів з реальними підшипниками ковзання з біметалічного сплаву, виконаними для визначення можливої заміни підшипника кочення підшипником ковзання. Важлива кореляція між результатами лабораторних експериментів з моделлю трибологічної системи і реальним вузлом цапфи може бути досягнута шляхом максимального підходу до симуляції функцій шляхом повторних умов роботи. Таким чином, дані умови експерименту є результатом моделювання конкретного вузла, а саме блоку сервоприводу рульового управління. Експерименти були виконані на випробувальній машині Tribotester M'06.

Ключові слова: трибологія, триботестор, підшипник ковзання, підсилювач рульового управління.

Конкурентоспособная среда вынуждает производителей автомобильной промышленности снижать затраты. Производителям, а также субпоставщикам необходимо найти возможность для экономии. В статье представлены результаты лабораторных экспериментов с реальными подшипниками скольжения из биметаллического сплава, выполненными для определения возможной замены подшипника качения подшипником скольжения. Важная корреляция между результатами лабораторных экспериментов с моделью трибологической системы и реальным узлом цапфы может быть достигнута путем максимального подхода к симуляционным функциям путем повторных условий работы. Таким образом, данные условия эксперимента являются результатом моделирования конкретного узла, то есть блока сервопривода рулевого управления. Эксперименты были выполнены на испытательной машине Tribotestor M'06.

Ключевые слова: трибология, триботестор, подшипник скольжения, усилитель рулевого управления.

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

Keywords: tribology, tribotestor, sliding bearing, steering booster.

Introduction. Nowadays, the experimental determining of tribological features is realised via devices with different configurations. It is very common that the experiment parameters are always chosen based on the 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. The data reached from the experiments have important influence for interpreting 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 loading on few Newtons. 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. There also exist minimum experimental devices which are able to realise an experiment with real journal node during real running conditions as they are usually used to provide durability tests. 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. Tribological experiments were performed using Tribotestor M'06 (fig. 1). This universal test device enables performing various types of tests such as limit load tests (seizing test), limit velocity tests (velocity seizing test), capacity tests for the determination of pv diagram and life (durability) tests.

The test device consists of three main parts. The test part itself is made of the drive units of samples rotational motion, vertical loading force and measuring head. Another part contains the elements of a pneumatic circuit and all of the electronic devices. The last part is a control-evaluating unit in the form of a connected desktop computer. Starting, check, control, data collection and test evaluation itself are performed on the connected computer.

Our experiment was performed through the seizing test.

The fig. 1 shows the measuring device with the detailed view on the measuring head and test sample.

The couple of bodies, creating a planar contact by rotating of the test shaft against sliding bearing, forms the sliding node. The sliding bearing marked as B10 (and B30) was moulded into the measuring head.

For the producer of a steering servo unit, the experiments with different kinds of journal bearings were elaborated for the purpose of replacing a rolling bearing by a 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.

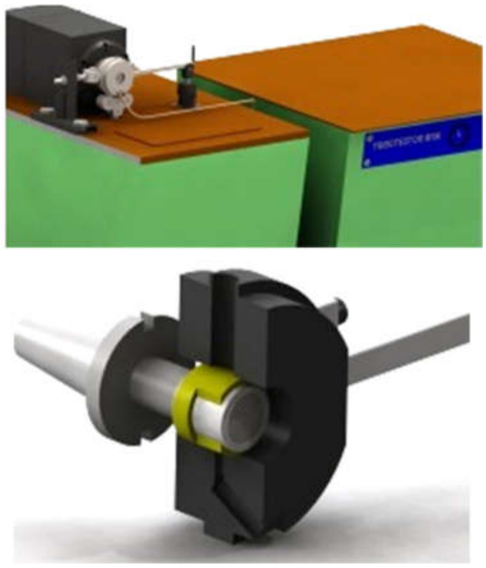


Fig. 1 – Measuring device Tribotestor M'06 and a detailed view of the measuring head and test samples

Table 1 – Chemical structure of materials (weight percentage)

Material	Cu	Pb	Sn	Zn	P	Fe	Ni	Sb	Others
B10	rest	9-11	9-11	≤0.5	≤0.1	≤0.7	≤0.5	≤0.2	≤ 0.5
B30	rest	26-33	≤0.5	≤0.5	≤0.1	≤0.7	≤0.5	≤0.2	≤ 0.5

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. The required shaft features are the same as for B10 material. The basic features of B10 and B30 materials are illustrated in table 2.

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

- shaft of Φ 10, cemented, hardened and edged – material ČSN 14 220 (each shaft used only for one measurement);
- bearing clearance of 0,02 mm;
- six tested samples of each materials;
- no lubrication.

Table 2 – Basic features of B10 and B30 materials

Material	Chemical structure	Tensile strength, MPa	Maximum load in static stress, MPa	Maximum load in dynamic stress, MPa	Maximum operation temperature, °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	2000
Measurement with constant speed	20	120	100	2000
	140	120	150	2000
	260	120	200	2000
	380	120	250	2000
Sliding node stabilisation	500	20	20	4000
Measurement with constant speed	520	120	150	4000
	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

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 test run of 600 s with revolutions of 2000 min^{-1} and load of 150 N. The phase was considered to be a preparatory phase and the reached results are not taken into consideration further. After the test run each node was stabilised, i.e. loaded by 20 N with revolutions of 2000 min^{-1}.

Consequently, the measurement with revolutions of 2000 min^{-1} was realised with load of 100 N, 150 N, 200 N and 250 N. Each measurement lasted 120 s. Before measurement with constant load each node undertook another stabilisation which lasted 20 s loaded by 20 N with revolutions of 4000 min^{-1}.

The measurement with constant load was undertaken with load of 150 N and revolution of 4000 min^{-1} or 500 min^{-1}. The both measurements lasted 120 s. The diagram of load and rotational frequency depending on time is illustrated in fig. 2.

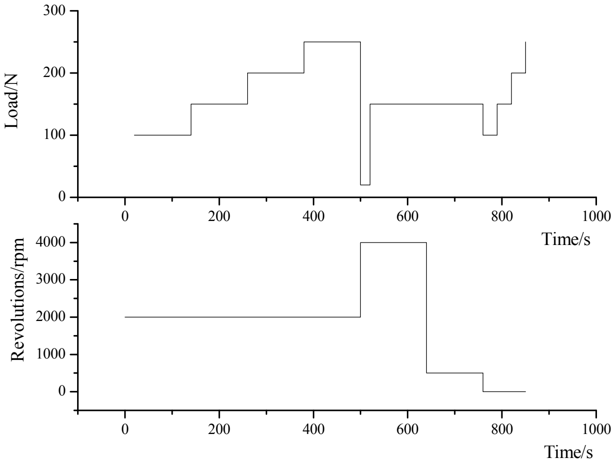


Fig. 2 – Load and rotational frequency depending on time

There are many statistical interpretations of friction measurement. For an application in tribology, the table data is used rather as additional information. The most important information is that in the unfiltered record during friction force measurement. Generally, the measured data reflects the reality in a tribological node. Further processed information in a table or a figure is only its interpretation. Thus, we have decided for a compromise, i.e. for a figure interpretation with illustrating average value of measurement and statistical interval of 95 %. The idea is supported by the fact that the total friction coefficient is not a measured value but a calculated one. At the same time, the rotational frequency was used rather than surface speed concerning the features of steering servo unit.

Results and Discussion. Within a sliding node B10 material had stable features in connection with the 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. 3.

At the load of 100 N, the friction coefficient was 0,05. When decreasing the load, there were only little differences of 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 of 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. 4.

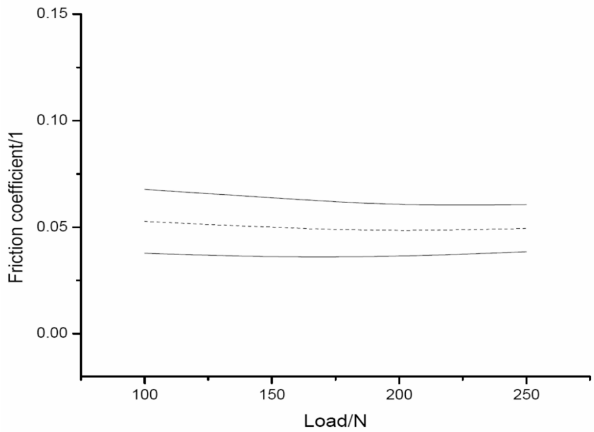


Fig. 3 – B10 material – friction coefficient depending on load

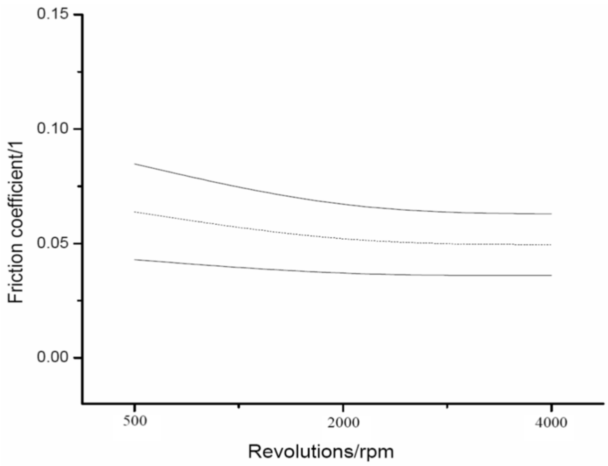
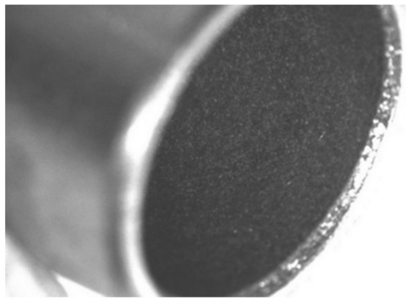
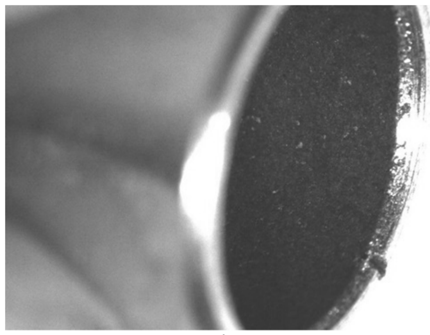


Fig. 4 – B10 material – friction coefficient depending on rotational frequency

At the frequency of 4000 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, local wear of thin film, i.e. sliding layer on a surface of the material (fig. 5).



a



b

Fig. 5 – B10 material – surface before (*a*) and after (*b*) the measurement

Regarding the surface of 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. 6).

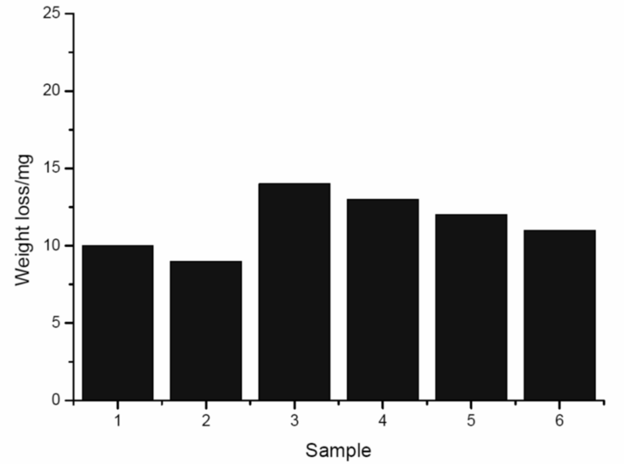


Fig. 6 – B10 material – weight loss

The test run or experiments themselves when using B30 material recorded more important vibrations. Fig. 7 illustrates the diagram of friction 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.

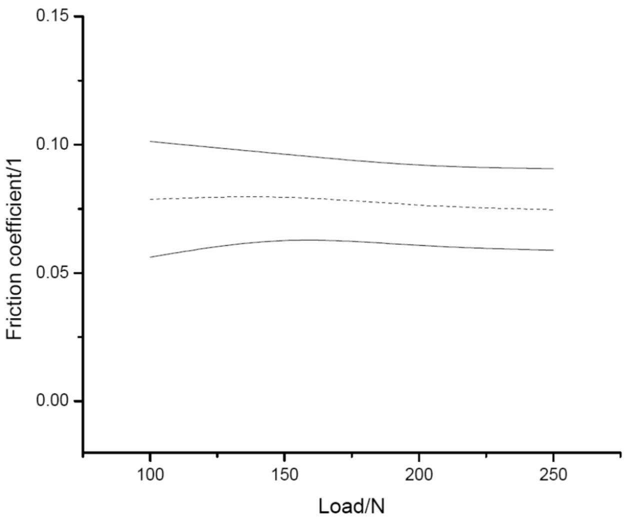


Fig. 7 – B30 material – friction coefficient depending on load

At the load of 150 N and lower frequencies the friction coefficient decreased to 0,10. It resulted from conditions in 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. 8. At the frequency of 4000 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. 9).

The 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 B30 material ranged from 11 to 16 mg (fig. 10).

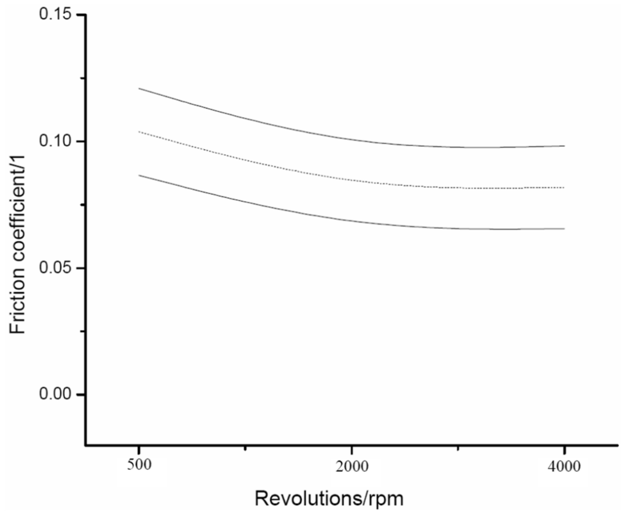
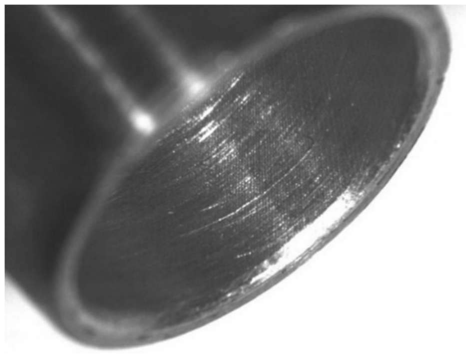


Fig. 8 – B30 material – friction coefficient depending on frequency



a

b

Fig. 9 – B30 material – surface before (*a*) and after (*b*) the measurement

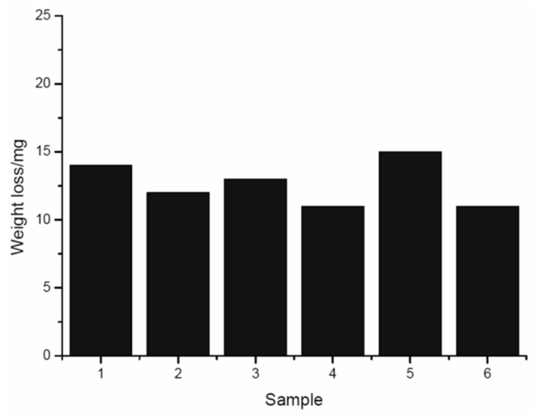


Fig. 10 – B30 material – weight loss

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

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