BEARING TESTING MACHINE WITH ROTATING LOAD VECTOR

Jan Sikora

Mechanical Engineering Faculty
Technical University of Gdańsk, Narutowicza Str. 11/12, 80-952 Gdańsk
Phone: +48 58 347 1844, Fax: +48 58 347 2742, e-mail: jsikora@pg.gda.pl

Abstract

A concept of new test stand for investigating fatigue strength of plain bearing surface layer has been presented in the paper. The machine with rotating load vector was designed, built and experimentally verified in tribological laboratory of the Technical University of Gdańsk. The machine of such a principle of slide bearing surface layer load generation is, according to ISO 7905 standard, a basic device to experimental evaluation of fatigue strength of bearing materials. Advantages and drawbacks of the test rig have been analysed.

1. Introduction

Mechanism of fatigue crack generation and propagation in dynamically loaded bearing surface layer has not been fully explained yet. A main source of information on bearing dynamic load-carrying capacity are results of experimental investigation of bearing bushes in test rigs under various operating conditions.

The wide range of practical application of plain bearings with different requirements led to the design of numerous bearing fatigue testing rigs. The testing machines in use differ mainly in the way how the dynamic load is produced. Two methods are usually applied in fatigue investigations:

a) loading with unbalanced masses acting on journal or housing of tested bearings,

b) loading with hydraulically generated forces.

Both methods have some advantages and drawbacks so research centres are usually equipped with both types of testing devices.

Regarding load generation principles all the rigs may be reduced to three essential schemes shown in fig. 1. These different principles of operation result in specific patterns of fatigue forcing exerted on the investigated objects.

A variant in fig. 1a is the test rig where a constant load vector rotates over the bearing circumference. This produces a pulsating load for each point of the bearing, theoretically equal for each line of the bearing circumference.

In the machine in fig. 1b, investigated bearings are located in big ends of perpendicular connecting rods and loaded only with the force component acting along connecting rod axis. Hence a rotating mass produces a bearing load of sine wave shape. The essential sinusoidal shape of load can not be modified for the lack of mechanism to change a load gradient. A machine shaft is usually relatively flexible and its deflection during testing may be considerable. The shaft deflection forces a maximum pressure at the edge of the bush, which often led to rupture of oil film continuity and to seizure of the bearing before fatigue damage is reached.

1) Sponsored in the research project (grant) no 7 T07B 033 19 KBN.
In the test rig in fig. 1c, the bearing load is produced hydraulically, either by valve governed pulse cylinder or by small eccentric movement of the test pin causing a pressure build-up in the hydraulic cylinder. Valves or other steering devices control the magnitude of load and its gradient. Thus it is possible to control two main parameters of bearing load - a maximum value of oil pressure and a speed of its change - to a large extent independently of the shaft rotational speed.

![Fig. 1. Principles of operation of plain bearing fatigue test rigs; 1 - tested bearing, 2 - unbalanced masses, 3 - eccentric journal of the shaft, 4 - piston, 5 - connecting rod](image)

The last two types of rig produce pulsating or alternating load vectors of constant direction. The position of the minimum film gap and of the oil pressure peak moves only over a narrow sector of the bearing. Pressure in oil film is produced by both rotation and displacement of the shaft.

For the systems shown in fig. 1, due to the different load conditions, the significant operating parameters of the test bearings vary widely for the same maximum specific load and identical bearing design features. A comparative computational analysis of these parameters has been made in [1].

In the paper a design of the bearing fatigue test stand based on the concept of machine with rotating load vector has been presented. It is the first testing device of that kind in our country. The machine was designed, built and experimentally verified in tribological laboratory of the Technical University of Gdańsk.
2. Design concept of the test rig

A scheme of the testing head unit of the MWO machine with rotating load vector is shown in the fig. 2 and general view of the whole test rig - in fig. 3.

Fig. 2. Scheme of head unit of the MWO machine with rotating load vector; 1 - investigated bearing, 2 - shaft, 3 - unbalanced masses, 4 - bearing housing, 5 - Cardan universal joint, 6 - stabilising bar, 7 - flexible joint, 8 - ball joint

Fig. 3. General view of the MWO machine
Two model bearing 1 housed in the supports 4 are simultaneously investigated. Bearing loading is produced by rotation of a dynamically unbalanced shaft 2 with masses 3. Load can be controlled by proper selection of masses 3 and rotational speed of the shaft 2. Bearing housings are fixed to a stabilising bar 6 supported on ball joint 8 and four flexible joints 7. Bearings are fed with lubricating oil through the system of holes in the shaft.

Distribution of normal stresses in bearing alloy layer in central cross-section of the bearing bush tested in the rig is presented in fig. 4. Since the bearing reaction rotates synchronously with rotation of test journal, identical variable normal stresses occur at each point of slide layer in given cross-section during the loading cycle. These are pulsating compressive radial stresses and variable tangential stresses that alternate from tension to compression.

Temperature of the investigated object is measured with thermocouples located on external surface of the bearing and inside the oil supplying passages. If allowable temperature is exceeded or excessive vibration of the head unit occurs a power transmission system of the rig is automatically switched off.

3. Methodology of bearing fatigue research

Fatigue investigation in the test rigs is based on the assumption that tested bearing lining reaches its critical state when any macroscopically visible disarrangement of the specimen surface structure, such as first local fissures, a net of cracks (no matter how long) or lining breakouts, appears. A fatigue rating $S_P$ was defined as a critical value of tangential stresses in the investigated bearing material layer. $S_P$ relates to a value of stresses for which probability of fatigue damage of bearing lining, after reaching a specified number of load cycles, is equal to $P=0.50$, which means that $S_P$ is a median of evaluated random variable. The basis of standard fatigue test in MWO rigs was assumed as $3.6 \times 10^6$ loading cycles.

Sequence of fatigue tests corresponds to two-point strategy of experimental design and fully formalised statistical procedure of fatigue test data handling [1,3]. An objective of the research is determination of bearing dynamical load carrying capacity expressed in terms of stress amplitude and number of load cycles. A result of the investigation may be presented as Wöhler’s diagram or as critical stress value for given number of load cycles. The latter case may be considered as the fatigue rating of investigated bearing material determined in the result of standard test procedure in the test rig under consideration.

4. Analysis of operation of the test rig with rotating load vector

Characteristic feature of test loading pattern in the rig is that the whole internal surface of tested bearing in given cross-section is identically loaded. Synchronously with journal it rotates not only a load vector but the whole oil pressure distribution as well. If the investigated bush is rotationally symmetrical and reference system (co-ordinate system in hydrodynamic analysis) is linked to the journal, the bearing may be analysed as statically loaded bearing with rotating bush, where hydrodynamic pressure is generated only by relative rotational velocity of journal and
bush. In that case calculation of hydrodynamic oil film parameters is relatively easy and reliable. The same concerns the stresses produced in bearing wall because at each point of slide layer (in cross-section of the bush) the tangential stress alternates from maximum tensile value to minimum compressive value resulting from oil pressure distribution for statically loaded bearing (fig. 4). Thus variation of stresses in bearing material in central cross-section of the bush can be unmistakably determined. It is an important advantage for analysis of fatigue test results based on this principle of load generation. Moreover, the bush does not change its cylindrical shape in consequence of wear, e.g. during running-in, that is identical for the whole circumference.

The machine with rotating constant load has, however, the following drawbacks:

- bearing load depends strongly on rotational speed,
- when testing half shells or bearing of real internal macrogeometry, there is a problem of lubrication gap discontinuity around the bearing circumference,
- concentration of friction energy on testing bearing surface is significantly lower (several times) than in the real bearing of the engine,
- fatigue failures are uniformly distributed over bush circumference, which hamper observation and investigation of phenomena accompanying fatigue, e.g. cavitation.

Moreover, it results from many research works (Buske [4], Gitter [5], Lawrowski [6], Grobuschek [7] and author’s own experience [1]) that in case of investigation of high-loaded bearing alloys, even for very rigid systems, the shaft deflection leads to change of oil gap shape. In consequence, oil pressure distribution in axial cross-section of the bearing (fig. 5) can differ from the parabolic one, usually assumed in model of bearing hydrodynamic calculation that is a basis for assessment of stresses in bush surface layer [8]. The most loaded zone it will not be a central cross-section of the bearing but a region near to the bush edge (fig. 5c and 5d). So fatigue cracks can appear first of all at that place. In the result a model of stress calculation based on the assumption of parabolic pressure distribution in oil film may prove not to be adequate for determination of critical stress values leading to fatigue.

It is curious that in technical literature there are no photos of bearing surfaces being destroyed during fatigue tests in machine with rotating load vector. All exhibited views of fatigue failures concern bearing bushes investigated in machines of different load pattern, which results from characteristic distribution of fatigue cracks over bearing surface. This fact seems to confirm doubts that published data about fatigue critical stress values (e.g. Lang [2]) for bearing materials investigated in test rig with rotating load might have been evaluated on the basis of not fully adequate calculation model.

![Fig. 5. Oil film pressure distribution in axial cross-section of hydrodynamic bearing under conditions of various shaft and bush deflections; a – rigid bearing with parallel journal and bush surfaces, b – journal deflection in bimetallic bearing (rigid bearing alloy), c - journal deflection in trimetallic bearing (rigid bearing alloy, flexible overlay), d - trimetallic bearing with flexible bearing alloy, e – bearing with profile housing (edge flexible support)](image-url)
In paper [1] there are discussed results of calculation of oil film parameters for bearings of various dynamic load patterns in test machines, for typical crank-shaft bearing of IC engine and for steady loaded bearing. The important information from the analysis is as follows:

- The maximum oil film pressure in bearing test rigs is significantly higher than in a similarly loaded real engine bearing. This is particularly so for the machine with rotating load that not only has the highest peak pressure but also the greatest gradient in the region of oil pressure descending. Therefore the bearing surface fatigue damage can be reached in this rig in the shortest time.

- The highest concentration of thermal energy on bearing surface, resulting from friction, is in steady loaded bearing. In all rigs with constant direction of varying load vector the energy concentration is higher than the one in the real connecting rod bearing, whereas in the rig with rotating load (variant \(a\) in fig. 1) is several times lower. So in case of scale-model experiments, where a simulation of thermal loading of investigated bearing system is an essential requirement of process similarity, the last rig is less useful than the other systems.

- The real engine bearings operate somewhere between the two extreme test rig conditions, (variants \(a\) and \(c\) in fig. 1) depending on their respective load characteristics.

The first conclusion has been experimentally confirmed by Lang [2]. The fatigue tests on the rig with rotating load always led to the lowest critical maximum specific loads based on fatigue criterion. This is easy to explain, because in this case the hydrodynamic pressure is built up only due to rotation alone. As a consequence there is the most steep pressure descent in the oil film, which in common opinion leads to tensile stresses in bearing material in that region of the bearing. This feature of oil film may be derived from simple combining wedge effect and squeeze effect of hydrodynamic pressure, which is shown in fig. 6.

![Fig. 6. Superposition (c) of lubricant wedge effect (a) and squeeze effect (b) and their influence on distribution of pressure in oil film (d) in hydrodynamic journal bearing](image)

In bearing tested in the machine with rotating load vector there is not a squeeze effect of the lubricant that always smooths the oil film pressure distribution over the slide surface. Therefore, this machine is ideally suitable for the investigation of the pure fatigue phenomena of bearing material as the main object, whereas machines with different load pattern are more convenient for scale-model research.

In view of the above analysis, it is not surprising that the results of different test rigs are not directly comparable. The various data published about fatigue parameters of bearing materials - in particular fatigue ratings - show similar sequences among most bearing materials but their relations and absolute magnitudes differ considerably. To the above mentioned principal load generation differences come the different, not precisely specified, test conditions under which data are obtained, the arbitrary definition of some failure criteria and the peculiarities of the test machines themselves.
5. Recapitulation

The MWO machine with rotating load vector, fatigue testers SKMR-1 [9] and SKMR-2 [10] and reactive bearing testing machine SMOK [11] (designed and built at tribological laboratory of the Technical University of Gdańsk) constitute a complete set of devices that ISO 7905/1,3,4 standards [8,12,13] require for experimental determination of fatigue strength of surface layer in plain journal bearings. A recommendation of the standards is that results of fatigue research should be expressed in terms of varying stresses in bush surface layer. A method of combining the results of fatigue experimental investigation with computational evaluation of stresses in bearing slide layer has been, until now, elaborated only for the test rig with rotational load vector. An appendix to ISO 7905/1 [8] standard contains a procedure of calculation of limit stress values on the basis of fatigue test results obtained with use of this machine. Since such a computation is not done routinely for test rigs with different load generation patterns, the stress dominant or responsible for rupture, is not fully known for these rigs. Due to the lack of stress data the bearing performance often has to be assessed on the basis of a load value. The value still predominantly used today is the maximum specific load, because it is quite easy to determine.

In the paper the peculiarities of the bearing fatigue research method by means of the machine with rotating load vector have been analysed because the results of fatigue tests carried out on this device may be considered as the reference level, due to completeness of procedures of stress computation and data handling, for results of tests performed in other bearing testing machines.

References