BEARING GEOMETRIC RELATIONS VS.FRICTION LOSS

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Abstract

The results obtained using the method presented in the paper prove that there is a possibility of bearing friction loss estimation on the grounds of the journal center path and computer simulation of bearing operation parameters. Using the presented method one can foresee such geometric relations of bearing that promise engine high mechanical efficiency. Geometrical relations should be understood as such parameters as specific clearance, relation of bearing main dimensions and subtle errors of ideally cylindrical form of journal and shell.

Streszczenie

W artykule przedstawiono metodę określania wpływu relacji geometrycznych łożyska głównego wału korbowego na przewidywane straty tarcia. Otrzymane rezultaty potwierdzają możliwość szacowania strat tarcia w łożysku na podstawie analizy drogi środka czopa oraz symulacji komputerowej parametrów pracy łożyska. Wykorzystując metodykę określania strat tarcia zaprezentowaną w artykule można na etapie projektowania silnika zaproponować takie relacje geometryczne, które dają szansę na uzyskanie dużej sprawności mechanicznej silnika. Poprzez relacje geometryczne rozumie się: luz względny, stosunek wymiarów głównych łożyska oraz subtelne odstępstwa od idealnie cylindrycznego kształtu czopa i panwi.

1. Introduction

The need for an increase in engine mechanical efficiency is quite obvious if the goal is the reduction in fuel consumption, not only for nominal but also for average conditions of engine run. The latter mean frequent idle run and rare extreme loads. A substantial reduction in friction losses has to concern those engine kinematic nodes that determine the mechanical efficiency, i.e. piston-cylinder group and crankshaft bearings. Principles of determination of friction loss have been already presented in literature [1,2]. As one can foreseen, the most faithful results can be achieved combining experimental and analytical methods. In the case of slide bearings the experiment consists in measurements of journal center path, while the computer simulation allows to estimate the bearing friction force moment using the measured path and calculated circumferential flows.

2. Test bed

The test bed consists of reconstructed two-cylinder engine coupled to the DC dynamometer. The last can operate both as generator and electric motor. In the first case the system operates as a dynamometer equipped with a typical set of measuring devices necessary for measuring the engine power and other basic parameters of run. The engine has been reconstructed through change or elimination of certain auxiliaries. Moreover, an external system of oil and cooling fluid temperature stabilization has been introduced.. The accuracy of temperature stabilization system was far higher than that of typical car engine. For points of sensor location the accuracy of temperature stabilization was not less than 0.5 ^{o}C. The engine has been deprived of the timing system gear. Such modernization made the independent engine run impossible. The only way of engine run was use of external drive. Elimination of timing system gear was necessary because of its high percent in total friction losses. The most impor-

tant engine reconstruction was installation of journal position sensors in main bearing. Fig. 1 presents the bearing furnished with non-contact sensors of oil film thickness.

A rod equipped with passive sensors has been located above the bearing. Those sensors reduce the sensitivity of oil film measuring arrangement to the variations of bearing temperature.

Literature [1] describes the way of processing the oil film thickness signal picked up by four sensors to the journal axis trajectory.

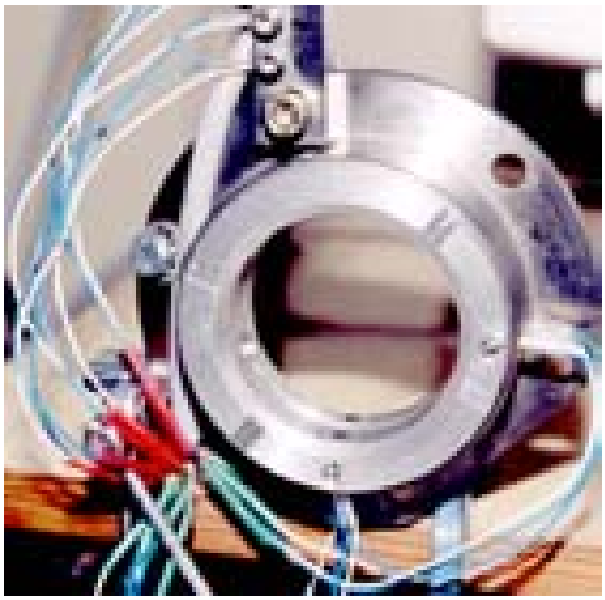

Fig. 1. The main bearing shell of the tested engine with journal position sensors

3. Results of bearing journal center path measurements

The measurements of journal center path have been carried out on an IC engine without a timing system, therefore the cycle repeats not every 720, as for regular 4-stroke engine but every 360 ^{o}CA. Such change made that each cyclic parameter including the journal center path repeats every 360 ^{o}CA and due to that diagrams presented in the further part of this paper consider only one crankshaft revolution.

Figs 2 and 3 present the journal center path for engine rotational velocity 500 rpm at inner (near engine's interior) and outer (opposite) bearing edge, respectively.

The low rotational speed has been deliberately assumed for the research purposes, because their results have to inform about the reduction in friction losses at the lowest possible velocity of idle run. Comparing both diagrams one can see the differences for two planes defined as the inner and outer ones. These differences result from the shaft deflection. If the shaft moved without any deflections, diagrams in Figs 2 and 3 should be identical. Differences in courses presented in Figs 2 and 3 can be utilized for calculations of shaft axis deflection, which is not the subject of this paper, but should be taken into consideration for the determination of frictional losses.

4. Estimation of clearance circle radius

Definition of the so-called clearance circle makes a basic difficulty of the analytical-measuring method of frictional loss determination. It is generally assumed that the clearance circle is the area where a point of journal axis can be found. If the journal axis happened to be

outside the clearance circle, diffusion of journal and shell sliding surfaces should occur. For actual systems the determination of clearance circle is almost impossible what results from the imperfections of both cooperating surfaces. Another definition of the clearance circle has been assumed for the needs of this paper. Namely, it is a circle selected so as to achieve an oil film reaction equal to all components loading the bearing. It means that this method consists in load determination on the grounds of the vector analysis of forces acting on the crank-piston system and using the method of successive approximations eventual selection of such clearance circle, which assures a comparable oil film reaction. Figs. 4 and 5 present the oil film reaction for the clearance circle of 34 μm and 30 μm in diameter, respectively.

Fig. 2. Journal center path close to the bearing inner edge (closer to the engine interior)

Fig. 3. Journal center path close to the bearing outer edge

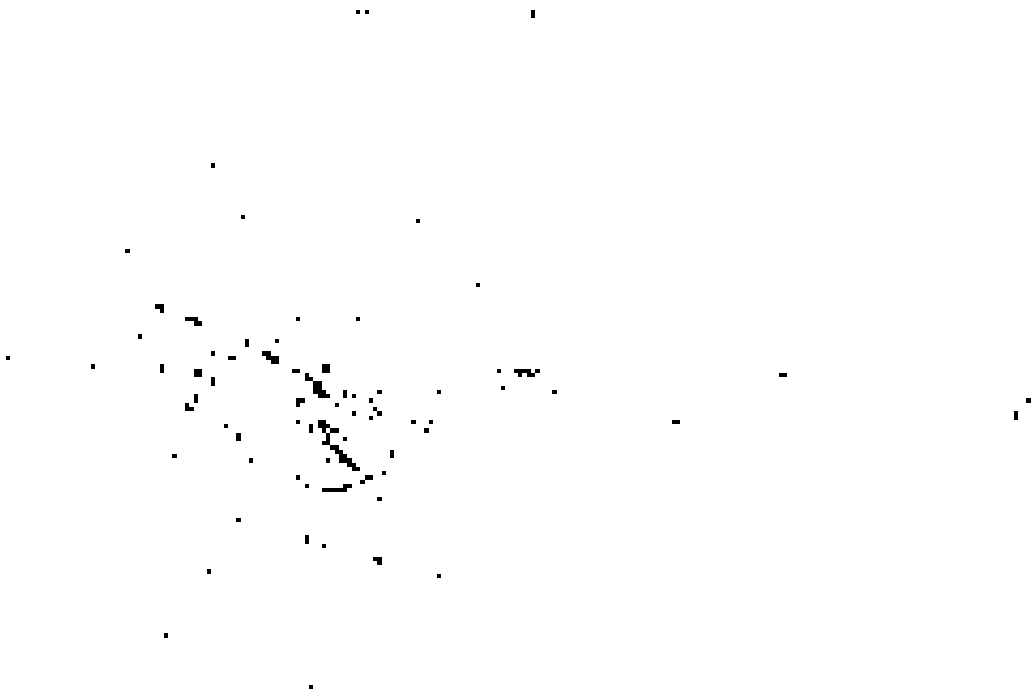

Fig. 4. Oil film reaction for the clearance circle of 34 μm in diameter

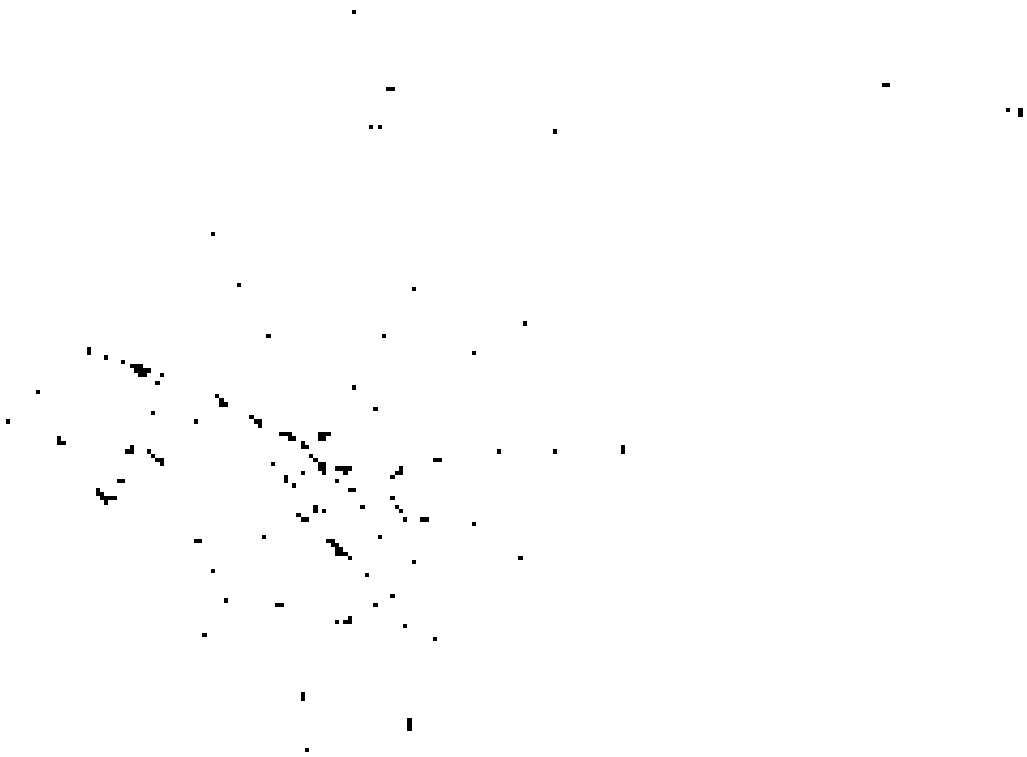

Fig. 5. Oil film reaction for the clearance circle of 30 μm in diameter

The scale of radius axis has been assumed so as the maximum reaction value occurs far from the beginning of coordinate system because of the necessity of presenting the character of oil film reaction course for the entire crankshaft revolution. Due to that, the diagram in

Fig. 5 had to be divided and the pressure maximum value had to be placed above the basic part, which makes reading the maximum value, equal to 19.5 kN, difficult.

Assumption of the clearance circle radius higher than 4 μm gives the oil film reaction value almost twice lower (see Fig. 4). Comparing the maximum value of oil film reaction (presented in Fig. 4) with bearing loading for this point it comes out that values obtained are fairly close and equal 12.2 kN. The clearance circle radius has been assumed as high as 34 μm for the bearing friction force analysis.

5. Determination of the bearing friction force moment

On the grounds of clearance circle and journal center path one can determine all parameters of oil film in the gap between journal and shell, including the friction force. However, the computational procedure is quite complex. It is worth mentioning that programs presented in references [1] have been used for this purpose. The course of instantaneous value of bearing friction power for the analyzed case has been presented in Fig. 6.

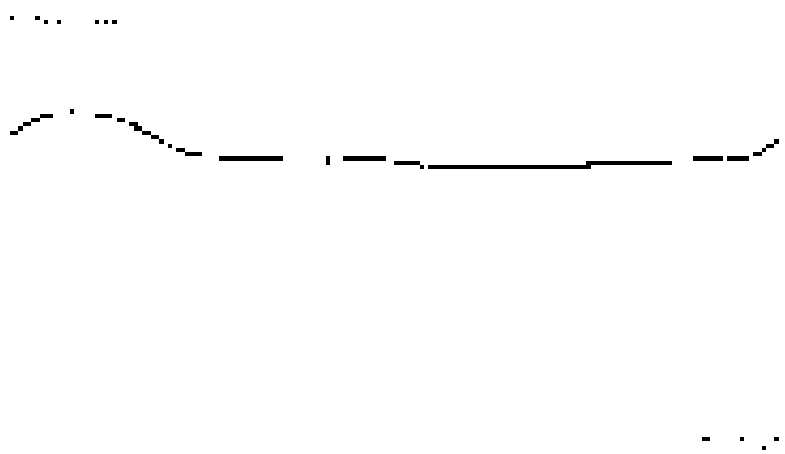

Fig. 6. Instantaneous value of bearing friction power vs. crank angle for n=500 rpm and clearance circle radius r=34 μm

The average value of bearing friction power, which directly affects the engine mechanical efficiency, can be easily determined on the grounds of momentary values of friction power. This average value of friction power can be found above the diagram in Fig. 6.

The effect of bearing geometric relations on friction losses can be clearly seen changing the radius of clearance circle. If this radius equals 30 μm, i.e. the same as in Fig. 5, the course of bearing friction power will be similar to that in Fig. 7.

6. Summary

The presented method of bearing friction loss estimation is the one of few methods, which take into account the instantaneous values of friction along the crankshaft revolution. The computer simulations of friction loss that do not take the journal center path measurement results into consideration are least reliable. The comparison between the oil film reaction and bearing loading should serve as a test for the journal center path measurements and resulting distribution of oil film.

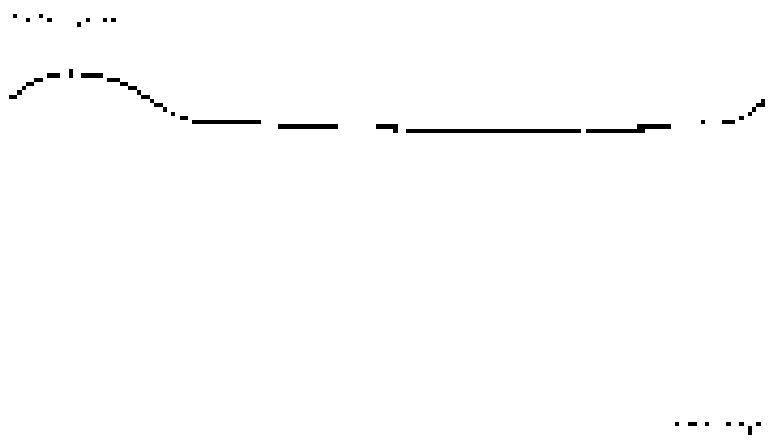

Fig. 7. Instantaneous value of bearing friction power vs. crank angle for n=500 rpm and clearance circle radius r=30 μm

References

[1] Iskra A. Parametry filmu olejowego w węzłach mechanizmu tłokowo-korbowego silnika spalinowego. Wydawnictwo Politechniki Poznańskiej, Poznań 2001

[2] Iskra A. Anomalia obciążeń łożysk wału korbowego. 28th KONES 2002. 8-11.09.2002