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MODIFICATION AND CORRECTIVE ACTION OF OIL GROOVE FOR HIGH PERFORMANCE BEARINGS

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Abstract—The article deals with the failure of bearings due to oil starvation due to absence of sufficient oil film between the bearing and the journal permits metal-to-metal contact this resulting wiping action causes premature bearing fatigue and High loads applied to the bearings result in a reduction of minimum oil film thickness so to prevent such failure specific load applied to the grooved bearing may be lowered by means of a reduction in groove width by modification in groove design.

I.INTRODUCTION

The loads acting on engine bearings are generated by the pressure of the fuel-air mixture combusting in the cylinders. This pressure, or combustion force, drives the piston down during the engine power stroke. The piston is connected to the crankshaft by the connecting rod, which transmits the load and converts the linear motion of the piston into crankshaft rotation. The upper connecting rod bearing is the first bearing supporting the load generated in the cylinder. Since the crankshaft is supported by the main bearings, they react to the load transmitted via the connecting rod bearings. In the simplest one-cylinder engine there are two main bearings supporting the crankshaft, and the load from the cylinder pressure is directed onto these two lower main bearing shells.

However, combustion force is not the only force generated by an internal combustion engine. Such engines contain parts performing accelerating/decelerating motion (either linear or rotating): pistons, connecting rods, crankpins, counterweights and webs of the crankshaft. These parts generate inertia forces that are added to the combustion forcesand therefore affect the support reactions of the bearings. The value of the inertia force produced by a moving part is proportional to the square of the rotation speed. At low and medium rotation speeds the inertia forces are relatively low. Therefore the most loaded part of a connecting rod bearing is the upper shell, and the most loaded part of a main bearing is the lower shell. However at high rotation speeds, which are characteristic of high performance engines, inertia forces become considerable. For example, an increase of rotation speed from 2000 to 6400 RPM raises the inertia forces by 10 times. In high performance engines inertia forces are comparable to combustion forces and may even exceed them.

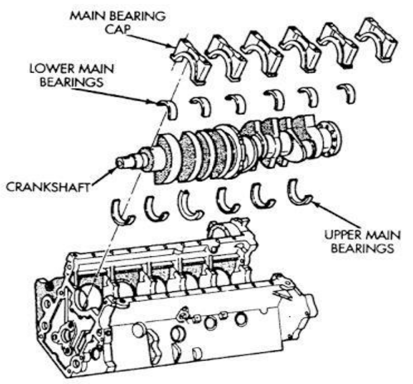

Fig.1 Assembly of Engine Bearing II.

LITERATURE REVIEW

In the hydrodynamic journal bearing groove affects the performance of the bearing. Many researchers have evaluated that the groove position, shapes and number of grooves have a direct effect on the performance of the bearing Kangetal.Analyzed static and dynamic characteristic of oil lubricated herring-bone grooved journal bearing having a groove profile and then compared the results with plain and herring-bone grooved journal bearing having a rectangular groove profile. It was evaluated that the optimum bearing parameters for the circular grooved bearing are different from that of the rectangular one. Circular grooved HGJB are much firmer than a plain journal bearing especially in the case of small eccentricities. They have performed a numerical analysis on the static and dynamic characteristics of circular grooved oil lubricated herring-bone grooved journal bearing. The designers have concluded that the optimum values of the groove width ratio as 0.25 and groove angle as 28° "Myung-RaeChoetal"[1] presented the effects of circumferential groove on the minimum oil film thickness in engine bearings. The fluid film pressures are calculated by using the infinitely short bearing theory for the convenience of analysis. Journal locus analysis is performed by using the mobility method. A comparison of minimum oil film thickness of grooved and ungrooved bearing is presented. It is found that circumferential 360° groove only reduces the absolute magnitude of the oil film thickness, but 180° half groove affects the shape of film thickness curve and position of minimum oil film thickness

Koichi Matsudaetal [2] suggested clearance configuration of fluid-film journal bearings and optimized in a sense of enhancing the stability of the full circular bearing at high rotational speeds. A performance index is chosen as the sum of the squared whirl- frequency ratios over a wide range of eccentricity ratios, and a Fourier series is used to represent an arbitrary clearance configuration of fluid-film bearings. An optimization problem is then formulated to find the Fourier coefficients to minimize the index. The designed bearing has a clearance configuration similar to that of an offset two- lobe bearing for smaller length- to-diameter ratios. The load capacity of the designed bearings is nearly in the same magnitude as that of the full circular bearing for smaller length-to-diameter ratios. The designed bearings successfully enhance the stability of a full circular bearing and are free from the whirl instability

Boedo and Eshkabilov [3] implemented a genetic algorithm scheme to optimize the shape of fluid film general bearings under steady journal rotation. They considered only one objective function, namely the maximization of load capacity. However, the optimum design of a journal bearing is a multi objective problem which must be solved with multiple objectives taking into consideration design constraints. Each objective function (i.e. minimization of temperature rise, minimization of oil feed flow, minimization of power loss) has a different individual optimal solution. Multiple optimal solutions exist because no one solution can be optimal for multiple conflicting objectives.

[4]The technical paper on " Measurement of Oil Film Pressure Distribution in Connecting Rod Bearing With Test Rig ",state how to use thin film pressure sensor which is based on the change of resistance in the pressure sensitive part. The lowest layer electrically insulates the sensitive film from the bearing shell. The uppermost layer protects the sensitive film from wear. Because the surface of the protection film is plain, the thin film pressure sensor resembles a common plain coating on the sliding surface of the bearing.

III. FUNCTION OF OIL GROOVE ON BEARINGS. The inertia force generated by a rotating part of the crankshaft is directed from the rotation center to the center of mass. Therefore it is transmitted to both the upper and lower main bearing shells. The oil groove is commonly made in the upper shell where the oil hole is located. A 180º groove is sufficient for providing the required amount of oil to the connecting rod bearing, which it reaches by flowing through passageways within the crankshaft.

The lower main shell has no groove. Therefore its effective area is greater than that of the upper grooved bearing. This design allows distributing the load applied to the lower shell over a greater area, reducing the specific load acting on the bearing material. Since the lower bearing shell is generally loaded heavier than the upper, the specific loads of the two are balanced.

However, at high rotation speeds in high performance engines the absolute loads applied to the upper and lower bearings may become close to each other. In this case the specific load (force per unit area) applied to the upper bearing may exceed the specific load to the lower bearing. Excessive loading of the upper bearing may cause the following two problems:

- Fatigue of the bearing material.[5] Internal combustion engines are characterized by cycling loading of the bearings. It is caused by alternating pressure of combustion gases in the cylinders and inertia forces developed by accelerating parts. The oscillating loads applied to a part may cause bearing failure as a result of material fatigue [1]. This occurs if the load exceeds the fatigue strength (load capacity) - the maximum value of cycling stresses that a bearing can withstand after an infinite number of cycles.

- Too low minimum oil film thickness. High loads applied to the bearings result in a reduction of minimum oil film thickness. This may cause non-uniform distribution of the bearing load (localized pressure peaks) [2] characterized by metal-to-metal contact between the bearing and shaft (mixed or boundary lubrication regime), high coefficient of friction (power loss), increased wear, and the possibility of seizure between the bearing and shaft materials.
- Worn Overlay surface with a dark green hue, in which pitting is evident, often indicates chemical corrosion due to fuel in the lubricant. Pitting is especially likely when combustion as show in Fig. 2

Fig.2 Worn Overlay Surface

-A rather large number of conditions can lead to oil starvation in the bearing areas. Wiping of the metals, blue discoloration, and welding of bearing metal to shaft or pin follow loss of the oil film as shown in Fig.3

-Broken or plugged oil passages, prohibiting proper oil flow

-A blocked oil suction screen or oil filter.

Fig.3 Oil starvation in bearing areas

Fig.3 shows a cross section of a bearing with a conventional oil groove design.

IV. Corrective action to prevent Lubrication Failure

-Double check all measurements taken during the bearing selection procedure to catch any errors in calculation.

-Check to be sure that the replacement bearing you is about to install is the correct one for application.

-Keep oil in pan at proper level, and be sure oil pan angularity arrangement is adequate for the installation. -Align bearing oil holes with supply drillings

- Use specified Grade of lubricant, Store carefully, and handle with attention to cleanliness.

-Maintain Crankshaft oil seal

- Check engine for possible blockage of oil passages, oil suction screen and oil filter

- Be sure that oil holes are properly indexed, when installing the replacement bearings.

Fig.4 Engine Bearing after corrective Action

V. New U-Groove Design of Oil Groove

The level of specific load applied to the grooved bearing may be lowered by means of a reduction in groove width.

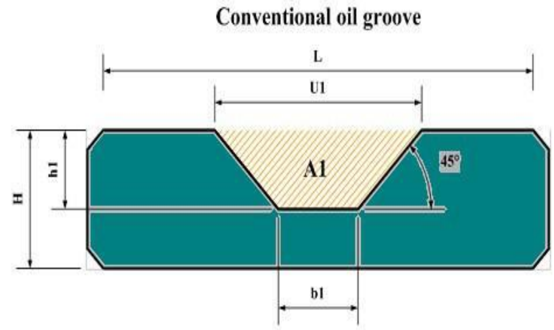

Fig.5 Conventional Oil Groove

The effective bearing length is L-U1. It may be increased by a simple decrease of U1, but it would reduce the cross sectional area of the groove. This is an extremely undesirable modification, particularly for high performance bearings generating high oil flow rates due to operation at high rotation speeds. The connecting rod bearing is lubricated by oil passing through the main bearing groove and then the oil passages in the crankshaft. The amount of oil entering the connecting rod bearing should be not lower than the oil flow produced by the hydrodynamic lubrication of the main bearing. A reduction of the cross sectional area decreases the passage capability of the groove, which may cause a formation of oil starvation conditions in the connecting rod bearing.

A new design should result in a reduction of the groove width without decreasing the groove cross sectional area. The modification according to these demands is presented in Fig.4.

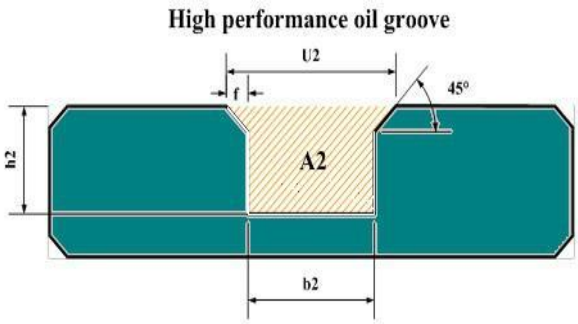

Fig.6 High Performance Oil Groove

The rectangular shape with small chamfers has allowed reduction of the groove width at the top (U) by at least 30%. On the other hand the cross sectional area A has not changed due to an increase of both groove width b and groove depth h.

Conclusions

A new design of oil groove for the upper main shells of high-performance bearings has been developed by King Engine Bearings.

- The modified design results in an increase of the effective bearing area without a decrease of the cross sectional area of the groove.

- According to the theoretical calculations, and observed rig and dyne test results done by the king bearings, the new design provides greater bearing durability due to lower specific loading and a more stable hydrodynamic lubrication regime: smaller peak oil film pressure and greater oil film thickness.

- The tests of the bearings in the test rig prove that, at loads close to the load capacity of the bearing material, the new esign is capable of preventing failure due to fatigue of the overlay.

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