

Effects of Connecting Rod Big End Stiffness on Oil Film Pressure

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ABSTRACT

A new thin-film sensor that enables accurate measurement of the oil film pressure of compact bearings for passenger cars was developed. Using this new sensor, the oil film pressure distribution was measured for a variety of connecting rod big end stiffness levels, and the effect of those changes was investigated. These results confirmed that the oil film pressure distribution changes according to variety of the stiffness. Whereas the oil film pressure occurs with one peak in the load direction for a rigid connecting rod big end, the oil film pressure for an elastic connecting rod big end is dispersed and the direction of the peak does not match that of the load. In addition, elastohydrodynamic calculations were performed with the same conditions as the measurements, and the reasonableness of the measurements and calculations were confirmed.

1. Introduction

The width of connecting rod big end bearings has tended to become smaller to reduce bearing friction in response to recent environmental issues, making it more difficult to ensure durability. Bearings in particular, which determine the engine framework, are being required to have their performance predicted to a level of precision that is approaching the limits of feasibility from the perspective of shortening development time.

Engine bearings are lubricated by fluid in the form of an oil film, and measurement of this oil film pressure is an important factor for evaluating durability. However, it is extremely difficult to directly measure the oil film pressure between sliding parts, and there are few cases of this having been successfully carried out.⁽¹⁾ Although the authors had previously applied a method in which a diaphragm-type sensor was embedded in a shaft, this method had the following drawbacks:

- (1) Restrictions on the space required for embedding the sensor in the case of a small engine bearing for a passenger car engine
- (2) Restrictions on the heat resistance of the semiconductor gauge
- (3) Measurement under high-pressure conditions
- (4) The effects of irregularities in the sensor pressure-receiving portion

We therefore decided to develop a thin-film sensor⁽²⁾⁽³⁾ that would be nearly able to completely overcome these problems and make it possible to measure the pressure between sliding parts that has been considered difficult to achieve in the past. In addition, oil film calculations (elastohydrodynamic calculations; EHD) are attempted that take into consideration the rigidity of the connecting rod big end. Such calculations have become practical with the

dramatic improvement that has taken place in the calculation environment in recent years, and several calculation results have been reported.⁽⁴⁾⁻⁽⁷⁾ Based on the combined results of the calculations and measurements, a discussion is presented about the mechanism by which the annular rigidity of the connecting rod big end affects the oil film pressure.

2. Measurement Method

2.1. Construction of a Conventional Diaphragm-type Pressure Sensor

Fig.1 shows the appearance of a diaphragm-type pressure sensor and the cross-sectional construction of the sensor unit. A sensor unit 14 mm in diameter and 6 mm thick, in which a semiconductor strain sensor is attached to a diaphragm having a diameter of 1.0 mm, is embedded in a hole provided in the shaft. Electron beam welding is used to secure the sensor unit in position.

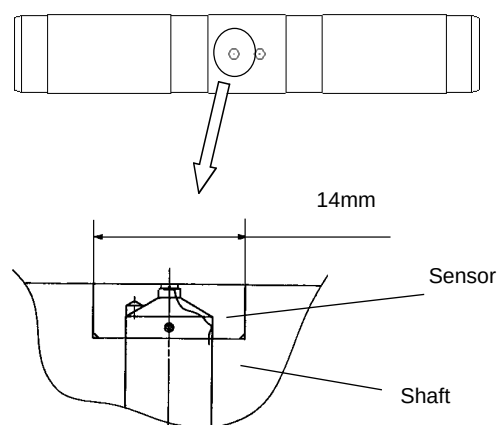


Fig.1 Diaphragm sensor

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2.2. Construction of the New Thin-Film Pressure Sensor

The appearance of the new thin-film pressure sensor is shown in Fig.2, while the cross-sectional construction of the sensor unit is shown in Fig.3. Manganin, whose resistance is sensitive to pressure, was used for the sensor material, and a pressure-receiving area measuring 1 mm in diameter and 20 μm in radial width was formed on the shaft sliding surface by sputtering. The total thickness of the sensor, resulting from the combined thickness of the insulating and protective films, is about 7.2 μm . The chemical ingredient of manganin is shown in Table 1.

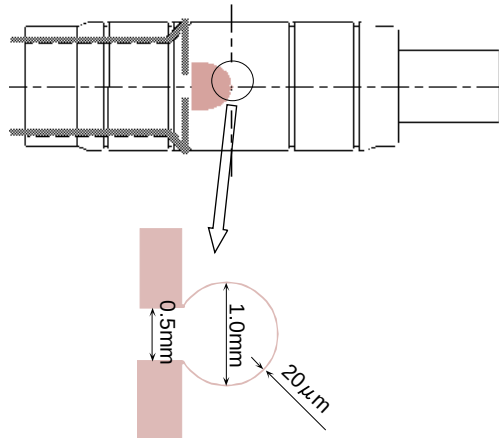


Fig.2 Thin-film sensor

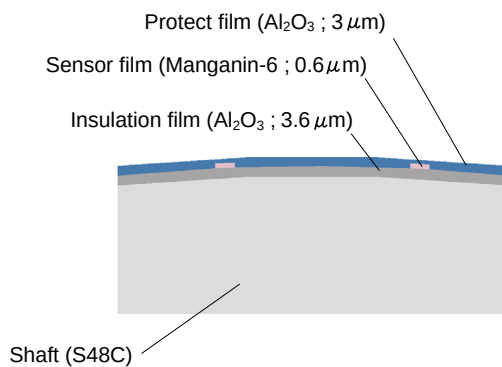


Fig.3 Sensor part section

Table 1 Chemical ingredient

	Cu	Mn	Ni
Manganin-6	87.6	9.2	3.2

Fig.4 shows a plot of the pressure versus percentage resistance change. This graph demonstrates the excellent linearity and low hysteresis performance of the sensor over a broad pressure range of 0-100 MPa.

The area of the shaft where the sensor is formed is ground by the amount of sensor thickness prior to processing, and a small circular shape with an extremely low level of irregularities is maintained even after the sensor is formed.

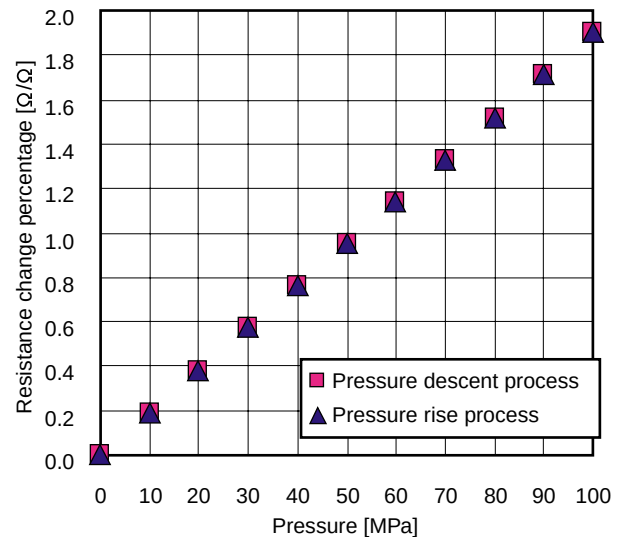


Fig.4 Thin-film sensor characteristic

2.3. Overview of the Experimental Device

Fig.5 shows the experimental device used in this study for measuring the oil film pressure. Roller bearings were used to support the shaft in order to minimize the thermal effects of the support bearings themselves on the test zone. A vertical hydraulic load was applied downward to the connecting rod. The output from the sensor is led outside the rotor by a slip ring, and a Wheatstone bridge is set up in front of the slip ring to increase the S/N ratio. An electromagnetic clutch is installed on the motor side to protect the apparatus from an overload torque in the event of bearing seizure or other high torque-generating events.

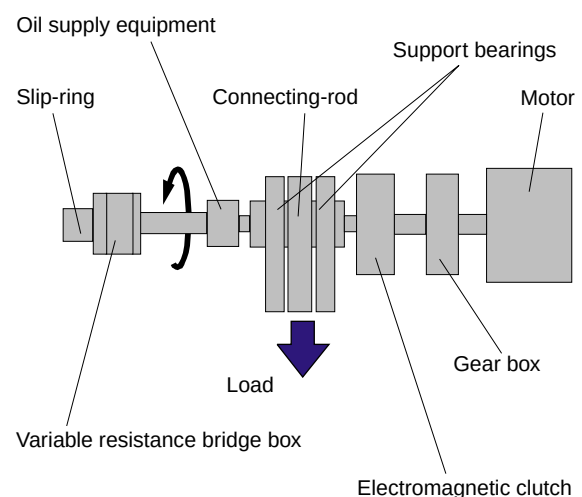


Fig.5 Assembled experimental device

2.4. Measurement Conditions

Fig.6 shows the appearance and Table 2 shows the dimensions of the test connecting rods. The two types of connecting rods were tested to identify any difference attributable to the annular rigidity of the big end in the distribution of oil film pressure. The Type A connecting

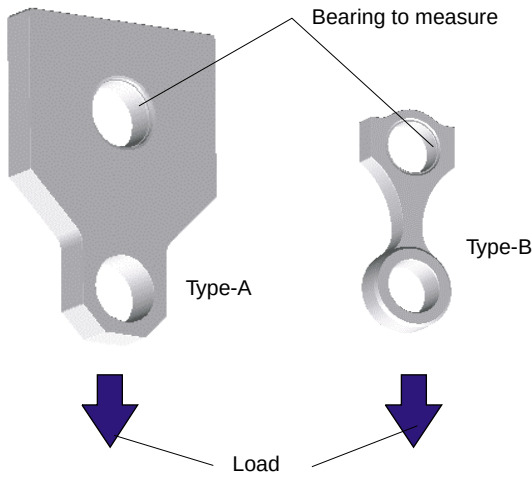


Fig.6 Connecting rod

Table 2 Test specification

Type		A	B	Unit
Conrod	Length	138.5		mm
	Big-end axial width	30	24	mm
	Material	SUS304		-
Bearing	In-diameter	50		mm
	Mean-width	20		mm
	Material	Cu + Pb		-
Oil		SAE10W-30		-

rod has a big end which is equivalent to a rigid body, while Type B, in which elastic deformation can occur more easily, has a shape resembling that of the big end used in passenger car engines.

The test connecting rods were each in one piece, without a separate big-end cap, and a completely circular bearing was gently press fitted into the big end.

3. Calculation Method

Commercially available software was used for the EHD calculations. The oil film characteristics were determined by using Reynold's equation, the fundamental equation of fluid lubrication. The elastic deformation of the connecting rod big end was incorporated into Reynold's equation as a mass-rigidity matrix contracted from an FEM model. Calculations were made by assuming the density, temperature and viscosity to remain constant at all times within tolerable limits. The connecting rod was used as the constraint in the calculations, the load being input from the shaft. The internal roundness of the bearing after it had been press fitted into the big end was measured and expressed in simple terms in the FEM model for input into the calculations to improve the overall accuracy. The FEM model used in the calculations is shown in Fig.7.

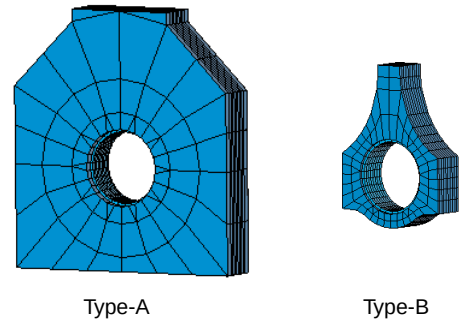


Fig.7 FEM model

4. Comparison Between the Measured and Calculated Results

4.1. Measured Results of the Oil Film Pressure

Fig.8 shows the results of measuring the distribution of oil film pressure around the bearing circumference at 3000 rpm for the Type A and Type B connecting rods.

The oil film pressure in the rigid body connecting rod big end of Type A was distributed over a narrow range around the bearing circumference, and increased with increasing load, while demonstrating a single peak coinciding with the direction of the load.

In the case of the Type B big end which can undergo

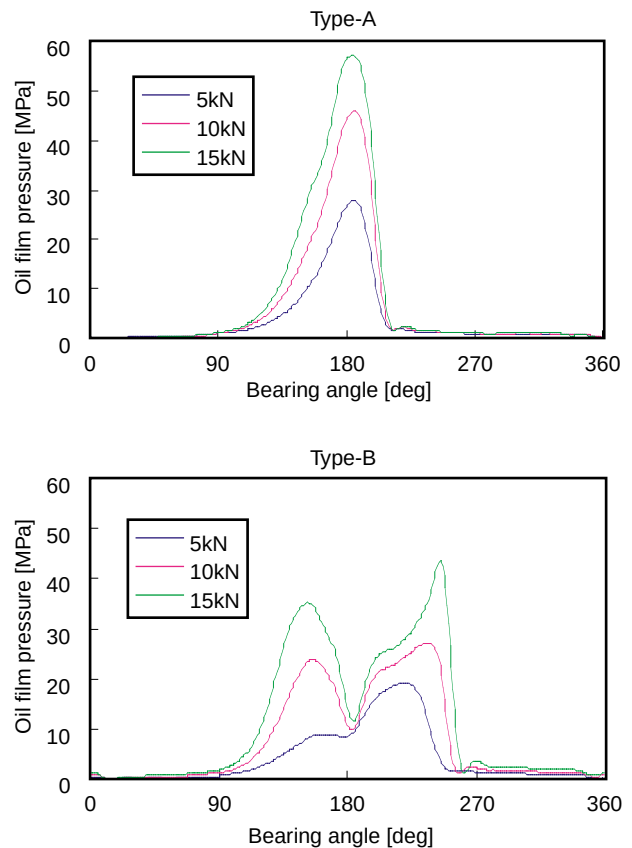


Fig.8 Measurement oil film pressure at 3000rpm

elastic deformation, the oil film pressure was distributed over a wider range around the bearing circumference than the case of Type A. As load was increased, the broad single peak was divided into two peaks, one either side of the direction of the load. In comparison with Type A, the increase in oil film pressure with increasing load was less pronounced.

4.2. Comparison Between the Measured Results and EHD- calculated Results for the Oil Film Pressure

Fig.9 shows the measured results and EHD-calculated results for the oil film pressure at 2000 rpm for the Type A and Type B big ends. The distribution of oil film pressure around the bearing circumference could be simulated with considerable accuracy for both the Type A and Type B big ends. These results made it possible to confirm the integrity of both the thin-film sensor measurements and EHD calculations.

In order to assess the feasibility limit for predicting bearing performance, it was thus determined that it is necessary to use EHD calculations that incorporate bearing elasticity, instead of using conventional oil film calculations that assume the connecting rod big end to be a rigid body.

4.3. Discussion on the Mechanism for the Change in Oil Film Pressure

We confirmed from the measured results of the thin-film sensor that the distribution of oil film pressure changed

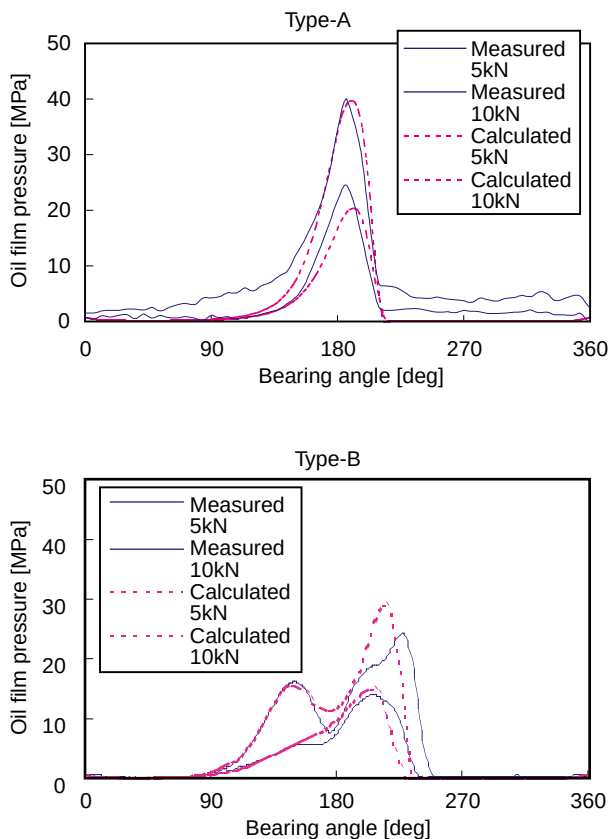


Fig.9 Comparison measurement and calculation at 2000rpm

according to the rigidity of the connecting rod big end. This is believed to have resulted from the load causing contraction of the big end diameter in the direction perpendicular to the load (referred to as close-in) when there was elastic deformation in the big end, while a corresponding increase in bearing clearance in the direction of the load occurs; this resulted in a reduction in the rate of increase in oil film pressure in the direction of the load, thereby leading to a corresponding increase in the distribution of oil film pressure in order to maintain the balance.

The mechanism for the change in oil film pressure can be deduced from the results of the EHD calculations. Fig.10 shows a 3D contour diagram of the oil film pressure distribution based on the calculated results, while Fig.11 shows the distribution of oil film thickness also based on the calculated results. In the case of the Type A big end, at the same time as the oil film pressure increases in the direction of the load, the oil film thickness can be seen to become thinnest at nearly the same location. This can be said to have resulted from no increase in clearance in the direction of the load due to the absence of close-in since the big end did not deform.

In the case of the Type B big end, however, oil film pressure in the direction of the load is relatively low, and oil film is consequently relatively thick. Locations where the

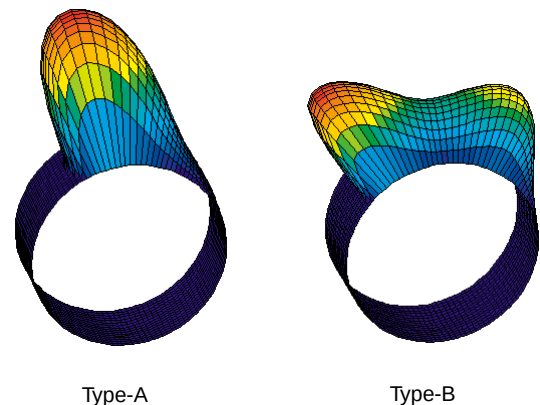


Fig.10 3D contour of oil film pressure distribution

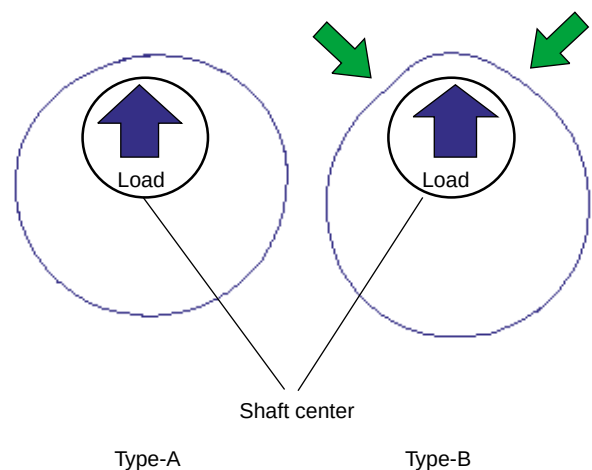


Fig.11 Oil film thickness distribution

oil film is relatively thin can be seen to exist on both sides. Increased clearance in the direction of load can be said to have occurred due to the close-in caused by elastic deformation of the big end.

Fig.12 shows a schematic representation of the mechanism for the change in distribution of oil film thickness.

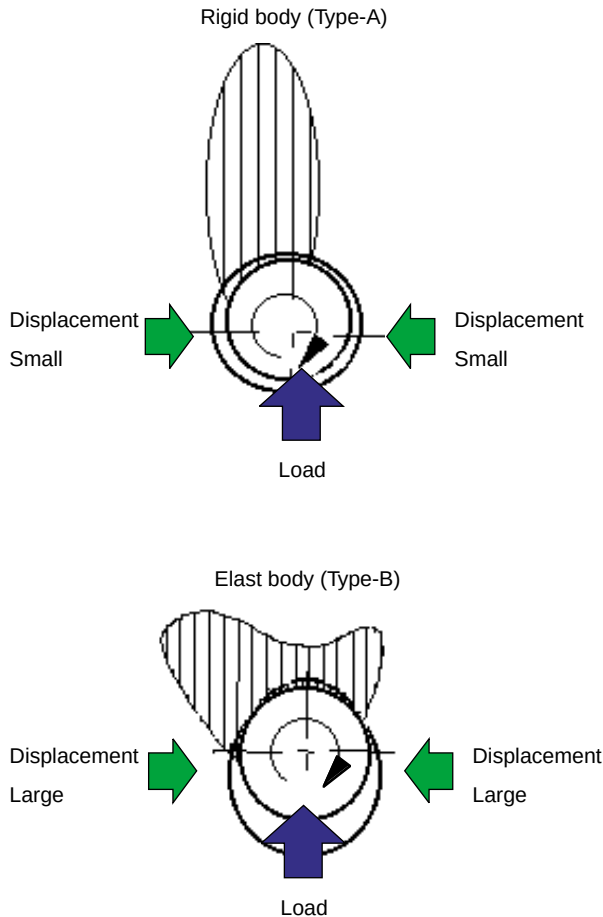


Fig.12 Mechanism of oil film pressure distribution

5. Conclusion

- (1) The development of a small thin-film pressure sensor made it possible to directly measure the bearing oil film pressure in gasoline engines used in passenger cars.
- (2) The distribution of the oil film pressure around the bearing was confirmed to be considerably affected by the rigidity of the connecting rod big end. The oil film pressure distribution in a big end with high rigidity demonstrated a narrow peak in the direction of the load, and the pressure increased with increasing load. On the other hand, in the case of a big end with relatively low rigidity like that used in passenger car engines, two lower peaks in the oil film pressure distribution resulted, one being each side of the load, while the increase in pressure was more gradual with increasing load.
- (3) Good correlation was obtained between the measured

results and those obtained by EHD calculations, verifying the integrity of both. It was confirmed that EHD calculations that allow for elastic deformation of the big end, and not conventional oil film calculations that assume the big end to be a rigid body, will be necessary in the future for precisely predicting bearing performance.

6. Acknowledgments

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