

Proposal of New Bearing Simulation Rig Tester for High Efficiency Engine Bearing Development

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ABSTRACT

To select bearing specifications for an engine using a bearing rig test, it is necessary to achieve sliding conditions that are near the actual operating conditions of the engine. A new bearing rig tester with low resonance and high rigidity permits high frequency excitation and has a mechanism to correct misalignment between the axle and the bearing. This rig tester can reproduce load fluctuations under the actual operating conditions of the engine. The test equipment makes it possible to clarify for the first time the mechanism that causes the difference in the break-in performance of lead alloy and aluminum alloy bearings. It was possible to evaluate the small end bearing fitted on an actual connecting rod, and the sliding characteristics caused by different surface treatment methods for the small end bearings. In addition, it was possible to measure the friction coefficient of the bearings with high repeatability, and efficiently evaluate the bearing material.

1. Introduction

As automobile engines have become smaller and lighter, bearing widths have decreased. Simultaneously the engine output, performance and combustion pressure have all increased. These and other factors impose a greater load on the bearings every year. Bearing materials have also changed, particularly with the introduction of eco-friendlier lead-free bearings⁽¹⁾⁻⁽³⁾. To develop an efficient engine, it is important to carry out conformance tests on many parts before they are used in an actual engine.

Because the function of a bearing is greatly affected by conditions in the vicinity of the housing, methods such as elastohydrodynamic lubrication (EHL) analysis have been used to analyze the causes of bearing housing deformation^{(4),(5),(9),(10)}, changes in the viscosity of the lubricant caused by the temperature^{(6),(8)}, shaft profile changes⁽⁷⁾,

and other factors. The more severe the sliding conditions become, the greater the necessity to take into account many factors including those mentioned above. Also, it is still difficult to account for transient deformation, which is necessary for expressing the break-in characteristics; transient changes in plasticity accompanying wear; and transient state changes accompanying wear. This means that it is still necessary to rely on actual engine tests to determine most of these factors. On the other hand, to develop the bearing vicinity more efficiently using conventional static load bearing testers, it is necessary to bring the sliding conditions, including bearing housing deformation, closer to those of an actual engine.

This paper describes the increased efficiency in evaluating bearing rig sliding performance using a newly designed bearing tester, as well as its influence in the selection of the actual engine specifications.

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2. Bearing Rig Tester

2.1. Conventional Tester

Figure 1 is a configuration diagram⁽¹¹⁾ of the standard static load bearing tester used until now.

Because this tester has been designed to produce a static load, it cannot generate a dynamic load. Figure 2 shows an example of calculating the connecting rod small end bearing load when the engine is operating, and the static load applied. In addition, it shows the position of the connecting rod small end of the piston pin versus load.

The magnitude and direction of a dynamic load change over time, and the thickness of the oil film between the piston pin and the connecting rod small end bearing changes too. However, in the case of a static load, these parameters remain constant with time. The oil film is maintained under a static load due to the wedge action (effect due to pressure generated by the relative velocity between the shaft and the bearing in the sliding direction). In the case of a dynamic load, the direction and magnitude of the load change, resulting in a squeeze action (effect due to pressure generated by the velocity of the shaft and the radial direction of the bearing). As a result of this effect, a comparatively thick oil film is maintained at the maximum load, and the main direction and magnitude of heat generated by oil shear due to the change in the load direction are dispersed, and the temperature of the sliding face also falls. The oil film pressure of the sliding face generated by the

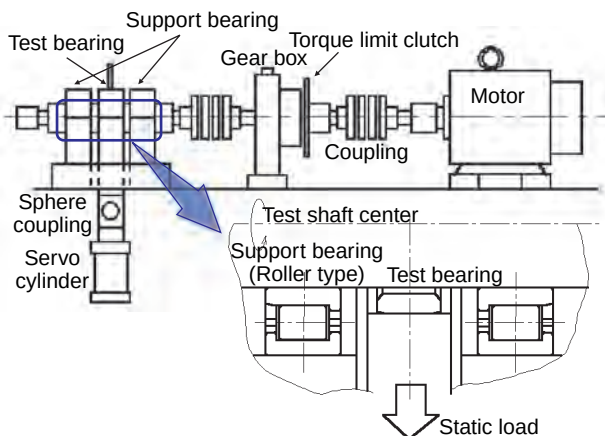


Fig. 1 Conventional system

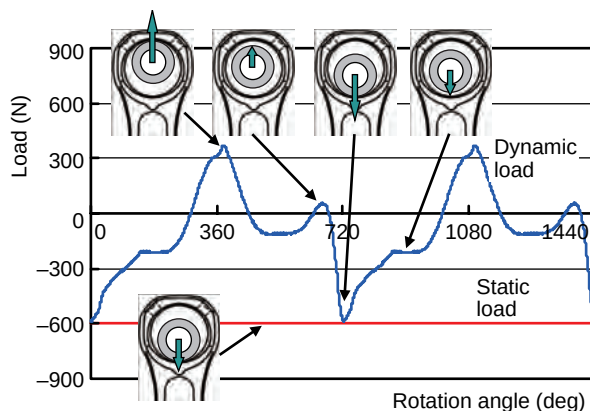


Fig. 2 Differences in sliding parts by load type

Table 1 Performance of new bearing rig tester

Item	Performance
Bearing slide conditions: Similar to actual engine	• Use of various entity connecting rods • Synchronization of X, Y direction loads • Generation of various load waves • Frequency: 300 Hz Load: ± 10 kN • Synchronization of bearing load with shaft angle
Tester protection from seizure	• Construction that can separate torque meter from motor torque
Repeatability of test results	• Adjusting mechanism of bearing-shaft tilt • Roller-type support bearing • Multi-point measurement of bearing temperature

wedge and squeeze actions causes elastic deformation of the piston pin and connecting rod small end bearing. This deformation is relatively large compared to the oil film thickness that is formed, so it changes the sliding conditions. Also, the lower the rigidity of the housing, the greater the deformation, and hence the greater its effect.

Given the above factors, the only way to make the sliding conditions closer to those of an actual engine is to use a dynamic load, and so it was necessary to develop a new tester.

2.2. New Bearing Tester

Table 1 shows the functions and features of the new tester. None of these functions are available in commercially available testers. To get the full performance out of the exciter, it is necessary to study not only the exciter but many other parts. Almost all parts are being newly designed with reference to previous research projects^{(12) - (24)} to enable the tester to be used for as wide a range of tests as possible.

2.2.1. Overall configuration of the tester

Figure 3 shows the overall configuration diagram of the developed tester.

The torque meter was designed to withstand high rotational speeds, and the gear box with torque limiter clutch can reliably protect the rotational system from the abrupt generation of friction. As a result, the torque meter can be left installed up to the highest speed test conditions, enabling friction to be measured at all times.

If plain bearings were used as support bearings, heat generated in the support bearings would be transmitted to the test section, affecting the results. Also, when plain bearings undergo temporal change as a result of wear, the friction and oil flow rates change, causing the temperature of the test bearings to change as well. To prevent this and other issues from occurring, roller bearings are used. In the new tester, a double row of roller bearings is used to ensure rigidity when a dynamic load test is being performed.

The actuator and servo valve, which generate the excitation pressure, are larger than those used in conventional static load tests, so the frame surrounding the actuator is being completely reevaluated.

The new tester also has a new control system that adjusts the tilt angle of the shaft and the bearing when tests are being performed.

2.2.2. Study of the tester frame

To ensure satisfactory excitation at high frequencies, it is essential that the frame is rigid to minimize displacement and nullify the effects of resonance. The target excitation frequency is 300 Hz, so it is necessary to keep the resonant frequencies above this value. Figure 4 shows the FEM model of the frame that was studied, and the resonant frequencies obtained from calculation.

Generally, the exciter has a columnar frame. However, a study

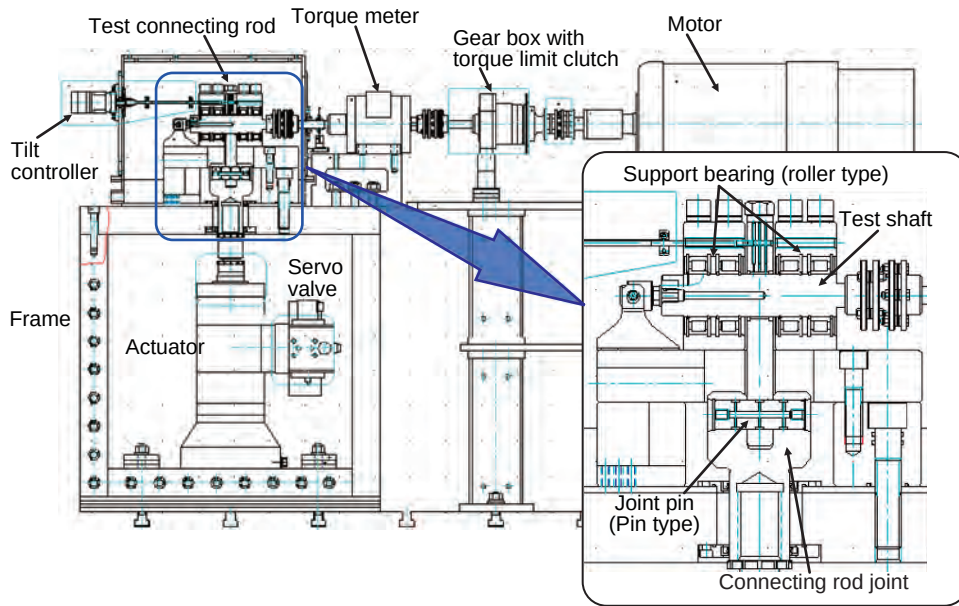


Fig. 3 Structure of new bearing rig tester

has shown that with the columnar construction, it is not possible to achieve the target frequency by either making the columns thicker or increasing the number of columns. Accordingly, a box structure using a sheet thickness that is highly rigid, albeit at the expense of maintainability, was used. Also, the frame structure of the newly fabricated tester was designed in such a way that the frame for the horizontal direction exciter can be joined to the frame for the vertical direction exciter, to enable the test bearing to be excited not only in the vertical direction but also in the horizontal direction as well.

2.2.3. Study of the bearing tester

To excite the test bearing at a high frequency and under a large load, it is necessary to minimize deformation under load of both the frame and the tester. This is because hydraulic oil pump capacity and servo valve flow characteristics are limited. The main items implemented for this purpose are as follows:

- (1) Use of a double row of support roller bearings and an increased interference setting for the outer race
- (2) Use of a more rigid support bearing housing

- (3) Elimination of the ball joint between the connecting rod and connecting rod joint and minimization of the clearance between them
- (4) Use of an increased number of clamping bolts for each part

Figure 5 shows photographs of the conventional tester and the new tester, and also the time system waveform of load versus output from the displacement sensor mounted on the actuator when these testers are used. As a result, it was found that with the conventional tester, the ± 100 kN-converted displacement was 1.60 mm, which is excessively large. It was found that there were two causes for this displacement; deflection of the various parts, and looseness due to clearances. By enhancing the rigidity of the parts concerned and minimizing the various clearances, it was possible to reduce the amplitude to a ± 100 kN converted value of 0.59 mm.

		Column-type frame	Box-type frame
FEM model			
Resonance frequency (Hz)	Front and rear	280.4	389.3
	Right and left	264.6	437.0
	Twist	222.3	750.0
	Above and below	480.6	787.1

Fig. 4 Resonance frequency difference by frame type

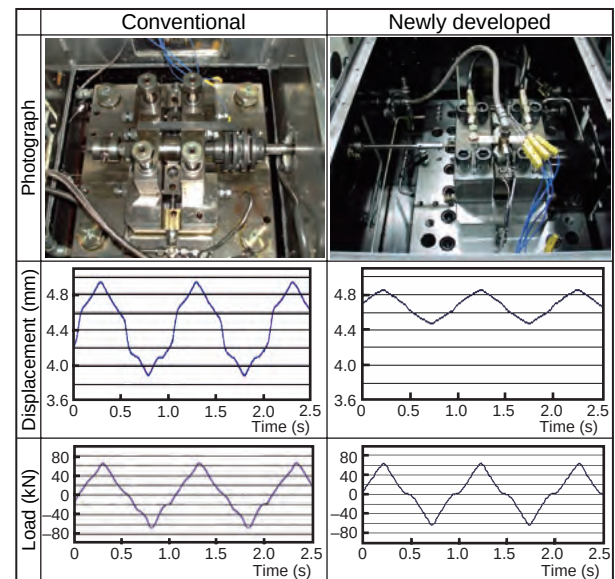


Fig. 5 Displacement of exciter piston by load

2.2.4. Tilt control system

The results of the bearing test are affected by a number of factors.

Compatibility of data is important when performing a comparison of the bearing itself, such as a comparative evaluation of the basic performance of the bearing material. In such a case, it is necessary to remove influencing factors other than the bearing, such as the rigidity of the housing, and so on, before carrying out the tests. The use of a highly rigid connecting rod such as that shown in Fig. 6 is effective.

However, when a highly rigid connecting rod is used, misalignment of the connecting rod and the shaft greatly affects the results of the test. It is thought that this is due to the fact that with a low rigidity connecting rod, parallel contact is achieved by a self-fitting process resulting from elastic deformation and bending of the stem of the connecting rod. With a highly rigid connecting rod, this self-fitting effect is minor. To address this issue when using the previously used static load tester, a ball joint was used at the connection point between the exciter and the connecting rod. It was found, however, that as the load increased, the effectiveness of these bearings was reduced and the displacement under load increased. Use of the ball joint was therefore discontinued.

Figure 7 shows the test bearing temperature measurement points and the connecting rod tilt adjustment mechanism when a highly rigid connecting rod is used.

A feed mechanism driven by a step motor is installed on the connecting rod, three thermocouples T_R , T_M and T_F are installed in the width direction of the bearing. T_M measures the temperature of the rear face in the middle of the bearing, and T_R and T_F measure the temperatures of the rear faces of the ends of the bearing.

When misalignment occurs, a greater amount of heat is generated on the side where clearance between the shaft and the bearing is smaller, hence the temperature rises. Accordingly, the temperatures of T_R and T_F at the ends of the bearing were measured, and both temperatures were maintained at a constant level, enabling the test shaft and bearing to make parallel contact with each other.

Figure 8 shows the conditions for the test performed using this adjustment mechanism, and Fig. 9 shows the differences in temperature when the misalignment adjustment mechanism was installed and when it was not installed.

When there was no misalignment adjustment mechanism, the difference between T_R and T_F increased, and in the zone where the temperature difference widens and the effect of misalignment becomes severe, the temperature waveform became irregular. It can be seen that when there is a misalignment adjustment mechanism, T_R and T_F are maintained at constant levels. It was found to maintain constant contact between the shaft and bearing, it is necessary to adjust the feed amount together with the load.

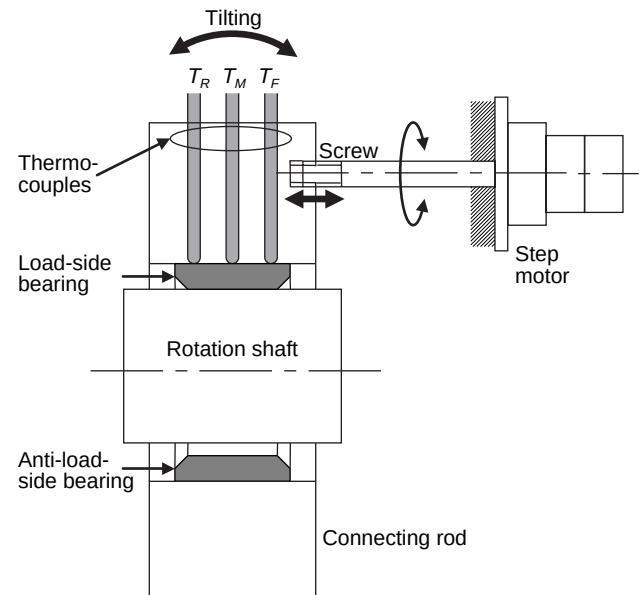


Fig. 7 Temperature measuring points and tilt control system of bearing

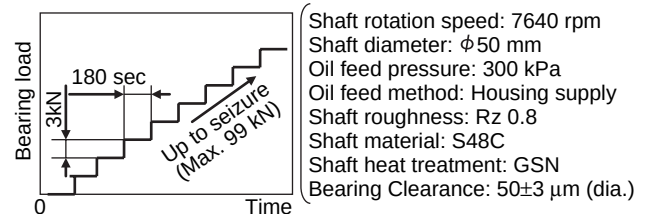


Fig. 8 Test pattern and example conditions in static load testing

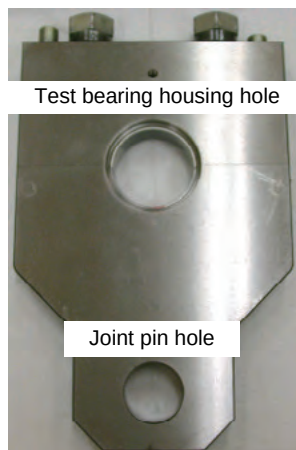


Fig. 6 Rigid-type connecting rod

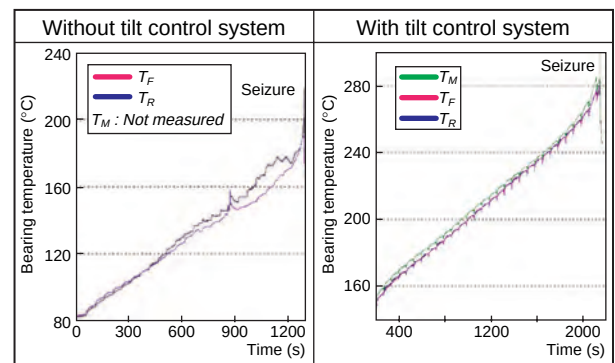


Fig. 9 Comparison of bearing temperature by tilt control system

3. Results of Tests using the New Bearing Tester

3.1. Results of Seizure Test

Figure 10 shows the results of changing the bearing material, and Fig. 11 shows the construction and surface hardness of the test bearing.

From the temperature results, it was seen that in the case of an aluminum alloy bearing, the temperatures of T_R and T_F at the ends of the bearing tended to be higher than the temperature of T_M at the middle of the bearing. It is thought that this is due to the fact that the sliding conditions at the ends were more severe than that at the middle of the bearing. With a lead alloy bearing, on the other hand, the temperature differences between T_R , T_M and T_F were not as large as in the case of an aluminum alloy bearing. Also, in a comparison of an aluminum alloy bearing and a lead alloy bearing, it was found that the lead alloy bearing seized more quickly.

To explain this fact, the temperature difference D_T was defined according to the following equation.

$$D_T = T_M - \frac{T_F + T_R}{2} \quad (1)$$

D_T : Temperature difference

T_M : Bearing middle temperature

T_F : Bearing front temperature

T_R : Bearing rear temperature

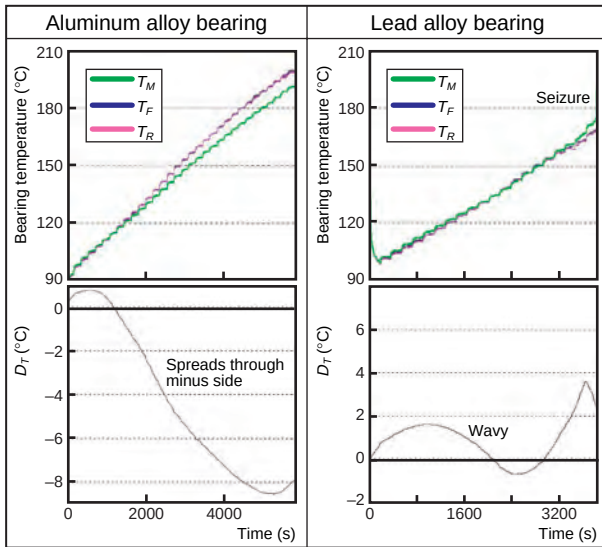


Fig. 10 Typical temperature pattern of aluminum alloy and lead alloy bearings

Bearing type	Aluminum alloy bearing	Lead alloy bearing
Construction	Aluminum alloy 200 μm Bonding layer 50 μm Back steel Reminder	Over-layer 15 μm Lining 200 μm Back steel Reminder
Surface hardness	50 (HV5)	13-17 (HVM10)

Fig. 11 Seizure test sample construction and hardness

The D_T of an aluminum alloy bearing was initially a positive value, and subsequently it tended to shift markedly to the negative. In contrast, the D_T of a lead alloy bearing did not increase significantly and tended to be wavy.

Figure 12 shows the form of wear of the bearing resulting from the test. The bearing wear width B_W after the test is defined as shown in Eq. (2).

$$B_W = B_A - B_B \quad (2)$$

B_W : Bearing wear width

B_A : Original effective width prior to test

B_B : No wear width after test

Figure 13 shows the relationship between the bearing wear width B_W and the bearing load.

The results demonstrated that the wear of a lead alloy bearing is greater than an aluminum alloy bearing when both are subjected to the same load. It can be said that the sliding face formed on the lead alloy bearing is caused mainly by wear, while that sliding face formed on the aluminum alloy bearing is caused mainly by elastic deformation. The sliding face is formed by the break-in

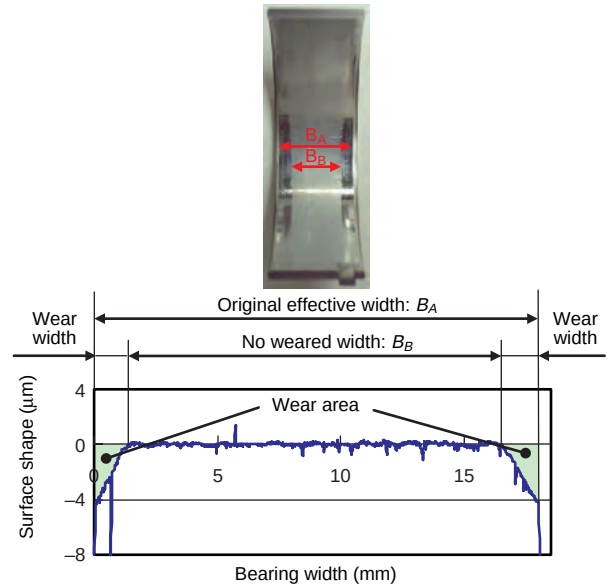


Fig. 12 Bearing shape features after testing

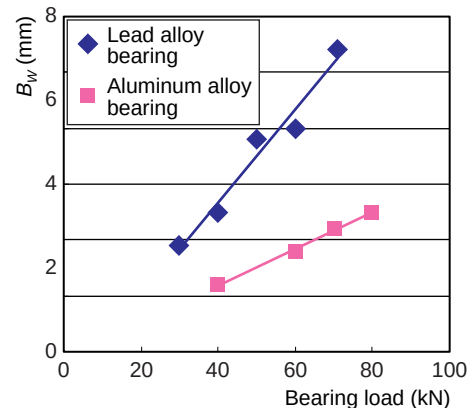


Fig. 13 Wear width of bearing by testing load

action. It is thought that the break-in form of a lead alloy bearing is due to wear, while an aluminum alloy bearing is largely due to elastic deformation. The break-in form was shown to differ greatly depending on the bearing material. It was also demonstrated that by measuring the temperature of the bearing it is possible to study one part of the break-in process of a bearing.

3.2. Friction Coefficient Test Results

Because the new tester measures temperature to provide the ideal conditions for maintaining the shaft and bearing in parallel, a test was performed to obtain the coefficient of friction of the bearing.

Figure 14 shows the test conditions. The test method used involved applying a 25 Hz amplitude load to gradually reduce the rotational speed, thereby making the sliding conditions more severe. Figure 15 shows the results obtained by repeating the test conditions described above.

As the number of rotational measurements increases, the friction decreases and stabilizes. It was found that if this process is repeated four or five times, the friction stabilizes, and a break-in condition is obtained. Figure 16 shows a graph of continuous data converted into a non-dimensional coefficient of friction, which was obtained by measuring the friction when different bearings (A, B and C) from the same lot were installed in the tester. To calculate the friction graph, the load was set to the maximum value of the

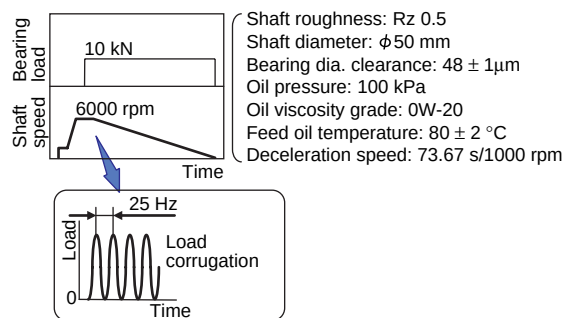


Fig. 14 Coefficient of friction test conditions

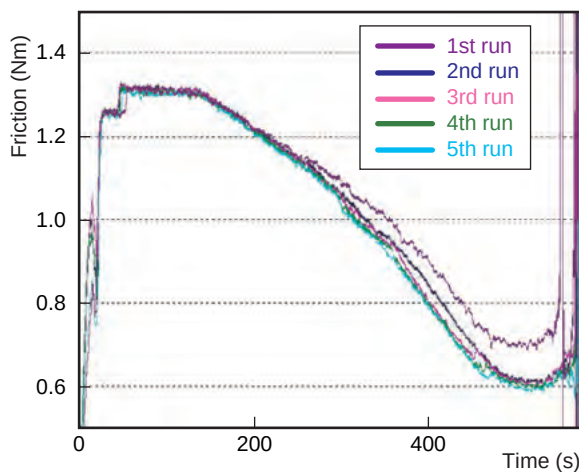


Fig. 15 Results of repeated friction test (using same bearing)

amplitude load, and the oil viscosity was measured using the mean value of the bearing rear face temperature for each sampling time. It was confirmed that even when the bearing installed in the tester was changed over, the same results were obtained, and by measuring the temperature and maintaining the shaft and bearing parallel to each other, it was possible to evaluate the coefficient of friction with high reproducibility.

3.3. Results of Test with the Bearings fitted on an Actual Connecting Rod

The new tester can perform basic tests using a rigid connecting rod as well as under sliding conditions that approximate those in an actual engine. The following is a description of the results of tests performed using actual engine parts.

In the engine used for this test, the connecting rod small-end bush was eliminated to reduce the weight of the reciprocating parts. When the bush was eliminated, the piston pin was surface-treated to provide better lubrication against the piston pin. In selecting this surface treatment, a test was carried out using the new bearing tester. Figure 17 shows the installed condition of the test connecting rod on the tester, and Fig. 18 shows the load condition. The piston pin was rotated at a constant speed of 2500 rpm by a motor. The load was applied from the big end of the connecting rod from 0 kN to a maximum compressive load of 40 kN at a frequency of 100 Hz, and then excited to the minimum tensile load of 20 kN over a period of 20 seconds. After 100 total cycles, the test was deemed complete. Oil was dripped into the hole at the small end of the connecting rod by a step-motor-driven oil pump.

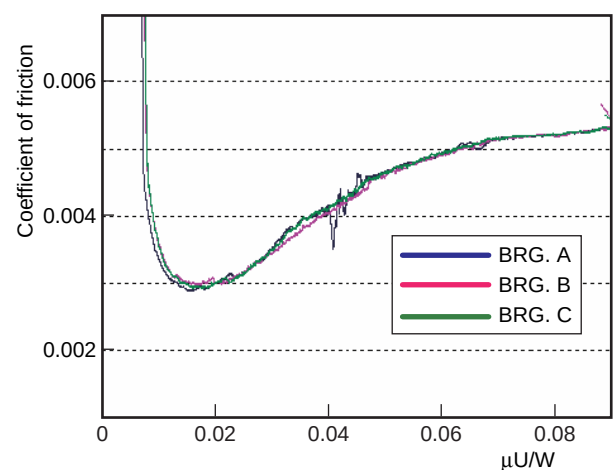


Fig. 16 Friction coefficient curve

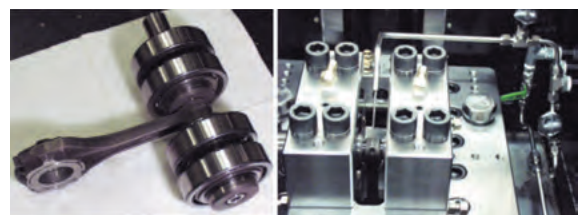


Fig. 17 Connecting rod (left); rod connected to test equipment (right)

Figure 19 shows a pattern diagram of the friction waveform obtained. The peak value of the stabilized friction is defined as F_m , and the peak value of the friction that increased midway is defined as F_p .

Figure 20 shows the types of surface treatment to which the piston pin and the hole in the small end of the connecting rod were subjected and photographs of the piston pin after the test. The surface treatments used were selected on the basis of high film strength, resistance to separation, and/or good lubricity. Figure 21 shows the measured friction results.

The condition of the lubrication was judged based on the following points:

- (1) Small amount of steady-state friction.
- (2) Small increase in friction due to scuffing.

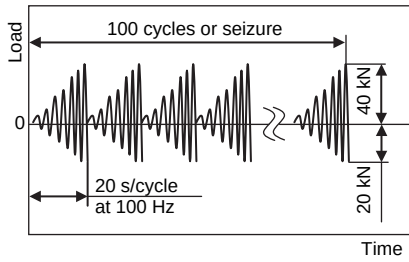


Fig. 18 Testing load conditions

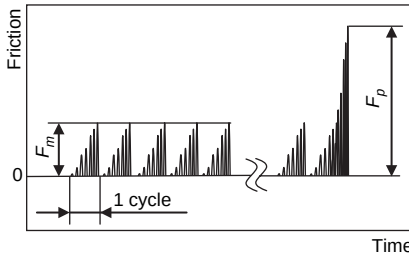


Fig. 19 Friction feature and definition of F_m and F_p

Regarding the results using different specifications, A was considered to be strong because of the nature of the film. However, it had large F_m and F_p values and exhibited low lubricity. B showed stable initial friction, but had a relatively large F_m value, and the friction was no longer stable after scuffing occurred. C showed little friction under steady-state conditions, and exhibited superior scuffing performance. In the case of D and E, scuffing occurred intermittently. F showed the smallest F_m value of all the lubricants tested, and exhibited good lubricity. The piston pin and connecting rod that had been surface-treated with F were tested in an actual engine. They met the standards of the actual engine test, so F was selected for use in commercial automobiles. This method was effective when applied to determine the bearing specifications for the engine.

Type	Surface treatment		Photograph after testing
	Piston pin	Piston pin boss	
A	Titanium nitride	Carburizing	Could not be pulled off
B	Prototype ferrous oxide coating (Fe_3O_4)	Carburizing	
C	Sulfurizing	Carburizing	
D	Prototype ferrous oxide coating (Fe_3O_4)	Sulfurizing after carburizing	
E	Prototype ferrous oxide coating (Fe_3O_4)	Shot blasting after carburizing	
F	Ferrous oxide coating (Fe_3O_4)	Carburizing	

Fig. 20 Surface treatment and surface status after testing

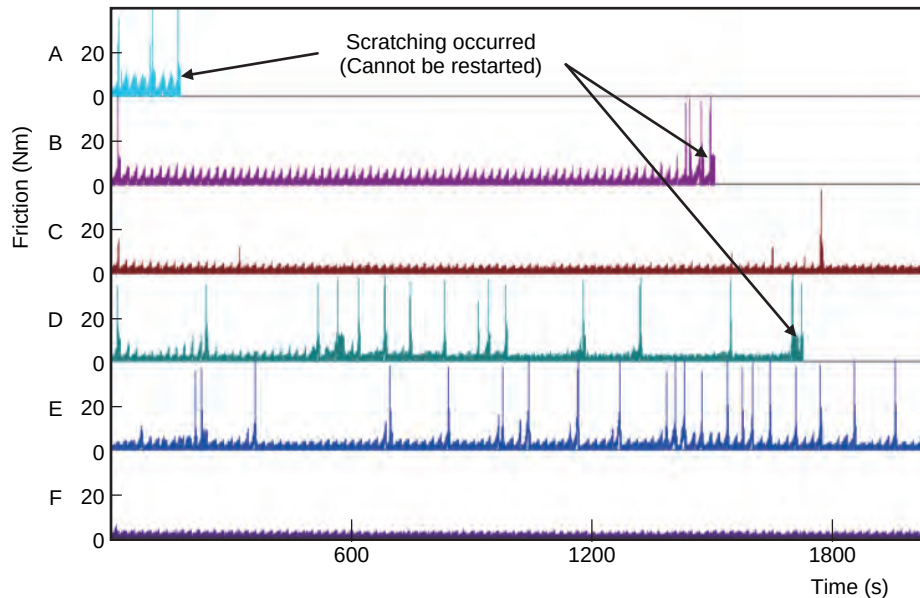


Fig. 21 Difference in friction by surface treatment

4. Conclusion

A unique dynamic load tester was devised that is capable of reproducing the sliding condition of an actual engine bearing and increasing testing efficiency by means of rig acceleration test, both issues for conventional static load bearing testers. The following results were obtained.

- (1) As a result of performing a comparative seizure test on lead and aluminum alloy bearings, it was found that the form of break-in of lead alloy bearings was due to wear, whereas the form of break-in of aluminum alloy bearings was largely due to elastic deformation.
- (2) It is possible to study the bearing break-in process by measuring the bearing temperature during break-in.
- (3) By measuring the temperature and maintaining the shaft and the bearing parallel alignment to each other, it was possible to evaluate the coefficient of friction with high reproducibility.
- (4) This method reproduced sliding conditions that approximate those of an actual engine, indicating that it is an effective method to select bearing specifications.

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