

Development of Rolling Bearing Model for Mechanical Dynamic Analysis Considering Elastic Properties

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ABSTRACT

To enhance the simulation accuracy for mechanical dynamic analysis of an assembled crankshaft often used in single cylinder, small displacement engines, it is important to make a detailed model of the rolling bearings simulating physical phenomena such as deformation, contact, and damping. A rolling bearing model that permits simulation of a three-dimensional contact considering elastic properties was developed, and the analysis accuracy was enhanced. By applying the rolling bearing model to the mechanical dynamic analysis of the assembled crankshaft, the correlation between the crankshaft displacement and the strain during actual engine operation was confirmed. As a result, it was confirmed that the rolling bearing model is accurate enough for a practical strength evaluation in the design stage of an assembled crankshaft system.

1. Introduction

In recent years, mechanical dynamic analysis of the crankshaft system has been an essential tool for the specifications of an engine to reach a level of maturity during the initial design stages. As for the one-piece crankshaft of multi-cylinder engines, the mechanical analysis technologies⁽¹⁾ have been constructed and applied to actual development as the investigation design tool. This is not only within Honda, many kinds of versatile CAE packages have been released and are widely used within the industry. On the other hand, the assembled crankshafts are widely adopted in small single-cylinder motorcycle engines, and differ from multi-cylinder engines in that rolling bearings are used in the main journal.

In the case of one-piece crankshafts, sliding bearings are used, where fluid film lubrication theory is incorporated in order to enhance analytical accuracy. On the other hand, conventional methods of rolling bearings consist simply of linear or non-linear simple springs and dampers. These are much too low to attain acceptable accuracy for analysis of crankshaft behavior, where the journal bearings are affected by heavy fluctuating loads.

In this paper, a new model for a rolling bearing is developed, considering the elastic properties of the rolling elements and the bearing rings and simulating three-

dimensional contact between them. This model was then applied to mechanical dynamic analysis of assembled crankshaft systems.

2. Model of Rolling Bearing

Figure 1 shows the structure of a single-cylinder motorcycle engine with an assembled crankshaft. Rolling bearings are installed in the main journal of the crankcase to support the crankshaft. At the running condition, rolling bearings show very complex behavior. They contain multiple rolling elements that are held between the inner and outer rings by retainers and are in constantly changing contact conditions due to the changing conditions of the crankshaft (engine speed, torque fluctuations). Many papers have been published by the bearing companies^{(2), (3)} about bearing behavior, but the purpose of these is to analyze behavior of the bearings themselves, detailed behavior such as influences by a rolling contact surface and tolerance of rolling elements are analyzed; they are not suitable for mechanical dynamic analysis of whole crankshaft systems. Therefore, this rolling bearing model was developed under several assumptions, while the behavior of the bearing itself was not considered because it would be relatively unaffected by heavy fluctuating loads.

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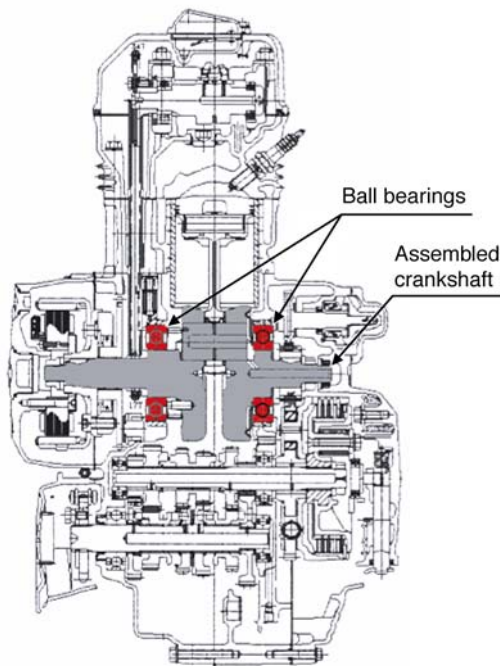


Fig. 1 Structure of single-cylinder engine

2.1. Requirements for Development

It was mentioned above that the conventional methods for a bearing model adopt simple linear or non-linear springs and dampers. In these methods, the properties of contact stiffness between the rolling element and the raceway of inner and outer rings, both radially and axially, are converted to simple spring properties. These methods were used in an initial attempt at mechanical dynamic analysis of an assembled crankshaft, but the accuracy of calculated crankshaft behavior was insufficient. Therefore, requirements for the new model were decided, with regard to all the factors necessary to provide enhanced analytical accuracy.

The rolling bearings chosen as the object for this development were deep groove ball bearings (ball bearings, hereafter). The requirements included the following:

- (1) This model (ball bearing model, hereafter) must be installed in the existing mechanical dynamic analysis of

crankshaft systems using general mechanical analysis software. It must enable strength evaluations of crankshafts during development of actual engines.

- (2) Three-dimensional contact between the rolling elements (balls) and the inner and outer ring raceway must be accurately calculated.
- (3) The elastic properties (stiffness and damping) of the balls and rings must be accurately calculated.

2.2. Outline of Model

The modeling method in which the ball bearing model is used for the mechanical dynamic analysis was comprised of two systems, the creation of the ball bearing properties and the main calculations. The overall system flow for creating the ball bearing model is shown in Fig. 2.

2.2.1. Creation system of ball bearing property

This defines the three-dimensional contacts for one ball in the ball bearing model and creates the property file. As shown in Fig. 2, the creation system of the ball bearing property consists of three component sub-systems, the pre-processor, the execution of linear static analysis, and the contact processor.

- Pre-processor

In this sub-system, the specifications for the ball bearing are input and the finite element method (FEM) data required for linear static analysis of the balls and rings are generated. The FEM data includes the geometric mesh, the loads and the boundary conditions.

- Execution of linear static analysis

This sub-system uses general structural analysis software to execute a linear static analysis of the bearing using the FEM data generated by the pre-processor. It creates the nodal stiffness data set used by the contact processor.

- Contact processor

This sub-system executes the analysis of the three-dimensional contact between the balls and the rings, using the nodal stiffness data created by the linear static analysis. As shown in Fig. 3, the orbital position angle α is determined by the number of balls, and the contact on a radial plane defined by the bearing axis and angle $\alpha^{(4)}$. The penetration δ and the contact angle β between the ball and the raceway are defined, and the contact load f , a function of δ and β , is calculated using the nodal stiffness data. Next, the

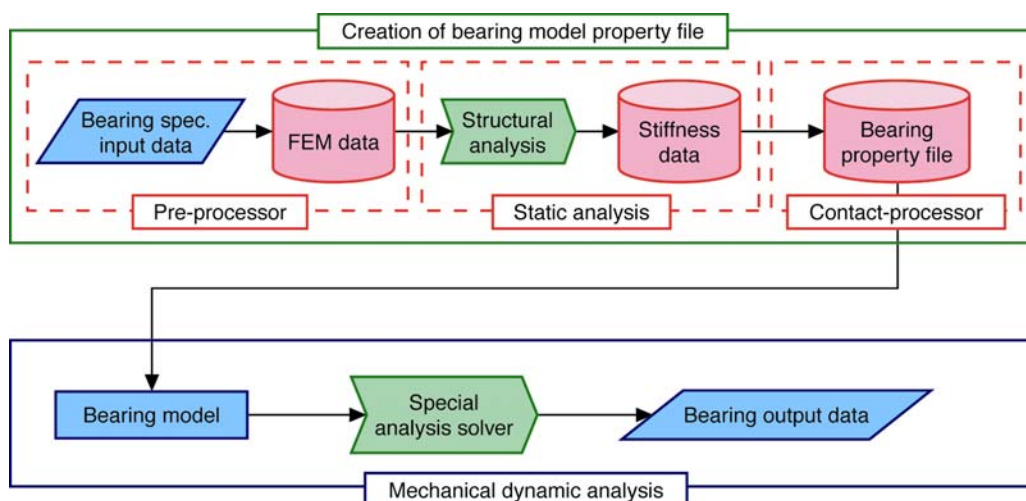


Fig. 2 System flow

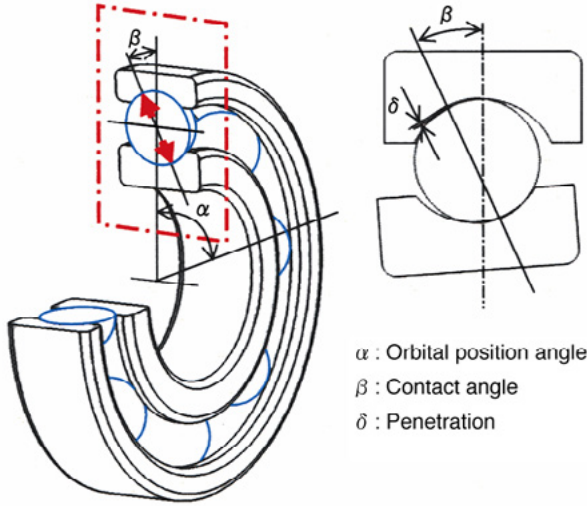


Fig. 3 Figure of contact method

three-dimensional contact loads in the inner and outer rings are calculated using δ , β and f . Then the contact processor finds the three-dimensional contact function of the inner and outer rings. This function is exported as a ball bearing model property file.

2.2.2. Main calculation system

The method to apply the ball bearing model to a mechanical dynamic analysis must be applied to numerical integration methods in the time domain. Also, the loads in six axes must be calculated considering the bearing loads caused by the deformation and the behavior of the crankshaft as they are transmitted to the crankcase. An exclusive built-in solver based on user subroutines was newly developed and applied to the existing general mechanical analysis software.

Output of the loads in six axes by the built-in solver follows this procedure:

- (1) The displacements, angles, velocities and angular velocities from the center of the case journal to the center of the shaft journal are calculated.
- (2) The contact loads and damping for one ball are calculated by using the ball bearing model property file.
- (3) The contact loads, torque around the axis and the damping for all the balls are calculated. Then the loads in six axes are exported as a ball bearing model.

A definitional Eq. (1) of the contact load vector $\{F\}$ of the ball bearing model is shown below. Here, $\{F_i\}$ is the contact load vector for one ball, f is the contact load obtained from the property file, C is the damping parameter, and $\{v_i\}$ is the translational velocity vector.

$$\{F\} = \sum_{i=1}^n \{F_i\} = \sum_{i=1}^n \{f(\delta_i, \beta_i) + C(f(\delta_i, \beta_i)) \cdot \{v_i\}\} \quad (1)$$

In the same way, a definitional Eq. (2) for the torque vector $\{T\}$ around the axis of the ball bearing model is shown below. Here, $\{T_i\}$ is the torque vector around the axis for one ball and $\{l_i\}$ is the distance vector from the case journal center point to the contact point.

$$\{T\} = \sum_{i=1}^n \{T_i\} = \sum_{i=1}^n \{ \{f(\delta_i, \beta_i) + C(f(\delta_i, \beta_i)) \cdot \{v_i\}\} \cdot \{l_i\} \} \quad (2)$$

3. Verification of Model

The function of the developed ball bearing model was verified. This process consisted of two steps, one is verification of the component function of the ball bearing model itself, and the other is verification of the assembly function of the model applied to the mechanical dynamic analysis of the crankshaft system.

3.1. Verification of Component Function

Static stiffness of the actual ball bearing and the FEM model had been verified in advance. By the results, it was clarified that there is good correlation between the load and the displacement. Therefore, verification of static stiffness analysis of the ball bearing model and the FEM model was applied to that of the component function.

3.1.1. Verification method

As shown in Fig. 4, two directional loads were applied and linear static analysis was executed. A radial load F_r was applied perpendicular to the axis, and an axial load F_a was applied parallel to the axis.

In the radial direction analysis, function was verified on layout (a) and (b) as shown in Fig. 5, because the initial number of contacting balls and radial clearance are changed by the ball orbital angles.

3.1.2. Verified results

Table 1 shows the specifications of the ball bearing used for the verification test of the component function. When the static load was applied to the shaft model installed inside of the inner ring, elastic deformation occurred in the balls and raceway surface at the contact areas. The elastic deformation

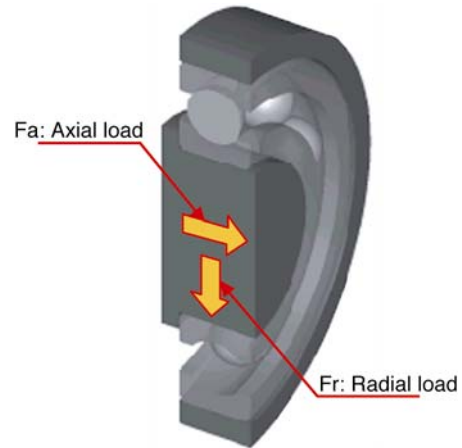


Fig. 4 Figure of component static analysis

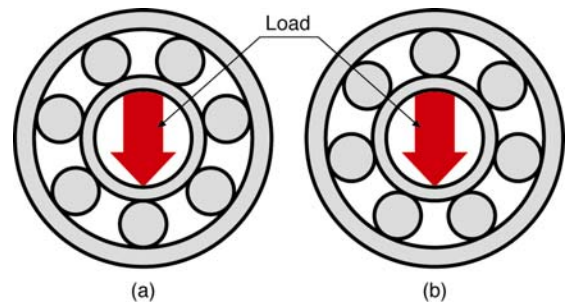


Fig. 5 Figure of balls layout

Table 1 Ball bearing specifications

Outside diameter	72 mm
Inside diameter	35 mm
Width	15 mm
Diameter of ball	7/16 inch
Number of balls	9

δr in the radial direction and the radial clearance c_{lr} were summed to define the radial displacement d_r . Similarly, the axial elastic deformation δa was summed with the axial clearance c_{la} to define the axial displacement d_a . The obtained displacement results are shown in Fig. 6.

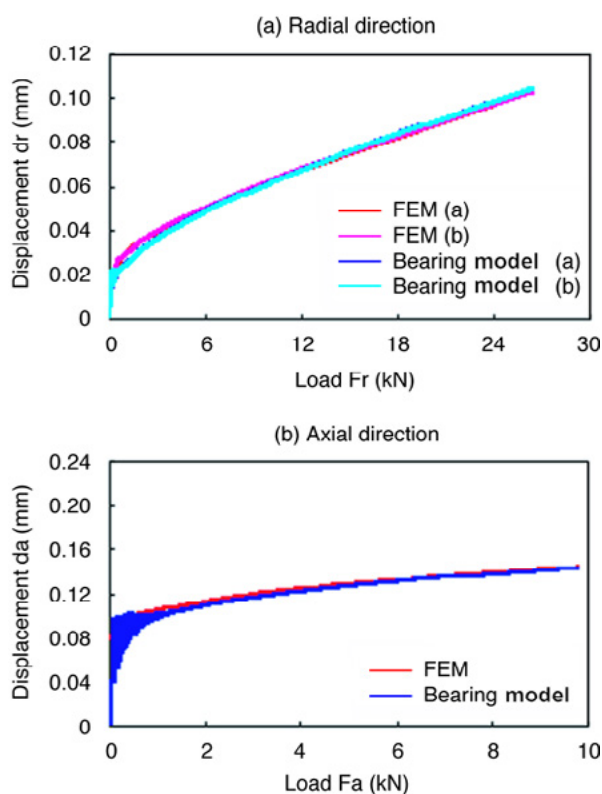


Fig. 6 Results of component static analysis

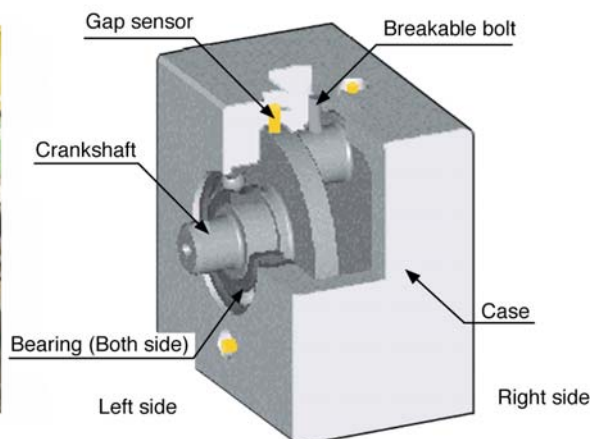
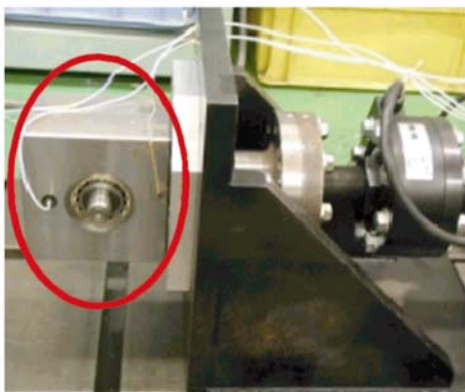


Fig. 7 Figure of test system and measuring device model

The displacement results from the FEM and ball bearing model showed good correlation, indicating the three-dimensional contact behavior properly by the new model, from low loads up to high loads. The ball bearing model satisfied the static functional requirement for mechanical dynamic analysis.

3.2. Verification of Assembly Function

The ball bearing model used in the component verification process was applied for the assembly verification. Test equipment with a simplified crankshaft and crankcase was made and correlation between experiments and calculations was verified. Figure 7 shows a picture of the whole test system and the model of measurement device.

“L” and “R” refer to the “left” and “right” sides of the measurement device below. The yellow components in the crankcase are gap sensors; these were fixed in the upper portion on the L and R sides, and next to the crank web on the L and R sides, for a total of four sensors.

3.2.1. Verification method

A bolt was screwed in the crank pin of the crankshaft and a tensile load was applied with a hydraulic tester. A notch was machined in the bolt before assembly so that the bolt would break at specified load; the impact load test is a method of using the impact caused by the break of the bolt as the exciting force. The pre-break behavior was defined as the static zone, and the post-break behavior, as the dynamic zone. The displacements of each area of the crankshaft were measured by four gap sensors fixed in the crankcase.

3.2.2. Static verification results

Figure 8 is a displacement result at the gap sensor on the upper L side of the assembly, until the bolt breaks under the tensile load.

There was good correlation between the experiments and the calculated results. The displacement increased with tensile load. The stiffness and three-dimensional contact conditions were calculated very accurately by the ball bearing model in the static analysis of the assembly. The function of the ball bearing model has successfully satisfied the requirement.

3.2.3. Dynamic verification results

The bolt breaks at the specified load, and the impact by the breaking and the elastic deformation excites the crankshaft. Figure 9 shows the time series of crankshaft displacement, indicating the dynamic behavior of this system.

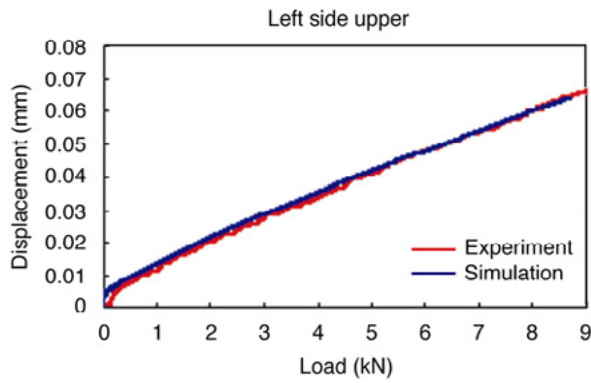


Fig. 8 Result of static assemble analysis

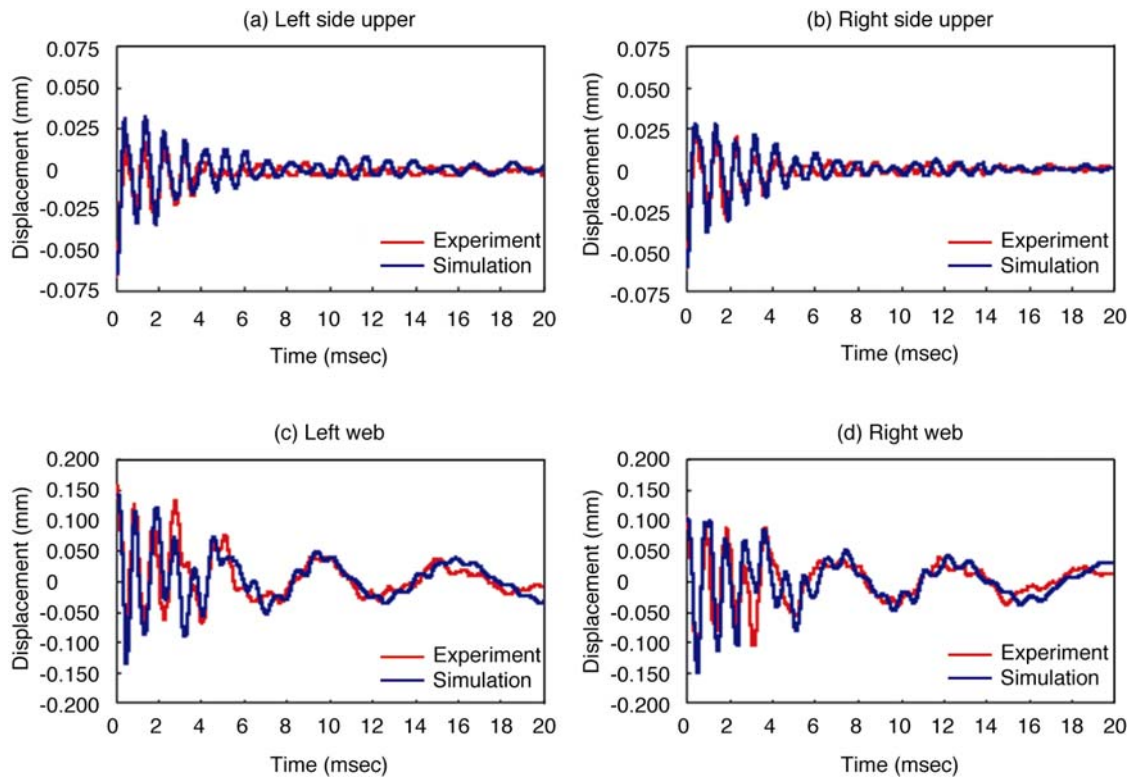


Fig. 9 Results of dynamic assemble analysis

The displacement results at the conditions where the impact excitation and vibration converge show a very good correlation.

Next, the time series of the displacement was used to analyze a power spectral density, which is a standard frequency analysis. Figure 10 shows power spectral density results for dynamic behavior of the crankshaft. The experiments and the calculation matched particularly well around 1 kHz, and the correlation is generally good. This indicated that the dynamic properties of the ball bearing model were also sufficiently accurate.

3.3. Summary of Model Verification

The function of the ball bearing model in the component and assembly tests satisfied the requirements in both static and dynamic properties. It was therefore applied to a mechanical dynamic analysis of an actual assembled crankshaft.

4. Application Example

The mechanical dynamic analysis model of the assembled crankshaft of a motorcycle engine was created as shown in Fig. 1, and the correlation was verified by an actual experiment. Table 2 shows the test engine specifications. Figure 11 shows the figure of the mechanical dynamic analysis model used for the analysis.

4.1. Details of Verification Test

The engine was operated in a constant speed mode from 4 000 to 10 000 r/min with increments of 2 000 r/min.

As shown in Fig. 12, the displacement of the crank-web was measured by a gap sensor fixed in the crankcase, with the Y direction defined from the crank center through the

crank pin, and X defined at a perpendicular angle to that. In order to measure crankshaft strain during the maximum combustion load, a strain gauge was attached on the crank pin fillet on the delayed angle side as shown in Fig. 13.

4.2. Displacement and Strain during One Cycle at Engine Operation

The calculated and measured waveforms for displacement of the crank web on the L side in the Y direction during one cycle at engine speeds of 4 000 and 8 000 r/min are shown in Fig. 14. Figure 15 shows the corresponding waveforms for strain at the crank pin fillet, L side, during one cycle.

At both the low speed of 4 000 r/min and the high speed of 8 000 r/min, the calculated displacement waveform, which included the considerations of the combustion load and inertia, showed a good correlation with the measured waveform. The calculated strain at the fillet during the combustion stroke also showed good correlation with the measured results.

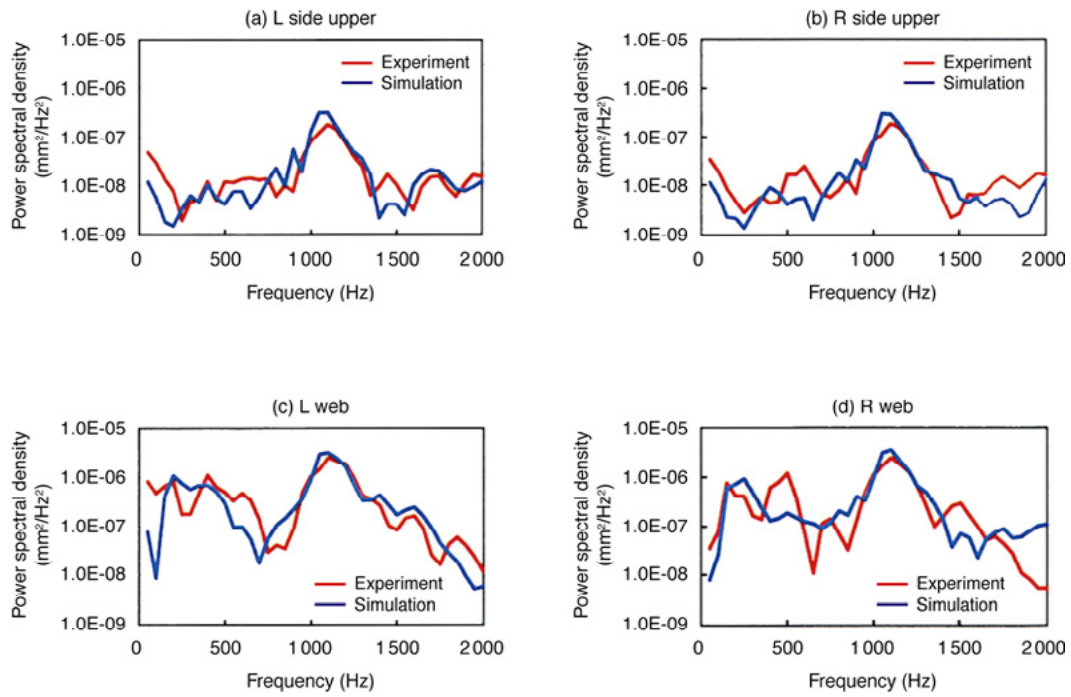


Fig. 10 Results of power spectrum

Table 2 Major specifications of test engine

Items	Specifications
Engine type	4-stroke, single cylinder
Displacement	149.4 cm^3
Valve train	DOHC 4-valves
Clutch type	Multiplate wet clutch
Transmission	6 speeds constant mesh

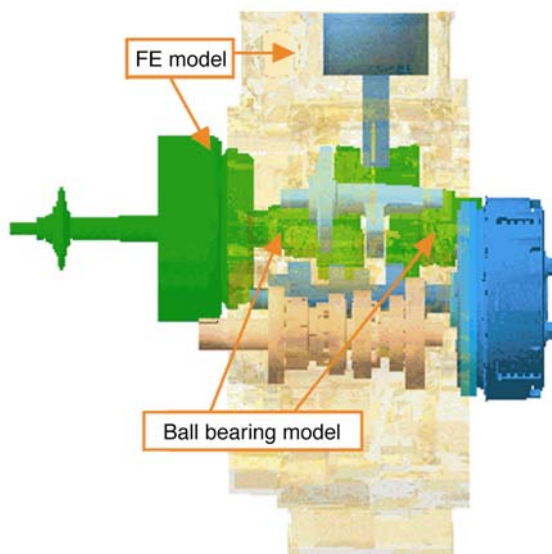


Fig. 11 Figure of mechanical dynamic analysis model

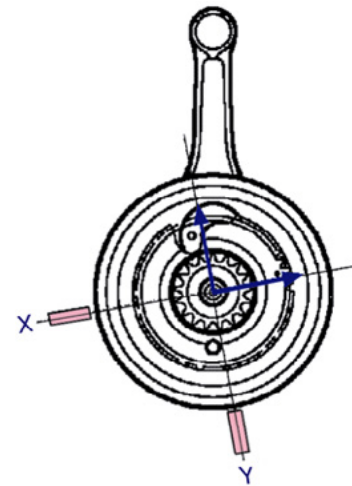


Fig. 12 Position of displacement sensor of Crankshaft-web

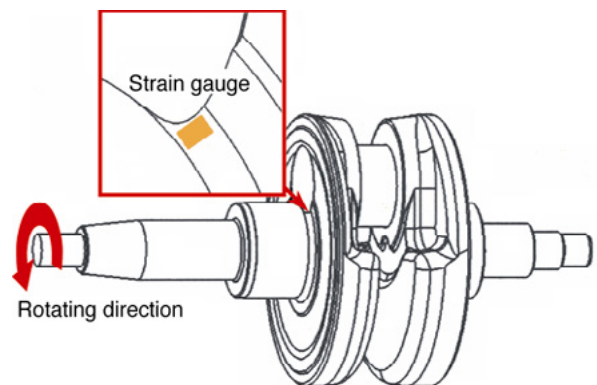


Fig. 13 Strain gauge position

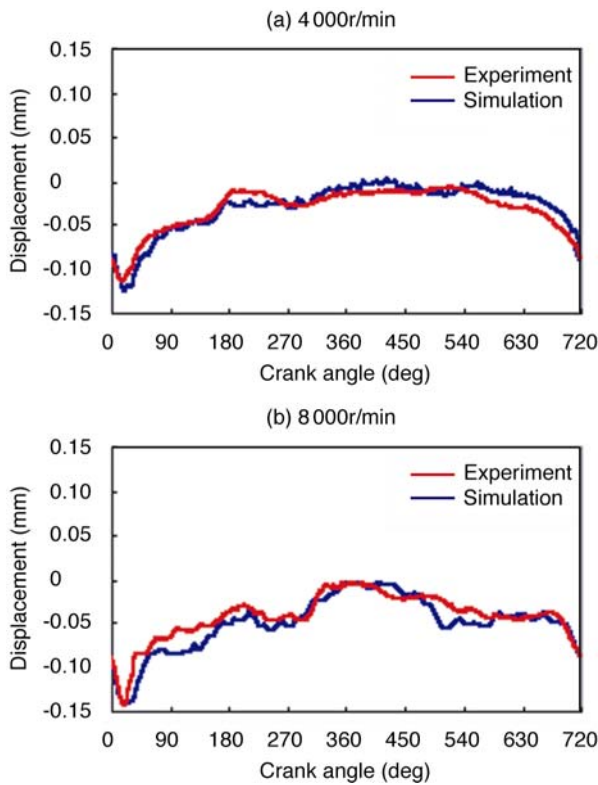


Fig. 14 Correlation results of displacement

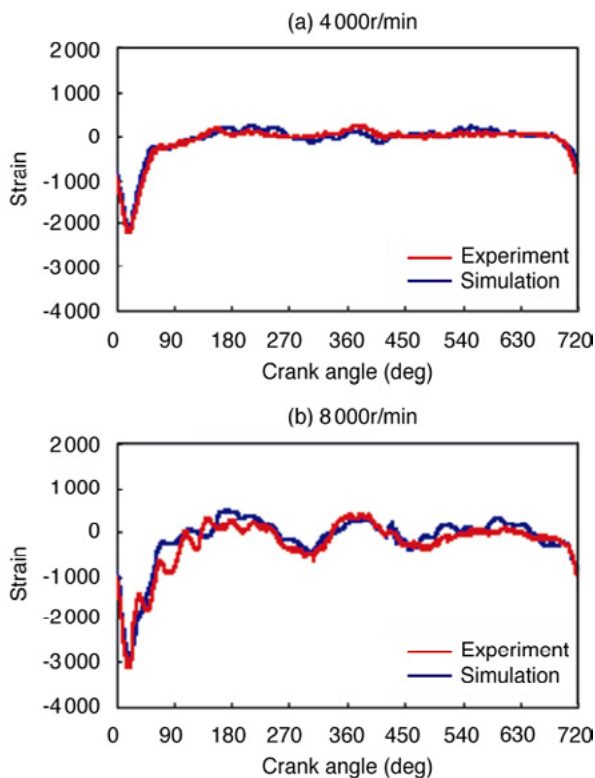


Fig. 15 Correlation results of strain

The above results showed good correlation between the calculated and measured values for displacement and strain at low and high speeds, and sufficiently high accuracy for the strength evaluation of assembled crankshafts.

4.3. Comparison of Bearing Model and Conventional Methods

Figure 16 shows the strain waveforms at the crank pin fillet, L side, during one cycle at 10 000 r/min. These data results were taken in an actual engine and calculated both with the ball bearing model and with a conventional spring-damper model.

The conventional spring-damper model indicated a maximum strain right after the combustion load about 30% higher than that calculated by the ball bearing model and measured in the actual experiment. Its high-frequency vibration results were also much more prominent than those in the actual experiment.

Thus, the ball bearing model showed more accurate results for mechanical dynamic analysis of the assembled crankshaft than the conventional model.

4.4. Summary of Application Results

The mechanical dynamic analysis of an assembled crankshaft, where ball bearings are used under severe condition, was conducted. The ball bearing model in this development has been verified that is the effective method to analyze with high accuracy.

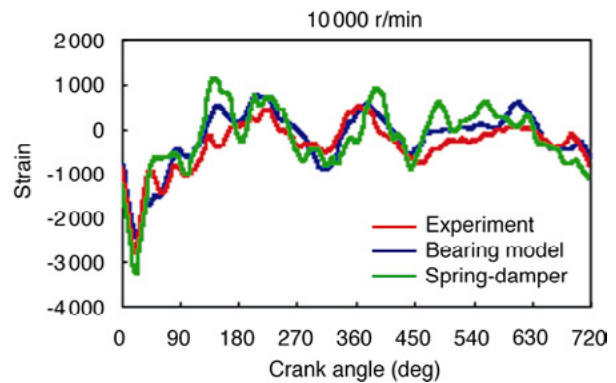


Fig. 16 Comparison of ball bearing model with spring-damper model

5. Conclusion

- (1) A new model of the rolling bearing (ball bearing model) was developed considering elastic properties of the rolling elements (balls) and the rings. The model simulates three-dimensional contact conditions between them.
- (2) The ball bearing model was applied to a mechanical dynamic analysis of the assembled crankshaft. The crankshaft displacement and strain in the firing operation was compared with those of an actual engine. Correlation between them was sufficiently accurate. It was verified that the analysis is suitable for use in a strength evaluation of the crankshaft in the design stages during development of engines.

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