

Development of Simulation Technology for Dynamic Behavior of Assembled Crankshaft Systems in Single-cylinder Engines

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ABSTRACT

When simulating an assembled crankshaft system of the type often used in single-cylinder engines, it is important, in terms of the accuracy of calculations, to adequately represent the influences of crankpin press fitting, the dynamic characteristics of the alternating current generator (ACG) rotor directly mounted to the crankshaft, and the characteristics of the rolling bearing. Actual measurements were performed to verify these characteristics, and a mechanical dynamic analysis model, which accurately represents the dynamic characteristics of the crankshaft system, was established. Also a rolling bearing model that represents the characteristics of three-dimensional elastic contact was developed. Using these in a mechanical dynamic analysis, the dynamic behavior of assembled crankshaft systems was simulated with enough practical accuracy for strength evaluations.

1. Introduction

Because assembled crankshafts are used in a wide range of models in single-cylinder engines powering motorcycles, all-terrain vehicles, and other vehicles, technology for predicting the durability of assembled crankshafts is important in development of single-cylinder engines. However, conventional simulation techniques for the assembled crankshafts offer inadequate accuracy due to the mounting of the crankshaft (in which one or both ends of the crankpin is press-fitted into the crank web) and due to the motion characteristics of the alternating-current generator (ACG) rotor, which is directly mounted to the crankshaft. Given the wide use of rolling bearings, simulation accuracy is also dependent on techniques for bearing modeling. In contrast, for the development of one-piece crankshafts used in multi-cylinder engines, durability prediction has been enabled early on by the dynamic simulation technologies^{(1), (2)} that employ mechanical analysis.

This paper discusses the enhanced prediction accuracy by simulation that is based on validation using unit measurements of the influences particular to assembled crankshafts, and based on mechanical dynamic analysis models that incorporate those properties.

2. Characteristics of Assembled Crankshafts

Figure 1 shows the structure of a typical assembled crankshaft. Assembled crankshafts used in single-cylinder engines exhibit the following five characteristics:

- (1) One or both ends of the shaft are press-fit between the crankpin and crank web.
- (2) Rolling bearings are mainly used for bearings.

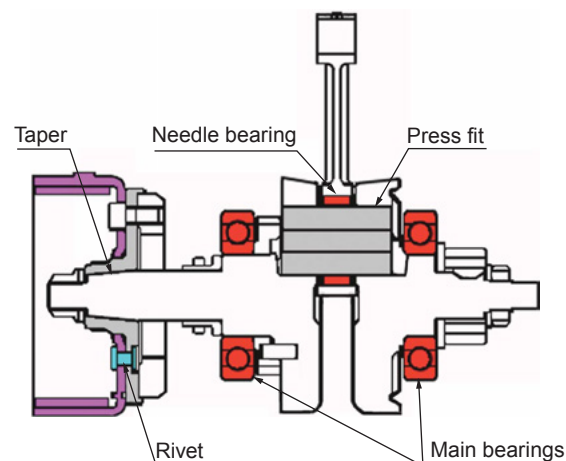


Fig. 1 Structure of assembled crankshaft

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- (3) With the ACG rotor having inertia mass almost equivalent to that of the crankshaft, it is thought that the dynamic properties of the ACG rotor affect crankshaft behavior.
- (4) Drive train units, such as a centrifugal clutch and a belt converter, are often directly connected to the crankshaft.
- (5) Needle roller bearings are used for the sliding portion between the connecting rod and crankpin.

Contact such as the press-fitted structure of the crankpin and rivet joints of the ACG rotor likely affect the natural frequency and damping more than in the case of one-piece construction. Also affecting crankshaft behavior is the dynamic characteristics of the drive train unit that is mounted to the shaft. In addition, because a rolling bearing is supported by a rolling element unlike a plain bearing, which is supported by an oil film, the bearing's quantity, size, and shape can affect bearing stiffness and damping characteristics.

After conducting validation bench tests on changes in dynamic characteristics with an actual crankshaft, a technique for using mechanical dynamic analysis models to represent the characteristics of press-fitting and rivet joints used in typical assembled crankshafts has been developed. To ensure applicability to the broad range of single-cylinder engines currently being mass-produced, mechanical dynamic analysis models for deep-groove ball bearings, cylindrical roller bearings, and needle roller bearings have been developed.

3. Results of Unit Measurements of Crankshaft Dynamic Characteristics

3.1. Characteristics Due to Press-Fitting of the Crankshaft

To determine how differences in crankshaft press-fit margins affect the crankshaft's dynamic properties, a test piece was fabricated and changes in dynamic properties in hammering tests were measured. Figure 2 shows the structure of the test piece. As for press-fitting of the pin, typical press-fit margins seen in mass-produced crankshafts were used to determine how changes in press-fit margins affected natural frequency and the damping ratio. Also, the original crank shaft that integrated the press-fit areas was fabricated, and the change between press-fit types and a one-piece type was verified.

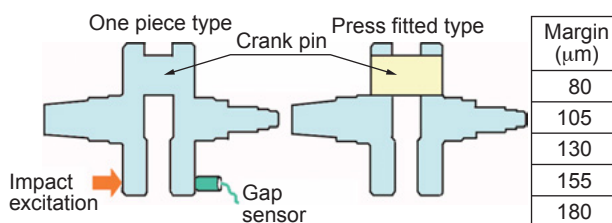


Fig. 2 Figure of press fit test piece

Figure 3 lists the rates of change in natural frequency relative to the one-piece crankshaft due to the differences in press-fit margins in the major vibration modes; Fig. 4 shows the changes in damping ratios. These results show that changes in press-fit margins produced almost no difference in natural frequency, regardless of the vibration mode, and that differences in natural frequency instead resulted from the presence of press-fit components. These differences were particularly noticeable in modes 1 and 4 (bending mode of pitching direction), with reductions of up to 2% compared to the one-piece crankshaft. Change in the damping ratio was greater in the press-fit crankshaft as compared to the one-piece crankshaft but small in absolute values, so that one can conclude that press-fitting had almost no influence.

3.2. Characteristics Attributable to Fastening Structure of the ACG Rotor

Structural features that greatly affect the ACG rotor's dynamic properties include the structure of taper that is bolted to the crankshaft and the rivet structure joining the boss and rotor. To verify the changes of dynamic properties by these connections, test pieces simulating an ACG rotor were fabricated and subjected to the same hammering test performed for press-fittings to measure frequency characteristics and the damping ratio. Figure 5 shows the shapes of the test pieces; Table 1 shows the specifications of test-piece combinations measured.

Figure 6 shows rates of change that each combination of connections produced in natural frequency relative to the one-piece equivalent in the major vibration modes. These results show that the tapered shaft generated the

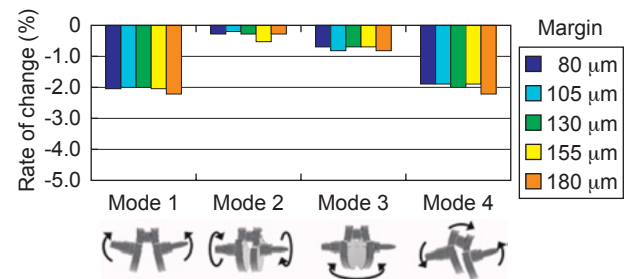


Fig. 3 Rate of eigen frequency with press fitting

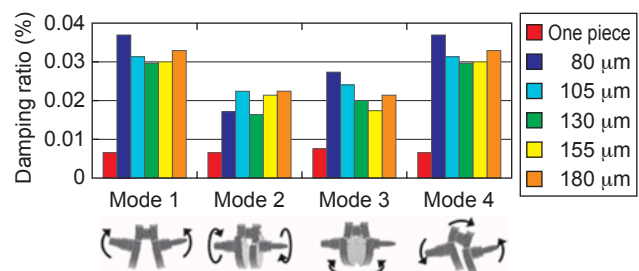


Fig. 4 Damping ratio with press fitting

Table 1 Specifications of ACG rotor test pieces

Type	Between rotor and boss	Between boss and shaft
I	One piece	One piece
II	One piece	Taper & Thread tightening
III	Riveting	Taper & Thread tightening

natural frequency to differ by only 2–3%, depending on the vibration mode. In contrast, the rivet joints generated a sharp reduction (relative to the one-piece equivalent) in the natural frequency. Varying according to mode, this decrease was the largest in Mode 3 (axial directional vibration), in which the rivet joints alone influenced a 14% reduction.

Figure 7 shows the results of unit measurements of damping ratio. Trends in the damping ratio were found to be similar to those of natural frequency, with the shaft's taper having almost no influence but the rivet connection

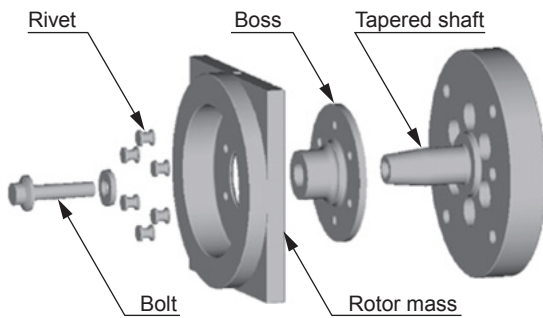


Fig. 5 Structure of ACG rotor test pieces

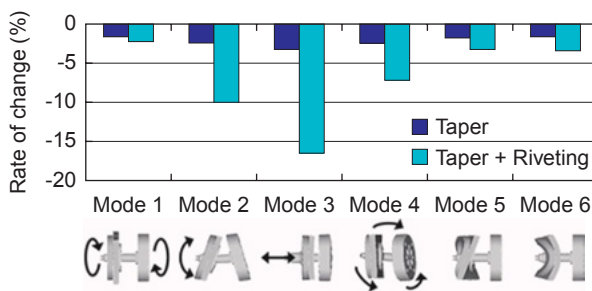


Fig. 6 Rate of change of eigen frequency with taper and rivet

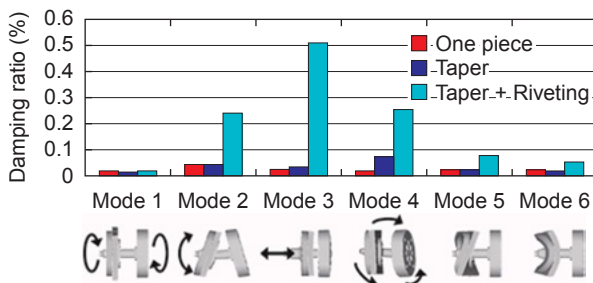


Fig. 7 Damping ratio with taper and rivet

having an influence. Change in the damping ratio was the greatest in mode 3 (in which change in natural frequency was also the greatest)—roughly 0.04% without the rivet versus an increase to 0.5% with the rivet, or an approximately tenfold increase.

4. Construction of the Mechanical Dynamic Analysis Model

The methodology with which an elastic model is constructed is an important issue when conducting a mechanical-dynamic-analysis-based simulation of the dynamic behavior of a crankshaft. Using the bench test described above to clarify the influences of press fitting and riveting on a crankshaft's dynamic characteristics, a finite element model (FE model) with elastic property was created to mimic the characteristics of press fitting and riveting to enhance accuracy of the simulation.

With regard to rolling bearings, a mechanical dynamic analysis model was developed to encompass not only standard ball bearings but also roller bearings and needle bearings (widely used at the big end of connecting rods, etc.) so as to assure applicability of this simulation to a diverse range of engine specifications.

4.1. Modeling of the Press-fitted Part of the Crankpin

Because actual testing showed that press-fitted components lower natural frequency, the elastic model was devised to account for the stiffness characteristics of press-fitted components. With the crankpin and right and left halves of the crankshaft represented with a separate FE model, the press-fitted surfaces were connected with spring elements to represent the change in stiffness due to press fitting. The spring elements were configured into a cylindrical coordinate system whose axes pass through the crankpin center, and stiffness characteristics were given in three axial directions. Regarding spring stiffness characteristics, actual test results were compared with the natural frequency determined by the FE model, and the optimum value was determined so as to minimize the error in each vibration mode.

Figure 8 gives the calculations made with this crankshaft FE model technique and the error in actual natural frequency. As the figure shows, this technique enabled an accuracy of within 1% in each major mode.

4.2. Modeling of ACG Rotors

Because of the considerable influence of riveting on the axial-direction vibration mode, the FE model developed to simulate the ACG rotor was made to represent the stiffness characteristics of the rivet–boss contact face. Three-axis spring connections were used for each rivet, and for the outermost edge of the boss–rotor contact face, a one-axis spring was added in the contact direction to make the connection. Single rivet stiffness characteristics were determined and used in defining the

rivet's three-axis spring characteristics. Optimum values for the uniaxial spring property at the boss outer diameter were determined according to correlation with the measurements. Figure 9 compares the calculated and measured results for the aforementioned ACG rotor measurement test piece. This shows that the technique is accurate to within 1.4%.

4.3. Treatment of Crank System Damping

Our measurements have shown that the ACG rotor and press fitting increase damping in the crankshaft system. However, in each actual test, measured damping values were small (Fig. 4) and not different from that of the one-piece crankshaft. As not only press-fitted components and the ACG rotor, but also such drive train units as the centrifugal clutch, belt converter, and torque converter are mounted directly to the crankshaft, the damping characteristics affecting the crankshaft can vary greatly, depending on the actual mechanism of each. Therefore, the damping value of the overall crankshaft system must be set according to each combination of drive train units. Currently, the only method for doing so is to estimate based on actual measurements. Measuring several types

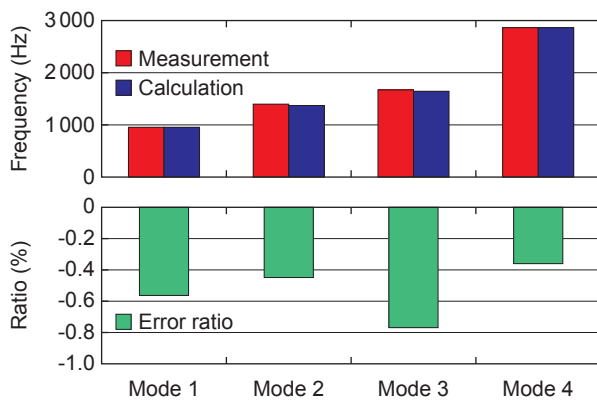


Fig. 8 Comparison of measured and calculated eigen frequencies with press fit

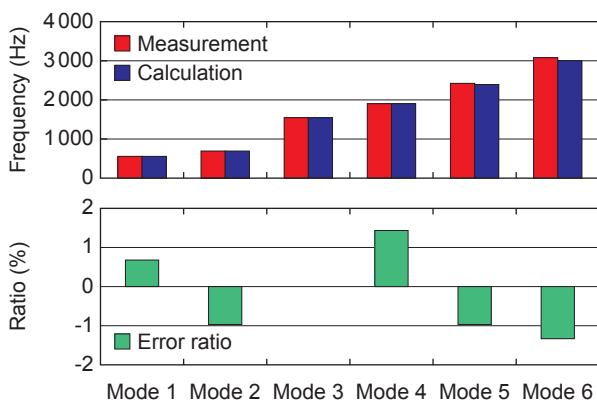


Fig. 9 Comparison of measured and calculated eigen frequencies with taper and riveting

of engines with different drive train units, it was found that single-cylinder motorcycle engine crankshafts had a damping ratio of approximately 1–5%. Based on these findings, simulations in which actual measured damping ratios for drive train units mounted to a crankshaft were assigned to the crankshaft system, were conducted.

4.4. Modeling of Rolling Bearings

Which method to use to model bearings is an important issue in simulating the dynamic behavior of crankshafts. For one-piece crankshafts, plain-bearing models based on elasto hydrodynamic lubrication theory are commonly used, but for assembled crankshafts, which contain rolling bearings, such three-dimensional elastic contact must be handled efficiently in the mechanical dynamic analysis^{(3), (4)}. The roller bearings used in single-cylinder engines are usually ball bearings, though roller bearings are also used, depending on the engine type. As the development of ball-bearing models has already been reported⁽⁵⁾⁻⁽⁷⁾, this paper discusses the development of a mechanical dynamic analysis model for roller and needle bearings.

4.4.1. Roller bearing model

For roller bearings, a crucial issue is how to represent large changes in contact points that occur with axis movement, as well as the corresponding change in stiffness. Because a roller bearing fundamentally has no component that constrains in the axis direction, for practical reasons it is treated as if no stiffness occurs until the rolling element contacts the side wall of the outer or inner raceway. As for the elastic characteristics of the contacted area, the finite-element method and general-purpose structural-analysis software were used to first calculate the relationship between load and reaction force for each contact surface in a single rolling element. Using these results, the variable stiffness function for each state as contact progresses was determined for different angles of contact β (Fig. 10). Saved as a bearing characteristic file, this data was referenced from the mechanical dynamic analysis software while determining load

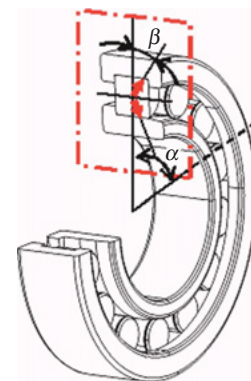


Fig. 10 Roller bearing model

corresponding to the displacement of each rolling element. Displacement of each rolling element was determined from the relative angle and distance between the inner and outer raceways according to the state of contact between them. Relative position along the circumference of each rolling element was determined geometrically according to the rotational difference between the outer and inner raceway⁽⁸⁾. In the mechanical dynamic analysis software, the resultant force of each rolling element's load was treated as a bearing reaction.

Next, an impact shaker tester (Fig. 11) was used to test the actual parts to verify that the bearing model was accurate. The crankpin was shaken by generating an impact load through breaking the bolt, and the resulting radius-direction displacement and axis-direction displacement on the right and left crank web were computationally compared with the results of measurement. The bearings used were a roller bearing in combination with ball bearings as the thrust bearings. Figure 12 lists the displacement data, which shows that the simulation accurately re-created actual behavior.

4.4.2. Modeling of needle bearings

The needle bearing used for the big end of the connecting rod can be treated as a case of extreme

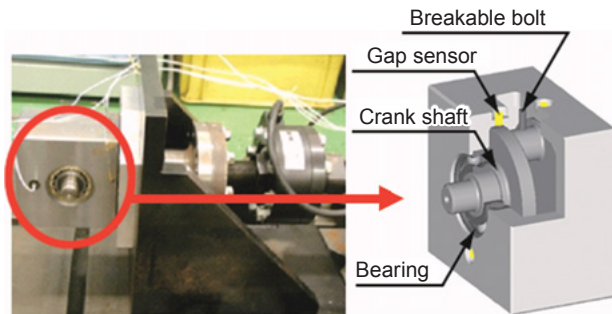


Fig. 11 Impact shaker tester

dimensions in the rolling element of a roller bearing. However, press fitting deforms the crankpin — which contains the bearing's rolling surface — into a barrel shape, with curvature in the cylindrical center axial direction. Consequently, for the inner raceway, the needle bearing model re-created this curvature in the axis direction.

5. Comparison with Actual Measurements

A single-cylinder engine typically contains two bearings, but because of differences in factors such as how crankshaft output is taken out, some single-cylinder engines contain three or more bearings. To verify the appropriateness of the mechanical dynamic analysis models discussed up to this point, engines with differing numbers of two types of bearings were used to test model accuracy.

5.1. Two-bearing Engine

Actual measurements and simulation results were compared for a single-cylinder engine with ball bearings on the right and left, the most common type of single-cylinder engine (Fig. 13). Because crankshaft output is applied directly through the gears, this type of engine is said to have the lowest damping attributable to drive train units.

Displacement in the right and left crank web in the normal direction of the center axis, as well as whirling displacement of the ACG, was measured with a gap sensor. Strain was measured by attaching strain gauges to the crank journal corner fillet on the ACG rotor side of the crankshaft.

For the simulation models, the aforementioned FE model designed to take crankpin press-fit characteristics and riveting characteristics into account was used, and a ball bearing model was used for the bearing.

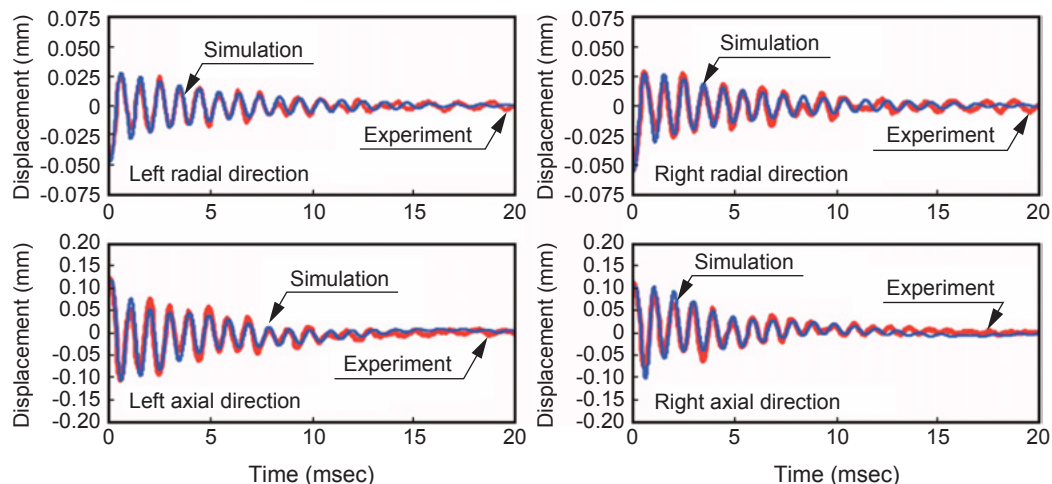


Fig. 12 Comparison of experiment and simulation of impact shaker test

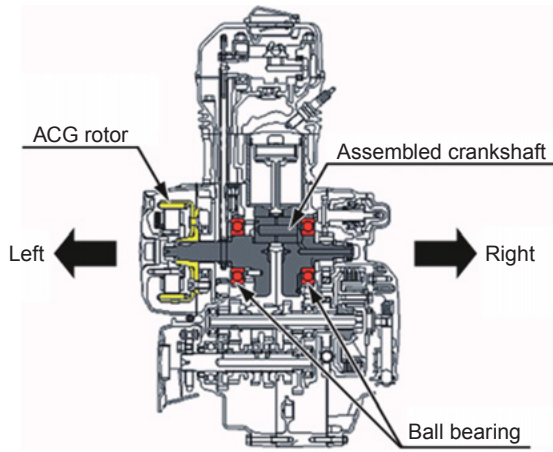


Fig. 13 Structure of 2-bearing type engine

Figure 14 shows the actual measurements of left web cylinder-direction displacement and strain in the crank journal corner fillet, along with the corresponding results for simulation. These results demonstrate the favorable correlation between simulation and actual measurements and, therefore, the appropriateness of this simulation technique.

In Fig. 15, actual strain measurements in the crank journal corner fillet are compared with simulations made with a conventional technique and those made with the developed method. Figures 15 (a) and (b) compare measurements with the results, respectively, of simulation using a model of conventional spring damper bearings and simulation that treated the press-fit part of the crankshaft as a single piece. These results show that the developed technique is superior in accuracy than that of the conventional technique.

5.2. Four-bearing Engine

This special type of single-cylinder engine, which is

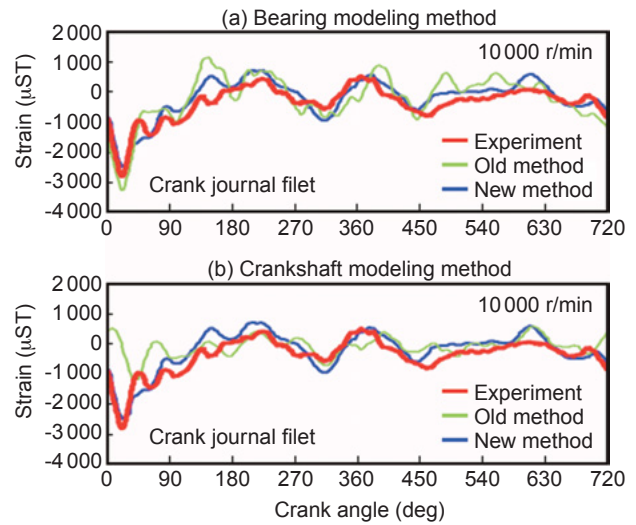


Fig. 15 Comparison of old and new methods of 2-bearing type engine

used in all-terrain vehicles, has two main bearings as well as two additional bearings that support the crankshaft at both ends. These engines sometimes have a torque converter as the drive train unit, and various additional mechanisms are installed on the crankshafts, resulting in a complex geometry. Consequently, actual measurements were conducted to compare the results for correlation with the results of simulation.

Figure 16 is a schematic of the engine in question. As with the two-bearing engine, displacement in the normal direction of the center axis at both crank webs, displacement in the ACG rotor in the normal direction of the center axis, and torsional strain in the front and rear axes were measured.

The simulation model is depicted in Fig. 17. This engine had a roller bearing as the front main bearing and ball bearing for the rear main bearing. At both ends, ball

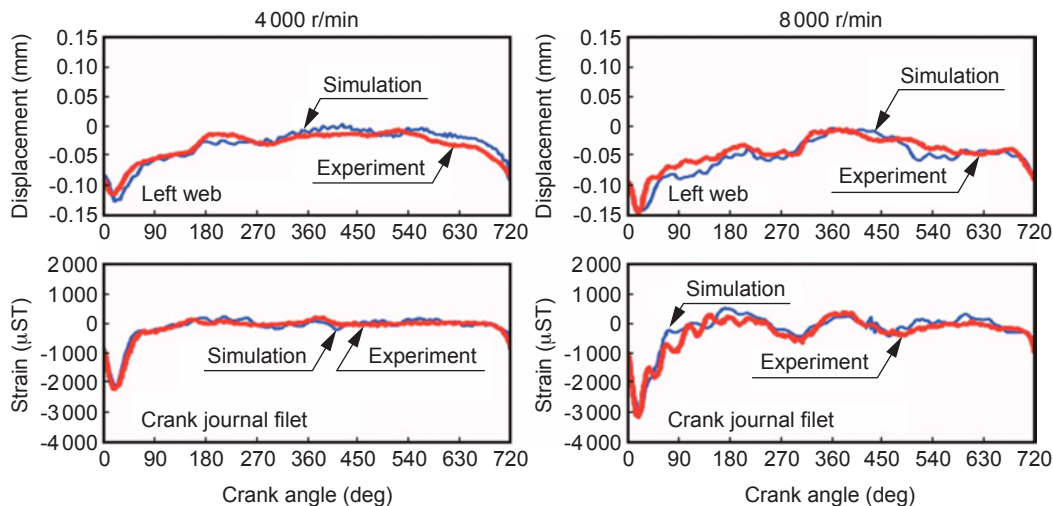


Fig. 14 Comparison of experiment and simulation results of 2-bearing type engine

bearings also supported the space between the ends and the engine case cover.

Measured and calculated displacements are shown in Fig. 18, which indicates the favorable correlation for front and rear crank web displacement and ACG rotor displacement. The larger measured displacement of the ACG rotor is due to the sensor passed through a balance-adjustment hole. Overall, these results demonstrated that this simulation technique accurately reproduces crankshaft bending behavior, regardless of the number of bearings.

Figure 19 shows measured and calculated torque converted from torsional strain. This data reveals a slight discrepancy between the measured and calculated values.

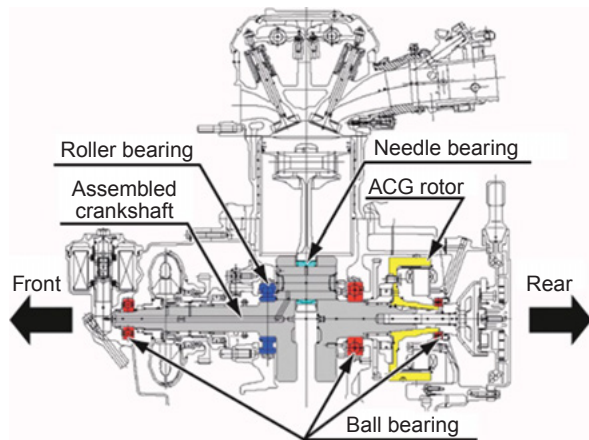


Fig. 16 Structure of 4-bearing type engine

The cause of which is most likely how the torque converter was represented in the model. In this simulation, the torque converter's mechanical dynamic analysis model was treated as a rigid model, thus failing to consider the influences on the torsional vibration system of the torque converter's own torsional stiffness and oil in the unit.

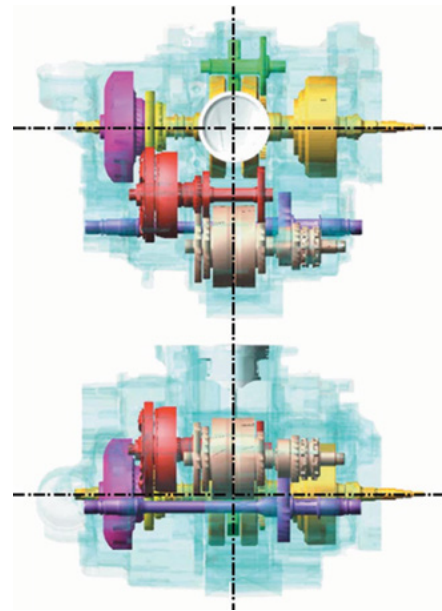


Fig. 17 Mechanical dynamic analysis model for 4-bearing type engine

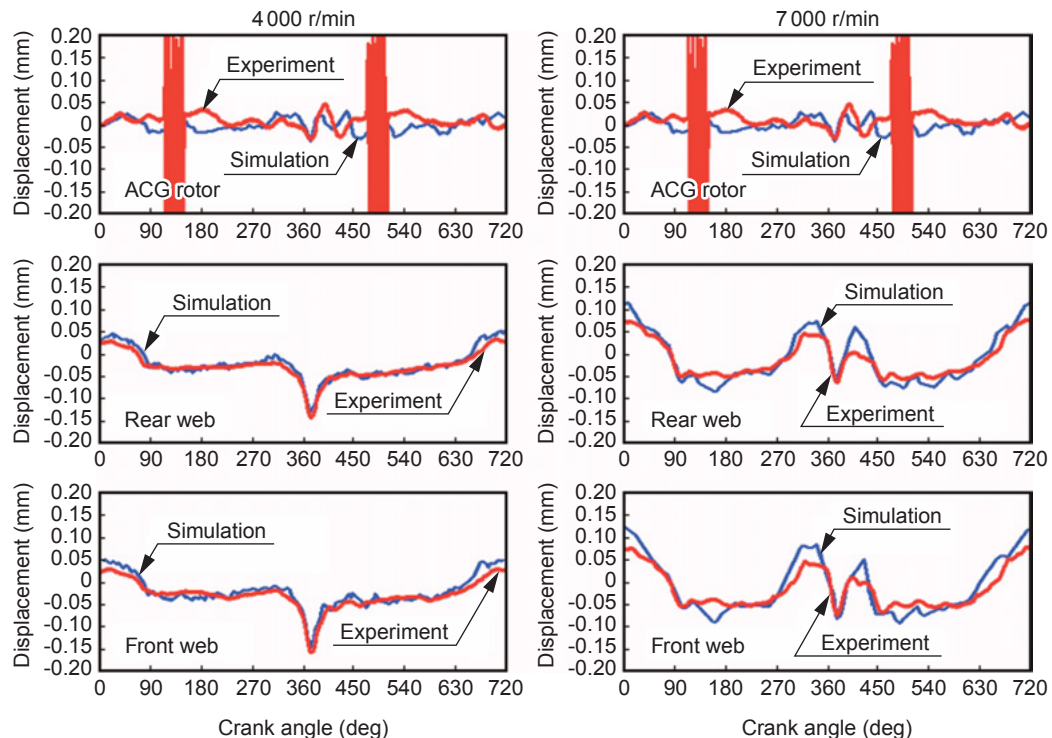


Fig. 18 Displacement of crank-web and ACG rotor of 4-bearing type engine

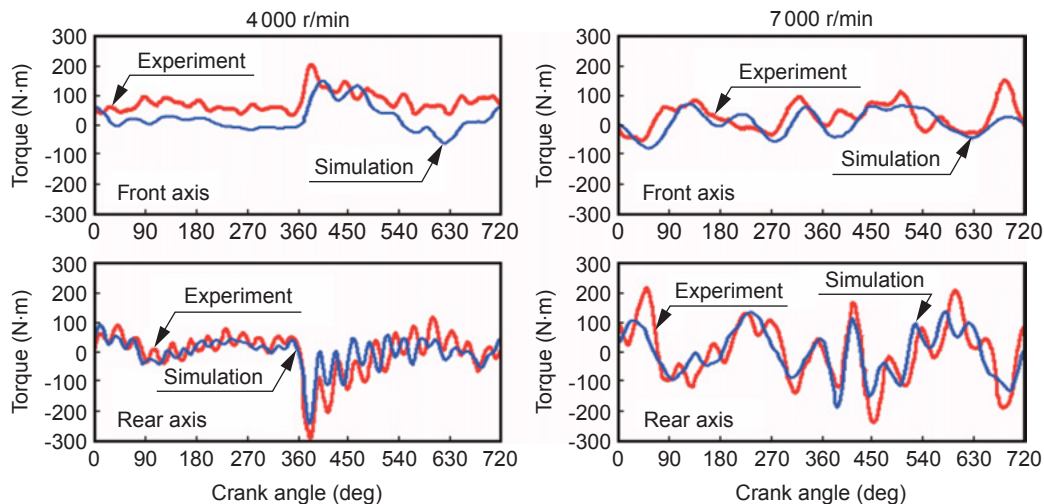


Fig. 19 Axial torque of 4-bearing type engine

6. Conclusion

- (1) The following actual characteristics necessary for simulating the dynamic behavior of assembled crankshafts were verified.
 - The natural frequency of assembled crankshafts was approximately 2% lower than that of one-piece crankshafts, regardless of the crankpin's press-fit margin. However, press fitting had almost no influence on the damping ratio.
 - The taper of the ACG rotor decreased the natural frequency approximately 3% compared to the one-piece crankshaft but had almost no influence on the damping ratio.
 - Rivet joints decreased natural frequency up to 14% and increased the damping ratio approximately tenfold.
- (2) A method was established to enable the construction of an elastic crank model for mechanical dynamic analysis that accounts for the dynamic characteristics of the crankshaft's press-fitting and rivet joints.
- (3) A rolling bearing mechanical dynamic analysis model considering the three-dimensional elastic contact characteristics of differing types of rolling elements was developed.
- (4) Using the mechanical dynamic analysis models referred to in (2) and (3) above, two types of single-cylinder engines were simulated, each with a different number of bearings, and compared the computed results with those of actual measurements, thereby demonstrating that the technique provides sufficient accuracy for practical applications in evaluating the strength of assembled crankshafts.

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