

A Combined Experimental-Numerical Approach for Motorcycle Crank Noise

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ABSTRACT

Riders prefer comfortable crank noise levels as they do for handling and riding vibration. A simulation modeling method for a single-cylinder motorcycle engine was established to identify the mechanism that is the cause of unpleasant crank noise. The combined approach of experimental and numerical analysis was applied to evaluate a design change's effectiveness in determining a cause of crank noise. An experiment is effective to identify an important engine behavior parameter when modeling the engine. The result of numerical analysis by using the established model was verified using experimental data from the prototype.

1. Introduction

Crankshaft noise is an unpleasant noise arising when the engine is under load. This is a noise occurring over a relatively wide spectrum, from 900 Hz to 2,000 Hz, according to the findings taken with a sound quality system, as shown in Fig. 1. Objectively, since this noise gives the impression of something rolling inside the engine, it is called a 'rattling' noise.

The purpose of this study was to gain some insight into the mechanisms that generate crankshaft noise and to offer specification improvements that will bring about an objective reduction in the noise. Essential pre-conditions were that the specifications could not involve changes in combustion or output power of the engine.

2. Process of the Generation of Crankshaft Noise

The process of crankshaft noise occurrence consists of three stages as shown in Fig. 2. These are called the 'source',

the 'transfer', and the 'emitter'. They are typical elements of crankshaft noise detected at the rider's ear. The approach to reducing noise can involve any of these stages.

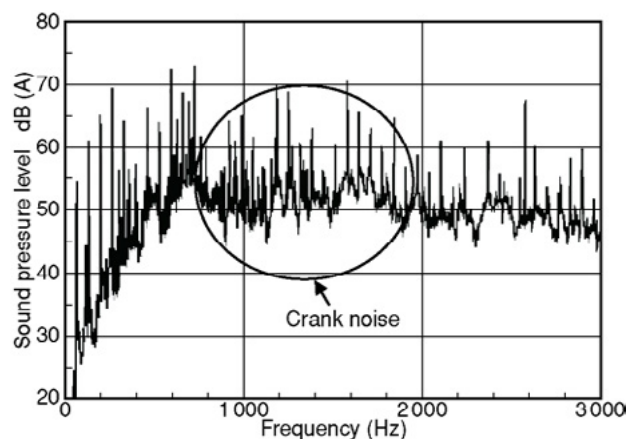


Fig. 1 Frequency range of crank noise

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Fig. 2 Process of crank noise generation mechanism

In the present case, the source involved dynamic behavior of the crankshaft in an operating engine. The forces on the bearings are the sum of inertial forces, combustion forces and dynamic properties of the crankshaft. They act on the crankcase and cause it to vibrate, giving rise to the radiation of noise.

In order to reduce crankshaft noise, the following interactions between the stages must be suppressed:

- Source: Changes to the dynamic characteristics of the crankshaft assembly.
- Transfer: Changes of bearing type.
- Emitter: Changes to the dynamic characteristics of the crankcase in order to reduce radiation noise.

The simulations for the investigation of dimensional revisions were mainly carried out for the transfer stage, but simulations of the other noise generation processes were also performed; for example, some changes were suggested in the dimensions for the crankshaft itself.

3. Modeling

It is necessary to model the mechanisms of each of the noise generation stages with the maximum possible precision. This is required in order to obtain a preliminary assessment of the effectiveness of performing an investigation into noise generation mechanisms and dimensional changes. The dynamic characteristics of the crankshaft and crankcase are important parameters for the source and emitter. Therefore, experimental excitation data for these components were used to update the FE models. The dynamic transfer characteristics of the bearings had to be identified for modeling of the transfer stage.

Certain assumptions are necessary in order to create a simulation model. The limitations posed by these assumptions must be taken into account. The model was verified in testing of an actual motorcycle. This enabled verification that the ultimate simulation model was precise enough to run qualitative comparisons with varying dimensions of the components.

3.1. Update of Modal Model

It is clear that the dynamic characteristics of the crankshaft play a key role in noise generation in the source, while the dynamic characteristics of the crankcase are equally important in the emitter; therefore, the dynamic behavior of these components must be well reflected in the model. In order to obtain precise results in the dynamic region, it was necessary to use methods based on experimental data from an actual engine while creating the model. This was particularly important for components with an uncertainty, such as press-fit components, which are often used in crankshaft assemblies in single-cylinder engines. Methods to update existing FE models with experimental data are key to this process. Since it is rare to have physical

models available that can actually be used during development under limited conditions, updating is often difficult. However, it is quite effective to build, during the final development stage of a machine, a virtual model of the next-generation machine in order to carry out updated investigations of that next model. This is because actual models exist at this stage and the reliability of the FE model can be assured with the updates.

Experimental modal analysis (EMA) was performed on an actual crankshaft (Fig. 3) and crankcase assembly (Fig. 4). Since experimental errors would influence the updates, special care was essential during these experiments; for example, the mass of the acceleration sensor is itself one of the influences on the crankshaft. The experimental modal parameters were compared with FE simulation parameters using correlation and update techniques. The FE model was divided into different sections in order to identify sections with modeling errors. A sensitivity analysis was done in order to update the material characteristics and connection characteristics of each component or connection (for

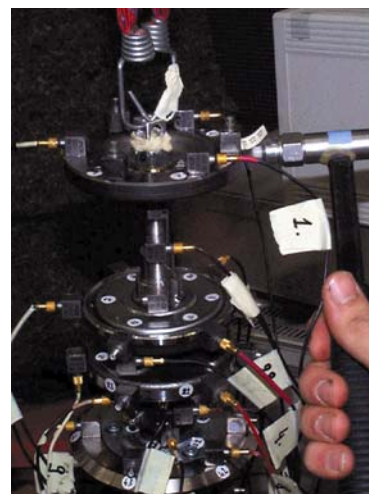


Fig. 3 Modal analysis of crankshaft

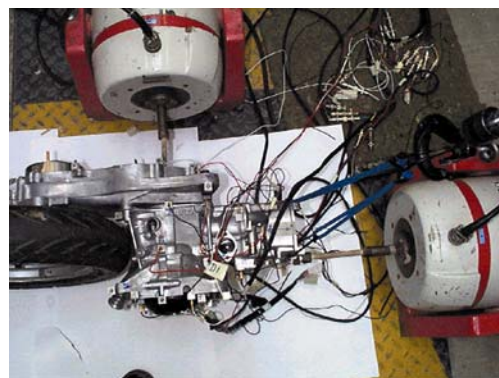


Fig. 4 Modal analysis of crankcase assembly

example, the press-fit connections in the crankshaft). Using these techniques, a good correlation was obtained between the modal parameters of the physical model and those of the FE model in the global modes of both the crankshaft and the crankcase assembly. The results are shown in Fig. 5 and Fig. 6.

3.2. Determination of Bearing Dynamic Stiffness

The inertial forces due to crankshaft rotation and the combustion forces are transmitted through the bearings to the crankcase. The bearing dynamic stiffness must be known in order to obtain the correct response in the crankcase to these excitation forces. A special tester was built to get these parameters. The inner race was placed on the test jig and the

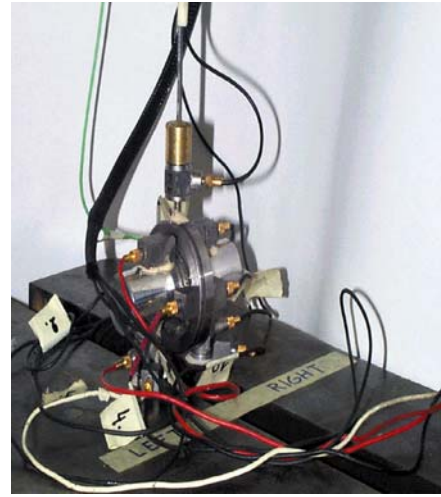


Fig. 7 Test setup for identification of dynamic bearing stiffness

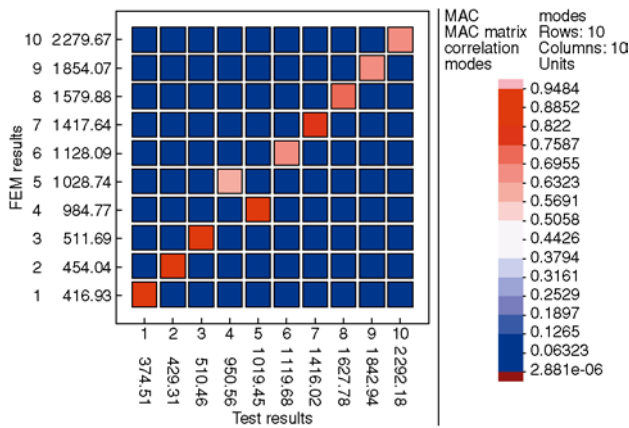


Fig. 5 MAC correlation matrix for crankshaft

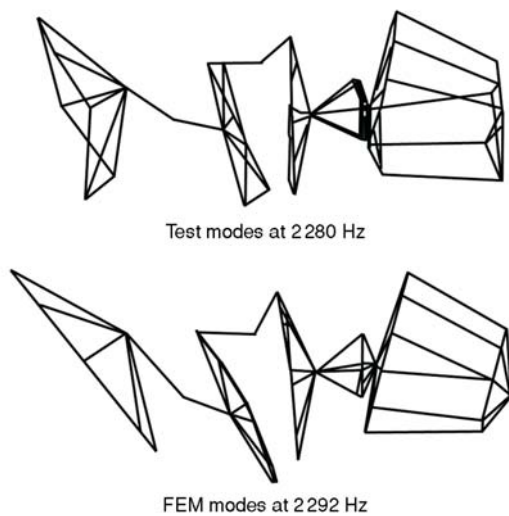


Fig. 6 Comparison of experiment and FEM mode

outer race was vibrated with an electromagnetic shaker under a random signal at frequencies up to 5 kHz. Figure 7 shows the testing apparatus. The dynamic stiffness of the ball bearing was extracted from the relative displacement between the races and from the reaction forces in the test jig. It was designed to measure both the stiffness in the radial direction and that in the axial direction. A bearing internal clearance is set to provide a balance of low noise and low friction during engine operation. This clearance resulted in noise during dynamic stiffness measurements that was the same as crank noise. It stopped immediately when the clearance was cancelled. This indicated that ball clearance is one of the key noise generation mechanisms.

In an engine like that in this study, where the outer races of the crankshaft ball bearings are press-fit into the crank case during assembly of the engine, nearly all of the clearance is cancelled at room temperature. However, this clearance increases with temperature and is significantly larger at the engine operating temperature. This occurs because the coefficient of thermal expansion of the aluminum crankcase is higher than that of the steel bearings. This clearance increase has been confirmed through experiments. Crankshaft noise worsens with an increase in temperature.

3.3. Conditions and Limits in Virtual Model

The purpose in the present virtual model was to observe how changes in various dimensions affected crankshaft noise in an attempt to elucidate noise generation mechanisms. To do this, it was necessary to create models that allow qualitative assessments of changes in various dimensions. For this reason, conditions for certain unknown parameters were set according to the following assumptions:

- The bearing clearance during engine operation is dependent on temperature, but in the simulation, the clearance was fixed. This clearance corresponded to some engine temperature.
- The CVT exerts a force on one end of the crankshaft; this was approximated as the belt tension when the engine is stopped.

- The effect of magnetic forces in the AC generator rotor, which is mounted directly on the crankshaft, was ignored. The excitation forces from the valve train were also ignored.
- The monolithic engine-transmission combination is attached to the frame with a bushing. The frame was assumed rigid and conditions for the engine-frame connection were chosen to avoid causing problems in the model.

Some limitations can be expected in the results due to the method of bearing modeling. If a bearing is perfect with no surface defects (flaws), it will show random noise due to the rolling motion over a wide spectrum of vibrations. However, when there are flaws or other defects in the outer or inner race or in the balls, noise of particular frequencies will be generated, as shown in Fig. 8. These frequencies and their harmonics will be seen in the vibration spectrum of the bearing. It has been reported that these frequencies will appear not only when there are defects in components, but also when the inner and outer races are tilted in relation to each other during operation. In a 4-stroke engine, the combustion forces exert a bending moment on the crankshaft once every two revolutions. This causes the outer races of the bearings to take a periodic tilt with respect to the inner races, as shown in Fig. 9. Thus, even if the bearings are perfect, a harmonic is generated at 0.5/rev, due to the combustion process. These effects were not considered in the present bearing model, so they were not reproduced in the frequency response obtained here.

3.4. Modeling

The virtual dynamic model of the engine is composed of the updated FE models of the crankshaft and the crankcase and the dynamic stiffness models of the bearings. This was subjected to assumptions and limitations described in section 3.3. Next, the internal pressures in the combustion cylinder were measured with in-cylinder sensors on a completed motorcycle mounted in a chassis dynamometer. These pressures were used for the combustion cylinder in the simulation with the virtual model. The completed virtual model is shown in Fig. 10. In this model, the engine and

transmission are supported on a rigid frame through bushings.

3.5. Credibility of Model

For the purpose of confirming the model, a completed vehicle was installed on the chassis dynamometer with an acceleration sensor embedded in the crankshaft to measure the acceleration of the crankshaft journal, and a non-contact displacement pickup in the shaft top to observe its displacement in the radial and axial directions. Figure 11 shows the test setup. The measured data were compared with the results of the simulation to verify whether they showed the same trends. Figure 12 provides an example, a comparison of the measured radial acceleration in the vicinity of the crankshaft journal bearing and the simulation. In view of the limitations and assumptions mentioned above, these graphs show quite a good match. This indicates that this virtual model can be used for qualitative comparisons with

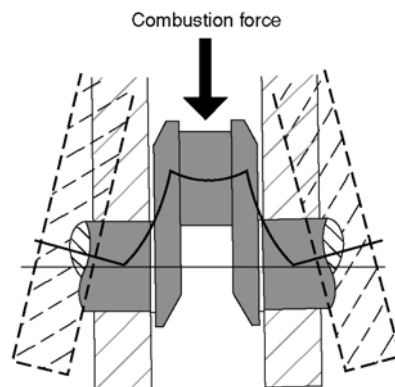


Fig. 9 Periodic bending of crankshaft due to combustion forces

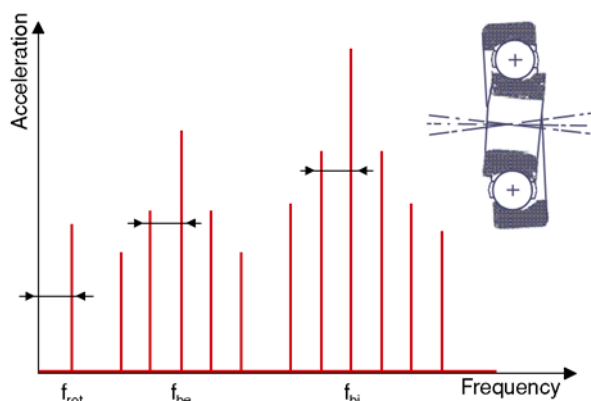


Fig. 8 Characteristic frequencies caused by tilting of inner and outer race

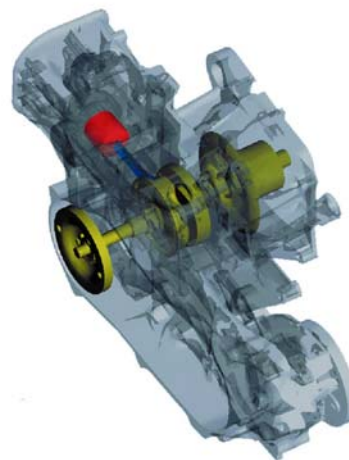


Fig. 10 Virtual mechanical model of engine with transmission

dimensions changed. Figure 13 displays the Lissajoux figure of the displacement of the crankshaft tip. The behavior is plainly visible, including the once-per-two-revolution pressure peak in the combustion cylinder.

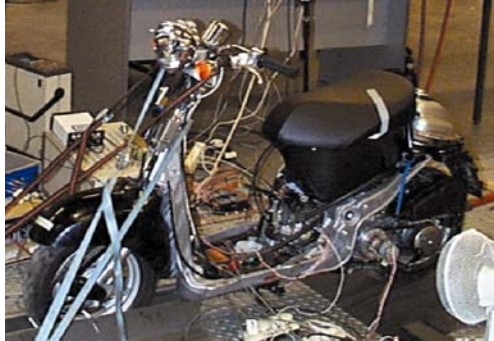


Fig. 11 Fixed motorcycle on roller bench

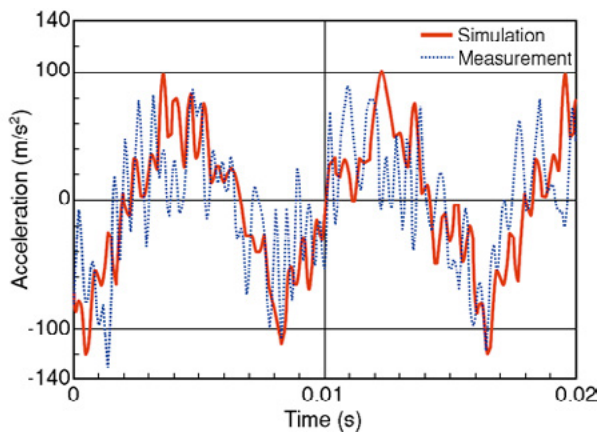


Fig. 12 Virtual acceleration on bearing

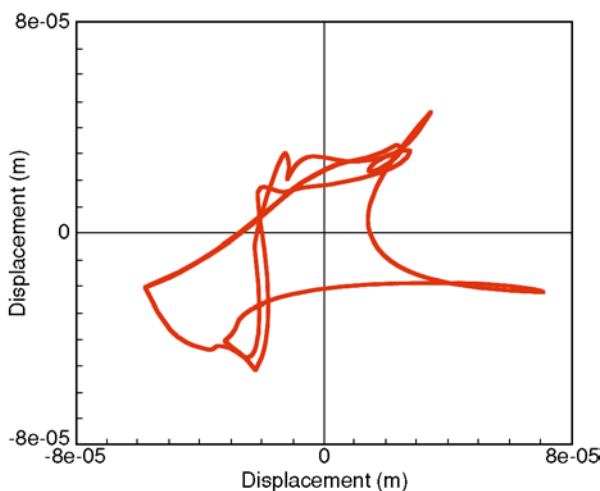


Fig. 13 Lissajoux figure of crankshaft displacement at bearing

3.6. Assessment of Changed Dimensions

The effect on noise from changes to dimensions was calculated with the qualitative virtual engine model. The changes were applied to all three stages of the noise generation mechanism.

- Source: Changes in crankshaft dynamic characteristics due to changes in inertial moment, addition of mass, and change in crankshaft type.
- Transfer: Changes in bearing clearance and bearing type.
- Emitter: Augmented stiffness of crankcase.

The first list of changes applies to the initial noise generation process in other words, to the source. The dynamic characteristics of the shaft were revised by changing the crankshaft dimensions. The post-revision crankshaft incorporated a smaller generator rotor 40% lighter than the previous rotor. Figure 14 shows that this change affected the frequency band of interest, in other words, the spectrum from 900 Hz to 2,000 Hz. The other two changes made in this study had to do with the transfer level of the noise generation process. One was to double the bearing ball clearance, and the other was to change to hydro bearings (plain bearings). Figure 15 and Fig. 16 show the results. As expected, increasing the clearance increased the impact force on the bearings, increasing vibration. Sufficient reductions in the noise level in the spectrum of interest were only obtained by switching to plain bearings.

4. Experimental Verification

All the specification changes were checked in chassis dynamometer tests of a completed motorcycle. Two stages of the noise generation process will be covered here, the source and the transfer stages. Two meaningful sets of information were gained from the results of the study on the virtual model simulation. The dynamic stiffness of the crankshaft is an

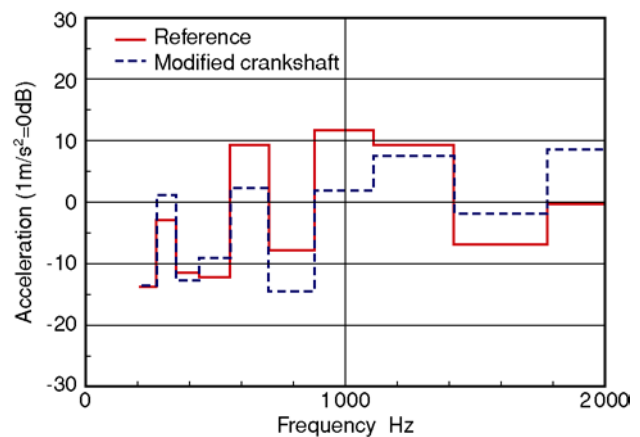


Fig. 14 Simulated comparison of reference acceleration in third octave between reference model and modified crankshaft

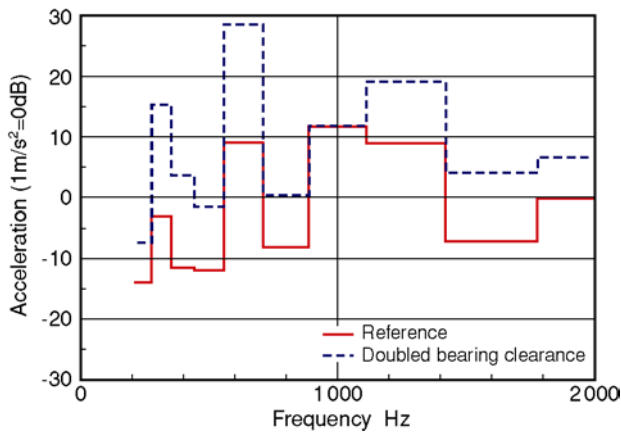


Fig. 15 Simulated comparison of reference acceleration in third octave between reference model and doubled clearance

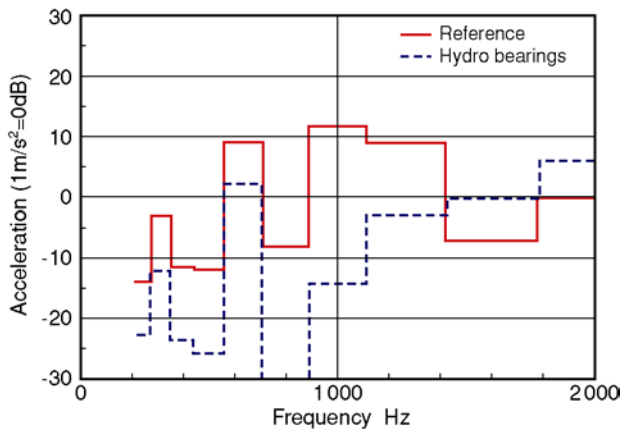


Fig. 16 Simulated comparison of reference acceleration in third octave between reference model and hydrobearings

important element in noise in the source stage, and the ball bearing clearance is an important element in the transfer stage.

4.1. Changed Dimensions at Source Stage

4.1.1. Change in crankshaft dynamic characteristics

In order for a better understanding of the effect of changes in the dynamic characteristics of the crankshaft, rather than designing and creating a whole new crankshaft, a radical modification (removal of large masses at each end) of the original crankshaft was carried out. The generator rotor was completely deleted and the CVT pulley radius and thickness were greatly reduced. Figure 17 shows the original and modified crankshafts. As a result, the resonant frequencies of the shaft were increased. The first torsional resonant frequency was increased from 511 Hz to 849 Hz, and the second, from 1,128 Hz to 1,728 Hz. As shown in Fig. 18, these dimensional changes greatly reduced the crankshaft noise measurement results at the rider's ear in the frequency band. The following is a simplified explanation of the mechanisms of reducing noise. Reviewing the spectrum of combustion cylinder pressures, it is easy to see, as shown

in Fig. 19, that the energy is considerably damped at higher frequencies. Due to this effect, if the crankshaft resonant frequencies are raised, the transfer efficiency of the excitation force will be suppressed.

4.2. Change in Dimensions at Transfer Stage

4.2.1. Reduction of ball bearing clearance using limitation on thermal coefficient of expansion

Because the ball bearing outer race is pressed into the aluminum alloy crankcase, after assembly, the clearance is smaller than in the free condition. But during operation, due



Fig. 17 Comparison between original and modified crankshaft

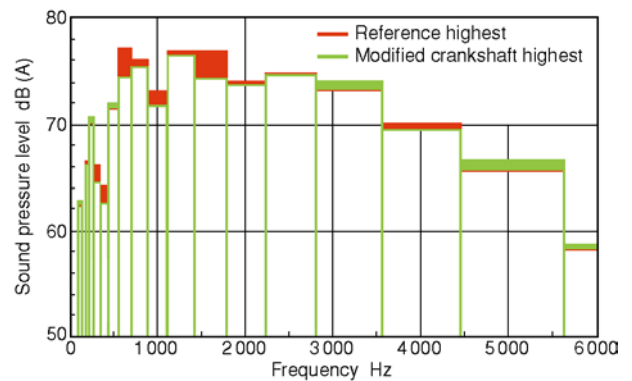


Fig. 18 Frequency analysis result by modified crankshaft

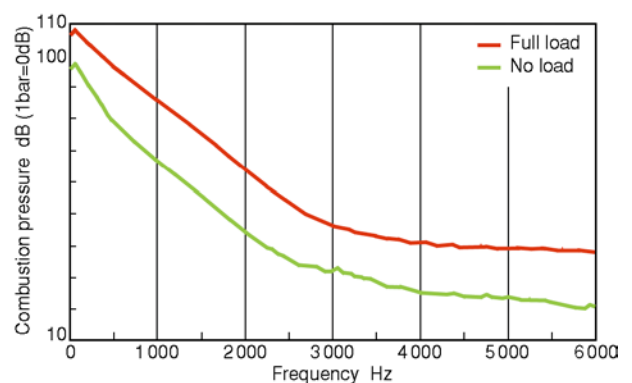


Fig. 19 Combustion pressure per one cycle

to the difference in thermal expansion coefficient between the aluminum alloy crankcase and the steel bearing, the clearance increases together with the oil temperature. In an attempt to prevent excessive clearance, a steel water jacket ring was placed around the outer race, as shown in Fig. 20.

4.2.2. Replacement of ball bearings with plain bearings

Another method to reduce the excitation force due to high bearing clearance is to change the bearing specification. As in Fig. 21, the bearings were changed from the ball type to the plain (hydro) type. Since the oil film damping is larger than that in a ball bearing, these bearings are expected to provide higher damping of the impact forces excited due to the clearance of the bearings in the crankcase.

Both versions showed reductions in noise, as indicated in Fig. 22 and Fig. 23.

These three tests proved the validity of this virtual model for qualitative comparisons of designs with varying specifications. Similar levels of noise reductions were seen in the simulation and experimental results for all the versions. However, sufficient objective noise reduction was only seen from the change of bearings to the plain type. Noise levels dropped in the other versions, but not enough, from the point of view of the rider.



Fig. 20 Crankshaft with water-cooled bearing



Fig. 21 Crankshaft with hydro bearing

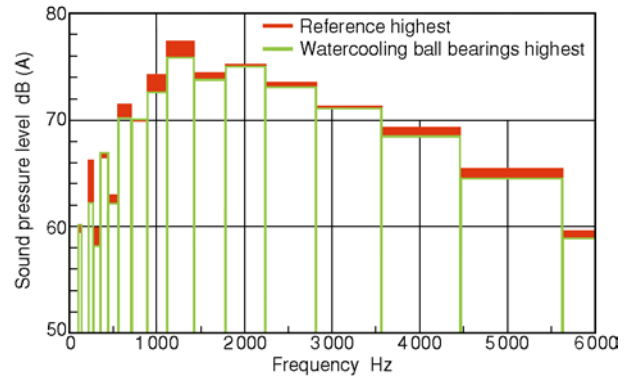


Fig. 22 Frequency analysis result by water-cooled bearing

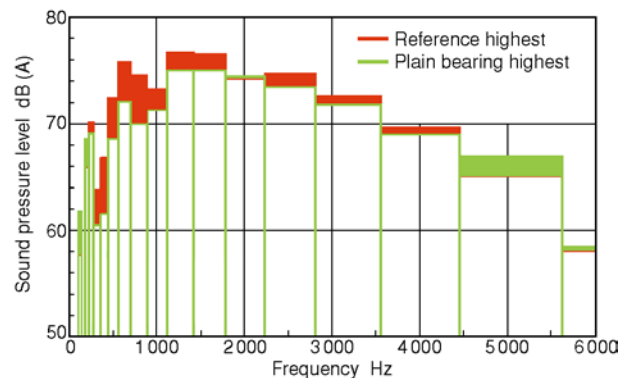


Fig. 23 Frequency analysis result by hydro bearing

5. Summary

A virtual model of an engine was constructed for qualitative assessment of designs for improved crankshaft noise characteristics and for relative comparisons of components with varying specifications. The model was shown valid in experimental tests using a completed motorcycle with components of altered dimensions.

- Crankshaft noise is caused by resonance of the crankshaft assembly.
- Excessive ball bearing clearance exacerbates crankshaft noise and causes the typical rattling noise.
- Noise is diminished by revising the design of the crankshaft.
- Noise is diminished by reducing bearing clearance.
- An objective improvement is obtained only by replacing some ball bearings with plain bearings.
- A combined approach of numerical analysis and experimental analysis contributes much to explaining the mechanisms of noise generation.

In future studies, it is necessary to develop a more precise model of bearings in dynamic mechanical simulation software.

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