

Development of Gear Train Behavioral Analysis Technologies Considered Nonlinear Elements

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ABSTRACT

It has been identified that the rotation behavior of gears is influenced by housing deformation, radial clearance, tooth elastic deformation, etc., in the modeling process of the behavior calculation model of a gear train in motorcycle engines. To maintain a high level of accuracy, influence factors were converted to the stiffness components in the rotational direction, and were built in the rotational transfer model of the gear train. Furthermore, a behavior calculation model of a crankshaft was combined as a drive excitation source of the behavior calculation model of gear train. Time domain numerical integration method was used for calculation of the transient response that changes widely by engine speed change. A jump phenomenon of response behavior of the driven gear was predicted that is a characteristic of a non-linear response. The calculation result accuracy was confirmed through correlation with an actual test engine. It was understood that this behavior calculation of gear trains could be used as a technological tool for engine development.

1. Introduction

There are various mechanisms used for revolution transfer in an engine such as gears, chains, CVT, and fluid couplings.

A small engine such as a typical general motorcycle engine shown in Fig. 1, contains a transmission mechanism. The requirements for low weight and miniaturization of motorcycle engines, are higher than those for passenger car engines as is the requirement for high engine r/min and high load factor. In order to meet the requirement, many gear train mechanisms are adopted for the valve train and accessory power train systems.

Incorrect behavior of the gear train system reduces durability and becomes a source of noise that can make a product less attractive. Undesirable gear train behavior includes an unstable revolution speed leading to a rattling

noise in the gears, thrust movement of helical gear, and inclination due to a lack of bearing rigidity. The prediction of gear train behavior becomes difficult when considering these unnecessary behaviors for rotation transfer.

When resonance occurs in a motorcycle engine gear train system consisting of small parts, the usual countermeasure is to increase the natural frequency of the system and to set the resonant frequency outside normal engine speed range. An alternative way is the use of springs and dampers that have a buffer function in the rotational direction. However, many prototype tests were required to decide the final specifications as there were no tools available for predicting gear train behavior even for the rotational transfer direction.

This paper reports the development of a simulation method for the rotational behavior of the gear train system, and its calculated results were agreed with experimental results including non-linear response.

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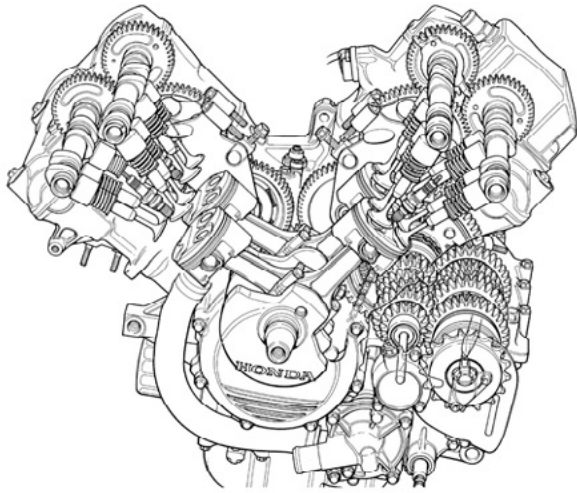


Fig. 1 Example of gear train system for a motorcycle engine

2. Modeling of Revolution Transfer Mechanism

2.1. Modeling of Revolution Transfer Mechanism

Generally, the driven gear is forced to oscillate by the angular velocity fluctuation of the crankshaft.

The basic model of the gear train system was made as a single degree of freedom model and forced vibration considered with backlash and friction. Composition of the basic model is shown in Fig. 2. A one-dimensional model produced by connecting basic models in-line or parallel was made as a model of gear train. Each basic model is combined in-line or parallel, and the one-dimensional gear train model was constructed by these combined systems. The number of the degrees of freedom is decided only by the number of transfer elements. The modeling approach for spring characteristics is explained in Subsection 2.2. Damping coefficient C is determined as a unique function of K by the parameter study.

The damper frictional resistance was dealt with as a continuous slide. The frictional force was supposed to be an almost constant value when the speed difference is above a certain fixed slight speed difference, and to be zero at no speed difference the discontinuous area has been interpolated as shown in Fig. 3. When angular velocity fluctuation is given to gear train, it shows the phenomenon that the driven gear drives the drive gear. This phenomenon is produced by the release of rotational inertia energy saved in a driven gear. When the backlash and angular velocity fluctuations are considered simultaneously, it is understood that the gear tooth flank repeatedly comes in contact and separates. Therefore, we used time domain numerical integration method according to Newton-Raphson Method for the behavior calculation of a gear train model.

2.2. Modeling of Spring Characteristics

The gear stiffness in the rotational direction of the simulation model is an important parameter that influences the accuracy of analysis. This gear rigidity consists of a composition of rigidities of each part, for example, shafts to support rotating units, bearings, housings, hub-tip twist, tooth flank slant, etc. Each gear has the radial clearance and can move within it. Such drive gear movement in the radial

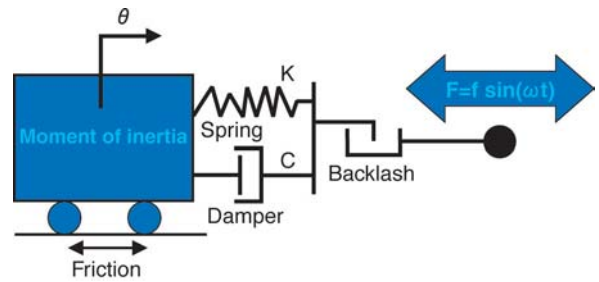


Fig. 2 Forced torsional vibration model of single degree of freedom system with clearance and friction

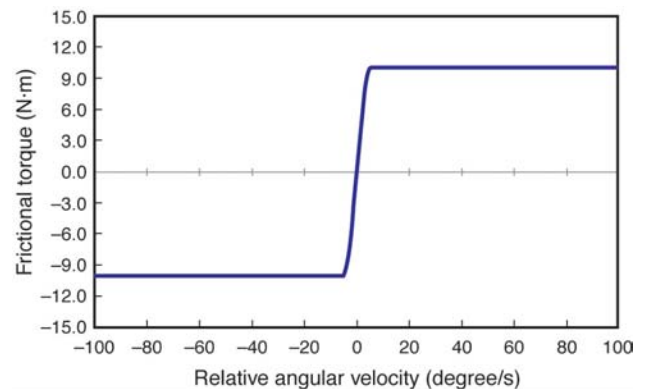


Fig. 3 Continuous friction torque

direction influences the rotational displacement of the driven gear. In the gear train, rotational displacement of the end driven gear could be integrated one by one. Therefore, it is very difficult to predict the rigidity in the rotational direction by simulation, the same as it is difficult to physically measure it.

Under the circumstances, we devised a method to obtain the rigidity of the rotation direction of each gear by combining a calculation result with a measurement result. Following is the explanation of this method as it applies to a cam gear train of a motorcycle engine used in a simulation model as shown in Fig. 4.

At first, we obtained load-displacement characteristics of the total gear train system by FEM static analysis. Next, we listed the configuration factors that influence displacement in the rotational direction. We carried out contribution ratio analysis of each listed factor for displacement of the rotation direction of the total gear train system. In addition, we obtained volume of deformation of each gear from a result of FEM static analysis beforehand.

We performed the actual measurement. It was carried out by stopping the revolution of the crankshaft. We measured the displacement of the cam gear tooth flank from when the backlash is zero. That tooth flank was caused by applying a known torque to the camshaft. The gear movement in the radial directions is also measured at normal and reverse rotation of gears. Finally, the simulated rigidity of the whole gear train system in the rotational direction was correlated with the measured rigidity, and the FEM analysis was verified with high accuracy. For the actual statical FEM analysis model, the gear holder, cylinder block and even the measurement equipment are modeled precisely. The influence of equipment was removed from the rigidity of the gear train system using the contribution ratio analysis.

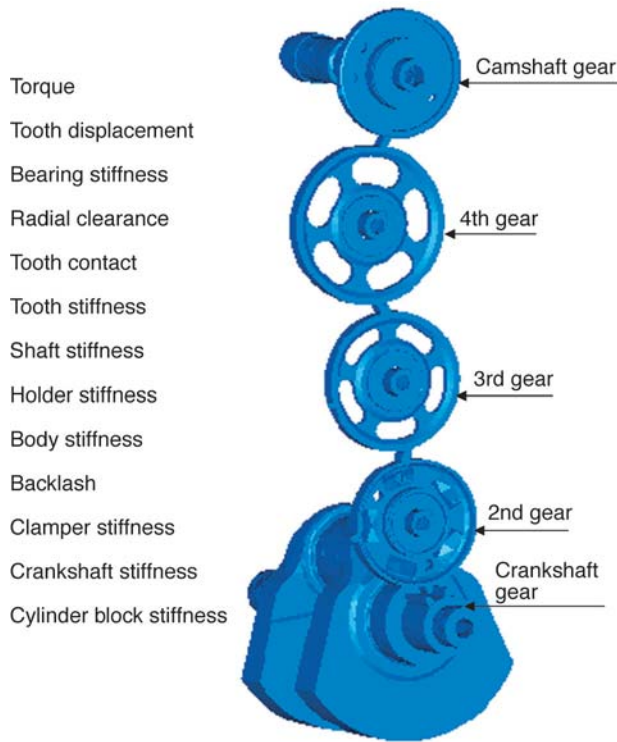


Fig. 4 Structure of gear stiffness calculation model

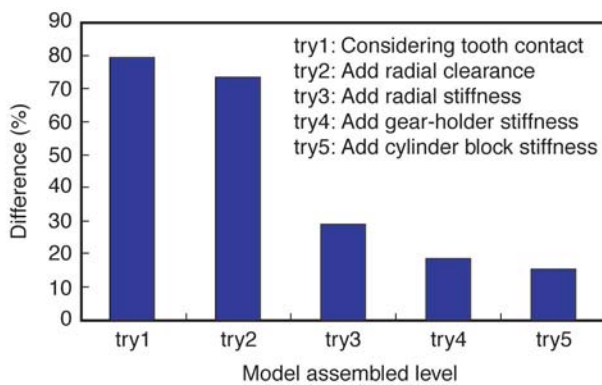


Fig. 5 Difference of gear train total stiffness between simulation results and physically measured values

Figure 5 is an example of data used for contribution ratio analysis of the factor that influenced displacement in the rotational direction. From this data, it was found that the difference between the physically measured value and the calculated one became smaller by adding factors that influence the rigidity in the rotational direction to the static FEM analysis model one by one. The rigidity data of an individual gear in the rotational direction were obtained from correlated FEM static analysis by the rigidity of the total gear train system, and inputted for the behavioral analysis model.

2.3. Modeling of Structure of Non Linear Damper

A gear train characteristic is that there is no phase difference between the drive side and the driven side when transformation and clearance are ignored. Deformation of parts and clearances in the actual gear train generate impact loads to the tooth flank. Therefore, we often use a damper which reduces the shock using the damper stroke operated by the deformation of springs, rubbers and frictional force occurring simultaneously. This operation angle of the damper

generates a phase difference in the gear train. Generally, the phase difference due to the damper stroke is larger than the gear deformation and clearance.

When applied to the simulation model, we used the physically measured value instead of the design value for damper characteristic. A typical damper characteristic has a hysteresis as shown in Fig. 6. As for the non-linear spring characteristics of the damper, we adopted the median value of the hysteresis curve as an input value, and made interpolation with the spline function of damper stroke angle - torque. The hysteresis was also adopted for the damper spring characteristic that was decided by the relative angular velocity vector occurring between parts of the drive side and the driven side of the damper.

3. Gear Train of Actual Machine and Model

In the case of the power train system of an actual machine, behavior in the rotational direction of each gear is complicated because they have the backlash of each part, the bumper mechanism acting like a damper, and stoppers to limit the stroke angle. Vibration within the backlash in the rotational direction causes the tooth flank impact load. In addition, if the engine speed is close to its natural frequency, rotational behavior becomes more complicated because the increase in damper stroke amplitude is forcefully limited by a stopper accompanied by shock.

There are some parameters in this analysis model, such as the amount of backlashes, non linear spring characteristics of each part, rigidity of damper stoppers, and each damping characteristic. These parameters were obtained by the method of combining calculated results with measured results. Fig. 7 shows the verification of correlation results between the physically measured value and calculated value for rigidity of the total gear train system in the rotational direction, when we decided each parameter.

In addition, Fig. 8 shows rigidity of each gear in the rotational direction obtained from the FEM static analysis model having correlation of rigidity of the total gear train system in the rotational direction. The gradient of each graph shows the rigidity in the rotational direction of the each gear. The rigidity was calculated from the rotational angle of each gear that includes the displacement caused by deformation. The small gradient in the graph means that not only the rigidity of the gear is high but also the rigidity of the bearing is high as well, which is the supporting part of the gear. Each backlash value was obtained from the graph. The cross points of the vertical axis, where torque equals zero, and the extension line of each gear stiffness indicates the movable range caused by the backlash and radial clearance.

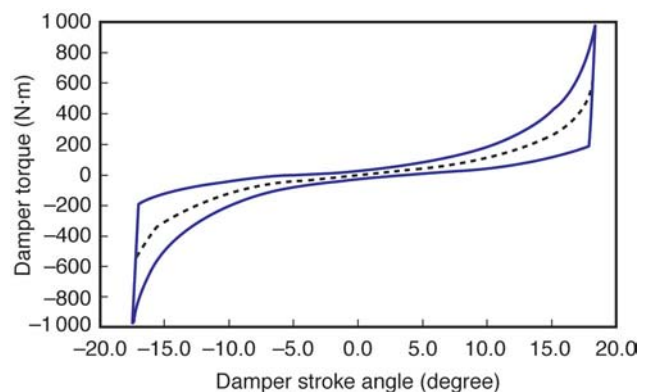


Fig. 6 Spring and friction damper characteristics

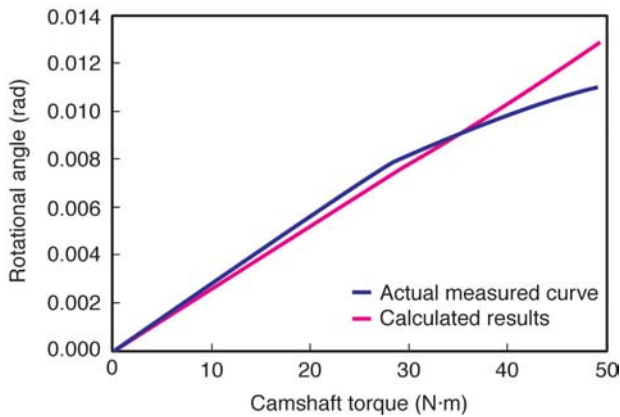


Fig. 7 Correlation between simulation and test about the gear train system torsional stiffness

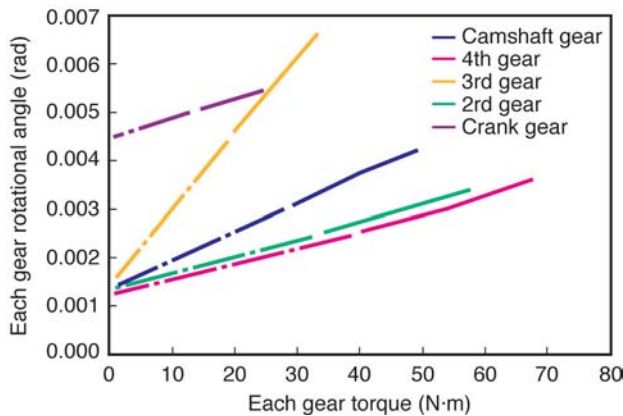


Fig. 8 Torsional stiffness of each gear

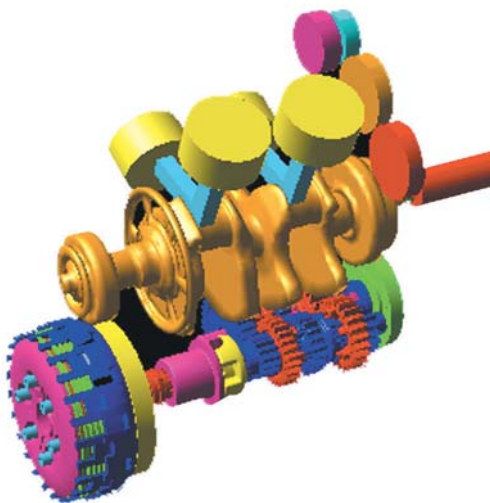


Fig. 9 Model of V4-type crankshaft behavior simulation

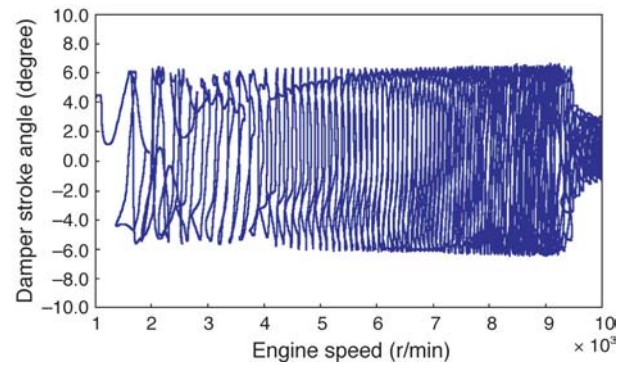


Fig. 10 Calculated damper stroke angle operating by WOT scanning mode

4. Application Example

4.1. An Application to Generator Power Train System

To adopt a gear train for the accessory drive train of the 90 degree V-type 4-cylinder engine, as shown in Fig. 9, we produced a simulation model (2) for behavior calculation of the crankshaft. The cylinder pressure value is determined as a function of the crankshaft angle, and is loaded on the piston. Therefore the piston drives the crankshaft and the fluctuation of the angular velocity provides the rotational vibration for the gear train. We carried out a response behavior analysis of the drive gear train system of a generator using this model simulating a firing operating condition. The operating condition was set to full load acceleration, at an engine speed range of 3,000-10,000 r/min referring to a physical engine test on a test bench. Figure 10 shows the calculation result that is the behavior response amplitude of the damper stroke angle. The graph shows that the stroke angle amplitude is limited by a stopper in the area of low and middle engine speeds. A jump phenomenon that the amplitude converges in discontinuity occurred when the engine speed was 300 r/min slower than the resonance engine speed (9,864 r/min) that was theoretically calculated using moments of inertia and spring characteristics in the current model.

The transfer torque fluctuation that is absorbed by the damper is shown in Fig. 11. It was observed from the calculation result that the amplitude of the damper torque increases with engine speed, but, suddenly converges in discontinuity at high engine speed. It was also found that the timing of the jump phenomenon was synchronized between the stroke angle and damper torque amplitudes. Therefore, we made an experimental damper having the same non-linear specifications as the current simulation model and installed it on an actual test engine. As a result, we could confirm that the jump phenomenon occurred at 8,500 r/min as shown in Fig. 12 and the calculation was verified.

From the result, we could simulate the jump phenomenon non-linear response behavior using the behavior calculation model of the gear train operated over a wide range of engine speeds that are specific to motorcycle engines.

4.2. Trial of Aggressive Practical Use of Jump Phenomenon

As a result of simulation and measurements, we understood that the amplitude of the damper stroke angle and the torque was reduced after the jump phenomenon occurred.

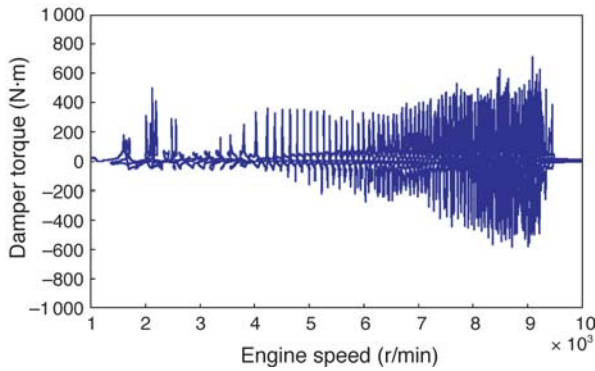


Fig. 11 Calculated results of damper torque operation by WOT, scanning mode

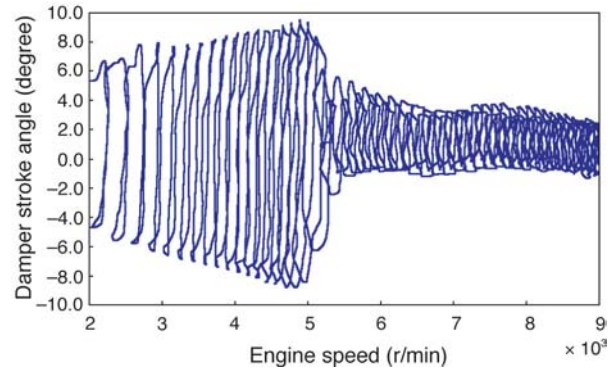


Fig. 13 Calculated damper response, in case of V4-type engine operating WOT, scanning mode

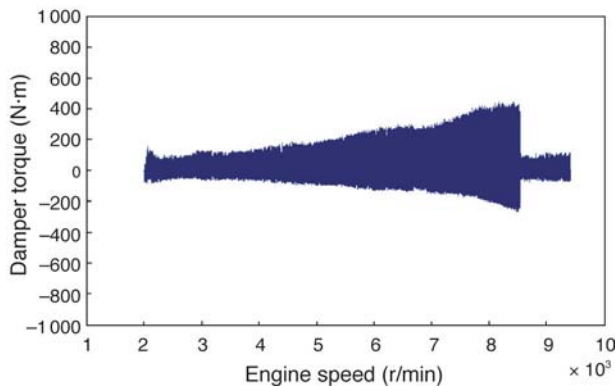


Fig. 12 Measured results of damper torque operation by WOT, scanning mode

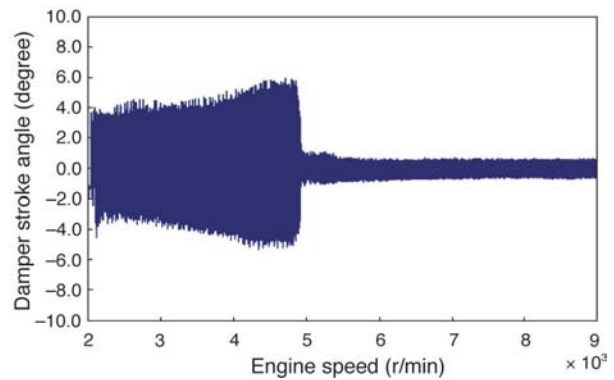


Fig. 14 Physically measured damper response, in case of V4-type engine operating WOT, scanning mode

We tried to effectively generate the jump phenomenon at engine speeds less than 3,000 r/min where the time of engine usage is very short, and at the normal engine speed range of 3,000-10,000 r/min where the damper behavior is stable.

Through the parameter studies, to generate the jump phenomenon at a low engine speed, we found that a big enough moment of driven part inertia and small enough spring rate are the requirements.

In order to improve the accuracy of predicting the engine speed at which a jump phenomenon occurs, we used the measured values of the damper characteristics and the in-cylinder pressure. Figure 13 shows the simulated transient response of the damper stroke angle amplitude and Fig. 14 shows measured result with an actual test engine. Although there is a difference in the absolute amplitude value, we could improve the accuracy of the motion simulation since the engine speed where the jump phenomenon occurred was 4,800 r/min in both cases. However, we also found that the structure to generate a jump phenomenon at engine speeds lower than 3,000 r/min requires a larger inertia of moment and smaller spring rate, which makes the engine layout too large.

4.3. An Example of Jump Phenomenon Practical Usage

Through extended parameter study, we found the important factor required to generate the jump phenomenon is not a faster engine speed but a larger application of the vibration frequency. We selected a horizontal opposed 6-cylinder engine, as shown in Fig. 15, having a combustion

stroke at every 120 degrees of crank angle for the study of multi-cylinder engines since the torque fluctuation frequency becomes high. The gear train system is installed as the generator drive line, and the damper was mounted on the drive side of the generator in this engine. This generator's equivalent inertia moment is equal to the crankshaft and offsets the moment of rolling direction that occurs when the motorcycle engine speed is changed. The stroke angle of the damper is large and the spring rate is small. The operation mode was set to full load acceleration of 1,000-5,000 r/min and the damper torque was calculated as shown in Fig. 16.

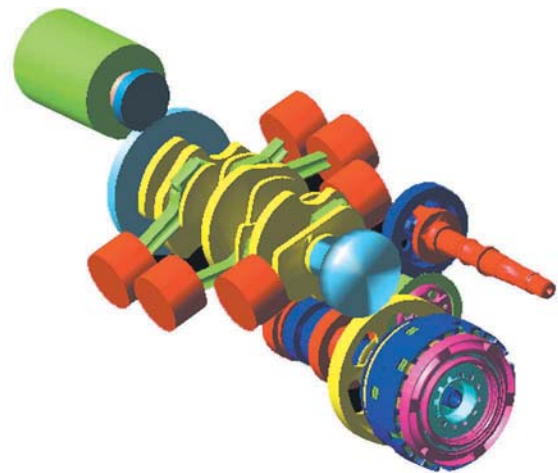


Fig. 15 Model of F6-type crankshaft behavior simulation

This graph shows the torque amplitude is already small enough at a low engine idling speed and the gear train is insulated from resonance through the range of calculation. This contributes to the creation of a motorcycle with low vibration.

4.4. Application Example of the Cam Gear Train

In the case of the engine running at a high speed, the gear train is adopted for the valve train system. It is understood that one advantage of the cam gear train is more precise dynamic valve timing at high engine speeds compared with a cam chain drive. However, there is a backlash in this cam gear train as well and output performance deteriorates when valve timing is lost in the case when rotational behavior becomes unstable. In addition, it gives a bad influence to the valve motion and greatly deteriorates the durability of the valve.

Therefore, by applying this calculation method to the experimental engine, we tried to correlate between the calculated and measured results for the cam gear train behavior. We used the crankshaft motion simulation model results as described in 4.1 and input it into the cam gear train model. Figure 17 shows the results of both the calculation and measurement of cam gear angular velocity fluctuation. The calculation has good correlation with the measurement for angular velocity fluctuation in amplitude and phase. As described above, this calculation method enables us to

examine the gear train specifications before building a physical prototype. The flexibility of determining specification was by clarifying the factors that contribute to the rigidity along rotational direction, as described in 2.2, although it was thought the rigidity of each gear should be higher in order to set the resonant frequency outside the normal engine speed range.

5. Conclusion

We established the behavior simulation method of rotational transfer mechanism including non-linear elements and the following was found:

- (1) The behavior of response calculation for a gear train has been completed over a wide range of engine speeds and the jump phenomenon that is the characteristic of a non-linear response was simulated. This technology has enough accuracy and is applicable to commercial engine development.
- (2) Analysis accuracy has been improved for performing behavior simulation of a gear train by adding elements such as deformation of bearing, radial clearance, and slant rigidity of the tooth flank.

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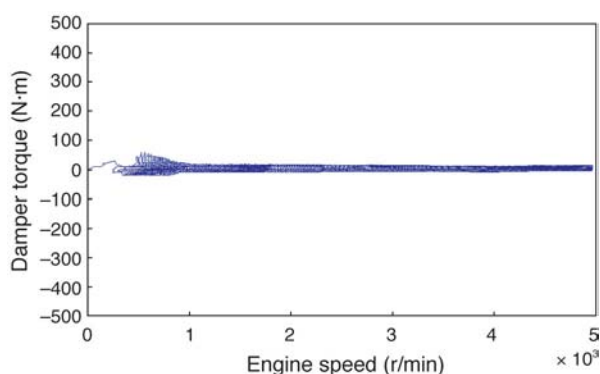


Fig. 16 Calculated damper response, in case of F6-type engine operating WOT, scanning mode

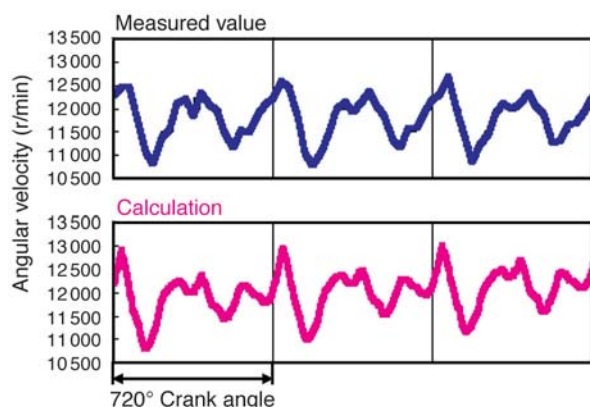


Fig. 17 Correlation between physically test and simulation about camshaft angular velocity

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