DEVELOPMENT OF GEAR TRAIN BEHAVIORAL ANALYSIS TECHNOLOGIES CONSIDERING NON-LINEAR ELEMENTS

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A numerical calculation method, which enables the analysis of gear train behavior including non-linear elements in a motorcycle engine, was established. During the modeling process, it was confirmed that factors such as bearing distortion, radial bearing clearance and elastic deformation of a tooth flank could not be neglected because they effect the rotation behavior. To keep a high accuracy, those factors were included in the simulation model, after they were converted into the rigidity elements along the rotational direction of each gear model. In addition, the model was combined with a crankshaft behavior calculation model for a driving and excitation source. A time domain numerical integration method was used to perform the transient response simulation across a wide range of engine speeds. A jump phenomenon of response behavior of the driven gear was predicted that is a characteristic of non-linear response. The phenomenon was also observed in a physical test. The simulation result was verified with an accurate physical and this technology could be applicable to the development of motorcycle engines.

Keywords : Gear Train, Modeling, Friction, Backlash, Finite Element Method (FEM), Forced Torsional Vibration, Nonlinear Damper, Numerical Analysis, Transient Response

1. Introduction

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 speeds. In order to meet these requirements, many gear train mechanisms are adopted for the valve train system and accessory power train system, as shown in Fig.1.

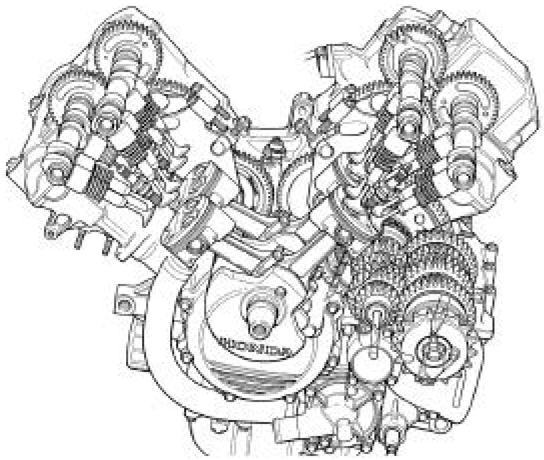


Fig.1 Example of gear train system for a motorcycle engine

Incorrect behavior of the gear train system reduces durability and becomes a source of noise that making a product
less attractive. Undesirable behavior of gear train includes an unstable revolution speed leading to rattle noise of gears, thrust movement of helical gear, inclination due to a lack of bearing rigidity, etc. The prediction of gear train behavior becomes difficult, by considering these unnecessary behaviors for rotation transfer.
When resonance occurs in the gear train system of a motorcycle engine consisting of small parts, the usual countermeasure is to increase the natural frequency of the system and to set the resonant frequency outside the 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 was no tool for predicting gear train behavior even for rotational transfer direction.
This paper reports how we developed a simulation method for the rotational behavior of the gear train system, and the obtained non-linear response solution has a good correlation with the experimental result.

2. Behavior Calculation Model

Fig.2 shows the composition of the basic model used for each transfer element of the gear train model. This basic model is a forced torsional vibration system which has a single degree of freedom, and of which spring, damper, backlash and friction are taken into account.
The model of the gear train system is a one-dimensional model that consists of many basic models, and each basic model is combined in-line or parallel. The number of the degrees of freedom is decided only by number of transfer elements. The rotational behavior of the total gear train system is managed by many factors

included in many basic models. And additionally, each element has non-linear characteristics.

In the engine, the driven gear is forced to oscillate caused by the fluctuating angular velocity of the crankshaft. Therefore, the gear train behavior model was combined with the existing model(1) that could analyze the behavior of the crankshaft, as a driving and excitation source as shown in Fig.3 and calculated simultaneously. That r iven by the cylinder pressure model. A c as used for the modeling and calculation.

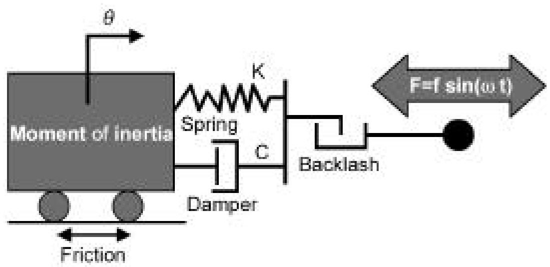


Fig.2 Forced torsional vibration model of single degree of freedom system with cl

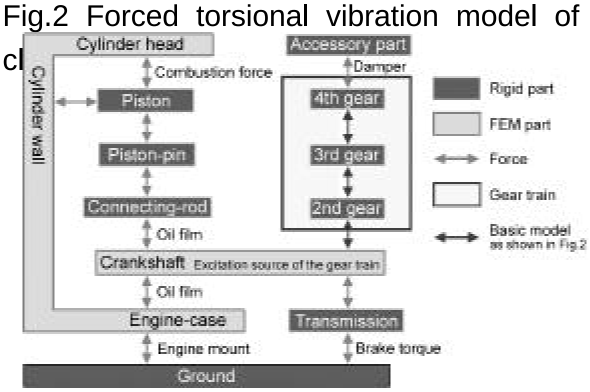


Fig.3 Schematic of the behavior analysis model of the gear train

3. Modeling of Non-linear Elements

3.1 Gear stiffness

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 radial clearance in its own bearing, and could move in it. Such movement of drive gear in the radial direction influences on the rotational displacement of the driven gear. In the gear train, these harmful influences are integrated one by one. Therefore, it is very difficult to predict the rigidity in the rotational direction of each gear by simulation, same as it is difficult to physically

measure it. We devised a method to obtain the rigidity in the rotational direction of each gear by combining calculation results with measurement results.
T od applied to the cam gear train of a m el shown in Fig.4 is applied.

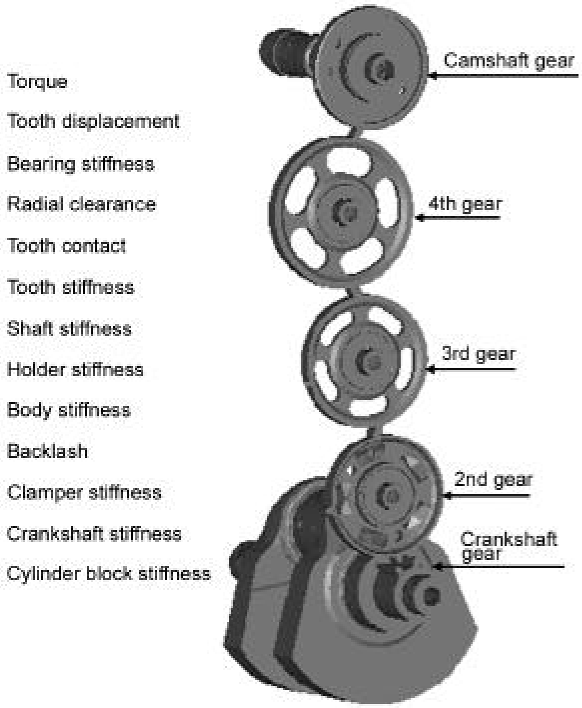


Fig.4 Structure of gear stiffness calculation model

First, we tried to correlate the statical stiffness of the whole gear train system along the rotational direction between one calculated by the static FEM analysis and the physically measured one. We listed the effective factors for gear train stiffness along the rotational direction, and added these factors one by one into the FEM static analysis model. Fig.5 is one of the results of contribution ratio analysis from the FEM result. It was found that the difference between the physically measured value and t ng factors that influence the rigidity in the r model one by one.

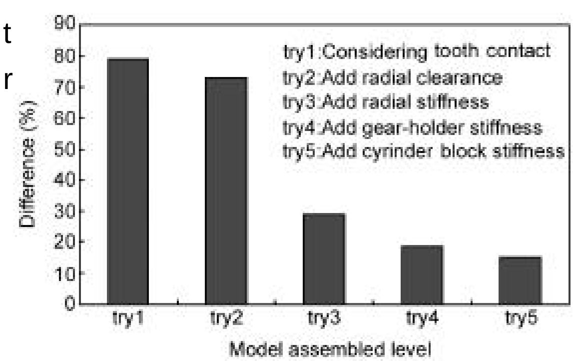


Fig.5 Difference of gear train total stiffness between simulation results and physically

measured values

Next, the stiffness obtained in the rotational direction for the whole FEM gear train models was correlated with the physically measured value as shown in Fig.6. On the r lial direction is also measured, at normal a

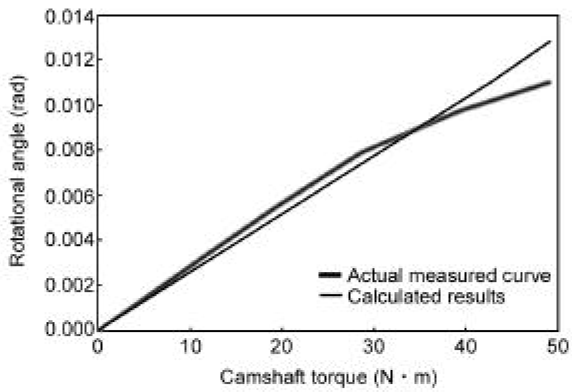


Fig.6 Correlation between simulation and test about the gear train system torsional stiffness

Finally, we separated the individual stiffness of each gear as shown in Fig.7. The gradient of each graph shows the rigidity in the rotational direction, K of each gear calculated by static FEM analysis. The small gradient in the graph means that not c he rigidity of the bearing is high as well, v

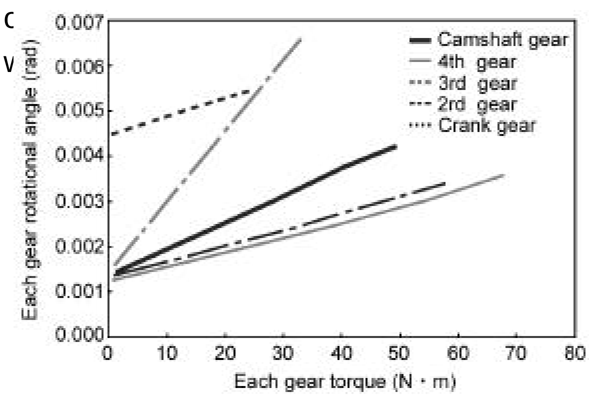


Fig.7 Torsional stiffness of each gear

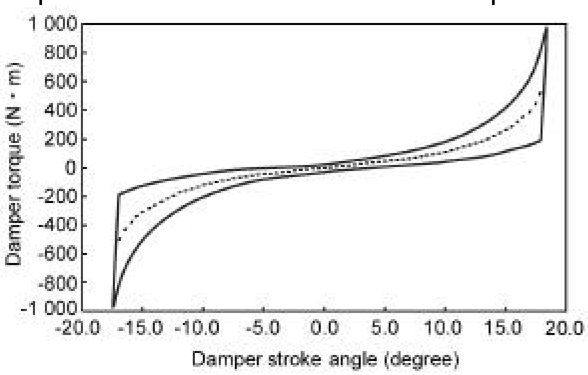
For the actual statical FEM analysis model, the gear holder, cylinder block and even the measurement equipment are modeled precisely. The influence of equipment was removed from the gear train system rigidity using the contribution ratio analysis. Damping coefficient C is determined as a unique function of K by parameter study.

3.2 Damper structure

There are deformations of each part and clearances in an actual gear train, which becomes a cause to generate impact loads to the tooth flank. Therefore, we often use a damper which is reducing the shock using the damper stroke operated by the deformation of a spring and rubber and frictional force occurring simultaneously. Generally, the phase difference due to the damper stroke is larger than the gear deformation and clearance.

And this operation angle of the damper generates a phase difference in the gear train.

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.8. As for the non-linear spring characteristics of the damper, we adopted the median value of the hysteresis curve. The hysteresis was also adopted for the damper spring characteristic that is decided by the relative angular velocity vector occurring between parts of drive side and driven side of the damper. Frictional resistance of damper was dealt with as continuous slide. The fr ... st constant value at more than certain fi ... ro at no speed difference, and to be c...



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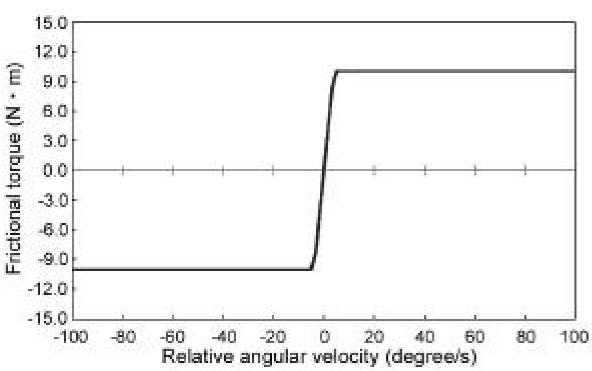

Fig.9 Continuous friction torque

3.3 The other non-linear elements

In case of the power train system of an actual machine, the backlash of each part and the stopper in the damper limiting the damper stroke must be considered.

We got each backlash value from the graph of gear stiffness as shown in Fig.7. The cross points of the vertical axis, where torque zero, and the extension line of each gear stiffness mean the movable range caused by the backlash and radial clearance.

The stiffness of the damper stopper was taken from the static FEM analysis. These non-linear elements have a discontinuity and we arranged that the damping coefficient does not occur rapidly in order to make the calculation go smoothly, at the moment of stopper impact.

When backlash and angular velocity fluctuation are considered simultaneously, it is understood that the gear tooth flank repeatedly comes into contact and separates frequently. Therefore, we used time domain numerical integration method according to Newton-Ruphson for the behavior calculation of a gear train model.

4. APPLICATION EXAMPLE

4.1 An application to generator power train system

For adopting a gear train system for an accessory drive line on a 90 degree V-type 4cylinder engine, as shown in Fig.10, we produced a simulation model for the b ft. We carried out a behavior reply analysis of the g using this model simulating a firing operating c

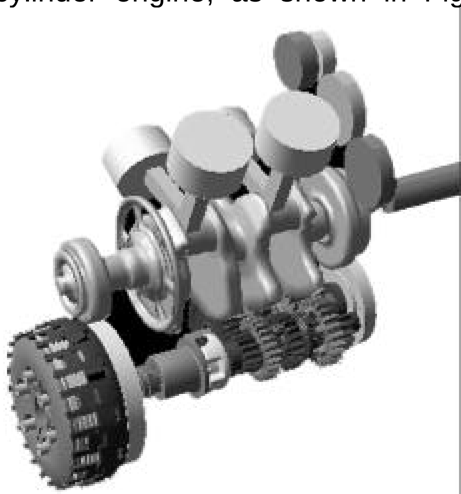

Fig.10 Model of V4-type crankshaft behavior simulation

The operating condition was set to full load acceleration, at an engine speed range of

3 000-10 000r/min. Fig.11 shows a 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 amplitude converges in discontinuity was observed. The engine speed when jump occurred was slower by 300r/min than the resonance engine speed (9 864r/min) that was calculated theoretically by moments of inertia and spring characteristics in the current model. The transfer torque fluctuation that is absorbed by the damper is shown in Fig.12. It was observed from the calculation result that the amplitude of the damper torque increases by engine speed, and suddenly converges discontinuity at high engine speed. And it was also found that the occurred timing of the jump phenomenon synchronized between stroke angle amplitude and damper torque amplitude. Therefore, we made an experimental damper having the same

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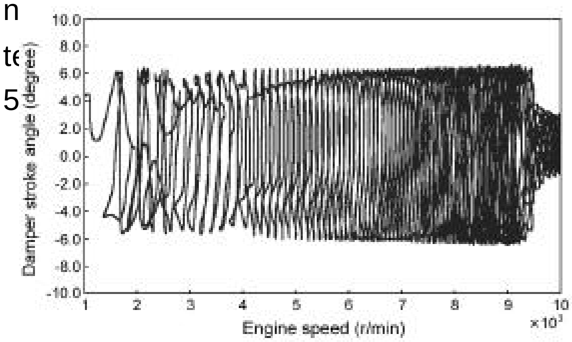

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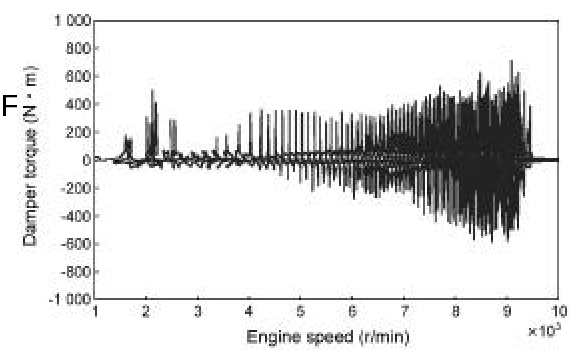

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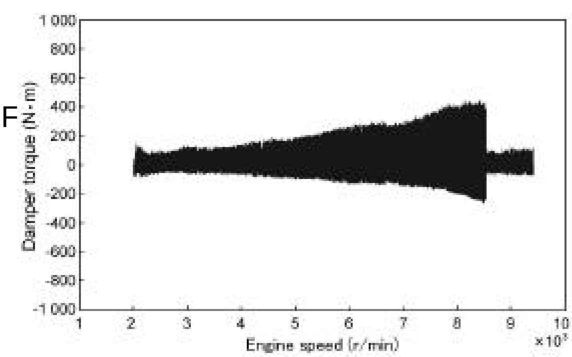

Fig.13 Measured results of damper torque operating by WOT, scanning mode

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

4.2 Trial of for effective 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. We tried to generate the jump phenomenon effectively in low engine speeds less than 3 000r/min where the usage of the engine concerned is seldom, and using time is very short, and to operate at the normal engine speed range of 3 000-10 000r/min where the stable area of damper behavior is.

Through the parameter studies, to generate jump phenomenon at 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 predicted engine speed at which jump phenomenon occurs, we used the measured values of the damper characteristics and the in-cylinder pressure.

Fig.14 shows simulation result of a transient response of damper stroke angle amplitude, and Fig.15 shows measurement result with actual test engine. Although there is difference in absolute value of amplitude, we confirmed the good correlation of the engine speed at which jump phenomenon occurs. However, we also found that th … n at engine speeds lower than 3 000r/min re … maller spring rate, that means too large la

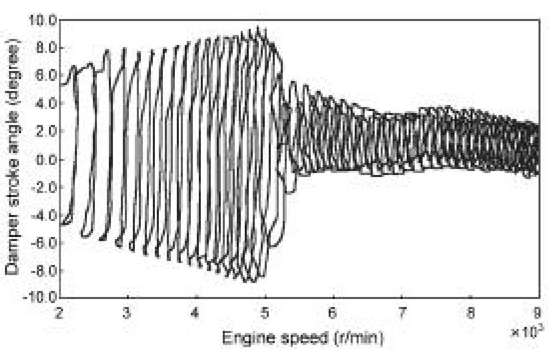

Fig.14 Calculated damper response, in case of V4-type engine operating WOT, scanning mode

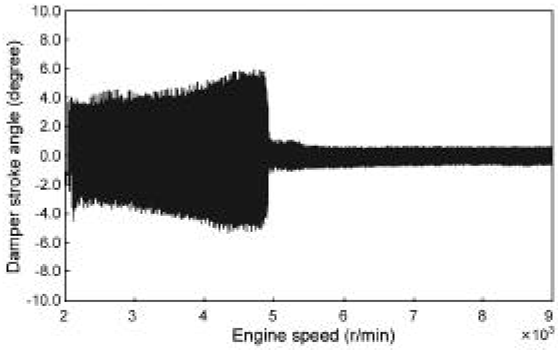

Fig.15 Physically measured damper response, in case of V4-type engine operating WOT, scanning mode

4.3 An example of jump phenomenon practical usage

This calculation method was applied to a 6-cylinder engine as shown in Fig.16 having a higher excitation frequency than 4-cylinder engine. The gear train system is installed as generator drive line and a damper was mounted on the drive side of the generator in this engine. The Stroke angle of the damper is large and spring rate is small. Operation mode was set to full load acceleration of 1 000-5 000r/min. Damper torque was calculated as shown in Fig.17. This graph shows that the amplitude of torque is already small enough at a low engine idling speed and this gear train sy[illegible] range of calculation. This contributes to re[illegible]

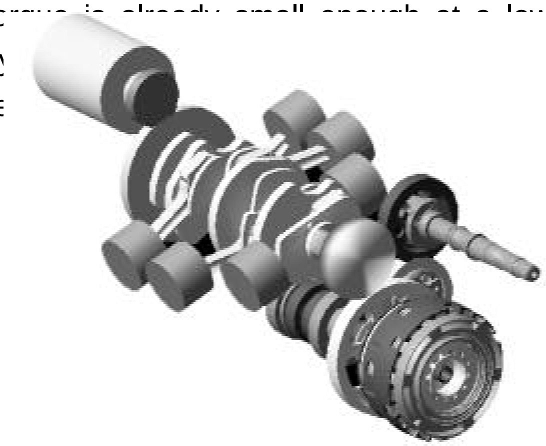

Fig.16 Model of F6-type crankshaft behavior simulation

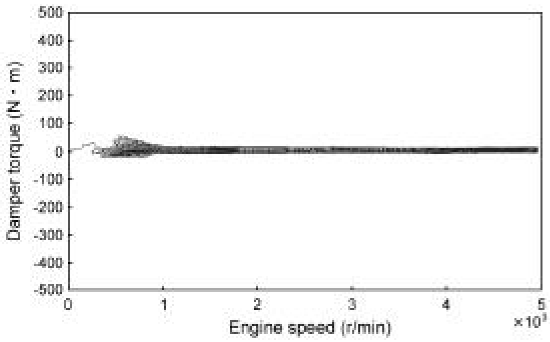

Fig.17 Calculated damper response, in case of F6-type engine operating WOT, scanning mode

4.4 Application example to cam gear train

In case of the high speed engine, the gear train is often adopted for the valve train system. It is understood that one advantage of cam gear train is the 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 case the rotational behavior becomes unstable. In addition, it gives a bad influence to the valve motion and deteriorates the durability of the valve remarkably. Therefore, by applying this calculation method to the experimental engine, we tried to correlate the behavior of the cam gear train between calculated results and measured ones. Fig.18 shows the results of calculation and measurement of cam gear angular velocity fluctuation. The calculation has a good correlation with measurement about 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. And the way to determine specifications become more flexible by clarifying the factors which contribute to the rigidity along rotational direction, although it was thought that the best countermeasure for resonance is to make the rigidity of each gear higher.

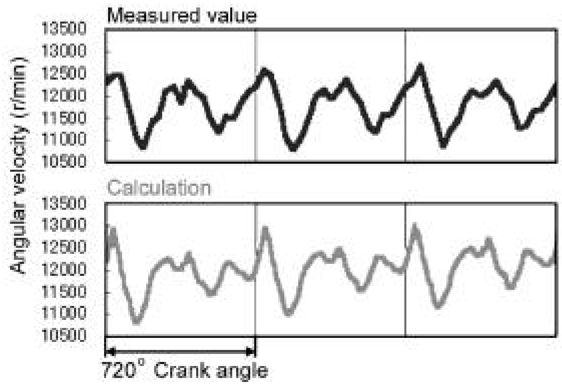

Fig.18 Correlation between physically test and simulation about camshaft angular velocity

5. Conclusion

A behavior simulation method of rotational transfer mechanism including non-linear elements was established, and the following was found.

(1) Behavior of response calculation of a gear train has been carried out at a wide range of engine speeds and the jump phenomenon that is the characteristic of non-linear response was simulated. This technology has enough accuracy and is applicable to the commercial engine development process.

(2) Analysis accuracy has been improved for performing behavior simulation of gear train, by adding elements such as deformation of bearing, radial clearance, slant rigidity of tooth flank.

References

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