

Prediction of Chain Load in Timing Chain System

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ABSTRACT

A chain load prediction technique was established to investigate timing chain durability under conditions of a variable valve lift timing mechanism and a continuously variable valve timing mechanism. Modeling a tensioner with a relief valve and valve train made it possible to predict the chain load at different relief pressures and valve timings. Furthermore, telemeter techniques were also developed enabling direct measurement of the chain load at high engine speeds, which was formerly impossible. The validity of the simulation was also verified using an actual engine. These developments made it possible to predict the chain load up to high engine speeds at different relief pressures and valve timings.

1. Introduction

Cases of engine cam drive using chains are increasing in recent years in order to improve durability and reduce size. This makes it necessary to predict the load produced on the timing chain from the design stage. For this reason, techniques for predicting timing chain loads have been actively researched by various companies for the past ten or so years. A wide range of efforts have been reported from simple analysis^{(1)–(3)} that performs calculations by adding the valve train torque to a simple model which views the cam and chain as a mass-spring system, to large-scale analysis^{(4)–(6)} that performs calculations by replacing each link plate of the chain with a particle. However, simple analysis cannot adequately take into account the effects of hydraulic tensioners which are the most important for development, and large-scale analysis also has problems such as the extensive calculation time that makes timely analysis impossible. Therefore, this paper reports the development of simulation technology capable of accurately predicting the chain load in a short time from the design stage.

2. Focus and Themes

Timing chain system behavior is governed by the inertia of the cams, sprockets and other parts, and by the attenuation of the spring comprised by the chain and subcomponents. The hydraulic tensioner in particular is a hydraulic mechanism with a built-in relief valve, and has a large effect on the chain load.

Also, in the case of gasoline engines, the valve train drive torque is the dominant force that excites the chain system, and the valve train parts have a large effect on the chain drive torque.

Therefore, a simulation technique was established focusing on the following points.

Focus 1: Modeling the characteristics of a tensioner with a relief valve

Theme: Assuring the accuracy of tensioner frequency dependent behavior due to relief valve flow channel resistance

Focus 2: Creating a model that includes the valve train which is the main exciting force on the chain

Theme: Assuring the accuracy of changes in the chain load due to differences in the valve train specifications

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3. Modeling the Characteristics of a Tensioner with a Relief Valve

3.1. Simulation Model

Fig. 1 shows the tensioner used in this study. The force applied to the tensioner plunger is maintained by the main spring and the oil taken into the high-pressure chamber. If the input load is high at this time, the oil leaks from the relief valve and the gap between the plunger and the housing opens to reduce the load.

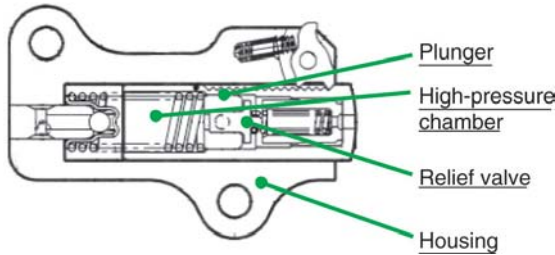


Fig. 1 Chain tensioner cross section

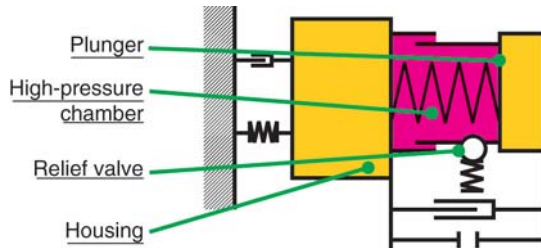


Fig. 2 Tensioner simulation model

This tensioner is replaced with the coupled vibration model of the plunger, spring, high-pressure oil chamber and housing shown in Fig. 2. The main spring and high-pressure chamber models are located between the piston and the housing, and the pressure-dependent nonlinear stiffness of the oil due to the influx of air is also taken into account. An oil intake check valve is also incorporated into the high-pressure chamber model. In addition to leakage from the gap between

the piston and the housing, oil discharge from the relief valve was also modeled. Taking into account the pressure loss due to the viscous resistance of the oil during discharge, the flow rate of the oil passing through the relief valve at this time was approximated using the following equation.

$$Q = \pi \cdot d \cdot l \cdot \sqrt{\frac{2 \cdot P_h}{\rho \cdot (1 + \xi)}}$$

Q : Flow rate

d : Valve sheet diameter

l : Valve lift

P_h : High-pressure chamber oil pressure

ξ : Coefficient of loss

ρ : Oil density

3.2. Verification Test

Discrete vibration tests were conducted using a hydraulic exciter to understand the dynamic characteristics of the discrete tensioner. Fig. 3 shows an outline of the test apparatus. The tensioner is fixed on the plunger and excited by pressing the plunger against a load detector. Excitation was performed by feeding hydraulic fluid from an external pump to the tensioner. Measurement assumed crank second order resonance up to 6,000 rpm, and was performed in the three steps of 30 Hz, 100 Hz and 200 Hz. Excitation consisted of displacement control, and was performed with a sine wave having a double amplitude of 0.4 mm. The validity of the simulation was verified by creating a Lissajous' waveform from the load generated and the tensioner displacement.

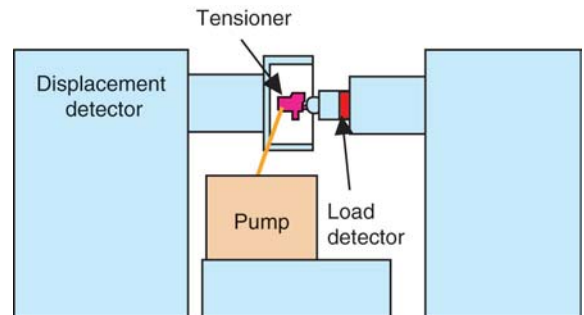


Fig. 3 Hydraulic excitation test apparatus

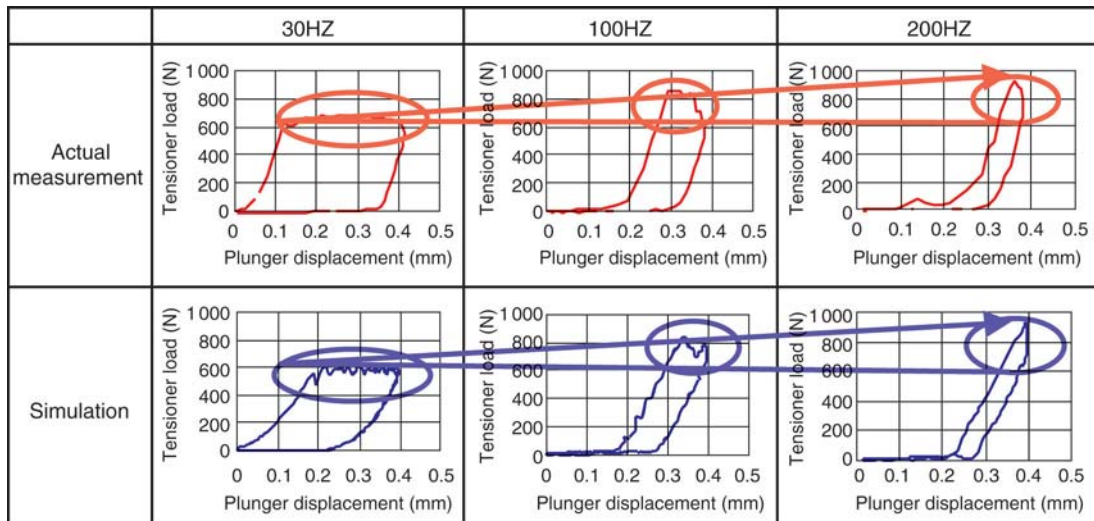


Fig. 4 Result of tensioner discrete excitation

3.3. Verification Results

Fig. 4 shows that the load and displacement could be accurately reproduced at 30 Hz, 100 Hz and 200 Hz. In particular, the reduction hysteresis and increase in the load at high frequencies could be accurately reproduced. This established that the simulation possessed sufficient practical accuracy for the discrete tensioner.

4. Creation of a Model Including the Valve Train which is the Main Exciting Force on the Chain

4.1. Simulation Model

The validity of the chain load simulation was verified using the engine shown in Fig. 5. This engine combined a variable valve timing lift mechanism and continuous variable valve timing mechanism, and featured high fuel economy, high output and reduced exhaust gas levels.

To perform the simulation, a calculation model was created of the chain system and the valve train in order to predict the timing chain load. Each valve system part was divided into particles, and a multi-body system approach that links each particle with stiffness and attenuation was adopted. Calculation used the second order Predictor/Corrector method⁽⁷⁾, and calculations were performed in the steady operation state in 200 rpm intervals.

Fig. 6 shows the overall model. The valve, rocker arm and cam shaft drive train model is driven by the chain model. Also, the chain model tension is maintained by the model of the tensioner with a relief valve described in section 3.1.

The details of each part are described below.

Fig. 7 shows the variable valve timing lift mechanism valve train, and Fig. 8 shows its model. Model creation focused on the case when using a high-speed cam with a large chain load. At high speeds the three rocker arms are coupled by pins. Therefore, the simulation also modeled the rocker arms in the same manner with the high-speed rocker arm in the center driving the left and right rocker arms.

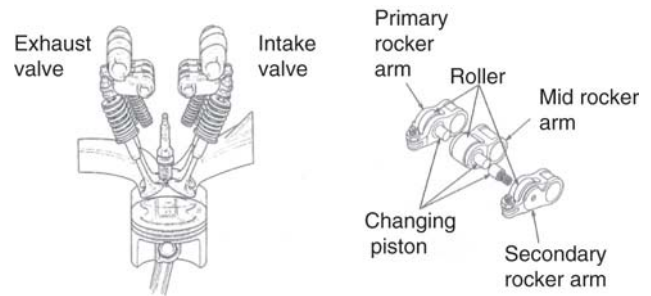


Fig. 7 Valve train

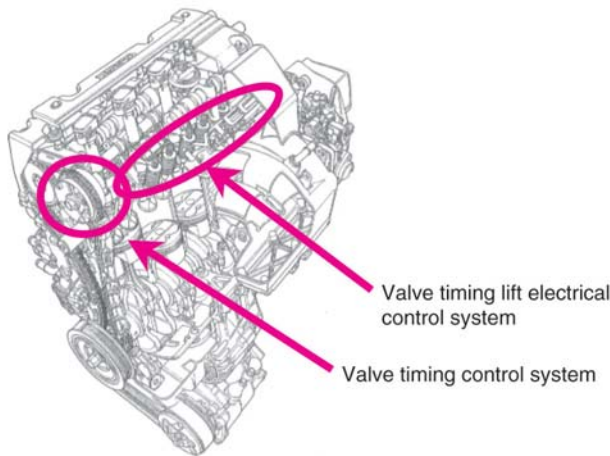


Fig. 5 Engine used for validation

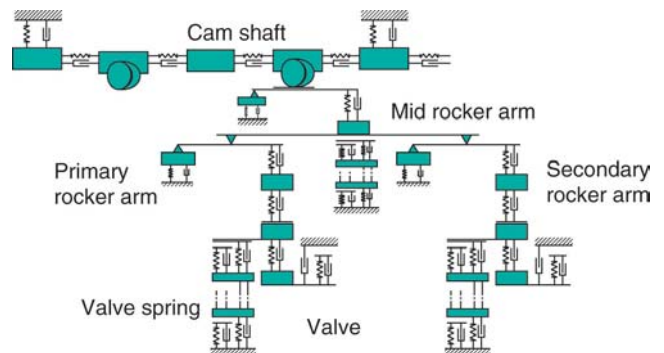


Fig. 8 Valve train simulation model

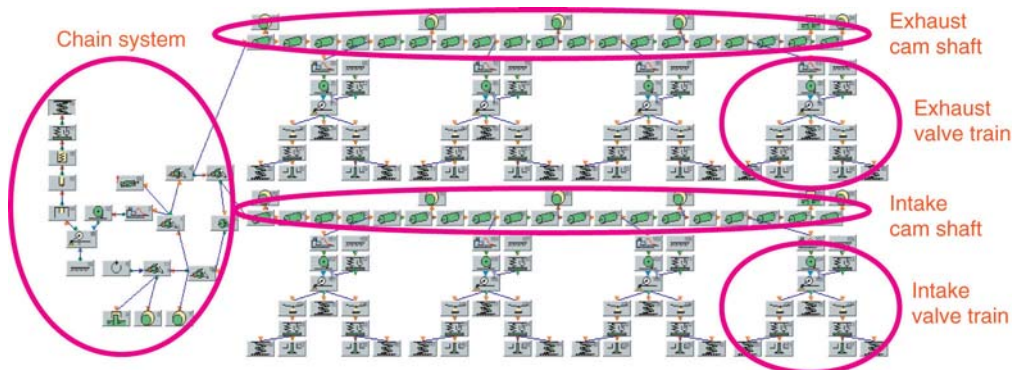


Fig. 6 Overall simulation model for timing chain load

Fig. 9 shows the structure of the continuously variable valve timing mechanism. This mechanism is a vane type actuator, and transmits driving force via oil filling its four chambers. This makes it possible to reduce the stiffness in comparison to steel parts.

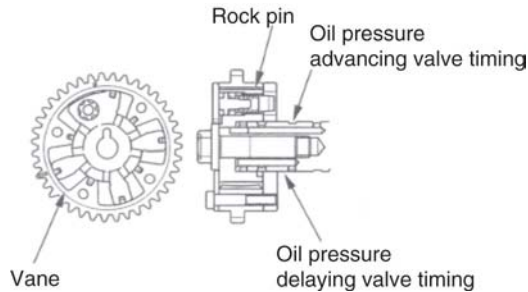


Fig. 9 Valve timing control actuator

Therefore, the stiffness was estimated and modeled using the following equation.

$$C = 4 * K * \frac{A^2}{V} * r^2$$

C: Stiffness

K: Bulk modulus of the oil (oil air content 10%)

A: Pressure receiving area

V: Oil volume per chamber

r: Effective radius

Fig. 10 shows the chain arrangement. The chain is a 6.35 mm pitch silent chain that drives both the intake and exhaust sprockets simultaneously.

Guides are provided to prevent lateral chain vibration on the tension side between the intake cam and the crank and between the intake and exhaust cams. The tensioner maintains the chain load on the slack side between the exhaust cam and the crank using an arm.

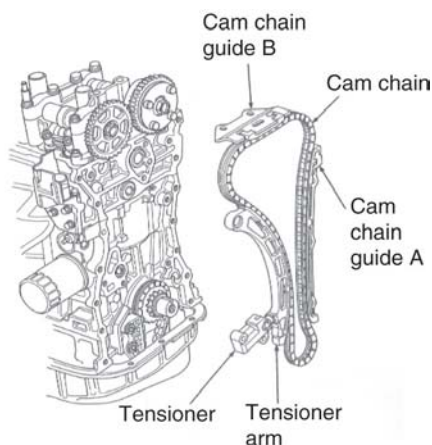


Fig. 10 Timing chain system

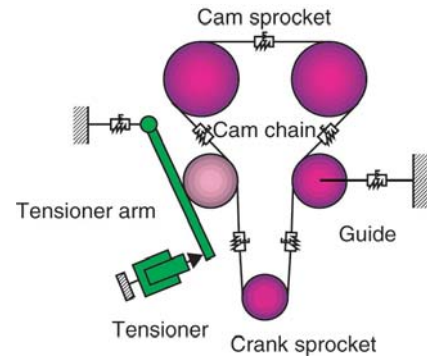


Fig. 11 Timing chain simulation model

The timing chain load is maintained in the direction perpendicular to the chain advance by guides and a tensioner in this manner. Thus, during normal operation, the tensioner plunger displacement is approximately 0.1 to 0.2 mm, and the lateral chain vibration can be ignored. Therefore, a one-dimensional model was created in the longitudinal vibration direction which is the main direction of chain tension.

Fig. 11 shows the chain model. The inertia of each part is coupled by the chain stiffness and attenuation, and an arm and tensioner are provided between the exhaust cam and the crank as in an actual engine. The model was created so that the tensioner maintains the chain load via this arm.

4.2. Tests to Verify the Validity of the Chain Load Simulation

Verification used an engine under development, and the simulation results were compared with the measurement results of firing tests.

4.2.1. Verification by torque measurement

First, the validity of the chain load simulation was verified by a relatively simple torque measurement.

The chain drive torque was measured by affixing a strain gauge to a thin tube attached to the crank sprocket. The simulation created a similar axle model and calculated the drive torque. This drive torque was divided by the pitch radius of the sprocket and used as the chain load for verifying the validity of the simulation. Comparison was performed in the high speed range using the high-speed cam at 6,000 rpm or more when the chain load is high. In addition the inlet valve timing advance was varied in the three steps of 0, 30 and 50 degrees as shown in Fig. 12 to check the tracking performance of the simulation. Fig. 13 shows the results of comparing the maximum chain load values.

The simulation accurately reproduced the maximum chain load over the wide range from 6,000 to 8,600 rpm. The simulation also accurately followed changes in the valve timing advance. Fig. 14 compares the time series data at a valve timing advance of 30 degrees at each speed of 6,000, 7,000 and 8,600 rpm. The waveforms match at each speed, thus confirming that the chain load can be reproduced with high accuracy.

Next, the tracking performance of the simulation at different tensioner relief valve pressures was verified. Fig. 15 shows the results when the relief pressure was lowered from 2.2 MPa to 1 MPa. These results verified that the developed simulation has excellent tracking performance even with respect to changes in the relief pressure.

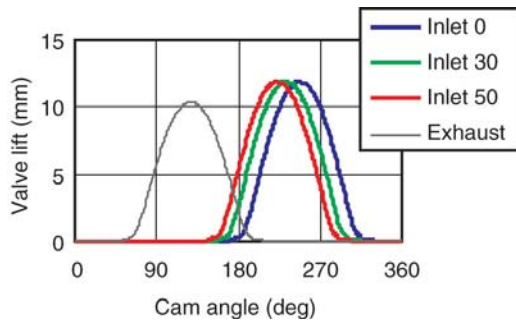


Fig. 12 Valve lift timing

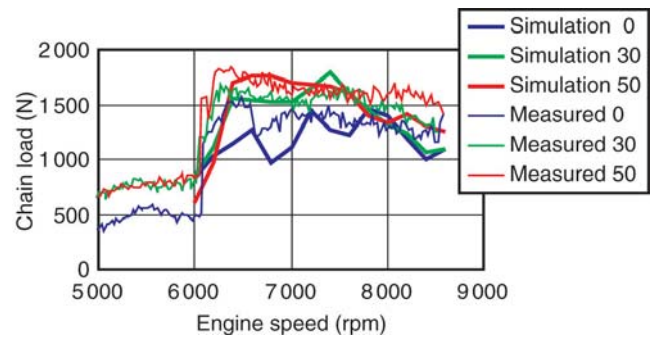


Fig. 13 Maximum value of timing chain load

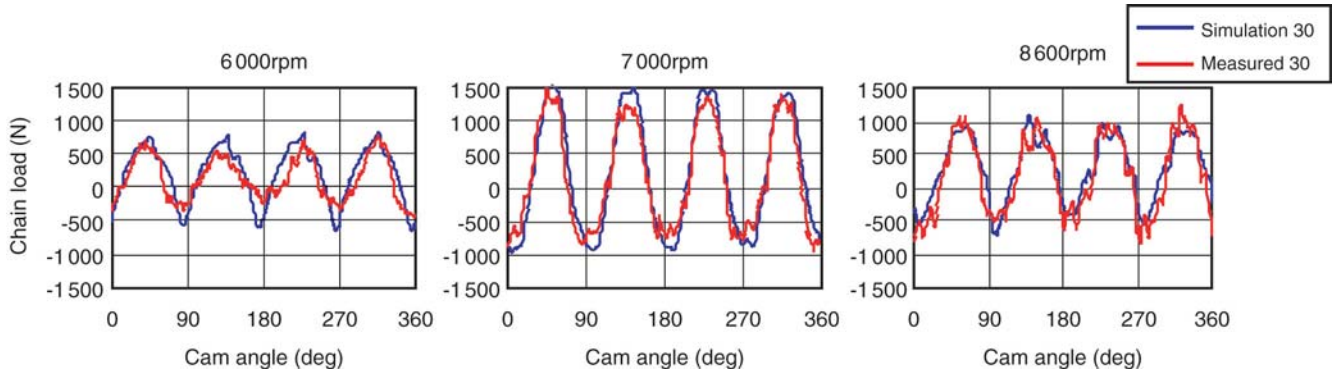


Fig. 14 Timing chain load fluctuation

4.2.2. Verification by chain load measurement

Estimating the chain load from the torque generated in the crank portion is a simple technique. However, this technique is unable to measure not only the pre-load applied to the chain, but also the chain load generated between the cams. Therefore, the signal from the strain gauge affixed to the chain must be extracted in order to directly measure the chain load. There have also been cases in the past where measurement was performed by connecting a piano wire or other auxiliary cable. However, this technique requires that the chain cover be removed for measurement, which not only

causes lubricating oil to splatter, but the piano wire also flies about when the piano wire mount comes off due to centrifugal force, which is extremely dangerous. Therefore, these problems were solved by using a telemeter.⁽⁸⁾ Fig. 16 shows a chain with the telemeter attached. The chain load is detected from a strain gauge affixed over the guide plate. The holder weight was reduced to allow measurement up to the

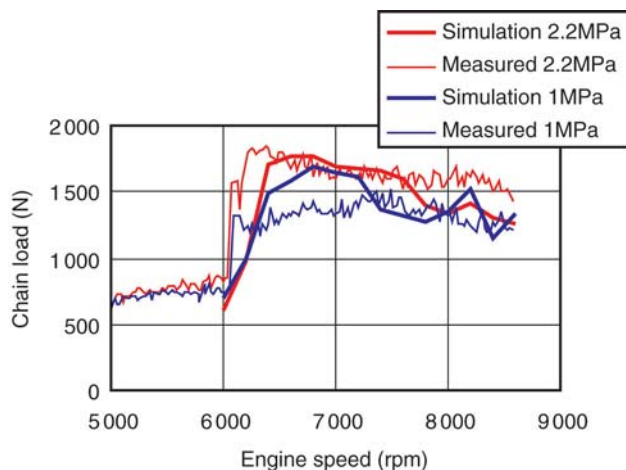
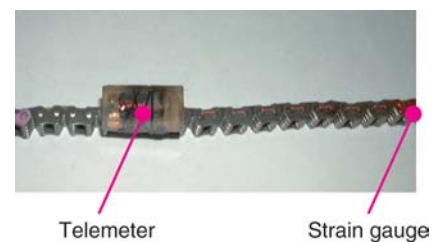

 Fig. 15 Maximum value of timing chain load
(Relief pressure 2.2 Mpa, 1 Mpa)


Fig. 16 Chain with telemeter

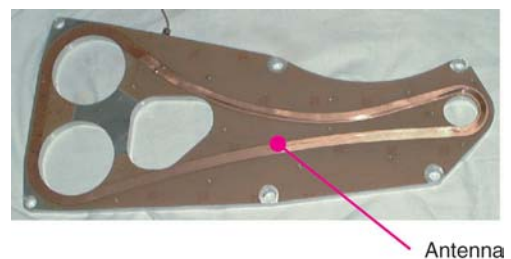


Fig. 17 Antenna for telemeter

high rpm range, and the strain gauge and holder were separated by an appropriate distance to prevent the strain gauge from being affected by centrifugal and inertial forces. Current was supplied from the antenna shown in Fig. 17 through the telemeter to the strain gauge, and the signal detected at the same time was extracted externally from the telemeter through the antenna.

Fig. 18 shows an example of the output signal. Changes in the chain load could be measured continuously as the chain passed through the tension side and slack side spans and the span between the cams. Using this telemeter system made it possible to directly measure the chain load up to the high rpm range of 8,000 rpm or more.

The load generated on the chain was also taken from the simulation and compared with these results.

Fig. 19 compares the chain load measurement results obtained using the telemeter and the simulation results for each span up to 7,000 rpm. These results confirmed that changes in the chain load can be accurately reproduced in each span.

5. Example of Application

The established simulation was applied to investigate measures for reducing the chain load. Fig. 20 shows the sprocket shapes before and after optimization, and Fig. 21 shows the chain load for each shape. The moment of inertia of the exhaust cam sprocket was increased from 0.00029 kgm^2 to 0.0007 kgm^2 , and the chain second order resonance rotations were adjusted to minimize the cam second order excitation force. This reduced the chain load by approximately 15%, and these effects were also confirmed using an actual engine.

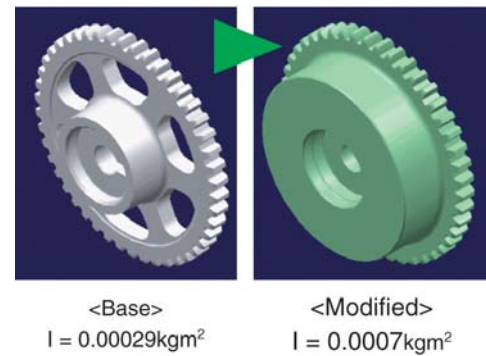


Fig. 20 Cam sprocket shapes

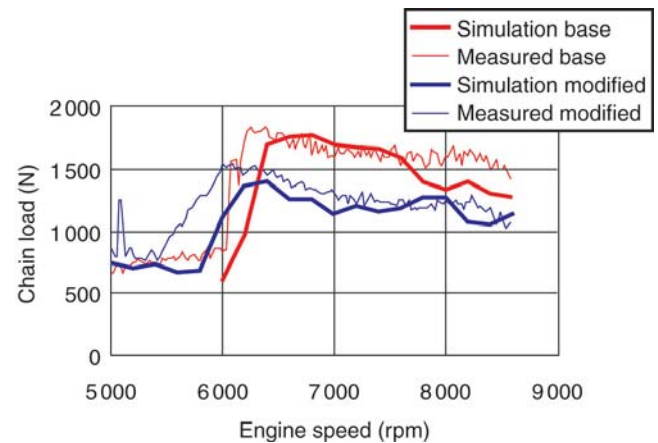


Fig. 21 Example of application

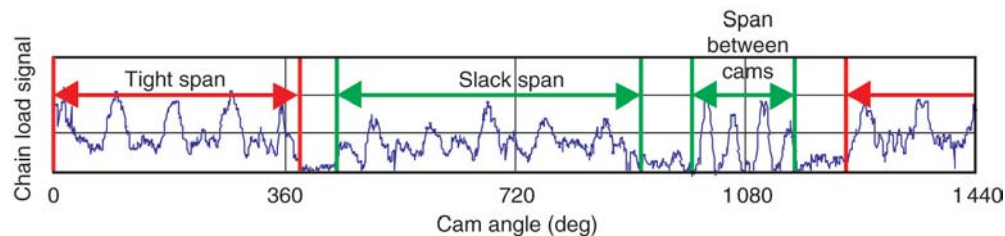


Fig. 18 Example output signal

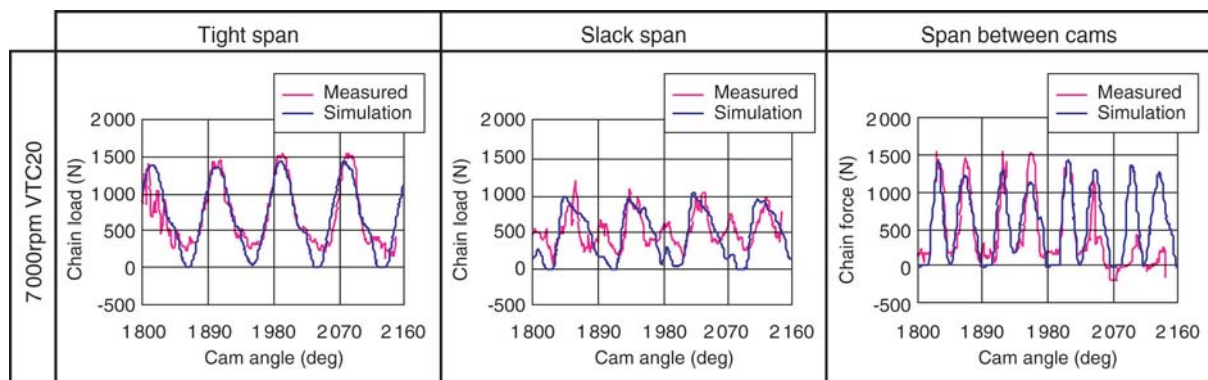


Fig. 19 Chain load comparison between simulation and telemeter measured data

6. Conclusion

This research yielded the following results.

- (1) A timing chain load prediction technique was established which takes into account tensioner frequency dependent behavior due to relief valve flow channel resistance and has sufficient accuracy for practical application to different valve timings.
- (2) This simulation was applied to an actual development, and confirmed to be effective for reducing the chain load.

7. Acknowledgements

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