

Chain Load Simulation for Turbo Diesel Engine

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ABSTRACT

Fluctuation of crankshaft speed in a turbo charged direct-injection diesel (i-CTDi) engine significantly increases the load on the timing chain.

A simulation method was developed that calculates the load on the chain caused by these fluctuations. All moving parts, centered on the crankshaft, are connected in the simulation, which is able to more accurately reproduce a working engine on a computer. The model was simplified by focusing on the frequency response of each component when it was being assembled, enabling calculation accuracy to be maintained with a reduction in calculation time.

This enabled the load on the chain in an i-CTDi engine to be both accurately and rapidly predicted. The virtual engine contributed significantly to the optimization of the design of the timing chain in an actual engine.

1. Introduction

The load applied to the rotating parts inside an i-CTDi engine is larger than that of a gasoline engine because of the high combustion pressure used. Also, if a fuel injection pump is installed on the end of the camshaft of an i-CTDi engine, the torque fluctuation on the camshaft will increase, causing the load on the timing chain and oil pump chain to increase as well. For this reason, technology for performing a simulation to accurately predict the transmission load generated inside the engine was necessary.

In a conventional chain load simulation, it was general practice to employ a method in which modeling was performed on a limited number of parts. For example, in the case of the timing chain, simulation was previously performed using a model in which the parts responsible for generating the main vibromotive force were limited⁽¹⁾, such as the chain and valve train parts. This method was effective in cases where fluctuations of the crankshaft speed were small, such as in the case of a naturally aspirated gasoline engine.

However, in actual engine development, maximum

priority is placed on obtaining an accurate grasp of phenomena concerning a working engine, so virtually all analysis consisted of behavioral and load measurement on a combustion operation engine. For this reason, analysis using the conventional simplified model produced a large discrepancy between measurement and simulation, and was inadequate for quantitatively predicting the transmission load.

Simulation has become large in scale. Figure 1 shows an example of large-scale analysis. Simulation includes calculations for determining the crankshaft behavior from the abridged results of FEM rigidity of the cylinder block and crankshaft, and takes into account various factors including each link in the chain, making it quite complicated. In order to simulate the chain load when linking these large-scale analyses, much calculation time was required and it was difficult to coordinate the results of simulation with actual development.

To overcome these issues, technology was established for predicting the chain load that can contribute to i-CTDi engine development. A description of this technology is reported below.

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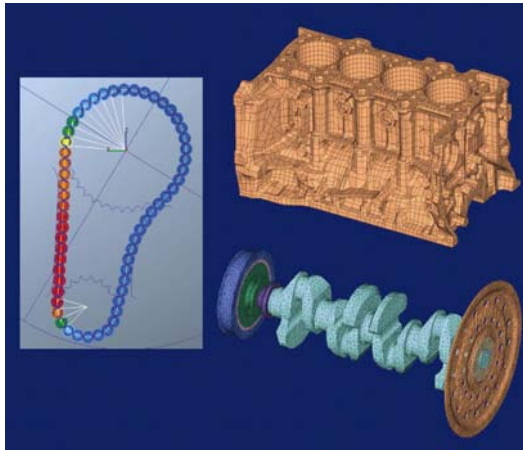


Fig. 1 Large-scale simulation model

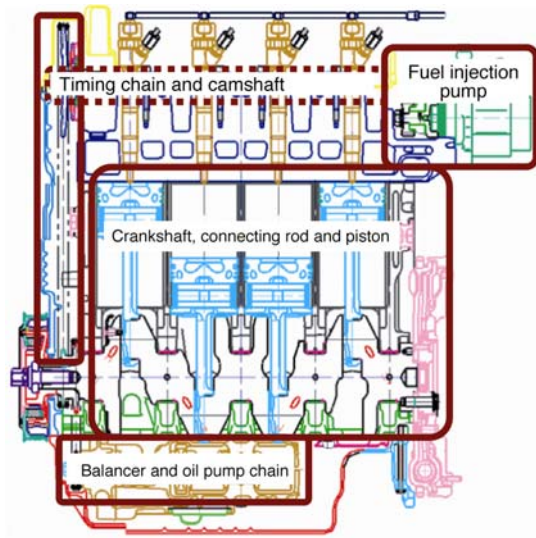


Fig. 2 i-CTDi engine

Table 1 General specification of i-CTDi engine

Number of cylinders	4 cylinders
Flywheel	Dual mass flywheel
Crank pulley	Isolation pulley
Balancer	Balancer driven by roller chain
Cam drive	Roller chain and gear
Number of cams	2 camshafts
Number of valves	Intake: 2 valves / Exhaust: 2 valves
Valve train	Roller rocker arm with hydraulic adjuster
Fuel injection pump	Common rail injection pump
Auxiliary belt	Serpentine poly-V belt

In the development, attention was paid to simulation technology that simultaneously coupled all moving parts centered about the crankshaft. Also, a study was carried out concerning the high prediction accuracy of the transmission system chain load and a practical calculation time.

2. Simulation Model

2.1. Configuration of a Virtual i-CTDi Engine

Table 1 shows the main specifications of the i-CTDi engine, and Fig. 2 is a cross-sectional diagram. Also, Fig. 3 shows the overall simulation model developed on this occasion. In this model, all of the moving parts were coupled, and a working engine was more accurately constructed on a computer. In the simulation model, the combustion pressure acted on the pistons, driving the crankshaft via the connecting rod, and the chains, gears, and belts coupled to

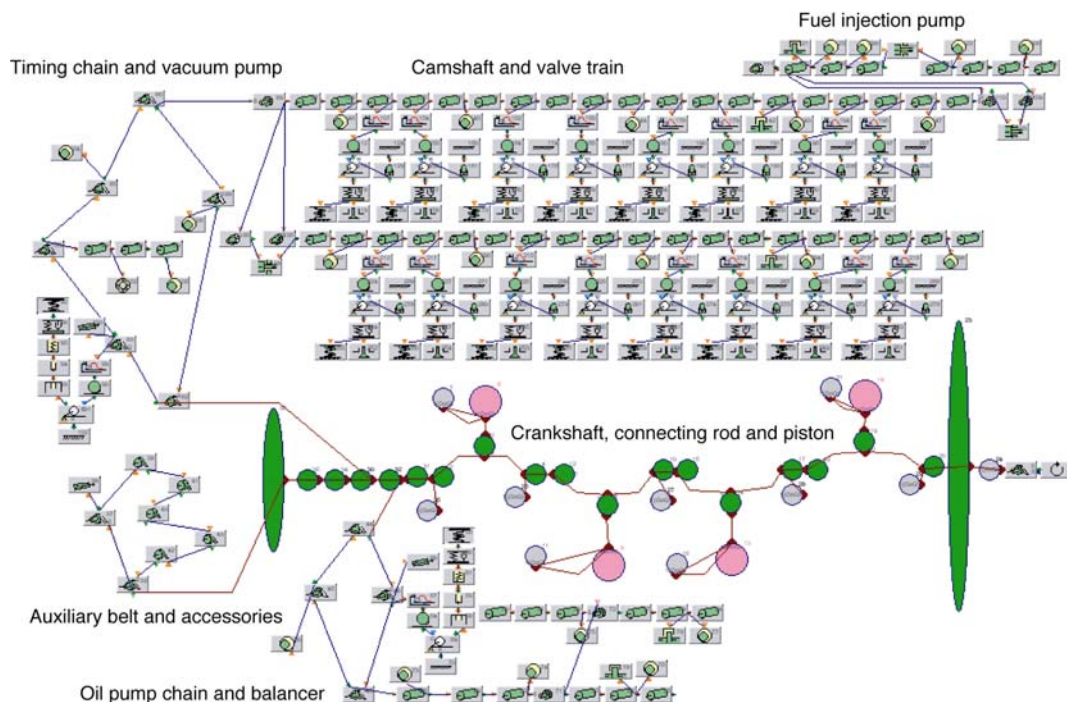


Fig. 3 Virtual i-CTDi engine

the crankshaft drove all of the parts including the cams, valves, fuel injection pump, vacuum pump, balancer and accessories simultaneously.

In this development, a highly accurate and unique model was constructed based on the TYCON software made by AVL, which most closely matched the objective. Details of this modeling technology are given below.

2.2. Crankshaft

The fluctuation of the rotational speed of the crankshaft of an i-CTDi engine is larger than that of a gasoline engine, and greatly increases the chain load. In the conventional simulation, the crankshaft model had six degrees of freedom and was intended mainly for studying details including stress evaluation. However, fluctuations of rotational speed exert the greatest effect on each part. For this reason, in order to reduce calculation time, it was decided to construct a crankshaft model that had degrees of freedom only in the direction of rotation and ignored bending deformation. Details of this crankshaft model are described below.

Figure 4 shows the crankshaft model. A link point for connecting the connecting rod was set on the crankpin, and the piston was connected to this point via the connecting rod. The cylinder was offset from the rotating axis of the crankshaft. A pressure load was applied to the piston, causing the crankshaft to be driven. Also, in order to simulate steady-state running, a constant brake torque was applied to the flywheel side.

The torsional rigidity of the crankshaft was calculated using the FEM-P method, as shown in Fig. 5.

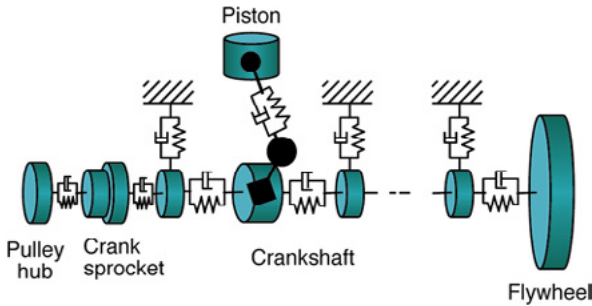


Fig. 4 Crankshaft simulation model

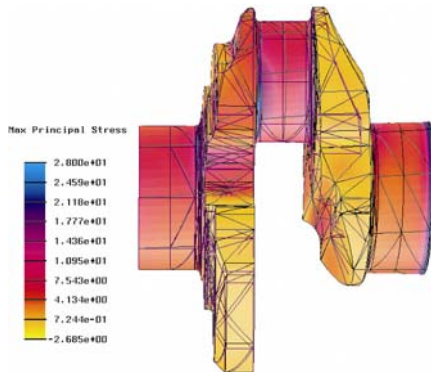


Fig. 5 Crankshaft rotational stiffness analysis

The dynamic behavior in the rotational direction of the crankshaft is determined by the torsional natural frequency characteristics of the crankshaft. The accuracy of the simulation model was guaranteed by confirming the accuracy of the torsional natural frequency of the crankshaft. The method of doing this is described below.

In order to check the frequency response, the crankshaft was excited by means of an impulse function to perform frequency analysis on the fluctuations of rotational speed. Eq. (1) shows the Fourier transformation equation for the impulse function.

$$F[\delta(t)] = \int_{-\infty}^{\infty} \delta(t) e^{-j\omega t} dt = 1 \quad (1)$$

Thus, the ideal impulse function when a Fourier transformation is performed is 1 over the entire area. Consequently, in an actual test, the eigen value is checked using a pseudo frequency response such as that produced by hammering. In the case of simulation as well, it is considered ideal to obtain the frequency response over the entire area. A pseudo impulse function should ideally have as narrow an impulse width as possible. For this reason, it is necessary that the time during which the impulse is applied be as short as possible, as shown in Eq. (2).

$$\delta(t) = \begin{cases} 0, & t \neq 0 \\ \infty, & t = 0 \end{cases} \quad (2)$$

$$\int_{-\infty}^{\infty} \delta(t) dt = 1$$

However, this simulation is calculated using a 2nd order predictor-corrector method. An extremely short step response brings about fineness of the calculation step, and as a result the calculation time increases. Here, the crankshaft was excited using step function. The Fourier transformation of the step function is shown in Eq. (3).

$$F[u(t)] = \int_{-\infty}^{\infty} u(t) e^{-j\omega t} dt = \pi \delta(\omega) + \frac{1}{j\omega} \quad (3)$$

The Fourier transformation of the step function is a function that is inversely proportional to frequency, so the input becomes large in the low-frequency area. However, it was considered that if the frequency range to be studied was limited, there should be no problem in identifying the frequency response. Also, like an impulse function, a step function enables the width definition to be omitted, and thus has the merits of simplifying the definition and reducing the calculation time.

The torsional natural frequency of the crankshaft of an in-line 4-cylinder engine is between about 300 and 500 Hz, so it is important that the simulation reproduces this frequency. The frequency was checked in two stages to take into account the effect of the crankshaft damper. First, a pure torsional resonance check of the crankshaft was performed using a fixed pulley model without a damper, and next the torsional resonance was checked using a damper model that simulated the condition in which a damper was fitted.

2.2.1. Fixed pulley model

The green line in Fig. 6 indicates the step response results. Because of the effect of the soft spring of the dual mass flywheel, a high value was obtained on the low frequency side, but a clear peak was seen at 437 Hz.

The green line in Fig. 7 indicates the results of measuring the 10th order component of the rotational acceleration of the crankshaft pulley hub in one rotation of crankshaft. It can be seen that torsional resonance of the shaft occurs at 2 600 rpm. In other words, $2\,600/60 \times 10 = 433$ Hz is the torsional natural frequency of the fixed pulley shaft, thus indicating that the simulation accurately reproduces the shaft torsional resonance.

2.2.2. Damper model

A damper is installed in this engine in order to reduce the torsional resonance of the crankshaft. Consequently, the main

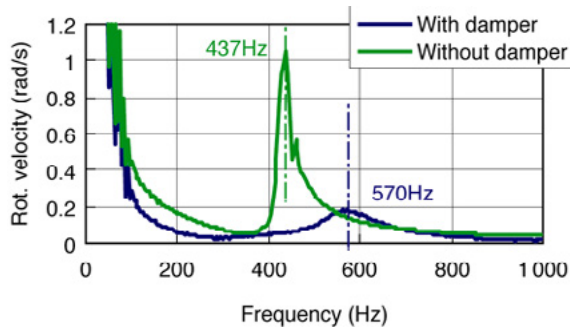


Fig. 6 Response of unit function

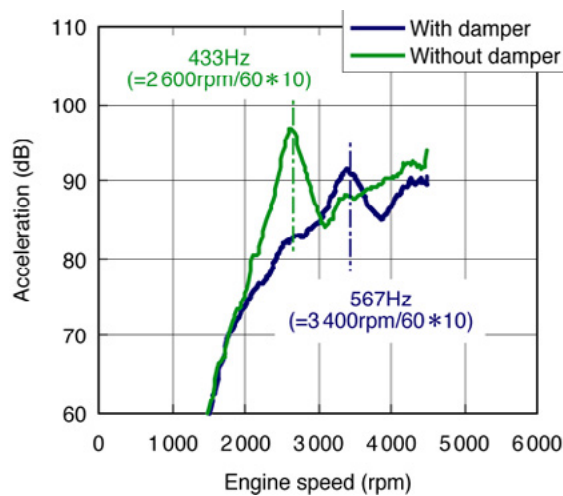


Fig. 7 Crankshaft rotational acceleration (order 10)

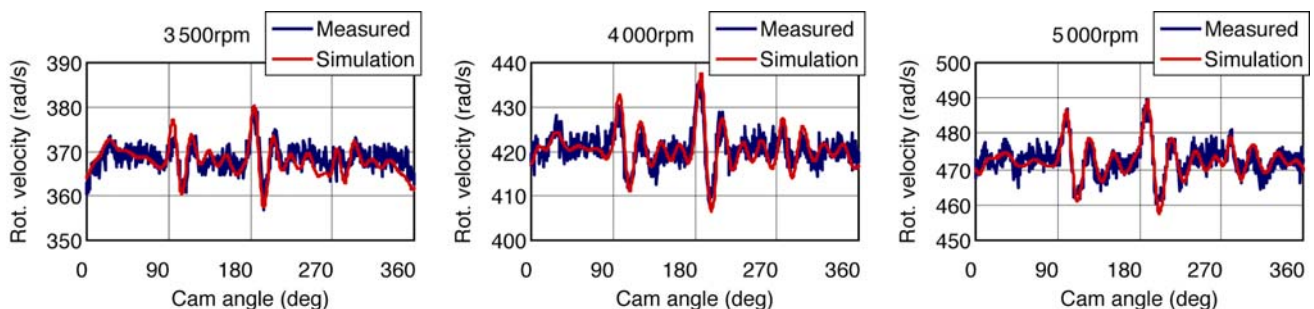


Fig. 8 Crankshaft rotational speed

resonance is reduced but a small resonance frequency appears instead. It was confirmed that this peak could be reproduced in the simulation.

The blue line of Fig. 6 shows the step response results. The peak has shifted to 570 Hz. It can be seen that even for the measurement results indicated by the blue line in Fig. 7, the resonance has shifted to 3 400 rpm. In other words, $3\,400/60 \times 10 = 567$ Hz is the torsional natural frequency, thus indicating that the simulation accurately reproduces the torsional resonance of a shaft on which a damper is installed.

In this way, it was possible to complete a simple crankshaft model that could reproduce the frequency response. The rotational speed fluctuation was verified using the completed crankshaft model. For the measurement, an optical rotary encoder was installed on the hub of the crankshaft pulley. Figure 8 shows the measurement results and the simulation results at 3 500, 4 000 and 4 500 rpm. In the i-CTDi engine, which produces a large explosive force, the effect of rotational speed fluctuation appears prominently when a combustion occurs in the cylinder nearest the crankshaft pulley. The torsional vibration of the crankshaft, which occurs as a result of this combustion, determines the speed fluctuation of the crankshaft. The simulation accurately reproduces not only the maximum speed fluctuation due to this combustion, but also the torsional vibration of the crankshaft as well. This was achieved by model verification based on the step response frequency. It was also found that the simulation accurately followed differences in rotational speed.

This model enabled the calculation for 10 revolutions of the crankshaft when rotating at 1 000 rpm in several minutes by a PC that had a CPU processing speed of 1.5 GHz. Because the model has degrees of freedom only in the direction of rotation, it was possible to construct a highly accurate crankshaft simulation model that permitted analysis to be performed in a short time such as this.

2.3. Balancer, Oil Pump and Oil Pump Chain

The balancer is excited by the fluctuation of the crankshaft speed. This speed fluctuation, which constitutes the excitation force, could be simulated accurately as mentioned above. Next in importance for predicting the transmission load is accurate prediction of resonance in the chain system. Accordingly, modeling was performed using a simple 1-dimensional chain model in the vertical direction that could accurately predict the transmission load. As a result, it was possible to reduce the calculation time.

Figure 9 shows the layout of the balancer and the oil pump. In order to cancel the 2nd order inertial force due to the mass of the reciprocating parts, such as the pistons, two balancer shafts are installed. The trochoid type oil pump that supplies oil to the entire engine is installed on the front shaft. It is driven by the rear shaft via gears. Figure 10 shows the balancer and oil pump model. Like the cams, the two balancers are supported by the journal. Also, a fixed pump drive torque is applied to the pump.

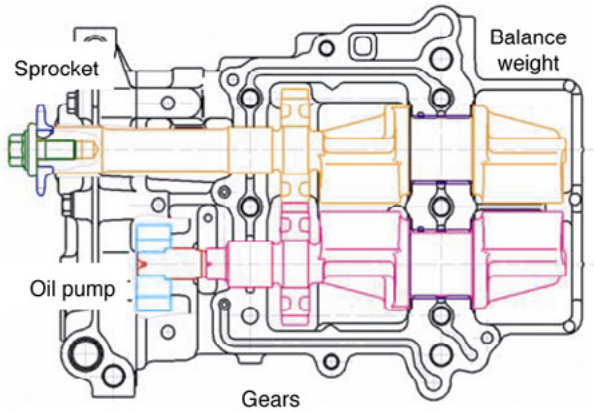


Fig. 9 Balancer shafts and Oil pump

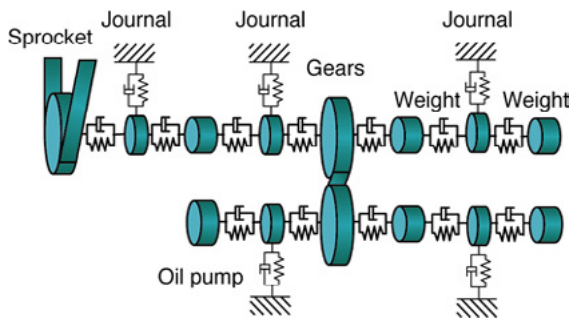


Fig. 10 Balancer shafts and oil pump simulation model

Figure 11 shows the layout of the oil pump chain. The sprocket drives the oil pump that is integral with the balancer shaft at twice the rotational speed of the crankshaft, via an 8 mm pitch roller chain. Figure 12 shows the oil pump chain model. If the tensioner is operating normally, the vertical vibration will constitute the main cause of chain load

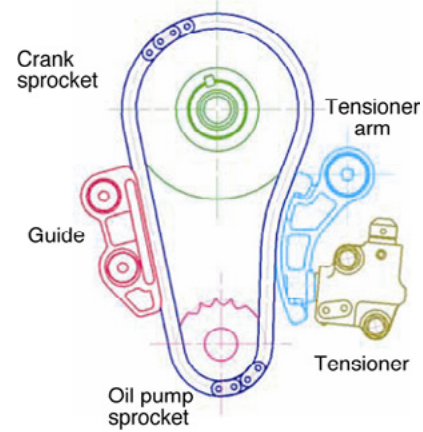


Fig. 11 Oil pump chain

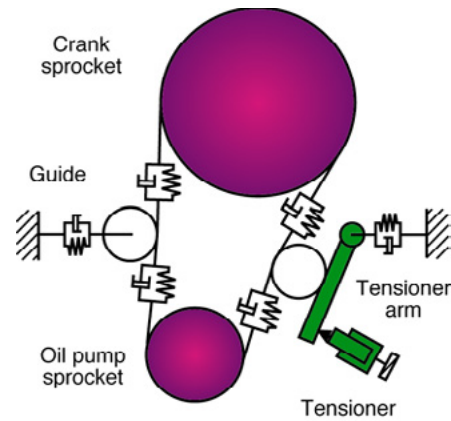


Fig. 12 Oil pump simulation model

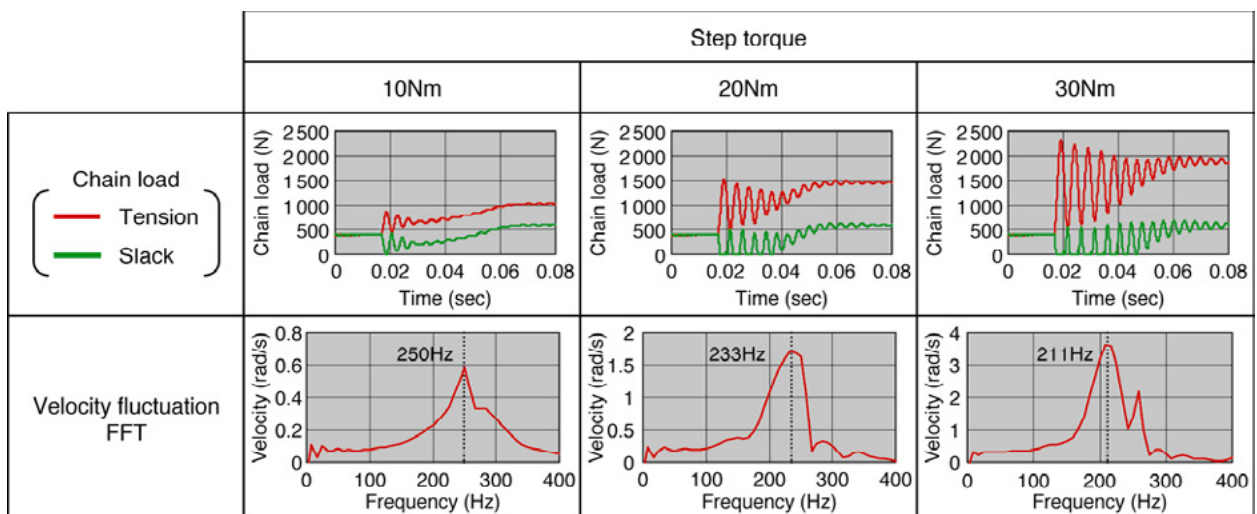


Fig. 13 Influence of step torque on chain resonance

fluctuation. For this reason, the chain was simulated by a vertical direction 1-dimensional model. The moments of inertia of each part were linked by the chain rigidity and an attenuation element. Also, in the model, the tensioner with relief valve maintains the chain tension via the arm.

The oil pump chain has a resonant frequency in the normal rotation range. This resonance exerts a large effect on the chain load. For this reason, the frequency constituents of the step response of the chain were checked in the same way as for the crankshaft. When checking the frequency response of the chain, it is necessary to be careful of the size of the step function. As shown in Fig. 13, the change in the resonance point due to differences in the step torque was verified by simulation. It was predicted that the resonant frequency would vary between 213 and 250 Hz depending upon the magnitude of the step torque. This is due to the non-linearity of the chain system. If the chain load vibrates within the preload, it is possible to maintain the chain tension at both the tension and slack sides. However, if the step function is too large, the chain load on the slack side will be lost, and the load will be maintained only by the chain on the tension side. In this case, as shown in Fig. 14, the spring constant of the chain system will decrease, causing the resonant frequency to fall as well.

Figure 15 shows the result of performing a check of measurement of rotational speed of the actual sprocket on the balancer side. From this graph it was possible to confirm that the resonant frequency in the high rotational speed range shifted to the low frequency side. In actual measurement, the resonant frequency varied between 215 and 250 Hz, which matched the simulation. Thus, when estimating the rotational speed at which the chain resonates, it is necessary to take into account the reduction of rigidity of the chain system due to the loss of the load. As described above, it was confirmed that the frequency response of the simulation model was correct.

The oil pump chain load was verified. Measurement was performed by providing a thin wall area on the crankshaft sprocket and mounting a strain gauge on it. The torque measured here was multiplied by the effective radius of the sprocket and converted into chain tension. Figure 16 shows the measurement results and the simulation results at 2 000, 3 000 and 4 000 rpm. It was found that the simulation could reproduce the measurement results with good accuracy. Consequently, it was verified that the load transmitted to the oil pump chain and the balancer could be reproduced with high accuracy using a simple model.

In addition, the reduction of the chain load was verified using this simulation model. It was established that the chain load could be reduced by optimizing the relief pressure of the

tensioner. Figure 17 shows the effect of the relief pressure on the chain load. By thus optimizing the relief pressure of the tensioner, it was possible to reduce the oil pump chain load.

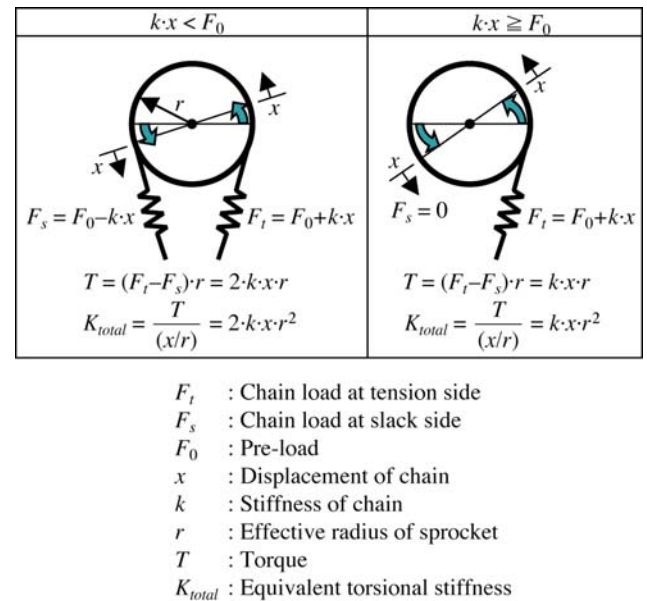


Fig. 14 Equivalent torsional stiffness of chain

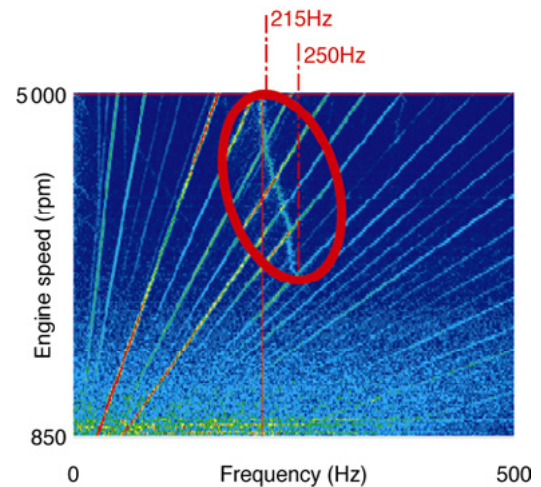


Fig. 15 Rotational velocity fluctuation FFT of balancer sprocket

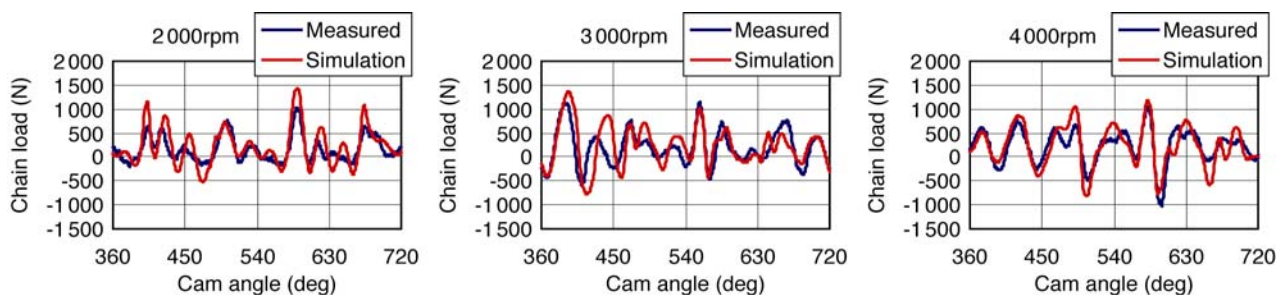


Fig. 16 Oil pump chain load

2.4. Valve Train

Conceivably, the next largest source of excitation following the crankshaft is the valve train. Particularly, in the case of the i-CTDi engine, the fuel injection pump is installed on the end of the cam, and also constitutes a large source of excitation. This excitation force is superimposed on the resonance of the timing chain, generating a high chain load. Details of the valve train model are described below.

Figure 18 shows the layout of the valve train. There are two intake valves and two exhaust valves for each cylinder. The cam has a roller bearing, and drives each valve via the rocker arm. The rocker arm is supported by a hydraulic lash adjuster for freedom from maintenance.

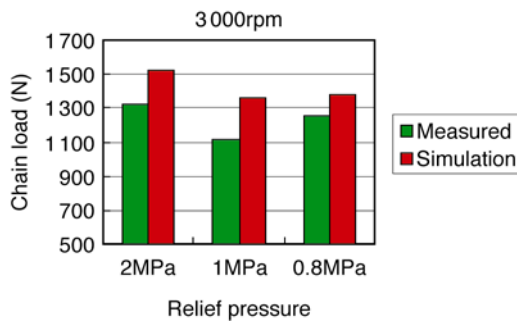


Fig. 17 Oil pump chain load

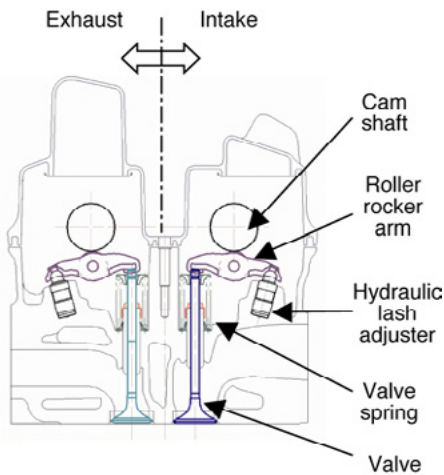


Fig. 18 Valve train

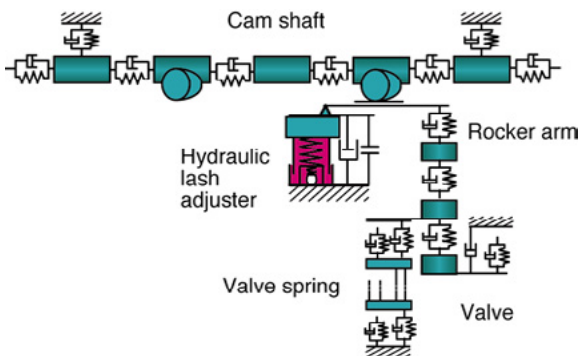


Fig. 19 Valve train simulation model

Figure 19 shows the valve train model. The calculation model consists of the rocker arm that is supported by a hydraulic lash adjuster, the valve and the valve spring. A detailed model was made because the hydraulic lash adjuster exerts a large effect on the valve train excitation force. The oil in the high-pressure chamber is supplied via a check ball by the motion of the plunger. The model also takes into account oil leakage from the gap between the plunger and the housing when load is applied, as well as the dependence on pressure of the oil rigidity due to ingress of air⁽²⁾.

Figure 20 shows the results of verifying the valve lift. An optical type measuring device was used for measurement. The simulation accurately reproduces both the timing and the maximum lift.

Figure 21 shows the cam layout. There are two cams, one for the intake and the other for the exhaust. First, the exhaust cam is driven by the crankshaft via the cam sprocket and the timing chain. In addition, the exhaust cam drives the intake cam via scissors gears located adjacent to the cam sprocket, and simultaneously drive the fuel injection pump via the intermediate shaft by means of the gear installed on the opposite side of the sprocket and the cam.

Figure 22 shows the camshaft model. In this model, the cam was divided into multiple mass points coupled together by beam elements. Regarding the gears, the meshing reaction is simulated 3-dimensionally, enabling the 3-dimension deformation behavior of the cam to be accurately simulated. In addition, the journal is supported by an oil film model to simulate fluctuations of the shaft in the clearance. Like the crankshaft, the frequency response of the camshaft was

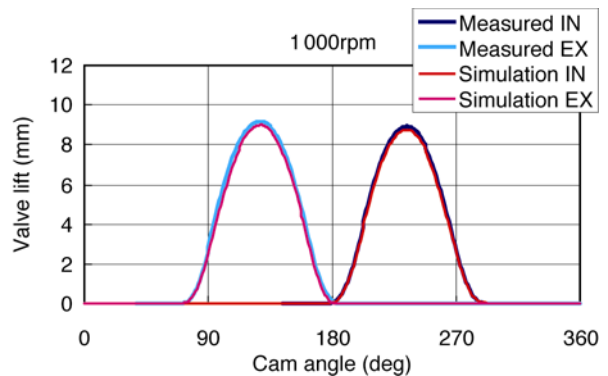


Fig. 20 Valve lift verification

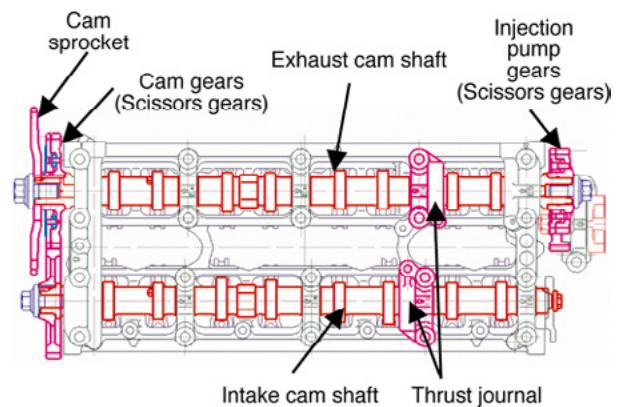


Fig. 21 Cam shafts

checked using the step response results in order to raise the accuracy.

Figure 23 shows the layout of the fuel injection pump and the drive system. The injection pump is mounted on the intermediate shaft installed obliquely beneath the exhaust cam, via an Oldham joint. This intermediate shaft is driven from the exhaust camshaft via scissors gears at a speed of $5/3$ times that of the camshaft.

Figure 24 shows the drive system model of the fuel injection pump. In the model, the Oldham joint transmits only torque. Also, the clearance in the rotational direction of the Oldham joint is equivalent to that in an actual engine. In the model of the intermediate shaft, the shaft is supported by the journal, as in the case in an actual engine, and like the camshaft, it is supported by an oil film. The drive resistance torque of the pump is applied to the shaft of the fuel injection pump.

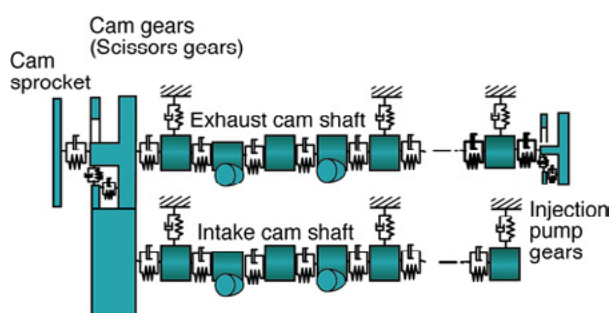


Fig. 22 Cam shafts simulation model

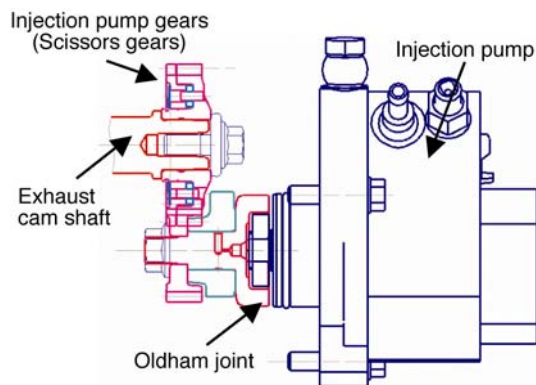


Fig. 23 Fuel injection pump driving system

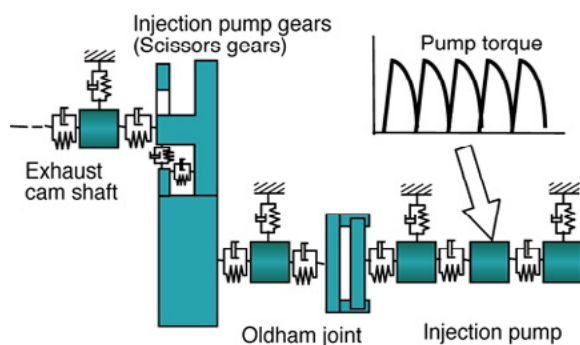


Fig. 24 Fuel injection pump driving system simulation model

By using these models, the thrust load generated in the camshaft holder was verified. The analysis results are shown in Fig. 25. In the simulation, the occurrence of a high impact thrust reaction was confirmed. The cause was found to be due to the high reaction of the fuel injection pump gears causing the cam to strike the holder. Similar measurement results were obtained from stress measurement of the cam holder as well, enabling the appropriateness of the simulation to be verified. The measurement results are shown in Fig. 26. Based on the impact load obtained, the optimum shape of cam holder was designed, and it was confirmed that the stress generated at the cam holder was within the guaranteed range of the material strength.

2.5. Timing Chain

Figure 27 shows the layout of an actual chain. The chain used was a roller chain that had a pitch of 9.525 mm. It

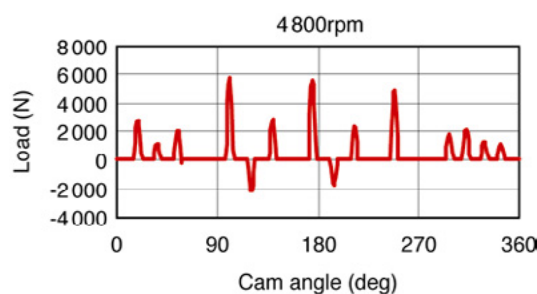


Fig. 25 Simulated camshaft thrust journal load

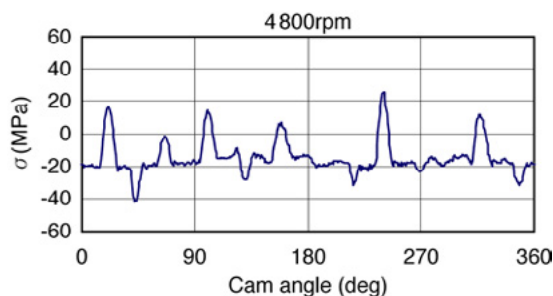


Fig. 26 Measured camshaft thrust journal load

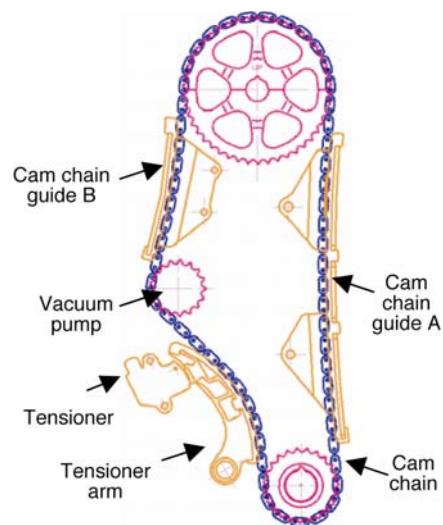


Fig. 27 Timing chain system

simultaneously drives the exhaust cam and the vacuum pump. There is a chain guide between the crankshaft sprocket and the camshaft sprocket, and also between the camshaft sprocket and the vacuum pump. Between the vacuum pump and the crankshaft sprocket, the tensioner maintains the chain tension via an arm. Because the chain is supported by the guide and the tensioner, a 1-dimensional model that ignored the transverse vibration of the chain took account only of vertical vibration in the chain tension direction.

Figure 28 shows the chain model. The inertias of each part are coupled by the chain rigidity and an attenuation element. The model simulated the maintenance of the chain tension by a tensioner with a relief valve via the tensioner arm^{(2), (3)}.

As in the case of the oil pump chain, the frequency response of the timing chain was verified from the step response.

Attention was focused on the timing chain, which is subjected simultaneously to the effects of these drive torques and crankshaft rotation fluctuations, and verification was done. Measurement was performed by providing a thin wall area on the crankshaft sprocket and mounting a strain gauge on it. The torque measured here was multiplied by the effective radius of the sprocket and converted into chain tension. Figure 29 shows the measurement results and the simulation results at 3 000 and 4 000 rpm. The 2.5th order

fluctuation of the fuel injection pump torque and the 2nd order component of the cam drive torque and the crankshaft speed fluctuation overlap each other, resulting in a 0.5th order wave of the timing chain load.

In addition, Fig. 30 shows the change in the minimum and maximum values of the chain load with the engine speed. The simulation was done by performing a comparison using a model without a crankshaft and a full-engine model. If there is no crankshaft model, the accuracy of the chain load in the low rotational speed range will be insufficient. On the other hand, it can be seen that the full-engine model accurately reproduces the measurement results through the entire range, thus verifying its accuracy. It was thus possible to construct a highly accurate timing chain model that enabled an analysis to be carried out in a short time.

Table 2 shows a comparison of calculation time using the large-scale model and that using the new model. This table shows an example in which calculation for 10 revolutions of the crankshaft rotating at 1 000 rpm is performed using a PC that has a CPU processing speed of 1.5 GHz. Compared to large-scale analysis, the new model enables the calculation time to be significantly reduced. By thus adopting independent simple models, it was possible to carry out a full-engine simulation, which is considered to require several weeks when using the conventional complicated method, in several hours. It also enables the load on the chain in the transmission system to be predicted with high accuracy.

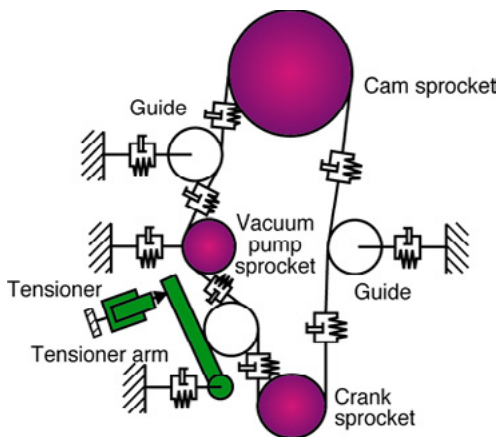


Fig. 28 Timing chain simulation model

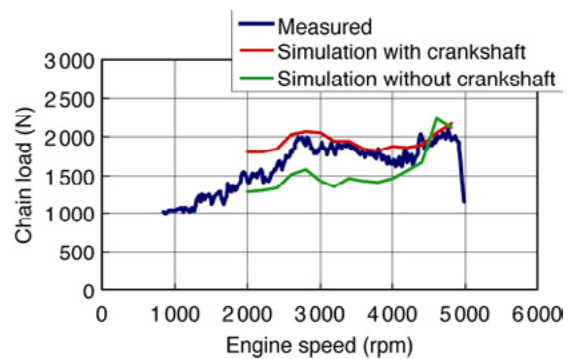


Fig. 30 Maximum and minimum load of timing chain

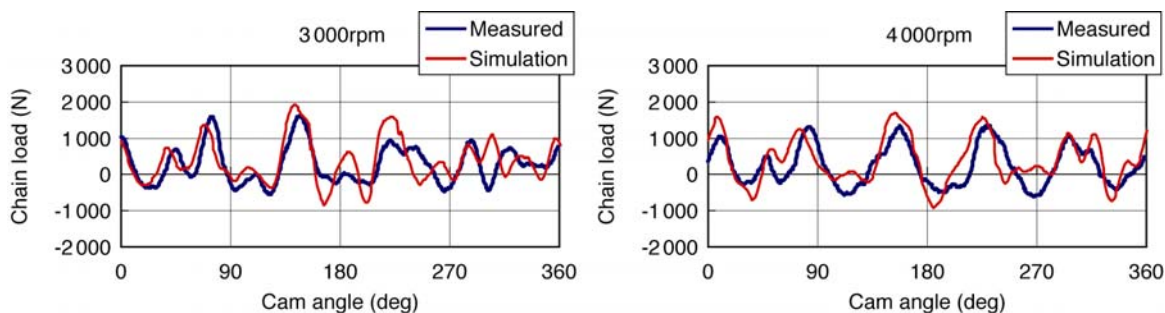


Fig. 29 Time history of timing chain load

Table 2 Calculation time

	T: Time (h)				
	Crankshaft fluctuation	Timing chain load	Oil pump chain load	Auxiliary belt load	Full engine
Large-scale model	$T > 48$	$T > 48$	$T > 24$	$T > 48$	—
New model	$0.1 > T$	$0.5 > T$	$0.2 > T$	$0.1 > T$	$1 > T$

3. Conclusion

The following facts were acquired through the establishment of simulation for predicting the load on the chain in the transmission system of the i-CTDi engine.

- (1) In order to obtain a grasp of the resonant frequency of the crankshaft, impact excitation simulation using a step function was carried out. The simulation results were compared with the actual measurement, and it was found possible to verify the torsional resonant frequency of the crankshaft. In this way, it was confirmed that the new model enables fluctuations of the rotational speed of the crankshaft to be accurately predicted.
- (2) In order to obtain an accurate grasp of the oil pump chain as well, impact excitation simulation using a step function was carried out. In the case of the chain, the resonant frequency varies according to the input torque. It was established that this phenomenon could be predicted using this simulation.
- (3) An accurate model of the valve train and the fuel injection pump, which are major sources of excitation of the timing chain, was made. As a result, it was possible to clarify the phenomenon of impact of the camshaft on the cam holder due to motion in the axial direction.
- (4) By coupling the crankshaft and all moving parts, it was possible to simultaneously and accurately predict the timing chain load and the oil pump load, which are affected by the crankshaft rotational speed fluctuation, the cam drive torque, and the fuel injection pump drive torque.
- (5) The calculation model could be simplified while maintaining accuracy, and realized a reduction in calculation time. As a result, it enabled the transmission system chain load to be predicted in a practical time.

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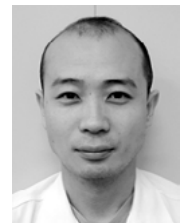
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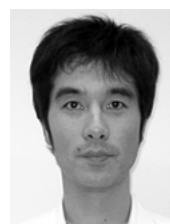
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