

Dynamic Simulation of Oil Pump Chain Using Link-by-link Method

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ABSTRACT

Simulation technology for predicting load on the oil pump chain of engines with high accuracy was established. In the development of this technology, attention was focused on the ability to take into consideration the centrifugal and inertial forces that occur in the chain, and the ability to predict the chain load from the blade tensioner, which undergoes large transversal vibration, to the system, which uses a hydraulic tensioner. To this end, a link-by-link model in which one link is replaced by one mass and adjacent masses are joined to each other by a spring was developed. As a result of this research, the features of the oil pump chain system were clarified, and the behavior of the oil pump chain, as well as the load on it, were able to be predicted with a high degree of accuracy.

1. Introduction

In recent years, there have been an increasing number of cases in which chain systems have been used as a means of linking the rotating shafts in the engine, in order to meet the needs of durability and compactness. One of these systems is an oil pump chain, which links the oil pump to the balancer and the crankshaft. When the vehicle is running, large load fluctuations occur in this chain. This is considered to be due to the combustion force resulting from the combustion of gasoline, and to the fluctuation of angular velocity of the crankshaft, generated by the inertia force that is due to the reciprocating motion of the pistons. These fluctuations interfere with the transmission of stable rotational speed, and cause wear of the chain and noise. In order to overcome such problems, tension is applied to the chain with tensioners, and the running path of the chain is restricted by guides. These tensioners are divided into types that use hydraulic pressure and types that use a spring.

Figure 1 shows an oil pump chain system that uses a hydraulic tensioner. The hydraulic method used is a system in which the end of the arm, which rotates about a pivot, is retained by a hydraulic tensioner. This method

positively prevents transversal vibration of the chain, but it requires a path for supplying working oil, has a complicated structure, and is expensive.

In contrast, the spring method applies a pre-tension force using only a spring, has a simple structure, and is inexpensive. Particularly, a blade tensioner is a spring-type

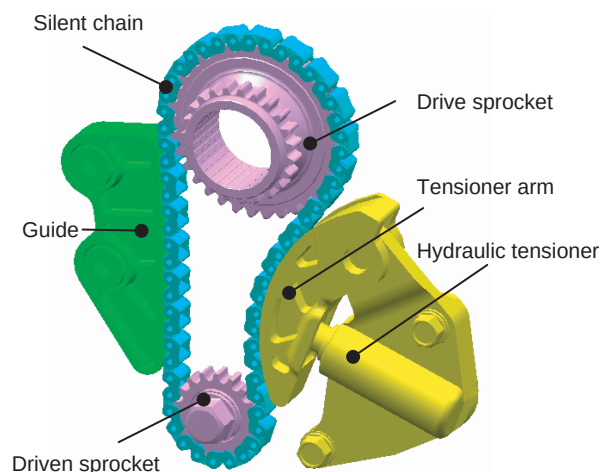


Fig. 1 Chain system with hydraulic tensioner

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tensioner, which functions as both a guide and a tensioner. Figure 2 shows the oil pump chain system, which uses a blade tensioner. A blade tensioner consists of a number of metal leaf springs that are fitted into a plastic guide and preloaded. As shown in Fig. 3, the leaf springs, which are fixed to a pivot, bend while rotating, thus reducing transversal vibration of the chain. The number of cases in which blade tensioners are being used is increasing from an economic viewpoint. However, due to increased combustion pressures and lightening of piston to enhance fuel economy in recent years, fluctuations in the angular velocity of the crankshaft have increased in the low speed region. As a result, the fluctuations in chain tension have increased, and there are cases in which the blade tensioner is unable to retain the oil pump chain. Accordingly, it is necessary to perform simulations that enable the applicable limit of the blade tensioner to be predicted.

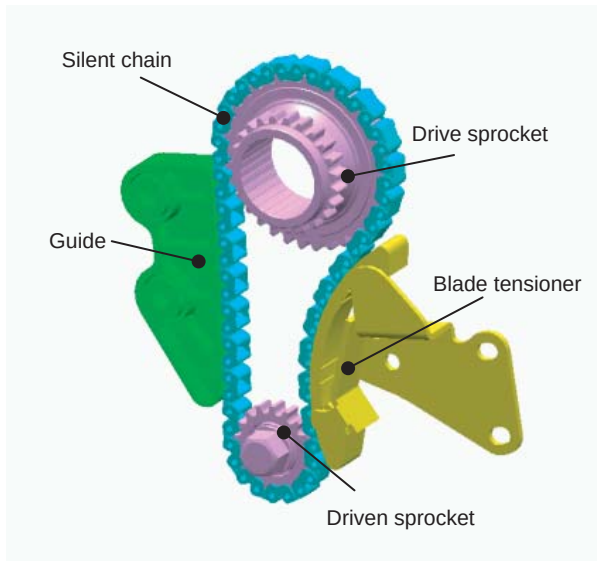


Fig. 2 Chain system with blade tensioner

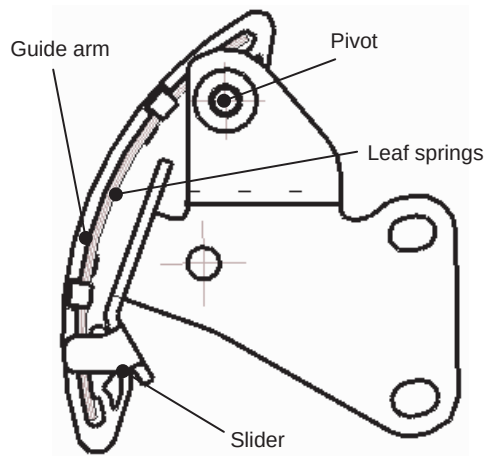


Fig. 3 Blade tensioner

Table 1 General specifications of verified engine

Engine type	Liquid-cooled 4-cylinder turbocharged engine
Displacement	2300 cm ³
Camshaft number	2
Number of valves	Intake: 2 Exhaust: 2
Valve system	VTEC
Maximum torque	353 Nm / 4500 rpm
Maximum power	179 kW / 6000 rpm
Balancer drive system	6.35-mm pitch silent chain with blade tensioner

2. Issues and Action Concerning Conventional Simulation

Until now, simulations of the behavioral load on a timing belt or a chain have generally employed a method using a longitudinal vibration model⁽¹⁾⁻⁽³⁾. However, in the conventional method, the following issues have existed in the prediction of the oil pump chain load.

- (1) It is not possible to reproduce the transversal vibration of the chain, which is generated in the case where a blade tensioner is used.
- (2) In the case of an oil pump chain that has a high speed, centrifugal force is added to the chain load, but a longitudinal vibration model cannot reproduce it.
- (3) The blade tensioner is greatly deformed by the chain load, but the longitudinal vibration model cannot reproduce this behavior.

Accordingly, in this research, a link-by-link chain model, a blade tensioner model consisting of a number of beam elements, and a hydraulic tensioner model, which can solve these issues, were constructed. As a result, a simulation method that cannot only reproduce the transversal vibration of the chain, but also take into account the effect of centrifugal force and inertia force, and the engagement with the sprocket was established. By using this method, the behavior and load of the chain were predicted with high accuracy, and the features of the oil pump chain system were clarified.

3. Making a Model of the Chain System

3.1. Chain

In order to validate the simulation, an inline 4-cylinder turbo supercharged gasoline engine having 2300 cm³ of displacement was used. An outline of the engine is shown in Table 1. A silent chain having 6.35 mm of pitch was used. The link-by-link method^{(4), (5)} was constructed using a 2-dimensional model in order to obtain the transversal vibration of the chain. In the modeling process, one link of the chain was substituted for by one mass point. These mass points have a degree of freedom that permits

horizontal and vertical displacement and rotation, and there are two connecting points at each pin position. In order to join adjacent links to each other, a rotatable spring and a damping element are defined at each connecting point. Also, a rear face and tooth profile are defined at each link, enabling the chain to touch the guide and mesh with the sprocket. This chain model is shown in Fig. 4.

3.2. Blade Tensioner

Figure 5 shows the blade tensioner model. The blade tensioner was modeled using a number of divided mass points and beam elements joining them to each other. These mass points have three degrees of freedom, as in the case of the chain. Each mass point was divided into five parts. This was decided upon taking into consideration the smoothness of the contact with the chain, the number of resonance modes that can be reproduced, the installation of the pivot and the slide part, and so on. Adjacent mass points were connected to each other by beam-type leaf springs, which can take account of bending deformation, and damping elements.

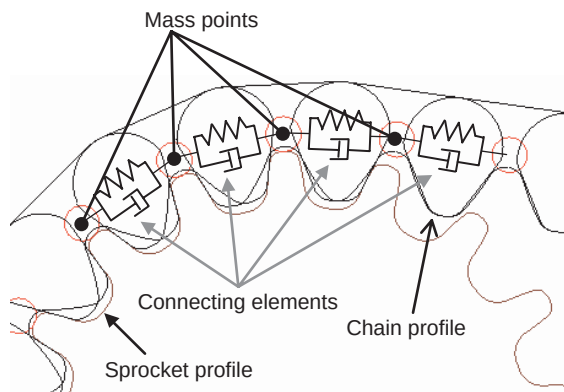


Fig. 4 Ling-by-link model

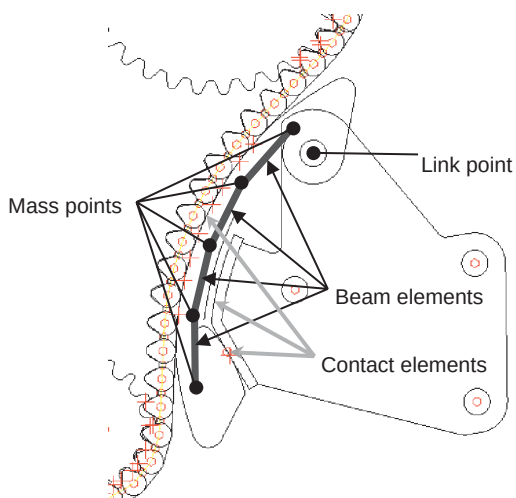


Fig. 5 Blade tensioner model

Connecting points were provided at the top ends of the connected mass points, and were retained in such a way that they could rotate. At the bottom ends, the contact elements where the blade tensioner slides along the holder were defined. As a result, the simulation enables the deformation of the blade tensioner to be reproduced in the same way as that of an actual blade tensioner. In addition, at each mass point, a profile that guides the chain was defined, and could apply tension to the chain. Also, preload was set by changing the curvature when a bending load is not applied to the beam. To increase the preload as in the case of an actual blade tensioner, the curvature is reduced. Conversely, to reduce the preload, the curvature is increased. The stiffness of the blade tensioner set in this way, as well as the appropriateness of the preload, were validated by comparing the results of the simulation with the quasi-static load test results. Figure 6 shows an outline of the test method. The center part of the tensioner constrained by the jig was pushed by the plunger, and the load and displacement at this time were measured. Also, the simulation model was calculated under similar conditions. Figure 7 shows a comparison between the measured and simulated results. As indicated by these results, it can be seen that the blade tensioner has high hysteresis characteristics. These characteristics were reproduced by adding a friction element to each sliding part. It can be seen that the hysteresis and load characteristics exhibited by the blade tensioner, which

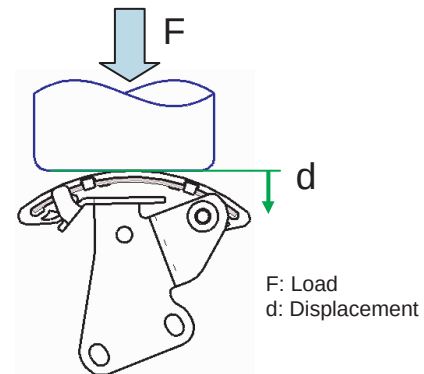


Fig. 6 Static load test of blade tensioner

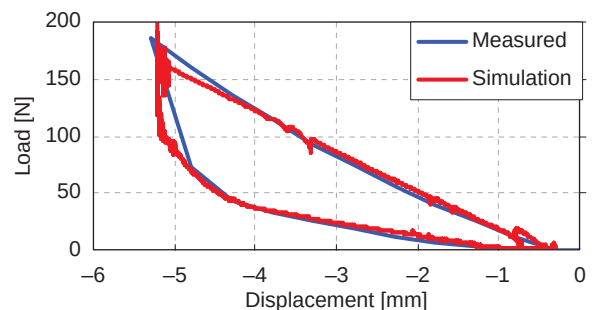


Fig. 7 Static stiffness of blade tensioner

bends when rotated about the pivot, are reproduced with adequate accuracy.

3.3. Hydraulic Tensioner

Figure 8 shows the hydraulic tensioner. The hydraulic tensioner is installed on the end of the movable arm and regulates the motion of the chain. This tensioner is substituted by a coupled vibration model of the plunger, spring, high pressure chamber oil, and tensioner body. The main spring and the high pressure chamber model are installed between the plunger and the tensioner body, and the nonlinear bulk modulus of the oil that is due to the inclusion of air is also taken into account. Also, the oil suction check valve is incorporated in the high pressure chamber model. The discharge of oil is added to the amount that leaked from the clearance between the piston and the housing, and the discharge from the relief valve is also modeled. The relief valve is contained in the plunger. The plunger operates according to the load balance indicated in Eq. (1), and the relief valve according to the load balance indicated in Eq. (2).

$$F_p = M\ddot{X} + C\dot{X} + KX + PS + F_{pr} + c\dot{x} + kx + f_{pr} \quad (1)$$

$$Ps = m\ddot{x} + c\dot{x} + kx \quad (2)$$

Where,

- F_p : Plunger load
- X : Plunger displacement
- M : Plunger weight
- C : Viscous damping between the plunger and the body
- K : Main spring stiffness
- P : High pressure chamber pressure
- S : Plunger pressure receiving area
- F_{pr} : Main spring preload
- x : Relief valve displacement

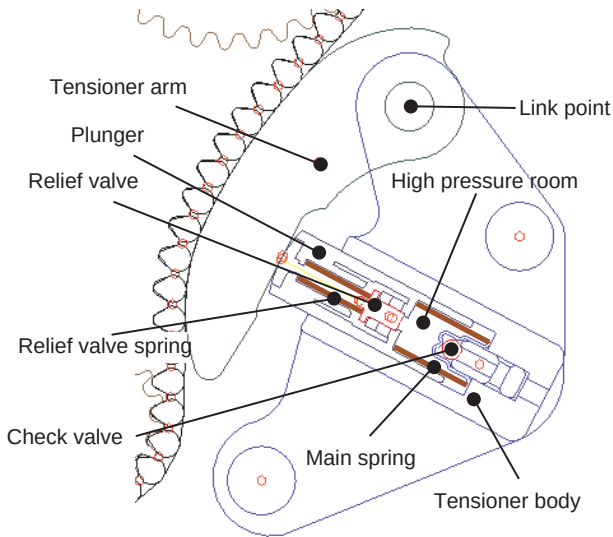


Fig. 8 Hydraulic tensioner model

- c : Viscous damping between the relief valve and the plunger
- k : Relief valve spring stiffness
- f_{pr} : Relief valve spring preload
- s : Relief valve pressure receiving area
- m : Relief valve weight

The pressure loss when the relief valve is operating can be approximated using Eq. (3).

$$\Delta p = \frac{\xi \rho}{2} \left(\frac{Q}{\pi d l} \right)^2 \quad (3)$$

Where,

- Δp : Pressure loss
- ξ : Loss coefficient
- ρ : Density
- Q : Flow rate
- d : Valve seat diameter
- l : Valve lift

The model created in the above way was subjected to a unit excitation test using a hydraulic exciter, and the accuracy was validated⁽¹⁾.

4. Chain System Resonance Frequency Characteristics

In the accuracy validation of the simulation model, the resonance frequency of the chain system that governs the dynamic characteristics of the chain was confirmed. The resonance frequency was identified by performing frequency analysis of the fluctuation of angular velocity when a step torque is applied to the oil pump sprocket. Figure 9 shows the results of identification.

The resonance frequency of the hydraulic tensioner system is 186 Hz. The value⁽⁶⁾ calculated from the chain stiffness and the inertia of the balancer and the pump agrees with this resonance value, and it can be seen that the link-by-link method reproduces the resonance frequency characteristics of the chain system. In contrast to this, the resonance frequency of the blade tensioner system is only 133 Hz. One possible reason for this is that the stiffness of the blade tensioner is lower than that of

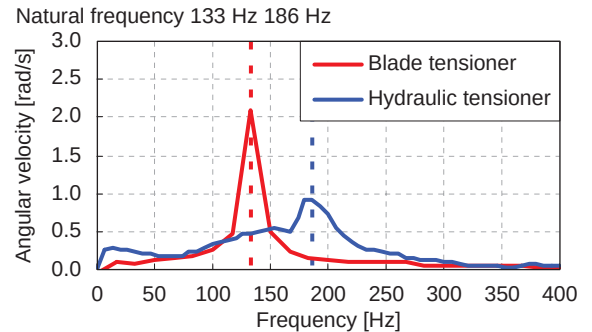


Fig. 9 Step response of chain system

the hydraulic tensioner. In the case of the hydraulic tensioner, the tensioner stiffness is higher than the chain stiffness. Consequently, the slack side chain stiffens, and together with the tension side chain, it governs the eigenfrequency. In the case of the blade tensioner, on the other hand, the tensioner stiffness is lower than the chain stiffness. For this reason, regarding the slack side chain, the chain load is lower and the chain cannot stiffen, and the eigenfrequency, which is determined by the chain stiffness on the tension side, and the inertia of the balancer, as well as the oil pump, govern the behavior of the chain system⁽⁶⁾.

In this way, if the blade tensioner is used, the apparent chain stiffness becomes 1/2, and the resonance frequency becomes $\sqrt{1/2} = 0.7$ times that of the hydraulic tensioner.

5. Validation of Dynamic Characteristics due to the Chain Load

5.1. Centrifugal Force

In order to reproduce the chain load under operating conditions, a model in which the crankshaft, chain, and balancer shaft were linked was constructed. Figure 10 shows the lump mass simulation model. The chain is wound between the crankshaft and the balancer shaft to transmit motive power. The two balancer shafts are coupled by a gear element that takes into account the gear reaction and backlash. The piston connecting rod is coupled to the crankshaft model, and is driven by cylinder pressure in the same way as in an actual chain. Also, the crankshaft model was that of a torsional model, which took into account rotation alone. Because the fluctuation of angular velocity of the crankshaft governs the chain load fluctuation, it was judged that there was no issue with accuracy, and hence the bending deformation was omitted for the purpose of simplification.

Compared to the timing chain, the sprocket pitch circuit radius of the oil pump chain on the crankshaft is

large, and the peripheral velocity is high. The fluctuation of angular velocity of the oil pump chain is small at the high rotation side. In contrast, the centrifugal force of the chain is large, and it is necessary to check the effect of centrifugal force when predicting the load. Accordingly, a validation was performed concerning whether or not a link-by-link model could reproduce the effect of centrifugal force. Figure 11 shows the results of a simulation of the chain load when only the rotational speed was raised, without applying fluctuation of angular velocity. The chain load increases parabolically according to the increase of the rotational speed. The chain load at 7000 rpm increased by 200 N compared to the chain load at 1000 rpm. This was compared with the results obtained using the theoretical equation. The increase in the chain load due to centrifugal force is indicated according to the following Eq. (4)⁽⁷⁾⁻⁽⁹⁾.

$$F_f = mv^2 \quad (4)$$

Where,

F_f : Chain load due to centrifugal force

m : Chain weight per unit length

v : Chain speed

According to this theoretical equation, the chain load at 1000 rpm is 4 N, and at 7000 rpm is 210 N. In the simulation model, preload and oil pump torque are applied to the blade tensioner. Consequently, the static chain load is 300 N on the tension side. As a result, the tension side chain load is 304 N at 1000 rpm, and 510 N at 7000 rpm, which agree with the results of simulation. It thus became clear that the chain load, which takes into account the centrifugal force in the high speed region by using the link-by-link model, could accurately be reproduced.

5.2. Chain Load when a Blade Tensioner is Used

In addition, the prediction accuracy of the chain load generated in the oil pump chain was validated.

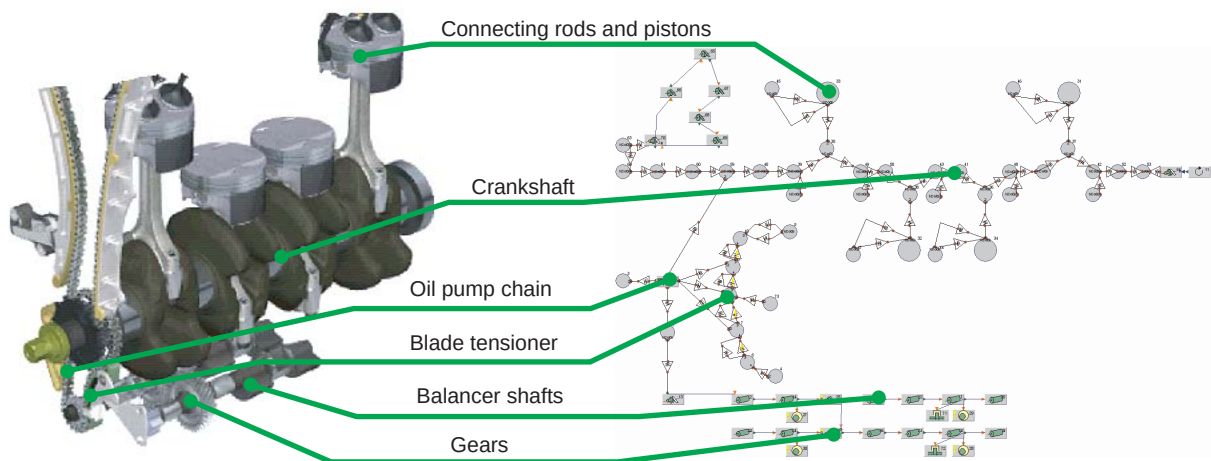


Fig. 10 Whole simulation model

Measurement of the chain load was performed by forming a thin part on the crankshaft sprocket and installing a strain gauge on that section. The torque measured here was multiplied by the effective radius of the sprocket and converted into a chain load.

Figure 12 shows a comparison of the chain load obtained by simulation and by measurement. Measurement was performed using a new chain. The chain load increased in the vicinity of 2000 rpm and then fell again at higher speeds. When the engine speed rose past 4000 rpm, the chain load increased again. The simulation also showed the same tendency.

Next, validation was performed with the blade tensioner exceeding the load retaining limit. If the blade tensioner has exceeded the load retaining limit and is

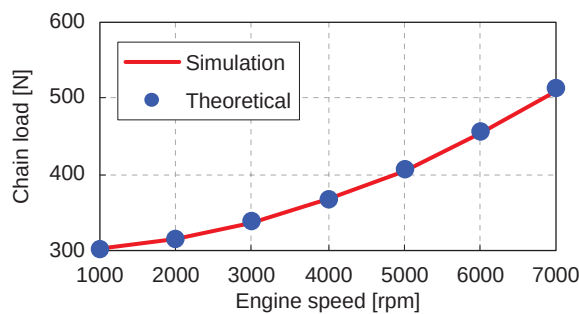


Fig. 11 Influence of rotational speed for chain load

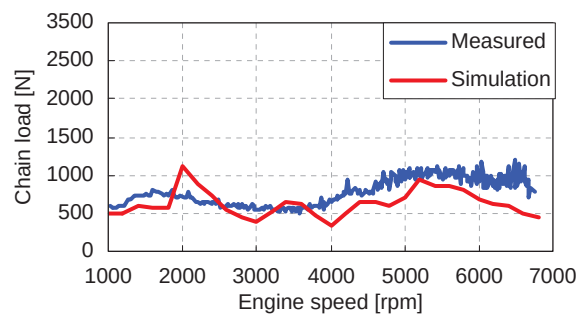


Fig. 12 Maximum chain load of normal chain with blade tensioner

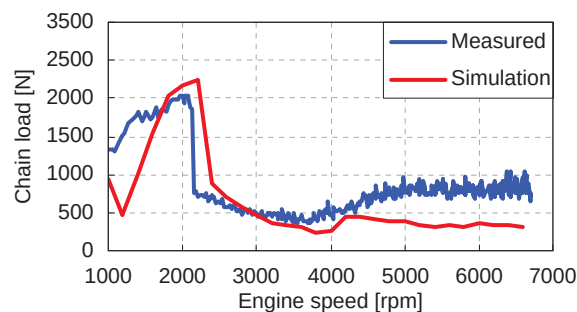


Fig. 13 Maximum chain load of elongated chain with blade tensioner

unable to reduce the transversal vibration of the chain, a high chain load that exceeds the allowable range may occur. Conceivable causes of this include the increase in the fluctuation of angular velocity of the crankshaft, insufficient preload of the tensioner, and the like. The prediction of the limit of occurrence of a high chain load due to transversal vibration is important from the viewpoint of the development of an engine.

A high chain load is likely to occur when the chain becomes worn. When the chain becomes worn and stretches, the geometry of the blade tensioner changes, thus reducing the preload applied to the chain. Reduction of preload causes the transversal vibration of the chain to increase. Accordingly, a chain stretched by 0.7% was used, and a comparative validation using specifications such that a high chain load is generated was carried out. Figure 13 shows the chain load when a 0.7% elongated chain was used.

With a new chain, the chain load was reduced in the vicinity of 1000 N. In the case of a 0.7% elongated chain, a peak chain load in the vicinity of 2000 N was generated at 2000 rpm. In this way, the simulation was able to reproduce the high chain load generated due to the elongation of the chain. Also, the engine speed at which the limit is exceeded and the chain load abruptly increases, the load generated, and the phenomenon whereby the chain load abruptly falls in the vicinity of 2200 rpm were also accurately predicted with this simulation.

Next, a measurement comparison of the chain load generated in a 0.7% elongated chain and the time series waveform of the simulation was performed. This comparison was performed at 2000 rpm, at which a high chain load occurred, and at 2600 rpm, at which the chain load fell. The results are shown in Fig. 14. A negative chain load occurs at 2000 rpm. A negative value expresses the chain load on the slack side. It is believed that on the slack side, the blade tensioner has not absorbed and buffered the chain load.

In addition, the detailed mechanism pertaining to why such a chain load occurs is estimated from the simulation. Figure 15 shows the chain load on the tension side and on the slack side. At 2600 rpm, the chain load on the tension side fluctuates, causing the load to become zero. On the tension side, the condition in which the chain is subjected to a load, and in which it is not subjected to a load, appear alternatively. In contrast, on the slack side, a soft blade tensioner receives the displacement of the chain, and compared to the tension side, the load is low and load fluctuation is constant. The absorption of chain load fluctuations using a soft spring is a feature of the blade tensioner. The fact that the blade tensioner can keep the chain load lower than the hydraulic tensioner can is considered to be an advantage of the blade tensioner. However, if the fluctuation of angular velocity of the crankshaft is large, causing the blade tensioner to strike the stopper, a large chain load will occur. By focusing

attention on the chain load at 2000 rpm, it can be seen that the slack side chain load rises abruptly. As a result of the reaction to this high slack side chain load, a large chain load occurs on the tension side as well. This chain load is generated by an impact between the chain and the guide, or between the chain, tensioner, and stopper.

Once an impact occurs, it occurs repeatedly and continuously using only a small amount of energy, resulting in a high chain load. This is shown in Fig. 16.

In addition, regarding the apparent chain stiffness, when the chain strikes the stopper, a symmetrical increasing stiffness of the spring abruptly occurs. This results in a jumping phenomenon of the resonance point at the high frequency side. The jumping phenomenon actually occurs on a stretched chain in the vicinity of 2200 rpm. As shown in Fig. 13, the simulation also accurately reproduces this jumping phenomenon.

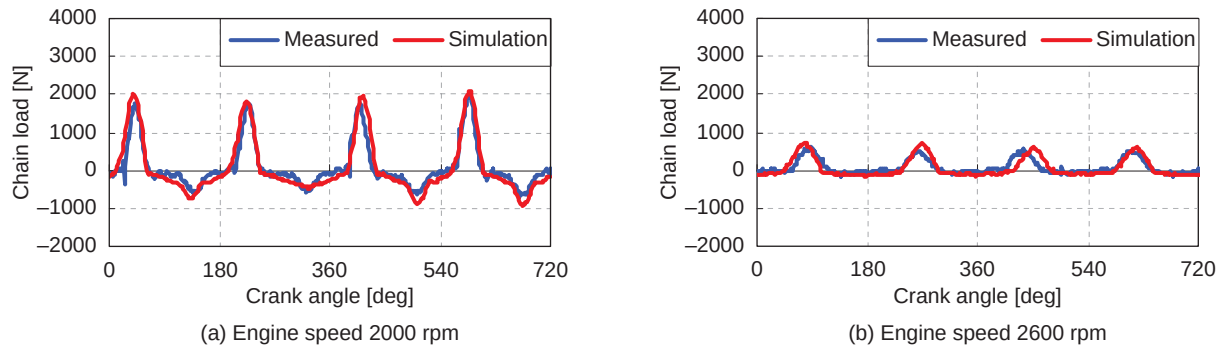


Fig. 14 Chain load of elongated chain with blade tensioner

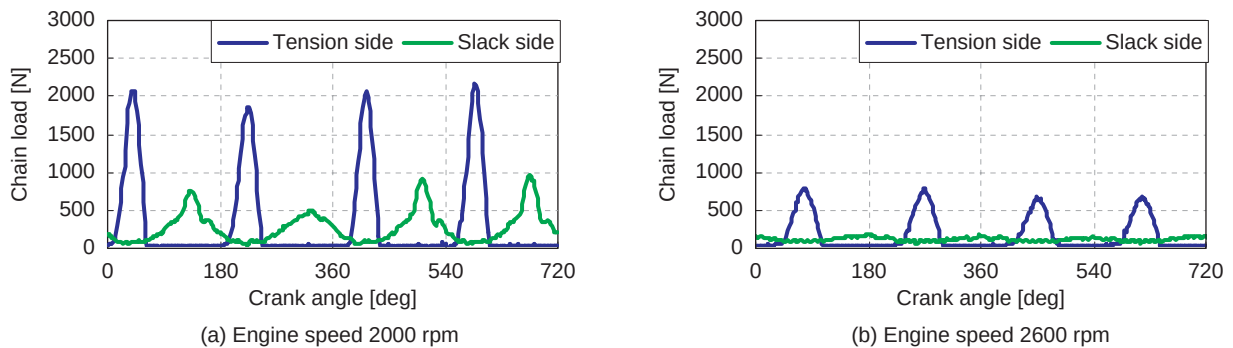


Fig. 15 Simulated chain tension with elongated chain and blade tensioner

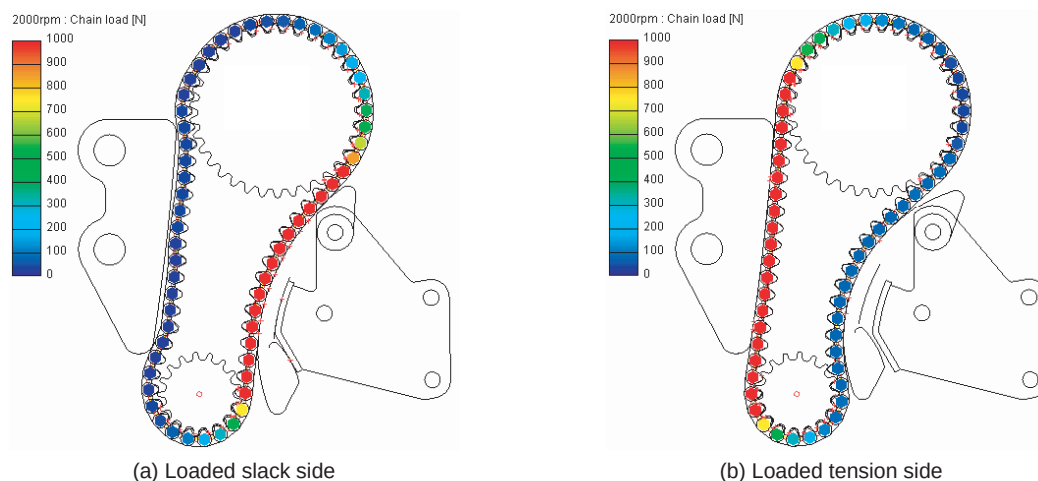


Fig. 16 Simulated chain behavior at 2000 rpm with elongated chain and blade tensioner

5.3. Chain Load when a Hydraulic Tensioner is Used

Conceivable methods of reducing the occurrence of a high chain load include increasing the spring load of the blade tensioner, reducing the fluctuation of angular velocity, and changing over from a blade tensioner to a hydraulic tensioner. Of these countermeasures, it is easiest to increase the spring load. However, there are space limitations, and the friction loss of the chain and guide shoe increases, so it is not always possible to increase the spring load as desired. The fluctuation in angular velocity can be reduced by a method in which vibration is isolated by a long torsion spring damper. It can also be canceled by using an elliptical sprocket. Also, changing over to a hydraulic tensioner with a strong damping force is an effective method of reducing transversal vibration.

Accordingly, Fig. 17 shows the maximum chain load for the case where a blade tensioner is replaced by a hydraulic tensioner. A hydraulic tensioner has greater stiffness than a blade tensioner. It can regulate the lateral vibration of the chain, and reduce the excessive chain load at 2000 rpm. It was confirmed by simulation as well that the chain load could be kept low. At the same time, the

maximum chain load at 5000 rpm could also be reproduced accurately, and the maximum chain load as well as the rotational speed at which it occurred could be predicted accurately. From the time series waveform comparison at 2000 rpm of Fig. 18 as well, it can be seen that the simulation can accurately reproduce the chain load.

Figure 19 shows the results of predicting the chain behavior. It can be confirmed that the chain slackness that had been seen in the blade tensioner has disappeared, and that the hydraulic tensioner keeps the chain load low.

In the above way, a simulation method that enabled the behavior of the chain and the load to be reproduced, even when using different tensioners, was established, and it became clear that this simulation method could be developed as a prediction tool at the layout stage.

6. Conclusion

Simulation technology for an oil pump chain was established using a link-by-link model, and the following information was acquired.

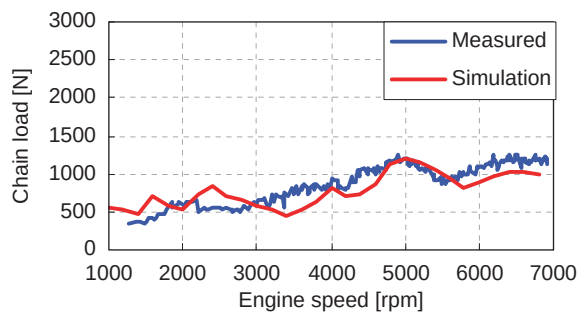


Fig. 17 Maximum chain load of elongated chain with hydraulic tensioner

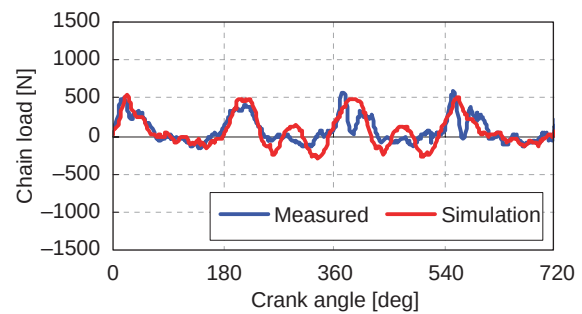
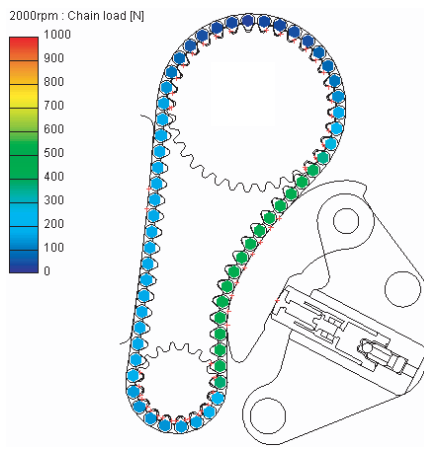
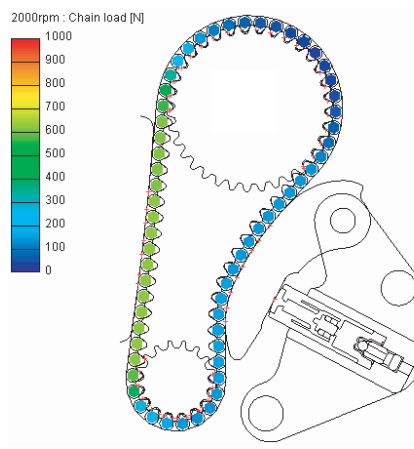


Fig. 18 Chain load of elongated chain with hydraulic tensioner at 2000 rpm



(a) Loaded slack side



(b) Loaded tension side

Fig. 19 Simulated chain behavior at 2000 rpm with elongated chain and hydraulic tensioner

- (1) By using the link-by-link chain model, the chain engagement, seating behavior, centrifugal force, and inertia force could all be reproduced, and the transversal vibration predicted.
- (2) The resonance frequency of the blade tensioner system is lower than that of the hydraulic tensioner. Also, the chain load sometimes increases abruptly, along with the increase of the fluctuation of angular velocity of the crankshaft. This indicates that the tension-retaining performance of the blade tensioner has reached its limit. Prediction of this limit is an important design guideline.
- (3) If the blade tensioner exceeds its tension-retaining limit, it is necessary to replace the tensioner with a hydraulic tensioner. By doing this, the transversal vibration of the chain will be limited, and the chain load could also be kept low.

A blade tensioner model consisting of a number of beam elements and a hydraulic tensioner model were constructed as described above, and technology for predicting the maximum chain load at the layout stage, and the rotational speed at which it occurs, was established.

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