

Simulation of Accessory Drive Belt Behavior Using Multi-body Dynamics

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ABSTRACT

A simulation method was proposed to predict the slippage and transversal vibration of the accessory drive belt. To reproduce these phenomena, the accessory drive belt was represented by a section-by-section model in which the belt was replaced by a finite number of masses and springs using multi-body dynamics simulation. In this model, the belt was able to vibrate in both the direction of advance and the direction perpendicular to it, and a friction contact element was defined between the pulley and the belt. The coefficient of friction was made variable with respect to the slippage speed to enable stick and slippage to be predicted. This method allows for accurate simulation of the amount of belt slippage and the amplitude of transversal vibration, thus enabling the optimum belt layout to be determined at the design stage.

1. Introduction

Recently, the method of driving multiple accessories (pulleys) by passing a single V-ribbed belt around each pulley in a serpentine form has been gaining wider use. This arrangement enables the width of the engine to be reduced, thus contributing to compactness. On the other hand, the load applied to the belt is higher than with the conventional arrangement, and slippage and transversal vibration are likely to occur, producing durability, noise and vibration issues. For this reason, until now, much research has been carried out on methods that predict these issues using complex theoretical equations and simulations^{(1),(2)}. Generally, the dominant factor involved in belt slippage and transversal vibration is the phenomenon of fluctuating belt tension. A belt system has resonance that is determined by rigidity in the longitudinal direction of the belt and by the inertia of each accessory, causing rotational vibration

in the pulleys. This rotational vibration will hereinafter be called longitudinal vibration. There are many cases in which, when the V-ribbed belt of this so-called “serpentine method” is used, the longitudinal vibration matches the fluctuation of the crankshaft’s angular velocity, resulting in large tension fluctuations in normally used areas. For this reason, the one-dimensional longitudinal vibration model has been the main modeling method used in simulations until now. The slippage and transversal vibration between the belt and the pulley has also been estimated from the tension fluctuation obtained using this longitudinal vibration^{(3),(4)}. However, though this method has the merit of reducing computation time, it does not directly predict the extent of slippage or transversal vibration. To solve these issues, simulation technology using a section-by-section model in which the belt is replaced by a finite number of springs was created, as reported here.

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2. Belt Model

The simulation model consists of the accessory drive belt and the crankshaft that drives it.

The belt is divided into a finite number of mass points that are coupled together by beam elements. Each mass point can vibrate not only in the belt's direction of advance, but also perpendicular to the direction of advance. Thus, the belt's transversal vibration, which could not be predicted using a one-dimensional longitudinal vibration model, can now be predicted. This model is called a section-by-section model⁽⁵⁾. Figure 1 shows a pattern diagram of this model.

Figure 2 shows the coupling element of the belt model. Because the filament is more rigid than the rubber, the filament transmits the load. Consequently, in the simulation, adjacent mass points are coupled to each other at the filament's position. The beam elements are used for coupling. As a result, the filament is the neutral surface of the bending deformation, and it is possible to reproduce behavior similar to that of an actual belt. A value identified from the belt-induced vibration test is used as the bending rigidity. In addition, the belt is vulcanized into a cylindrical shape at the time of manufacture. Consequently, because the belt is initially cylindrical, the simulation is also set so that the belt is cylindrical when it is not wrapped around the unloaded pulley.

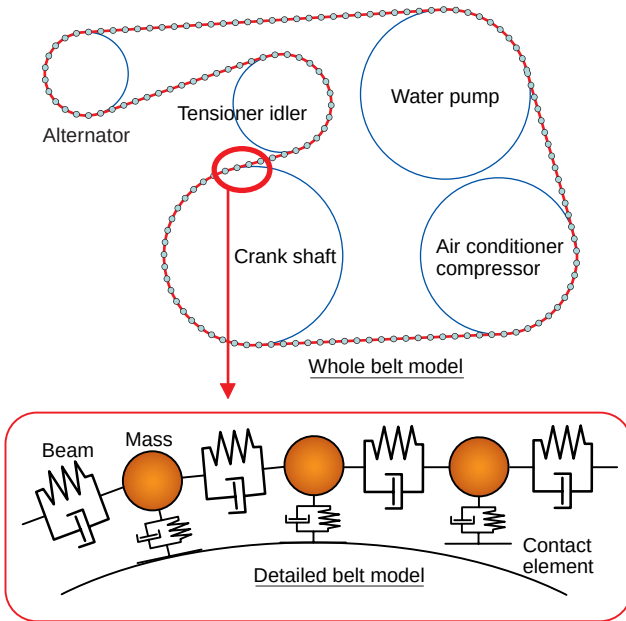


Fig. 1 Section-by-section model of belt

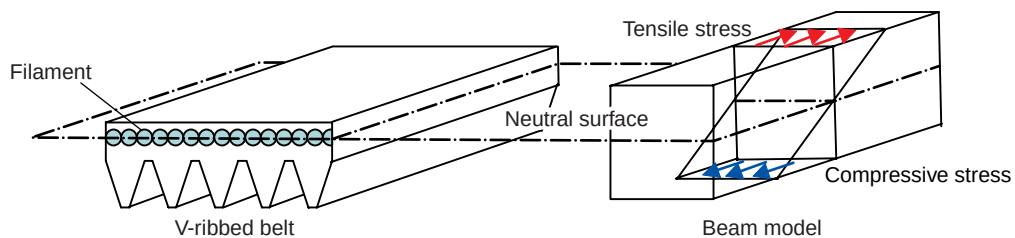


Fig. 2 Neutral surface of V-ribbed belt

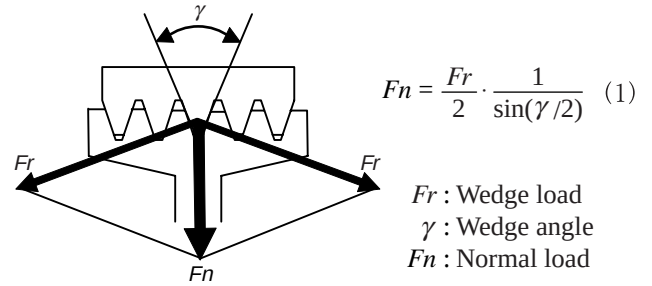


Fig. 3 Wedge load between belt and pulley

Compared to a flat belt, a V-ribbed belt generates high friction force due to the wedge effect when it engages with the pulley as shown in Fig. 3. Therefore, a friction force expressed by equation⁽¹⁾, which takes into account the wedge effect, is also assigned to the simulation. In addition, the coefficient of friction is assumed to be speed-dependent, increasing at low speeds to enable the stick-slip between the belt and the pulley to be reproduced. The coefficient of friction is computed from three parameters using interpolation Eq. (2)⁽⁶⁾. Figure 4 shows an example of a computation equation and the coefficient of friction.

$$\mu = A \cdot \sin \left\{ 2 - \frac{2}{\pi} \cdot \sin^{-1} \left(\frac{B}{100} \right) \right\} \cdot \tan^{-1} \left\{ S \cdot \frac{100}{C} \cdot \left(\tan \frac{\pi^2}{4 \cdot \left(\pi - \sin^{-1} \left(\frac{B}{100} \right) \right)} \right) \right\} \quad (2)$$

μ : Friction coefficient

S : Slippage

$A = \mu_{\max}$

$B = \mu_{100}$

$C = S_{\max}$

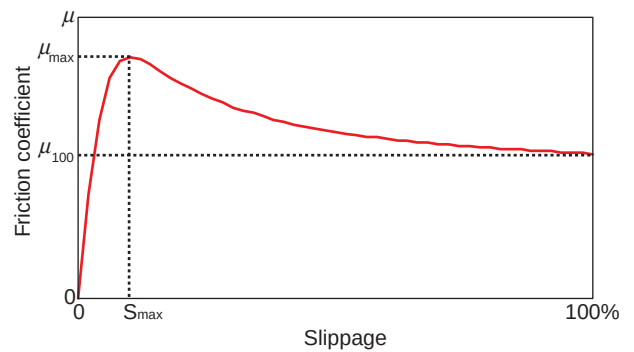


Fig. 4 Friction coefficient

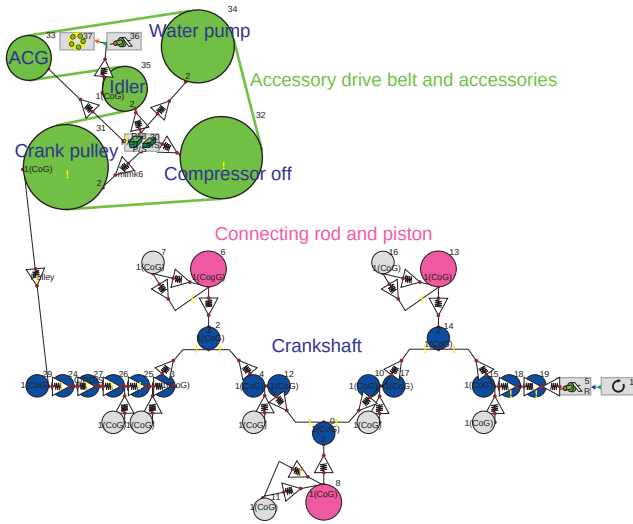


Fig. 5 Simulation model

Thus, the belt in the model is installed on the crankshaft pulley and drives multiple accessory pulleys. Each pulley in the model takes into account the inertia equivalent to each accessory. The load generated by each accessory is defined for each speed. The inertia and load of the pulley that drives the air conditioner compressor vary according to whether the compressor is operating or not. These parameters are also taken into consideration when determining the analysis conditions. To reproduce the crankshaft pulley's fluctuation in angular velocity, a model of the crankshaft that is driven by the cylinder pressure of the engine is created and used. Because fluctuation in the belt tension is caused mainly by fluctuation in the angular velocity of the crankshaft, the crankshaft model is a torsional model that takes into account only rotation. Figure 5 shows the simulation model.

3. Outline of the Test Equipment

The engine used to verify the simulation was a 660 cm³ inline 3-cylinder turbocharged gasoline engine. Figure 6 shows the layout of the installed accessory drive belt. A five-rib V-ribbed belt with a PET filament was used. The driven accessories were the alternator, the water pump and the air conditioner compressor. The belt tension

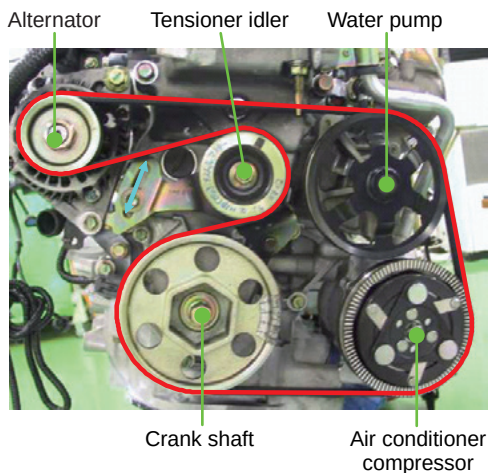


Fig. 6 Accessory drive layout

Table 1 General specifications of test

Engine type	Liquid cooled 3-cylinder turbo charged engine
Flywheel	Single mass flywheel
Crank pulley	Rigid pulley
Displacement	660 cm ³
Accessory drive belt	Serpentine V-ribbed belt
Filament of belt	Polyethylene terephthalate (PET)
Number of ribs	5 ribs
Material of rubber	Alkyl chloro sulfonated polyethylene rubber (A-CSM)
Tensioner	Fixed tensioner

was adjusted by changing the fixing position of the idler. This idler is hereafter called a fixed tensioner. The detailed test specifications are shown in Table 1.

While verifying belt slippage, it was necessary to measure the belt and crankshaft pulley velocities accurately and without making contact. Accordingly, these parameters were measured using a laser Doppler method wherein a laser beam is directed at the object of measurement and the velocity is calculated from the difference in wavelength between the incident and reflected waves. Figure 7 shows the measurement apparatus.

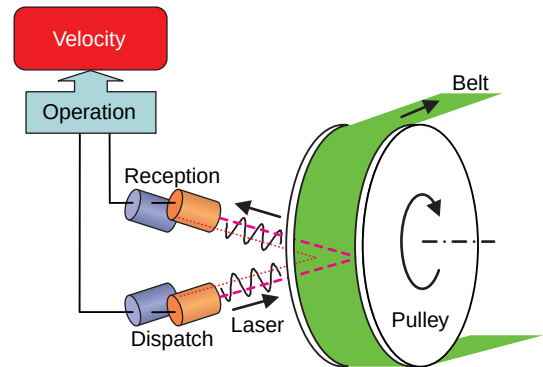


Fig. 7 Measurement of slippage

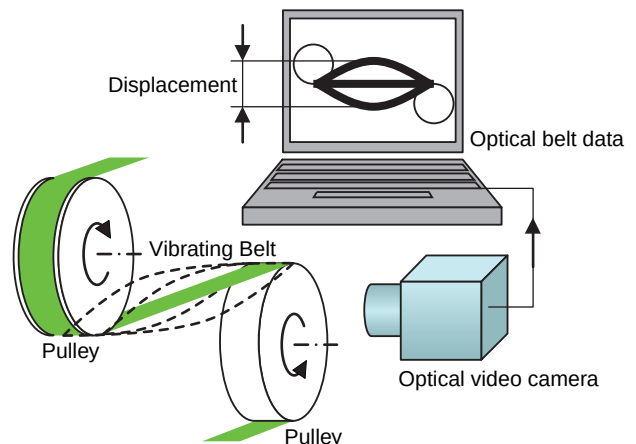


Fig. 8 Measurement of transverse vibration

It was also necessary to measure the displacement of the belt accurately and without making contact during the verification of transversal vibration, so the belt motion was measured using an optical displacement gauge. In this method, a photographed image of the belt was analyzed, and the displacement computed. Figure 8 shows an outline of the measurement apparatus.

4. Results of Verification

In the verification of the simulation, attention was focused on tension fluctuation, the slippage and the transversal vibration of the belt. The values of these items were difficult to predict in the case of a longitudinal vibration model because of their effect on the durability of the belt and because of the noise that was generated. Accordingly, verification was carried out according to the following three methods.

4.1. Belt Tension Fluctuation

Tension fluctuation is thought to be caused by the fluctuating angular velocity of the crank pulley and the resonance of the belt. In the case of the 3-cylinder engine used for the verification, the 1.5th order component was the main component of the fluctuating angular velocity of the crankshaft at low speeds where transversal vibration is liable to occur. This is because the explosion force is larger than the inertial force. Figure 9 shows the results of time series data verification of the fluctuation of angular velocity caused by the crankshaft's torsional vibration, and Fig. 10 shows the results of the analysis of the frequency of this vibration. In addition to this 1.5th order component, a 10.5th order angular velocity fluctuation component due to the 575 Hz torsional intrinsic frequency of the crankshaft was also reproduced. Because this high frequency component has a large effect on belt slippage, the prediction

accuracy must be high. The simulation measures the angular rotational fluctuation, including that in high frequency components, with high accuracy.

Next, attention was focused on the resonance of the belt. The belt resonance consists of longitudinal vibration, which is belt vibration in the direction of advance, and transversal vibration, which is perpendicular to the longitudinal vibration. First, to investigate the belt system's longitudinal vibration resonance, angular velocity fluctuations were applied in steps to the pulley of the crankshaft, and the response was investigated. The stepped angular velocity fluctuations generate impulse angular acceleration. Consequently, the same frequency response results as those of the impulse response were obtained. Figure 11 shows the results of frequency analysis of the angular velocity of the alternator pulley acquired due to acceleration. It can be seen that, of the several longitudinal vibration resonance modes possessed by this belt system, the resonant frequency showing the greatest amplitude is in the vicinity of 75 Hz. This is the primary mode in which shaking mainly by the alternator occurs due to the large equivalent inertia of the alternator. Belt tension fluctuation reaches a peak at a frequency of $75 \times 60 / 1.5 = 3\,000$ rpm, where the resonance coincides with the 1.5th order angular velocity of the crankshaft. Figure 12 shows the maximum value of the fluctuation of belt tension between the alternator and the fixed tensioner versus the rotational speed of the engine. This graph shows that the peak in the tension fluctuation occurs between 3 000 and 3 300 rpm for both the measured and simulated values, thus indicating that the simulation accurately reproduced the actual tension fluctuation. The causes of the peak at 3300 rpm are the closeness of the longitudinal vibration resonance to the primary component of the fluctuating angular velocity of the crankshaft and the closeness of the transversal resonance of 180 Hz between the alternator and the fixed tensioner to the 3rd order component of the fluctuation of crankshaft's angular velocity.

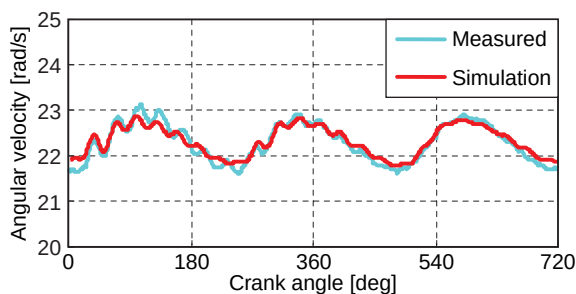


Fig. 9 Measured and simulated crankshaft speed fluctuation

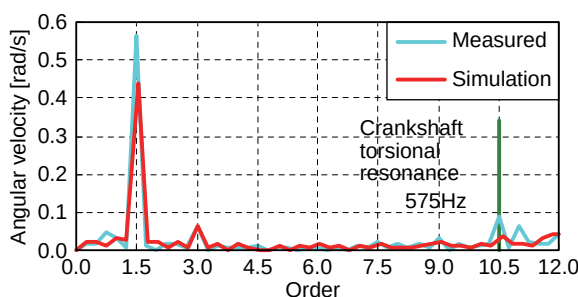


Fig.10 FFT result of crank shaft rotational speed

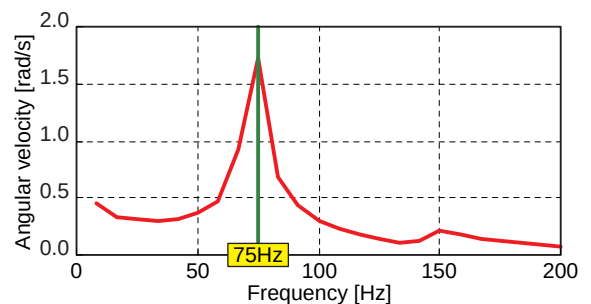


Fig. 11 Step response of angular velocity on alternator

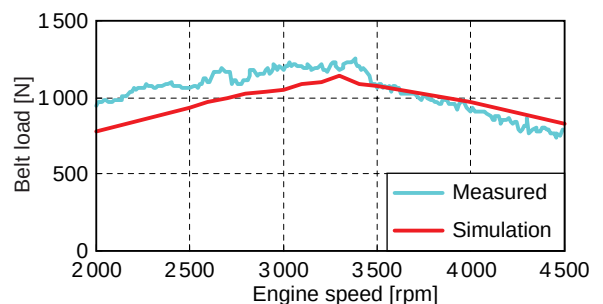


Fig. 12 Maximum value of belt load

4.2. Belt Slippage

Slippage between the belt and the pulley is a factor that has a large effect on the durability of the belt, generating unusual noise.

Excessively large slippage causes belt wear. If the static friction between the belt and the pulley is higher than the dynamic friction, stick-slip of the belt and also unusual noise are liable to occur. Accordingly, attention was focused on the belt slippage and a verification was performed.

The verification was carried out for a belt in which stick-slip occurred and unusual noise was emitted and for a normal belt in which stick-slip did not occur. The comparison was performed at 2 800 rpm, where fluctuations in tension and in angular velocity are large and stick-slip is likely to occur. The results are shown in Fig. 13. The figures on the left are the results for a belt in which stick-slip occurred, and the figures on the right are for a belt in which stick-slip did not occur. In the case of the belt in which stick-slip occurred, an instantaneous deceleration in the belt's angular velocity occurred in the vicinity of maximum speed fluctuation, causing the belt to slip. In the simulation as well, the belt slippage was reproduced at the same timing as that of the test results, and thus, it can be seen that the torsion vibration characteristics of the crankshaft affect stick-slip.

In the case of the belt in which stick-slip did not occur, the small amount of slippage in the vicinity of the point where the fluctuation of rotational speed was at its maximum and the smooth-slipped condition, were accurately reproduced. The coefficient of friction at this time is shown in Fig. 14.

It was possible to reproduce the actual phenomenon in a belt in which stick-slip occurred by applying a high coefficient of friction to the belt at a lower speed compared to that of a belt in which stick-slip does not occur. The coefficient of friction used here was a much higher value than that obtained from the results of general measurement. The conceivable cause of stick-slip lies in the setting

of the friction coefficient. From a computation standpoint, when the sign of the relative speed reverses, it is necessary to reverse the sign of the friction value as well, and when the speed is zero, the coefficient of friction is also zero. For this reason, it is necessary to apply a high friction coefficient at lower speeds to cause sticking to occur when the belt starts moving. On the other hand, in general measurement of the friction coefficient, the fluctuation in speed is less than that of actual stick-slip, and it is conceivable that the change in the friction coefficient at high frequencies is unknown. Consequently, it is considered appropriate when predicting stick-slip to use a friction coefficient that has been inferred so that the simulation agrees with the actual behavior.

4.3. Transversal Vibration

One conceivable cause of belt noise is transversal vibration of the belt. Accordingly, attention was focused on the part of the belt between the alternator and the fixed tensioner, where transversal vibration is liable to occur, and a verification of transversal vibration was carried out.

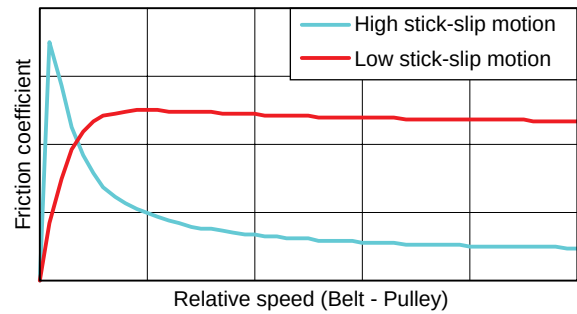


Fig. 14 Slip characteristics for investigated belt designs

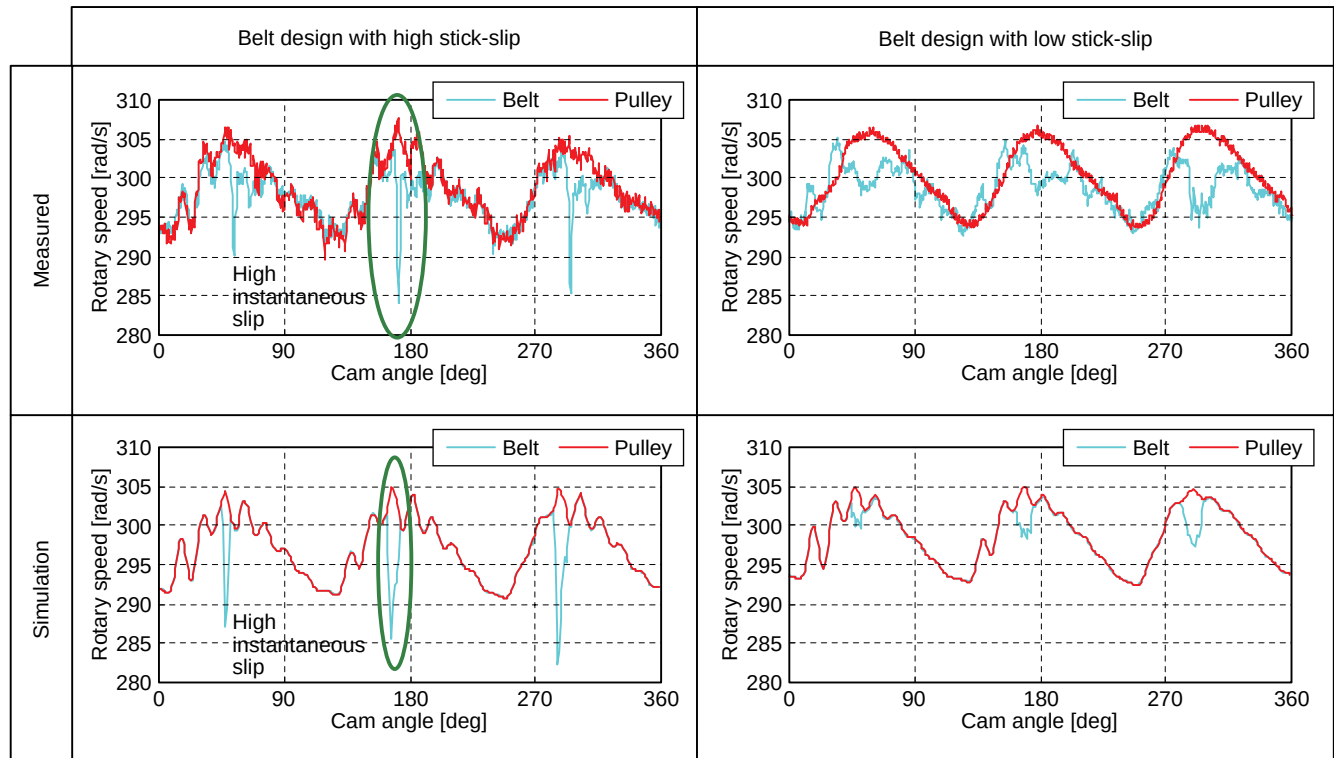


Fig. 13 Stick-slip motion at crankshaft pulley for two different belt designs

It is thought that transversal belt vibration is caused by tension fluctuation and transversal resonance. When the angular velocity fluctuation period of the crankshaft coincides with the belt system's longitudinal resonance, the tension fluctuation increases, and when the duration of this tension fluctuation coincides with that of the transversal vibration, transversal vibration is generated. Here, if the tension fluctuation is large with respect to preloading, the unstable region of the transversal vibration will increase, resulting in the generation of transversal vibration in the belt over a longer period of time. Other conceivable causes of increased transversal vibration include eccentricity of the pulley, friction force when the belt disengages from the pulley, belt inertial force, spring-back due to the belt's bend-rigidity, and engine vibration. Of these, belt inertial force and spring-back due to the belt's bend-rigidity are taken into account in the section-by-section model. Also, in the development of the simulation, predicting engine vibration at the initial stage of design was an important goal. For this reason, engine vibration that cannot be predicted unless the cylinder block layout is determined was not taken into consideration. Figure 15 shows computed results that do take the above factor into account.

At 2 600 rpm and 2 900 rpm, the simulation can accurately reproduce the actual transversal belt vibration. In addition, the results of analyzing the transversal belt vibration frequency at 2 900 rpm are shown in Fig. 16. Because the engine used in the verification was a 3-cylinder type, a $1.5n^{\text{th}}$ order angular velocity fluctuation of the crankshaft occurs. As a result, it can be seen that at 2 900 rpm, the $1.5n^{\text{th}}$ components (n is an integer), namely 72.5 Hz (1.5^{th} order), 145 Hz (3^{rd} order), 217.5 Hz (4.5^{th} order), 290 Hz (6^{th} order), and 362.5 Hz (7.5^{th} order) dominate transversal vibration. The transversal resonance of the moving belt is expressed by Eqs. (3) and (4) below⁽⁷⁾. The preloading during measurement was 300 N, so the transversal vibration frequency of the section of belt between the alternator and the fixed tensioner is $f_0 = 215$ Hz.

$$f_0 = \frac{1}{2L} \left(c_0 - \frac{v^2}{c_0} \right) \quad (3)$$

$$C_0 = \sqrt{\frac{T}{\rho}} \quad (4)$$

f_0 : Natural frequency of transverse vibration

L : Span of belt

v : Velocity of belt

c_0 : Transverse wave propagation speed

T : Tension

ρ : Belt mass per length

It is thought that representative tension fluctuation periods, at which transversal resonance occurs, are $2/3 \cdot f_0 = 143$ Hz, $f_0 = 215$ Hz, and $2 \cdot f_0 = 430$ Hz⁽⁷⁾. In fact, peaks can be observed in the vicinity of 143 Hz and 215 Hz. However, marked peaks can be seen at other frequencies as well. It is thought that this is due to the large tension fluctuation. Figure 17 shows tension fluctuation. As shown in the figure, if the fluctuation component is large compared to the mean tension, the region of transversal resonance will increase, and transversal vibration will occur over a wide frequency range. It is thought that this is the reason why the peak of transversal vibration occurred over a wide period between the 1.5^{th} and the 7.5^{th} orders.

It is thus possible to accurately reproduce the amplitude of the transversal vibration using this simulation model, which takes into account the respective excitation forces and resonances.

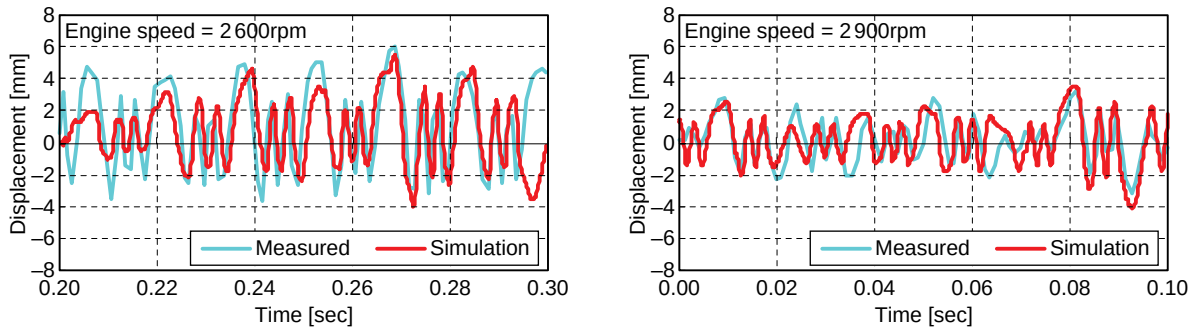


Fig. 15 Transverse belt motion

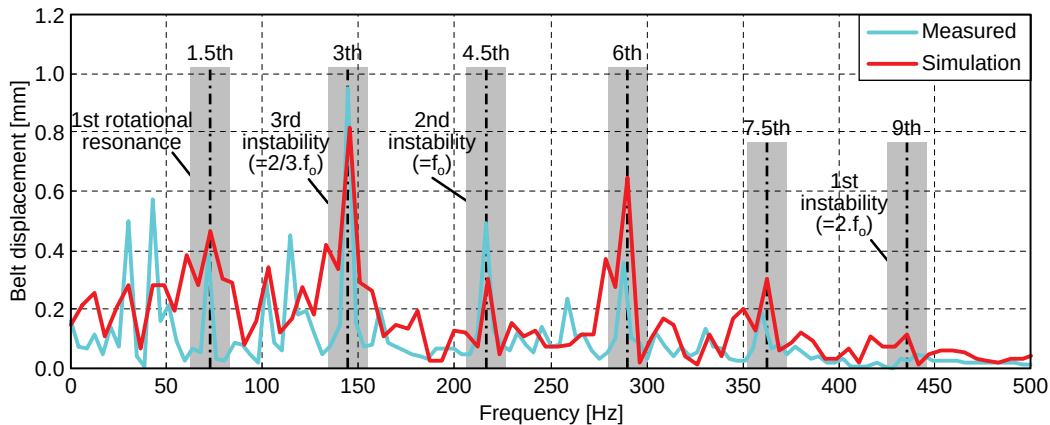


Fig. 16 Frequency contents transverse span vibration at 2 900 rpm (300 N belt pre-load)

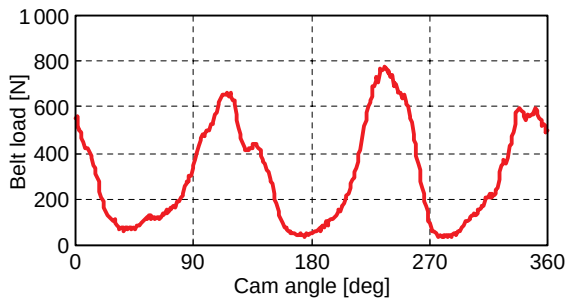


Fig. 17 Belt load between alternator and tensioner

5. Conclusion

The following results were obtained from the simulation using a section-by-section model, and technology for predicting the behavior of the belt was established.

- (1) A V-ribbed belt model, including a crankshaft, was created, enabling accurate reproduction of the fluctuations in angular velocity and belt tension.
- (2) By using a speed-dependent friction coefficient, it was possible to predict a wide range of slippage conditions, including the stick-slip condition.
- (3) By taking into account such factors as the eccentricity of the pulley and the frictional force when the belt starts disengaging from the pulley, it was possible to accurately predict transversal vibration of the V-ribbed belt.

By thus predicting the behavior of the accessory drive belt, it should be possible to establish the optimum belt layout at the design stage.

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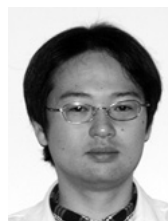
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