

Method of Estimating Instantaneous Safety Factor for Self-optimizing Control of Clamping Force in CVT

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ABSTRACT

When fluctuations in the drive pulley speed are transmitted to the driven pulley via the metal V-belt, the transmitted fluctuations are attenuated according to the belt slippage. The research discussed in this paper focused on these fluctuations in the pulley speeds and developed a method for estimating the instantaneous safety factor for belt clamping force. First, a vibration model was built by regarding the drive and driven pulleys as a one-dimensional vibrating system connected by elastic bodies. Based on this vibration model, the instantaneous safety factor was then estimated from changes in the pulley speeds. In order to verify the method's validity, the actual safety factor was calculated by measuring the driveshaft torque and belt clamping force in a test vehicle and compared with the value estimated using the proposed method. In verification testing, changes in vehicle body weight and left-right wheel differential were applied to the test vehicle. With the assumption that aging may affect the results, verifications were also performed with two test pulleys with different coefficients of friction.

The results verify that, in the required estimation accuracy range of a safety factor of 1.25 or lower, instantaneous safety factor can be estimated to within an error range of 2%. Using the proposed method, belt clamping force can be adapted even for changes in friction coefficient, a traditionally unknown variable.

1. Introduction

In recent years, with the ever-increasing demands for fuel efficiency improvements, it has become more important to increase the power transmission efficiency for transmissions. While automotive transmissions come in many different formats, the continuously variable transmission (CVT), which can optimize the brake specific fuel consumption (BSFC) of the engine, has rapidly spread in recent years⁽¹⁾. Structurally, as mainstream CVTs use metal V-belts, hereinafter "CVT" will refer to the metal V-belt CVT. In a CVT, the transmission adjusts steplessly to the desired speed by changing the winding radius for belts on a pair of pulleys. A breakdown of power loss in a CVT is shown in Fig. 1. The sample breakdown is with the CVT running in JC08, a mode for measuring fuel efficiency. The majority of CVT power loss is attributable to running the oil pump and internal frictional loss in the metal V-belt.

In a CVT, oil pressure is used to generate clamping force with the belts, which in turn creates friction between the metal V-belt and the pulleys to transmit power. An oil pump is required to generate this oil pressure.

The metal V-belt is comprised of many elements held

together by two pairs of laminated rings (Fig. 2). As the elements and rings can both move freely relative to one another, slip will occur between the elements and rings of a metal V-belt while the vehicle is running. This slip is the main cause of frictional loss within the metal V-belt.

Reducing the belt clamping force is an effective means of enhancing the efficiency of a CVT's power transmission. Reduced belt clamping force will reduce the tensile force

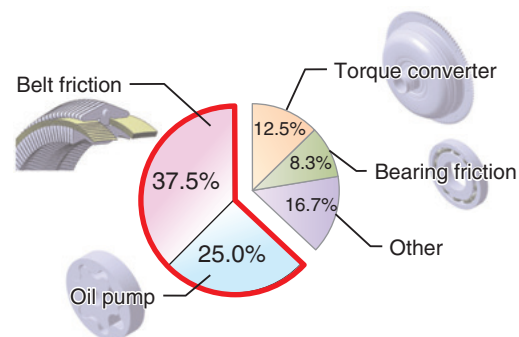


Fig. 1 Breakdown of power consumed in CVT (during operation in JC08 mode)

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acting on the rings, which can attenuate internal frictional loss in the metal V-belt. It can also reduce the amount of work done by the oil pump.

However, it is important to determine the minimum force needed to clamp the belt as excessive reduction will result in greater slippage between belt and pulley. The friction characteristics of metal V-belts are shown in Fig. 3⁽²⁾. While the belt will experience micro slips (elastic slip) due to its own elastic deformation, a region of the belt partially adheres to the pulley [area shown in Fig. 3(A)]. When belt clamping force is insufficient, the belt will slip at all points on the pulley (sliding slip). In this case, there is no adhesion area with the pulley [area shown in Fig. 3(B)]. Sliding slip will actually lower power transmission efficiency and could also damage the belt. Thus, in terms of

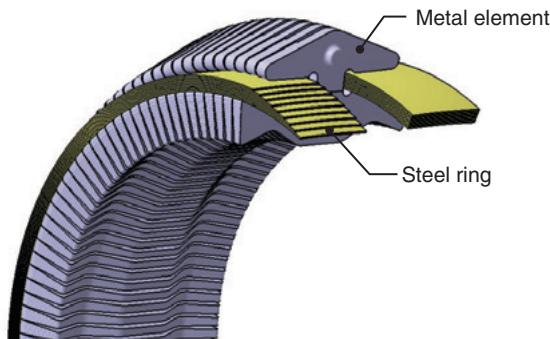


Fig. 2 Structure of metal V-belt

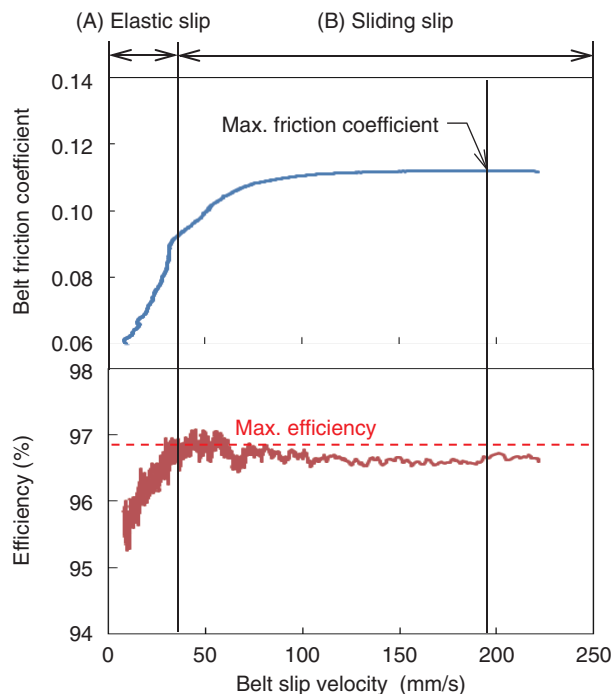


Fig. 3 Characteristics of belt friction

both power transmission efficiency and wear resistance, it is preferable to keep reductions of belt clamping force within the range of elastic slip. This makes the issue to address changes in the coefficient of friction. The friction between metal V-belt and pulley is boundary friction. The coefficient of friction is affected by roughness of the frictional face and the shear strength of the boundary lubrication oil film, which is formed by lubricants⁽³⁾. Also, as these factors change with operating load, changes in the coefficient of friction over time should be factored in. Accordingly, when setting a belt clamping force, a margin is built in, assuming the minimum coefficient of friction of the conceivable usage cases. For a low-mileage CVT, this clamping force margin will be excessive.

Methods have been researched for learning the limits of belt clamping force with a vehicle in motion⁽⁴⁾⁻⁽⁷⁾. In all cases, however, sudden increases in belt slip are used to determine the limits for belt clamping force. While brief, this generates sliding slip, which raises concerns of abrasive wear advancing. Additionally, surging oil pressure and response lag could also result in adhesive belt wear.

Power transmission efficiency is maximized at the elastic slip limit [the boundaries in Fig. 3(A)-(B)]. Given the required tolerances for oil pressure surge and response lag, however, the clamping force margin cannot be completely eliminated in actual controls. Determining the safety factor at the moment the elastic slip limit is reached should allow for this clamping force margin to be minimized. This safety factor, used to reveal the actual clamping force margin, will be called the instantaneous safety factor.

The purpose of this study is to circumvent the sliding slip in estimating this instantaneous safety factor. In a previous study, a method was proposed for predicting the elastic slip limit⁽⁸⁾. This paper explores using predicted elastic slip limit to estimate instantaneous safety factor. Further, the proposed method is also applied on an actual vehicle for validation.

2. Modeling Slip and the Proposed Safety Factor Estimation Formula

2.1. Preconditions and Assumptions

The goal in this study is to increase power transmission efficiency while cruising (in a steady state). The test vehicle is powered by an internal combustion engine, and engine cylinder pressure fluctuation is input to the CVT. The vehicle is assumed to be driving straight ahead with no slip between the tires and road surface. The CVT shift state is assumed to be at a speed ratio of $i \leq 1$ (acceleration side, $i = N_{DR}/N_{DN}$), where i is a steady-state $i = \text{const}$.

Based on these preconditions, the following assumptions were introduced in developing a safety factor formula.

(i) Belt slip occurs on the driven side

Slip from elastic deformation of the belt (elastic slip) is the dominant factor in belt slip⁽²⁾.

Figure 4 represents schematically the change in belt elastic strain energy associated with pulley rotation⁽⁹⁾. When the belt is wound in range $\beta_{DR,DN}$, frictional force acts in range $\alpha_{DR,DN}$. The drive pulley spins the driven pulley by pushing the belt out. Compressive force here generates compression strain in the belt. In terms of energy, input torque from the drive pulley is stored in the belt as elastic strain energy E_1 . Meanwhile, when torque is transmitted, compression strain is released in the driven pulley, dissipating elastic strain energy E_2 . Thus, the CVT system as a whole only loses E_2 of energy per belt revolution. This loss is synonymous with frictional loss between belt and pulley. Assuming a flat distribution of frictional force in the pulley, E_2 is proportional to α_{DN} in $\alpha_{DN} \leq \beta_{DN}$. As torque input is increased, α_{DN} will be enlarged to reach the required frictional force. When $\alpha_{DN} = \beta_{DN}$, sliding slip will occur. From the above, an equivalence conversion is possible, where belt slip on the driven side occurs at the energy-equivalent average slip speed V . This equivalence is assumed as follows:

$$\begin{aligned} V &\propto E_2 \\ &\propto \alpha_{DN}/\beta_{DN} = T_{DN}/T_{max,DN} \end{aligned} \quad (1)$$

where $T_{max,DN}$ is the maximum transmittable torque on the driven side and $T_{DN}/T_{max,DN}$ is the usage rate of torque capacity.

- (ii) The left and right driveshafts can be represented by equivalent rigidity

As the vehicle is driving straight ahead with no slip between the tires and road surface, it is assumed that torsional rigidity k_{2L} , k_{2R} of the left and right driveshafts can be expressed as an equivalent rigidity k_2 in the following equation:

$$k_2 = k_{2L} + k_{2R} \quad (2)$$

- (iii) The wheels are considered fixed as a vibration transfer path

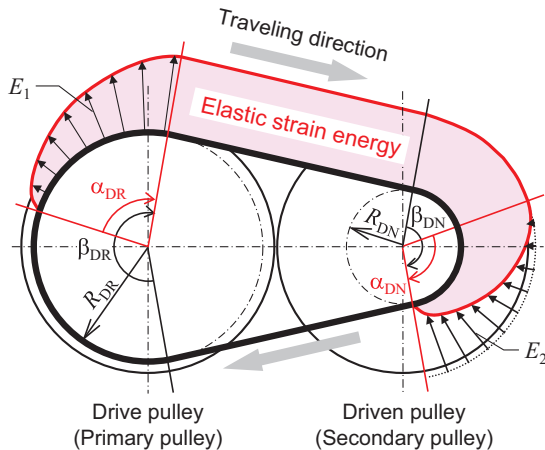


Fig. 4 Schematic diagram of transmitting power

As vehicle body weight is generally much greater than the rotating mass, it is assumed that irregularities in wheel rotation can be ignored.

2.2. Prediction of Elastic Slip Limit

In order to calculate the instantaneous safety factor, the elastic slip limit should be predicted in addition to estimating the slip state. In a previous study, the author proposed a method for predicting the elastic slip limit using only pulley speeds⁽⁸⁾. This method involved preparing a vibration model, taking into account friction attenuation from slip, and estimating attenuation parameters based on the measured pulley speeds. Thus, an appropriate vibration model for plant equipment is required. However, the previous research was only verified on a belt tester; the vibration model should be rethought for the proposed method to be applicable to actual vehicles.

The torsional rigidity of the mechanical parts comprising an actual vehicle powertrain is given in Fig. 5. In the figure, torsional rigidity for each mechanical part is shown in order of engagement and from engine to tires, from left to right. In the proposed method, the modeled range is set as being from drive pulley to the tires. This sets fluctuations in the drive pulley speed as the boundary condition, as will be described below.

In general, elements of torsional rigidity that are lower relative to the moment of inertia tend to contribute more to the natural frequency. In the interest of predicting the elastic slip limit, at the least belt rigidity will need to be considered.

Accordingly, the critical elements in setting a vibration model are as follows (red frame in Fig. 5):

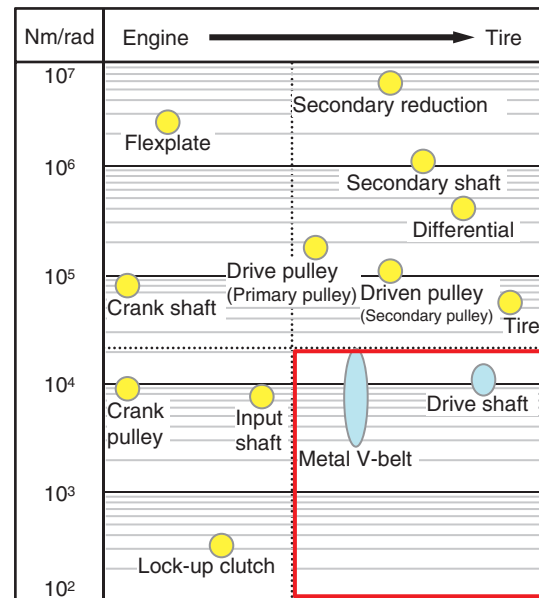


Fig. 5 Torsional rigidity of powertrain

- (i) Elements from drive pulley to tires
- (ii) Elements where torsional rigidity is equivalent to or less than belt rigidity

The belt and driveshaft were treated as torsion spring elements, and other elements were treated as rigid bodies. The vibration model as set is shown in Fig. 6.

With the assumptions outlined in the previous section, and a rotational perturbation θ around the operating point, the variational equations are as follows:

$$\frac{d}{dt} \begin{bmatrix} \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_3 \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ -\rho_1^2 \omega_1^2 & -2v_1 \rho_1^2 \omega_1 \zeta_1 & 0 & 2v_1 \rho_1^2 \omega_1 \zeta_1 \\ 0 & 0 & 0 & 1 \\ 0 & 2v_1 \omega_1 \zeta_1 & -\omega_2^2 & -2v_1 \omega_1 \zeta_1 \end{bmatrix} \begin{bmatrix} x_2 \\ x_3 \\ x_3 \\ \dot{x}_3 \end{bmatrix} + \frac{\rho_1^2 \omega_1^2}{\Omega} \begin{bmatrix} 0 \\ a_1 \sin \Omega t \\ 0 \\ 0 \end{bmatrix} \quad (3)$$

Here,

$$x_2 = i\theta_2, x_3 = i\theta_3, i = R_{DN}/R_{DR}, i_a = i_{sec} i_{fin}$$

$$I_2 = WR_{DN}^3 \beta_{DN}, I_3 = I_{DN} + \frac{1}{i_{sec}^2} \left(I_{sec} + \frac{I_{fin}}{i_{fin}^2} \right)$$

$$v_1 = r_{DN}/R_{DN}, \rho_1 = \sqrt{I_3/I_2}$$

$$\omega_1 = R_{DN} \sqrt{\frac{k_1}{I_3}}, \omega_2 = \frac{1}{i_a} \sqrt{\frac{k_2}{I_3}}, \zeta_1 = \frac{c_1 Q_{DN} R_{DN}}{\sqrt{I_3 k_1} \cos \psi}$$

The operating point refers to the average value (expected value) of periodic solution $x = (x_2, \dot{x}_2, x_3, \dot{x}_3)$ for all frequency components (See Ref. ⁽⁸⁾ for exact derivation).

In Eq. (3), there are two unknowns: (ω_1, ζ_1) . All other parameters are either constants or calculable known entities. As the unknown ζ_1 expresses the impact of belt slip, slip

state can be evaluated by estimating ζ_1 . Thus, engine cylinder pressure fluctuations can cause periodic fluctuation in the drive pulley speed. Assuming this speed fluctuation as amplitude a_1 and angular frequency Ω , the speed fluctuation can be expressed as $\dot{x}_1 = \dot{\theta}_1 = a_1 \cos \Omega t$. This speed fluctuation input is expressed by the second term on the right in Eq. (3). Now, a speed fluctuation of $\dot{x}_3 = a_3 \cos(\Omega t - \phi)$ is transmitted to the driven pulley, yielding $a_3/a_1 = m$. Substituting $\dot{x}_3 = a_3 \cos(\Omega t - \phi)$ for Eq. (3) and solving for (ω_1, ζ_1) , (ω_1, ζ_1) can be expressed as (m, ϕ) .

$$\omega_1(m, \phi) = \Omega \frac{\left[\left(\frac{\Omega}{\omega_2} \right)^2 \left(1 + \frac{1}{\rho_1^2} \right) - 1 \right] \left[\left(\frac{\Omega}{\omega_2} \right)^2 - 1 \right]}{\sqrt{\left(\frac{\Omega}{\omega_2} \right)^2 \left[\left(\frac{\Omega}{\omega_2} \right)^2 - 1 \right] \left(1 - \frac{\cos \phi}{m} \right)}} \quad (4)$$

$$\zeta_1(m, \phi) = \frac{1}{2v_1} \cdot \frac{\omega_2}{\omega_1} \cdot \frac{\left[\left(\frac{\Omega}{\omega_2} \right)^2 - 1 \right] \left[1 - \frac{1}{\rho_1^2} \left(\frac{\Omega}{\omega_1} \right)^2 \right]}{\frac{\sin \phi}{m} \cdot \frac{\Omega}{\omega_2}} \quad (5)$$

Thus, the current slip state ζ_1 can be estimated using Eq. (5) to evaluate amplitude ratio m and phase lag ϕ from the observed fluctuation in the drive and driven pulley speed. In addition, Eq. (4) indicates that the unknown belt rigidity k_1 can be estimated as well.

The above method illustrates how to estimate current slip state ζ_1 . The value for ζ_1 at the elastic slip limit, which is ζ_c , can be shown unambiguously.

$$\zeta_c = \frac{\rho_1}{v_1} \left[\frac{\rho_1^4 + 20\rho_1^2 - 8 + \rho_1(\rho_1^2 - 8)^{\frac{3}{2}}}{32(\rho_1^2 + 1)^3} \right]^{\frac{1}{2}} \quad (6)$$

This can be achieved by solving the Jacobian matrix in Eq. (3) for its maximum real number ζ_1 . From Eq. (6), elastic slip limit threshold ζ_c can only be solved for known values. Therefore, if current slip state ζ_1 can be estimated, the margin levels can be predicted up to the elastic slip limit.

2.3. Estimation of Instantaneous Safety Factor

Transmitting power with no belt slip demands a belt clamping force capable of transmitting the input torque. In an actual vehicle, oil is supplied to the pulley piston chamber to clamp down on the belt. With oil pressure providing the control power, oil pressure surging and response lag should be accounted for in the control system design. With that, controls should anticipate a clamping force margin, and safety factor s_f is defined by the following equation:

$$s_f = \frac{T_{\max}}{T} = \frac{Q}{Q_{\min}} \quad (7)$$

Here, $T_{\max}$ is the maximum transmittable torque for currently applied belt clamping force Q (axial thrust of the pulley), and $Q_{\min}$ is the minimum transmittable belt

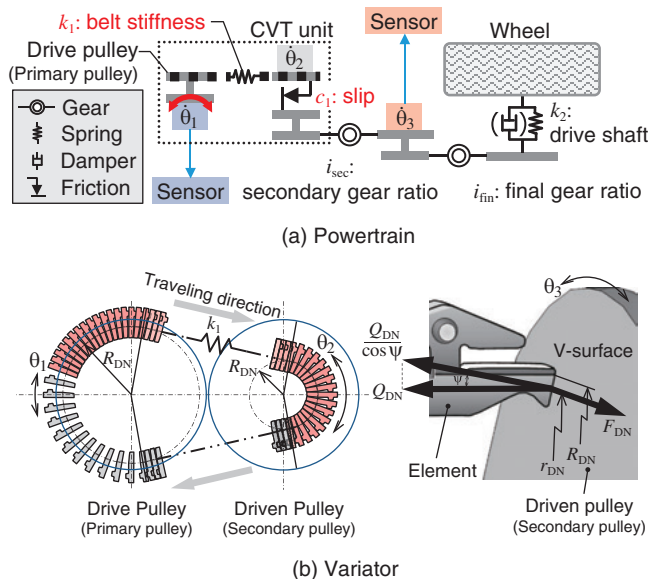


Fig. 6 Simple model of transmitting power

clamping force for current input torque T . “Transmittable” here refers to power transmitted within the elastic slip range. With this paper’s criteria, safety factor s_f was solved for the driven side as sliding slip occurs at the driven pulley.

From the previous section, the elastic slip limit at the point of slip velocity can be determined by comparing ζ_1 and ζ_c . However, the clamping force margin for the elastic slip limit cannot be calculated directly. As such, we will attempt to express safety factor s_f as a function of ζ_1 using the known characteristics, which are:

- (i) At $\zeta_1 = \zeta_c$, $s_f = 1$ (the elastic slip limit)
- (ii) $\zeta_1 \propto T_{DN}/T_{max,DN} = 1/s_f$ (From the assumptions in Eq. (1) and that ζ_1 correlates to slip velocity)

Graphing the above relationships yields Fig. 7. Sample test results are also plotted on Fig. 7. As ζ_c is known from Eq. (6), the relationship in Fig. 7 can be represented using ζ_1 in the following form:

$$s_f(\zeta_1) = \frac{1}{K_\tau(\zeta_1 - \zeta_c) + 1} \quad (8)$$

Here, K_τ is a coefficient. As the unloaded slip velocity will depend on the belt structure, the safety factor was estimated by first calculating K_τ with Eq. (8) for the belt used (with different specifications).

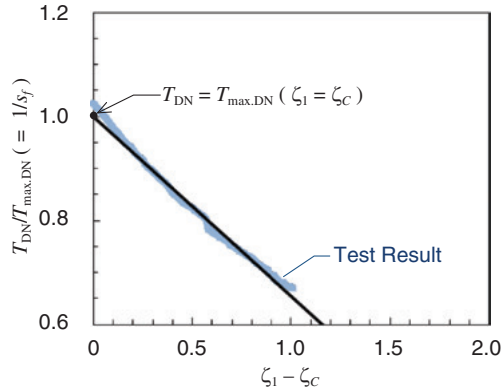


Fig. 7 Relationship between torque capacity and ζ_1

3. Testing Method

The previous section discussed a method for estimating instantaneous safety factor using the fluctuation in the drive and driven pulley speed. In order to verify this method as valid, instantaneous safety factor as measured in a moving test vehicle was compared with estimates. The test vehicle was configured as given in Fig. 8. The test vehicle used was a front-engine, front-wheel-drive vehicle. The power transmission path from engine to CVT consisted of a torque converter and planetary gear mechanism, through which power could be cut or switched to drive either forward or backward. Power output from the CVT was passed through

a two-stage deceleration mechanism with secondary and final reduction, then transmitted to the wheels.

CVT operation is represented by the maximum driven transmittable torque $T_{max,DN}$ with belt clamping force Q_{DN} applied to the driven pulley, which is given by the following equation:

$$T_{max,DN} = \frac{2\mu_c Q_{DN} r_{DN}}{\cos\psi} \quad (9)$$

Here, μ_c is the coefficient of friction at the elastic slip limit, r_{DN} is the effective radius of friction at the belt winding position, and ψ is the semi-vertical angle of the V-groove. From its definition and substituting Eq. (9) in Eq. (7), the safety factor is:

$$s_f = \frac{2\mu_c Q_{DN} r_{DN}}{T_{DN} \cos\psi} \quad (10)$$

Current load torque T_{DN} and belt clamping force Q_{DN} for the driven pulley were measured and substituted into Eq. (10) to solve for s_f , which was then treated as the measured instantaneous safety factor. The effective radius of friction r_{DN} can be solved geometrically from current speed ratio i ($= N_{DR}/N_{DN}$). Given the challenges in directly measuring driven pulley load torque T_{DN} , torque $T_{L,R}$ acting on the left and right driveshafts was measured. The torque, as measured with a strain gauge, was converted to driven pulley load torque T_{DN} using the following equation:

$$T_{DN} = \frac{T_L + T_R}{i_{sec,fin} \eta_{sec,fin}} \quad (11)$$

$i_{sec,fin}$ is the reduction mechanism gear ratio, and $\eta_{sec,fin}$ is the gear transmission efficiency. $\eta_{sec,fin}$ was 0.99 for both. Belt clamping force Q_{DN} was calculated by measuring the oil pressure as supplied to the piston chamber and adding the centrifugal oil pressure component. The coefficient of friction μ_c was pre-measured for the belts and pulleys before installing them in the test vehicle.

The test conditions are given in Table 1. Tests were conducted while cruising at speeds of 70-100 km/h. Thus,

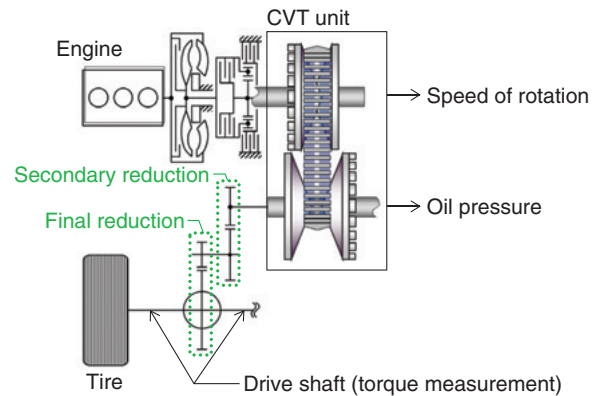


Fig. 8 Schematic view of test vehicle

Table 1 Test conditions

Vehicle speed	70, 80, 90, 100 km/h
Speed ratio, i	0.468
Vehicle body weight	1440, 1651 kg
Differential diameter rate, ε_{LR}	0.9-1.1 (used chassis dynamometer)
Oil temperature	85-105 degC

speed ratio on the acceleration side was nearly constant. Vehicle body weight was 1,440 kg when empty and 1,651 kg at seating capacity. The vehicle was warmed up before test driving.

Also, while straight driving is assumed for this study, note that a differential between left and right wheels could occur even while driving straight if the rolling radius of the left and right tires differs. As this differential could impact estimation accuracy, the test vehicle driving wheels were secured on a chassis dynamometer, and the left and right wheel rotation speed was changed. With a rolling radius $R_{L,R}$ for the left and right wheels, the differential diameter rate ε_{LR} between left and right is defined by the following equation:

$$\varepsilon_{LR} = \frac{R_R}{R_L} \quad (12)$$

Here, rotating speed $N_{L,R}$ for the left and right wheels changes according to differential diameter rate ε_{LR} as follows:

$$\begin{aligned} N_L &= \frac{2\varepsilon_{LR}}{1 + \varepsilon_{LR}} \cdot \frac{N_{DN}}{i_{sec}i_{fin}} \\ N_R &= \frac{2}{1 + \varepsilon_{LR}} \cdot \frac{N_{DN}}{i_{sec}i_{fin}} \end{aligned} \quad (13)$$

Where N_{DN} is the driven pulley rotating speed. For cases where a spare tire is equipped, such as with a flat tire, a left-right wheel differential of $\varepsilon_{LR} = 0.9-1.1$ was simulated.

4. Test Results and Discussion

With the vehicle cruising at a speed of 70-100 km/h, slip state ζ_1 was calculated based on Eq. (5). The relationship between ζ_1 and the measured safety factor at ζ_1 is shown in Fig. 9. The standard test conditions (Empty vehicle at 1440 kg, no left-right wheel differential: $\varepsilon_{LR} = 1$) from Table 1 were used. From the results of Fig. 9, K_τ in Eq. (8) was determined with the least squares method. As approximated with Eq. (8), the results were superimposed with a solid line in Fig. 9. Instantaneous safety factor was estimated based on this K_τ and ζ_1 , which was calculated from the speed fluctuation while driving.

Estimation error for safety factor as estimated with the proposed method under the above standard conditions is shown in Fig. 10. Safety factor estimation error ε_{sf} was defined as follows:

$$\varepsilon_{sf} = \frac{s_{fest} - s_f}{s_f} \quad (14)$$

where s_{fest} is the estimated safety factor value and s_f is the measured value. Estimation error was broken down into error from vehicle speed [Fig. 10(a)] and error from the safety factor range as measured [Fig. 10(b)]. Looking at Fig. 10(a), while the maximum observed estimation error was roughly 30%, no correlation was found between estimation error and vehicle speed. Meanwhile, in Fig. 10(b), estimation error did tend to decrease as safety factor approached 1. Presumably, this suggests that estimation is easier as the safety factor decreases because belt slip (i.e. variation in pulley speed) increases. Estimation error was 15% or lower for the $s_f \leq 1.67$ range ($1.50 \leq s_{fest} \leq 1.93$) and 2% or lower for the $s_f \leq 1.25$ range ($1.22 \leq s_{fest} \leq 1.24$). At the least, this confirms that, in the $s_f \leq 1.67$ range, safety factor can be estimated up to the elastic slip limit without involving sliding slip.

However, note that the above estimation accuracy is for identifying K_τ for the standard conditions. In terms of method verification, it remains to be confirmed whether the above estimation accuracy can withstand changes in vehicle conditions. Once again, the assumed vehicle conditions used in deriving the proposed method are:

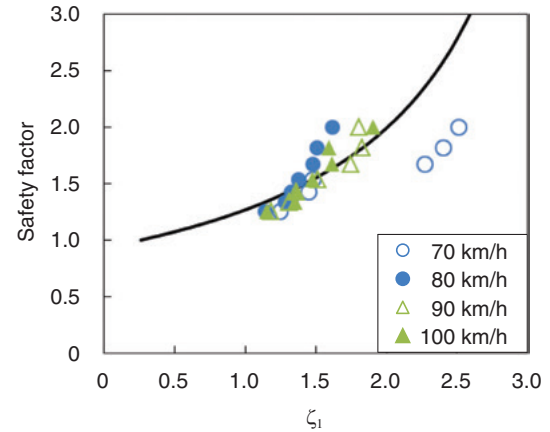
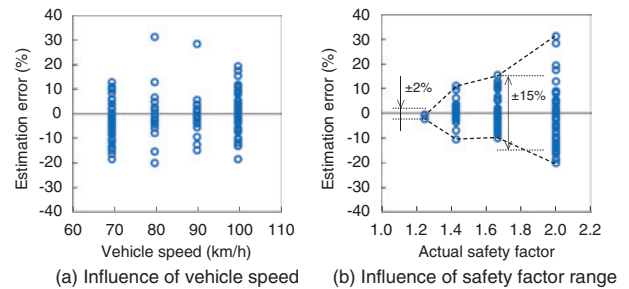
Fig. 9 Relationship between safety factor and ζ_1 

Fig. 10 Accuracy of estimating instantaneous safety factor

- (i) The left and right driveshafts can be represented by equivalent rigidity
 - (ii) The wheels are considered fixed as a vibration transfer path
- (i) is satisfied with no difference between left and right wheels, and (ii) is satisfied at sufficiently large vehicle body weights.

For (i), driveshaft rigidity for front-engine, front-wheel-drive vehicles generally differs from left to right (left and right are of unequal lengths).

With a left-right wheel differential, equivalent stiffness k_2 will change (Fig. 11). In Fig. 11, the top chart shows the rate of change for equivalent stiffness k_2 against differential diameter rate ε_{LR} , and the bottom chart shows the predicted values for estimation error as calculated from error propagation. In the range of $\varepsilon_{LR} = 0.9$ -1.1, estimation error is expected to be 4% or lower, which is negligible. Estimation error was similarly predicted for assumption (ii) (Fig. 12). The predicted values for estimation error when vehicle body weight is changed are shown in Fig. 12. At vehicle body weights of 400 kg and higher, the projected estimation error was similar to (i) at 4% or lower. Body weights for the test vehicle were 1440 kg (when empty) and higher. As such, the impact of changes in vehicle body

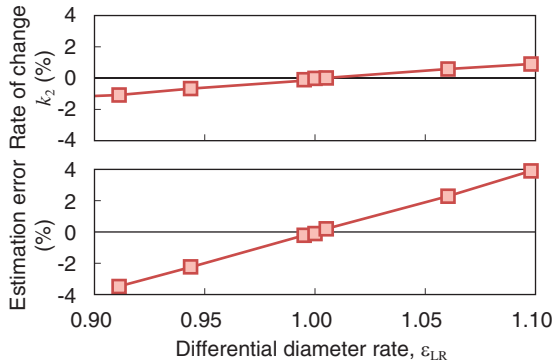


Fig. 11 Predicted influence of equivalent torsional rigidity of drive shaft

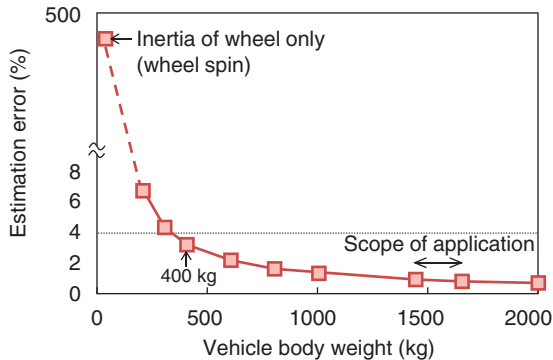


Fig. 12 Predicted influence of vehicle body weight

weight on estimation error can presumably be ignored. From the perspective of the rotating and vibrating systems, reduction in vehicle body weight is synonymous with wheel spin, where the wheels slip on the road. Figure 12 illustrates that estimation error increases sharply as vehicle body weight approaches the wheel unit mass from 400 kg. While pointless to reduce belt clamping force during wheel spin, care should be taken when constructing the controls to help prevent malfunctions.

The test results for assumptions (i) and (ii) are given in Fig. 13. Figure 13(a) shows the estimation error with the weight of the vehicle at nominal riding capacity (1651 kg). Figure 13(b) shows the estimation error with a differential diameter rate of $\varepsilon_{LR} = 0.9, 1.1$. In the figures, the dashed lines represent the maximum and minimum values (linear interpolated) for estimation error with the standard conditions [from Fig. 10(b)]. For both vehicle body weight and differential diameter rate, the estimation error falls within the dashed lines, confirming that estimation accuracy remains equivalent to the standard conditions. Focusing on the $s_f \leq 1.25$ range, where the risk of sliding slip is increased, the estimation accuracy was quite high, with an error range of 2% or less. Also, with left and right wheels of equal rolling radius, the lateral acceleration was 23.5 m/s^2 (2.4 g), corresponding to a differential of $\varepsilon_{LR} = 0.9, 1.1$. From this, instantaneous safety factor can presumably be estimated for cornering, assuming normal cornering under standard conditions.

To this point, the discussion on validation has concerned the aspect of changes in the vehicle. Another aspect is validation for change in coefficient of friction. In order to simulate how coefficient of friction μ_C declines over time, a test V-belt pulley with mirror-finished groove surfaces was prepared. The test vehicle was mounted with the test pulley and then subjected to the same tests again. The test results are given in Fig. 14. Figure 14(a) shows coefficient of friction μ_C for the test pulley. The mirror-finished surfaces reduced coefficient of friction μ_C by approximately 8%. Figure 14(b) shows the estimation error with the reduced coefficient of friction μ_C . In the graphs, the dashed line

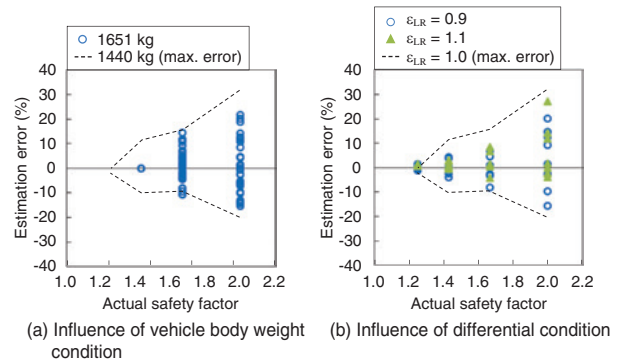


Fig. 13 Verification test results of estimating instantaneous safety factor

represents the estimated error range for the standard conditions. At $s_f = 1.67$, while the resulting estimation error does increase past that for the standard conditions at some points, there is sufficient marginal clamping force at these points. At $s_f = 1.43$, where the risk of sliding slip is elevated, estimation accuracy was confirmed to be equivalent to that under standard conditions. There were also no positive or negative offsets for the estimation error range.

As shown above, estimation accuracy was confirmed to remain constant, independent of changes in vehicle state or coefficient of friction. In the required estimation accuracy range of $s_f \leq 1.25$, this verifies that instantaneous safety factor can be estimated to within an error range of 2%.

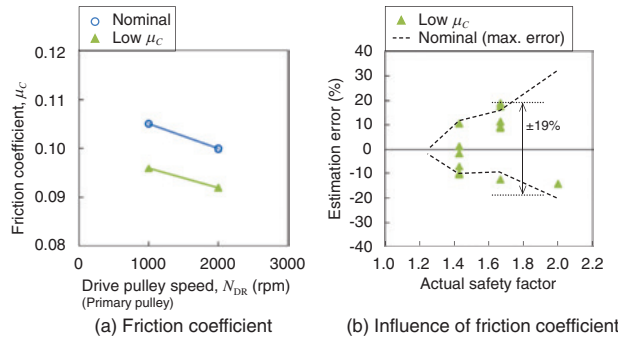


Fig. 14 Changes in estimated instantaneous safety factor over time

5. Conclusion

We have proposed a method for estimating instantaneous safety factor using only the fluctuation in the drive and driven pulley speed. With the proposed method verified as valid for cruising, we obtained the following results:

- (1) Instantaneous safety factor can be estimated independently of changes in vehicle status (vehicle body weight and left-right wheel differential) and coefficient of friction.
- (2) In the required estimation accuracy range of $s_f \leq 1.25$, estimations were within an error range of 2%.

In future studies, we aim to estimate instantaneous safety factor under sudden acceleration and other transient states. It will be important to validate responsiveness for these cases as we continue to expand on our estimation theory.

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List of Symbols

$a_{1,3}$	Amplitude of speed fluctuation (1: drive pulley; 3: driven pulley)
c_1	Damping factor with belt slippage
$F_{DR,DN}$	Tangential friction force on the V-surface (DR: drive pulley; DN: driven pulley)
i	Speed ratio, N_{DR}/N_{DN}
i_{fin}	Gear ratio of final reduction
i_{sec}	Gear ratio of secondary reduction
$I_{2,3}$	Moment of inertia
I_{DN}	Moment of inertia of driven pulley
I_{fin}	Moment of inertia of differential
I_{sec}	Moment of inertia of secondary shaft
k_1	Compression stiffness of belt
k_2	Equivalent torsional rigidity of drive shaft
$k_{2L,R}$	Torsional rigidity of drive shaft (L: left side; R: right side)
m	Amplitude ratio, a_3/a_1
$N_{DR,DN}$	Speed of rotation of pulley (DR: drive pulley; DN: driven pulley)
$N_{L,R}$	Speed of rotation of wheel (L: left wheel; R: right wheel)
$P_{DR,DN}$	Applied oil pressure for pulley thrust (DR: drive pulley; DN: driven pulley)
$Q_{DR,DN}$	Pulley thrust (DR: drive pulley; DN: driven pulley)

$Q_{\min,DR,DN}$	Minimum pulley thrust for transmitting load torque on pulley (DR: drive pulley; DN: driven pulley)
$r_{DR,DN}$	Effective radius of friction (DR: drive side; DN: driven side)
$R_{DR,DN}$	Belt pitch radius (DR: drive side; DN: driven side)
$R_{L,R}$	Effective rolling radius (L: left wheel; R: right wheel)
$T_{DR,DN}$	Load torque on pulley (DR: drive pulley; DN: driven pulley)
$T_{\max,DR,DN}$	Maximum transmittable torque on pulley (DR: drive pulley; DN: driven pulley)
s_f	Safety factor
V	Mean slip velocity between belt and pulley
W	Linear density of belt
$\alpha_{DR,DN}$	Active arc (DR: drive pulley; DN: driven pulley)
$\beta_{DR,DN}$	Belt wrap angle (DR: drive pulley; DN: driven pulley)
ε_{LR}	Differential diameter rate
ε_{sf}	Relative error of estimating safety factor
ϕ	Phase lag
η	Power transmission efficiency
μ_c	Mean friction coefficient on the V-surface of pulley
θ_1	Degree of perturbation of angle of rotation of drive pulley
θ_2	Degree of perturbation of angle of rotation of elements wrapped around driven pulley
θ_3	Degree of perturbation of angle of rotation of driven pulley
$\omega_{1,2}$	Natural angular frequency
Ω	Excitation angular frequency
Ψ	1/2 the V angle
ζ_1	Quantity of slip state
ζ_c	Critical state quantity (elastic slip limit)

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