

Prediction of Friction Drive Limit of Metal V-Belt

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ABSTRACT

When fluctuations in the speed of rotation of the drive pulley are transmitted to the driven pulley via the metal V-belt, the transmitted fluctuations become attenuated as friction force approaches a state of saturation. The research discussed in this paper focused on these fluctuations in the speed of rotation and developed an index for the slip state between the belt and the pulleys.

The drive and driven pulleys were regarded as a one-dimensional vibrating system connected by elastic bodies, and changes in the state matrix of the system were focused on. It was determined that when all of the eigenvalues in this state matrix become real numbers, slip speed between the belt and the pulleys increases sharply. A method was proposed of estimating this behavior of the eigenvalues from changes in the speed of rotation of the drive and driven pulleys, and indexing the current slip state.

Torque fluctuations reproducing the fluctuations produced by an engine were input to the drive pulley, and pulley thrust was gradually reduced in order to measure changes in slip speed. The friction drive limit found using the proposed method matched the point of increase in slip speed and the point of maximum power transmission efficiency as obtained from measurements, verifying the effectiveness of the method.

Using the proposed method, it is possible to predict the friction drive limit, a parameter that is strongly affected by individual differences between parts and age-related changes, and this will contribute significantly to the optimization of pulley thrust.

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INTRODUCTION

Demand for the realization of greater fuel efficiency is increasing today, and in the area of transmissions the achievement of increased power transmission efficiency is an urgent issue. Automotive transmissions take a variety of forms, but continuously variable transmissions (CVT), which make it possible to optimize the engine's brake specific fuel consumption (BSFC), and among CVT transmissions using metal V-belts, are in widespread use. [Figure 1](#) shows a breakdown of the consumption of power when a vehicle fitted with a CVT is operated in JC08 mode. Of the power supplied by the engine, 24% is consumed in the transmission, indicating that increasing power transmission efficiency can be expected to make a significant contribution to increasing automotive fuel efficiency.

Oil pressure pump work and friction loss in the metal V-belt represent the greater part of the breakdown of power consumption in a CVT. Metal V-belt CVT generate thrust in pulleys using oil pressure, and transmit power by driving the metal V-belt using this pulley thrust. The use of oil pressure pumps for the generation of oil pressure is therefore essential to the systems.

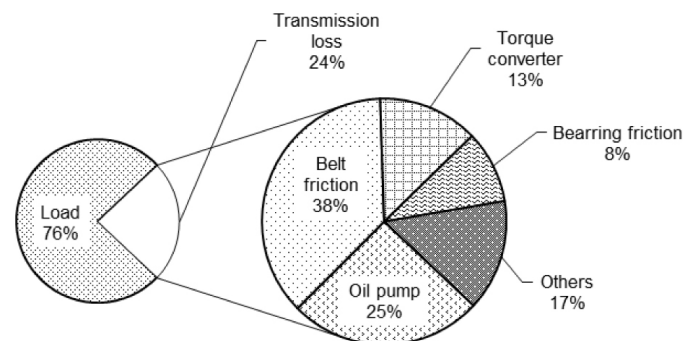


Figure 1. Breakdown of power consumed in a vehicle fitted with a CVT (during operation in JC08 mode)

As [Fig. 2](#) shows, metal V-belts are made up of multiple elements supported by a pair of layered rings. Because the configuration allows the elements and the rings to move freely, slip occurs between the elements and rings when the metal V-belt is rotating. This is the main cause of the friction loss that occurs in the metal V-belt.

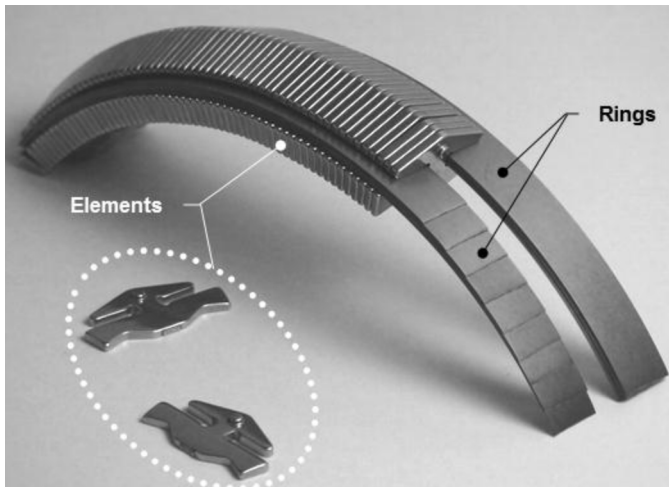


Figure 2. Configuration of metal V-belt

Given this, the reduction of pulley thrust is known to be an effective method of increasing power transmission efficiency in metal V-belt CVT [1]. As Fig. 3 shows, if pulley thrust is reduced, ring tension is also reduced, and friction loss between the elements and rings declines, increasing power transmission efficiency. The reduction of pulley thrust also reduces oil pressure pump work, making it possible to further increase fuel efficiency.

However, excessive reduction of pulley thrust can result in macro-slip between the pulley and the metal V-belt, which may cause damage to the CVT unit. In addition, the rapid increase in friction loss between the pulleys and the metal V-belt would result in a decline in power transmission efficiency. The coefficient of friction between the pulleys and the metal V-belt is significantly affected by individual differences between parts, the state of lubrication, and age-related changes, making it challenging to predict in an actual transmission. Designers therefore have no choice but to set pulley thrust that offers a considerable safety factor.

In response to these issues, a method of detecting limit pulley thrust from sudden increases in the speed of slip between the pulleys and the metal V-belt has been proposed [2]. For the application of this method, it is necessary that the system experience macro-slip, or a state of slip very close to macro-slip. While this is only for an extremely brief period, there is concern that it might promote abrasive wear. It also increases the risk of serious damage due to adhesive wear of the metal V-belt as a result of delay in the oil pressure response. Given this, the ability to predict points at which macro-slip will occur in advance would represent an ideal situation.

The research discussed in this paper therefore defined the friction drive limit from the perspective of increase in the speed of slip, and set out to develop a method of determining the margin for the current state of slip in relation to this friction drive limit. If this margin could be measured, the minimum safety factor in response to disturbance could be predicted, making it possible to develop practical pulley thrust control.

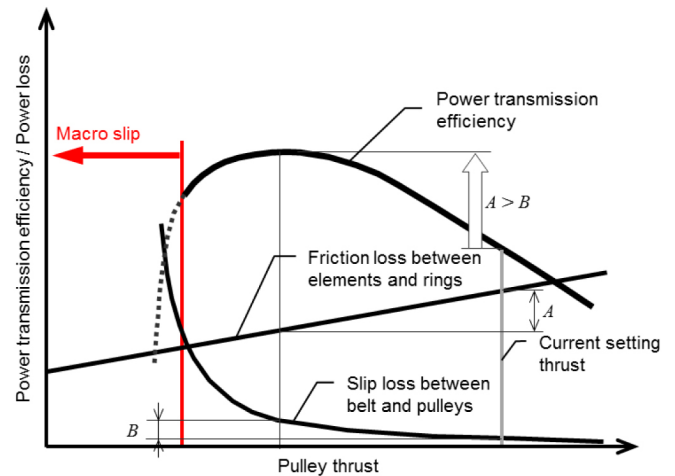


Figure 3. Slip between pulley and metal V-belt in relation to pulley thrust and change in power transmission efficiency

THEORY

Basic Equations

Metal V-belt CVTs are made up of a drive and a driven pulley and a metal V-belt. The speed ratio is changed steplessly by changing the winding diameter of the belt around each pulley.

In an actual vehicle, the CVT is driven by the engine. The torque input to the CVT is subject to fluctuations resulting from fluctuations in engine in-cylinder pressure. Because of this, it is necessary to consider slip between the pulleys and the belt dynamically. It has been reported that when conducting dynamic analyses of timing belt transmission, it is possible to calculate fluctuations in the angle of rotation of the pulleys by treating the longitudinal stiffness of the belt as a spring element and analyzing the vertical vibration of the belt [3,4]. It is known that metal V-belts, unlike rubber belts or chains, transmit power chiefly by means of compressive force between the elements [5]. A simple model, in which the compression stiffness of the belt is represented by a one-dimensional spring element (Fig. 4), will therefore be considered.

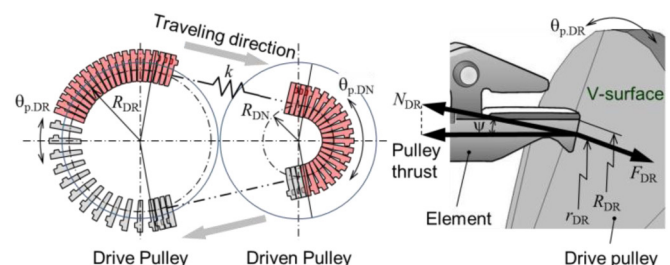


Figure 4. Simple model of power transmission in Metal V-belt CVT

It is assumed that load torque is acting on the drive and driven pulleys, and that an equilibrium point, θ_0 , exists. The degree of perturbation of the angles of rotation of the drive and driven pulleys from this equilibrium point will respectively be termed $\theta_{p,DR}$ and $\theta_{p,DN}$. If the elements wrapped around the drive and driven pulleys are

each considered a single mass point and the degree of perturbation of their angles of rotation from the equilibrium point termed $\theta_{b,DR}$ and $\theta_{b,DN}$, the equation of motion for the pulleys and the belt is as follows:

$$\begin{cases} f_1 = I_{p,DR} \ddot{\theta}_{p,DR} = -2F_{DR} r_{DR} \\ f_2 = I_{b,DR} \ddot{\theta}_{b,DR} = 2F_{DR} r_{DR} - kR_{DR} (R_{DR} \theta_{b,DR} - R_{DN} \theta_{b,DN}) \\ f_3 = I_{b,DN} \ddot{\theta}_{b,DN} = kR_{DN} (R_{DR} \theta_{b,DR} - R_{DN} \theta_{b,DN}) - 2F_{DN} r_{DN} \\ f_4 = I_{p,DN} \ddot{\theta}_{p,DN} = 2F_{DN} r_{DN} \end{cases} \quad (1)$$

Here, I_p is the moment of inertia of the pulleys, I_b is the moment of inertia of the sections of belt wrapped around the pulleys, F is the friction force acting between the pulleys and the belt, k is the compression stiffness of the belt, R is the belt pitch radius, and r is the effective radius of friction. The subscripts DR and DN respectively indicate the drive side and the driven side.

The friction force acting between the pulleys and the belt is the product of the friction coefficient, $\tilde{\mu}$, and the normal force acting on the V-surfaces of the belt, N .

$$\begin{cases} F_{DR} = \tilde{\mu}_{DR} N_{DR} \\ F_{DN} = \tilde{\mu}_{DN} N_{DN} \end{cases} \quad (2)$$

The coefficient of friction $\tilde{\mu}$ is the average coefficient of friction, and is defined using the following equation.

$$\begin{cases} \tilde{\mu}_{DR} = \frac{T_{DR} \cos \psi}{2Q_{DR} r_{DR}} \\ \tilde{\mu}_{DN} = \frac{T_{DN} \cos \psi}{2Q_{DN} r_{DN}} \end{cases} \quad (3)$$

T is pulley load torque, Q is pulley thrust, and ψ is 1/2 the V angle.

Because the elements and rings are not constrained, power is transmitted while micro-slips occur independently among individual elements on the pulleys. It is known that the slip speed between the pulleys and the belt increases macroscopically at this time because the transmission of friction approaches saturation and the number of elements transmitting power on the pulleys increases [6,7,8].

Terming the average slip speed between the pulleys and the belt $\dot{\tilde{s}}$, $\tilde{\mu}$ can be expressed as a function of $\dot{\tilde{s}}$: $\tilde{\mu} = f(\dot{\tilde{s}})$. Accordingly, $\tilde{\mu}$ can be expressed as a second-order approximation equation by Taylor expansion around the average slip speed at the equilibrium point, $\dot{\tilde{s}}_0$ (Fig. 5).

$$\begin{cases} \tilde{\mu}_{DR} = c_{1,DR} \left(\dot{\tilde{s}}_{DR} - \dot{\tilde{s}}_{0,DR} \right) - c_{2,DR} \left(\dot{\tilde{s}}_{DR} - \dot{\tilde{s}}_{0,DR} \right)^2 + \mu_{0,DR} \\ \tilde{\mu}_{DN} = c_{1,DN} \left(\dot{\tilde{s}}_{DN} - \dot{\tilde{s}}_{0,DN} \right) - c_{2,DN} \left(\dot{\tilde{s}}_{DN} - \dot{\tilde{s}}_{0,DN} \right)^2 + \mu_{0,DN} \\ \dot{\tilde{s}}_{DR} = R_{DR} (\dot{\theta}_{p,DR} - \dot{\theta}_{b,DR}) \\ \dot{\tilde{s}}_{DN} = R_{DN} (\dot{\theta}_{b,DN} - \dot{\theta}_{p,DN}) \end{cases} \quad (4)$$

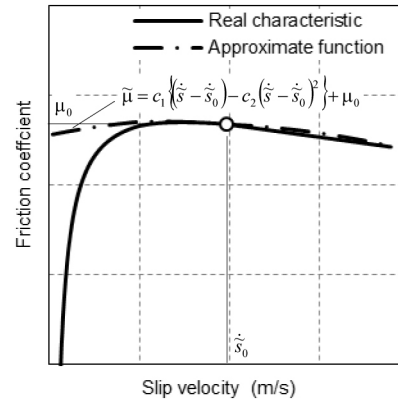


Figure 5. Friction characteristic between pulleys and belt

Because, from the definition of the equilibrium point, $\mathbf{f}(\theta_0) = \mathbf{0}$ and $\mathbf{f} = (f_1, f_2, f_3, f_4)$, Equation (1) ultimately becomes

$$\begin{cases} I_{p,DR} \ddot{\theta}_{p,DR} = -2N_{DR} r_{DR} c_{1,DR} \times \\ \quad \left\{ \left(R_{DR} \dot{\theta}_{p,DR} - R_{DR} \dot{\theta}_{b,DR} \right) - c_{2,DR} \left(R_{DR} \dot{\theta}_{p,DR} - R_{DR} \dot{\theta}_{b,DR} \right)^2 \right\} \\ I_{b,DR} \ddot{\theta}_{b,DR} = 2N_{DR} r_{DR} c_{1,DR} \times \\ \quad \left\{ \left(R_{DR} \dot{\theta}_{p,DR} - R_{DR} \dot{\theta}_{b,DR} \right) - c_{2,DR} \left(R_{DR} \dot{\theta}_{p,DR} - R_{DR} \dot{\theta}_{b,DR} \right)^2 \right\} - \\ \quad kR_{DR} (R_{DR} \theta_{b,DR} - R_{DN} \theta_{b,DN}) \\ I_{b,DN} \ddot{\theta}_{b,DN} = kR_{DN} (R_{DR} \theta_{b,DR} - R_{DN} \theta_{b,DN}) - \\ \quad 2N_{DN} r_{DN} c_{1,DN} \times \\ \quad \left\{ \left(R_{DN} \dot{\theta}_{b,DN} - R_{DN} \dot{\theta}_{p,DN} \right) - c_{2,DN} \left(R_{DN} \dot{\theta}_{b,DN} - R_{DN} \dot{\theta}_{p,DN} \right)^2 \right\} \\ I_{p,DN} \ddot{\theta}_{p,DN} = 2N_{DN} r_{DN} c_{1,DN} \times \\ \quad \left\{ \left(R_{DN} \dot{\theta}_{b,DN} - R_{DN} \dot{\theta}_{p,DN} \right) - c_{2,DN} \left(R_{DN} \dot{\theta}_{b,DN} - R_{DN} \dot{\theta}_{p,DN} \right)^2 \right\} \end{cases} \quad (5)$$

and pulley and belt motion that produces slip is described as a non-linear simultaneous differential equation.

Table 1 shows the main parameters in Equation (5). In the present research, the possibility of measurement of the main parameters was defined as shown in Table 1. Because it is necessary to deliberately cause macro-slip and potentially damage the CVT in order to determine the limit value of the coefficient of friction $\tilde{\mu}$, unless measurement of the value itself is the purpose, measurements in actual CVT are challenging to conduct.