

Design of Safety Factor for Slip of Half-toroidal Variator by Dynamic Behavior Analysis

Masahiro TANAKA*

Toshihiro SAITO*

Koji MIYAMOTO**

ABSTRACT

In order to determine the power transmission characteristics of half-toroidal variators, a dynamic behavior analysis technique was developed to predict the traction coefficient. The approach uses the finite element method so that the elastic deformation of the parts in actual operation can be reproduced in variator modeling. As the technique reflects the shear characteristics of the traction fluid on the traction surface, frictional force was identified based on a four-roller test and used for the contact surface in the behavior analysis model. To control ratios, feedback control was applied based on the shifting mechanism of the toroidal variator. Using this technique to quantitatively predict the change in slip rate for the traction coefficient of the half-toroidal variator allows for the corresponding safety factor to be established.

1. Introduction

Demands continue to increase in recent years for response to environmental issues such as global warming. With these demands, fuel consumption of automobiles is decreasing, thanks to factors such as enhanced efficiency of engines and application of electrified powertrains. For transmissions, extension of the ratio coverage is demanded in order to achieve both better fuel economy when cruising and the required acceleration force for launching. For example, the nine-speed AT covers the ratio spread of 9.823⁽¹⁾. In metal pushing V-belt-type CVTs as well, transmissions with a ratio coverage of 8.7⁽²⁾ have seen practical applications incorporating an auxiliary gearbox.

The infinitely variable transmission (IVT)⁽³⁾ presents possibilities as a transmission for allowing even wider ratio coverages. In the IVT, the ratio of transmission output rotation speed to input rotation speed can be set to zero. This ratio is called the geared neutral (GN) point. At near this ratio, a lower speed ratio can be achieved for takeoffs. This renders the torque converter and start clutch unnecessary and is expected to reduce both slip loss and number of parts.

The authors have developed an IVT utilizing a half-toroidal variator in the shift mechanism. In order to shift the variator speed ratio from the maximum speed side to the minimum side, the pulley on a metal pushing V-belt-

type CVT requires a stroke of approximately 20 mm. Meanwhile, the power roller on a toroidal-type CVT only needs to be offset by approximately 1 mm, offering the advantage of faster shifting. This characteristic can enhance response levels during kick down and reduce time lags during acceleration. Further, reports show that transmitting efficiency is high during usage in the high-speed range⁽⁴⁾.

The power transmission of the toroidal variator is through traction drive, a mechanism by which power is transmitted through the shear force of the traction fluid interposed between two bodies. Pressing the two surfaces together at high pressure causes the traction fluid to undergo a glass transition into a solid phase⁽⁵⁾. Determining the traction coefficient, or ratio of shear force to contact force, and setting appropriate its values for use when operating will enhance transmitting efficiency and reliability against slipping. However, whereas operation with a high traction coefficient will enhance transmitting efficiency, a state of gross slip will result if the limit value is exceeded, and no power will be transmitted.

To date, the traction coefficient has been predicted⁽⁶⁾ and measured⁽⁷⁾. These studies, however, focus on the maximum value and do not go into detail on the relationship with the slip rate. In this paper, therefore, a dynamic behavior analysis technique was developed for half-toroidal variators to predict the traction coefficient in actual operation. This allows for quantitative prediction of the relationship

* Automobile R&D Center

* ITO CORP.

between the transmitting efficiency and slip rate for the traction coefficient used. This paper will report methods for establishing the safety factor.

2. Transversely Installed Half-toroidal IVTs

2.1. IVT Configuration

Figure 1 depicts a cross-section of the prototyped IVT for a front-wheel-drive vehicle, and Fig. 2 gives the relationship between the speed increasing ratios of the IVT

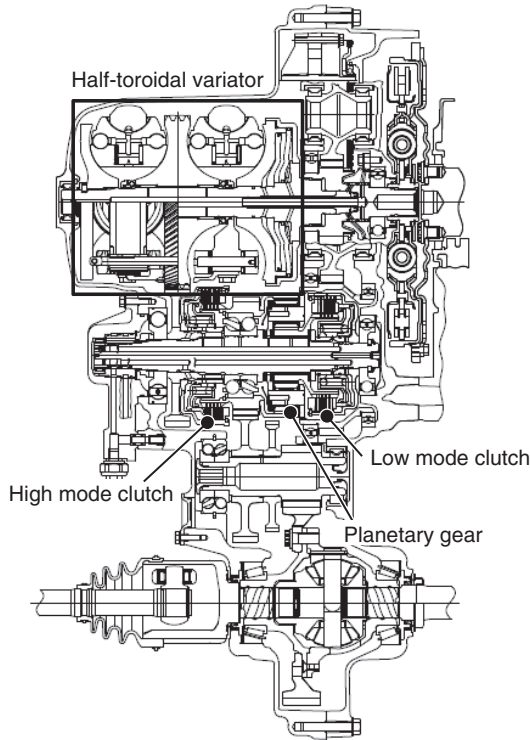


Fig. 1 Cross section of half-toroidal IVT

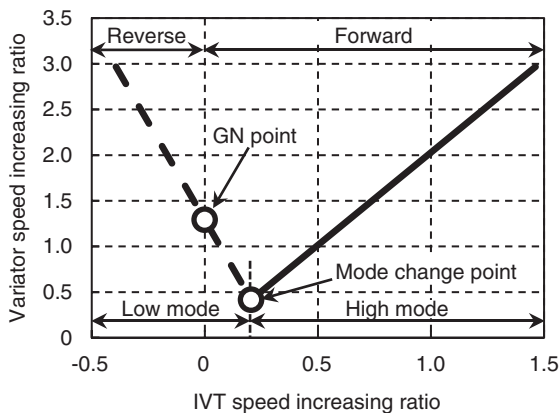


Fig. 2 Relationship between variator and IVT speed increasing ratio

and variator. A double-cavity half-toroidal variator was positioned on the input shaft, and the planetary gears for splitting power and the low- and high-mode clutches for shifting modes were positioned on the countershaft. Here, low mode is used mainly during takeoffs and reversing, and is thus set to a ratio width including the GN point. High mode, also called the CVT mode, includes cruising conditions.

2.2. Mechanism of Power Transmission for Half-toroidal Variator

Figure 3 shows an external view of the double-cavity half-toroidal variator, and Fig. 4 shows its cross-section. In order to allow for transverse installation on the vehicle, total length is shortened compared to longitudinal installation⁽⁸⁾ with changes including integration of the front and rear cavity output disks and the output gear⁽⁹⁾, and involute splines to fit the input shaft and input disk. The power rollers are supported by trunnions via thrust bearings. The power rollers are driven by axial force from the hydraulic loading piston to rotate between the input and output disks, and driving power is transmitted by the torque flow. This



Fig. 3 Double-cavity half-toroidal variator

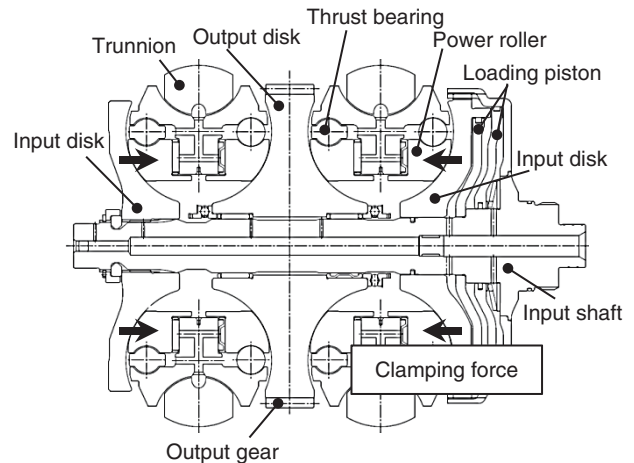


Fig. 4 Cross section of double-cavity half-toroidal variator

process is depicted in Fig. 5.

Tilting the power rollers to alter the ratio of r_1 and r_2 will in turn alter the toroidal variator ratio (see Fig. 6). Here, r_1 and r_2 are the radii of the respective points of contact on the power roller for the input and output disks.

2.3. Shifting Mechanism

Figure 7 depicts the shifting mechanism for the toroidal variator. Here, V_{id} , V_{od} , and V_{pr} are the circumferential

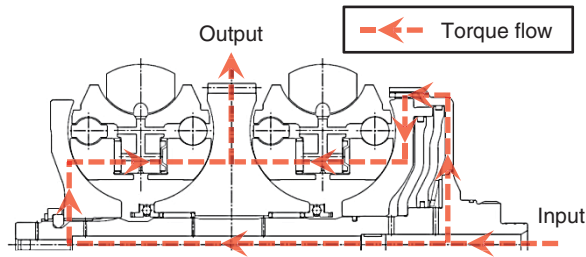


Fig. 5 Torque flow of double-cavity half-toroidal variator

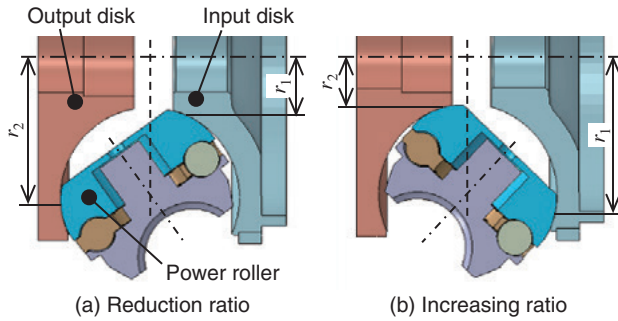


Fig. 6 Relationship between ratio and contact radius

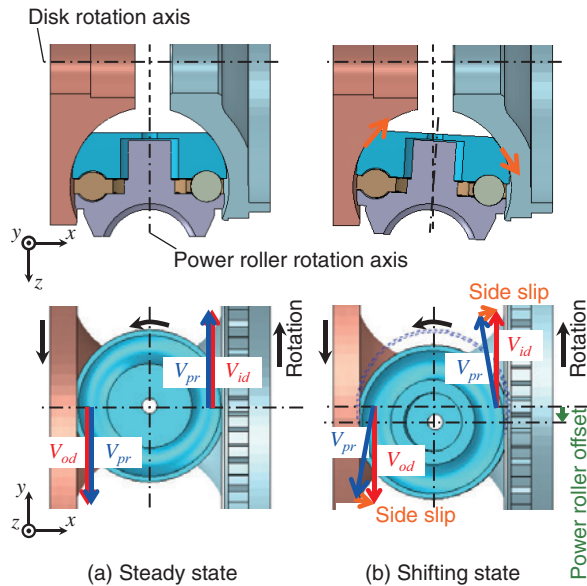


Fig. 7 Shifting mechanism

velocity at the respective points of contact for the input disk, output disk, and power roller. As depicted in Fig. 7 (a), when the rotation axis of the power roller is on plane zx including the input and output disk rotation axes, the directions of V_{id} and V_{pr} are aligned and the power roller does not tilt. This puts the ratio in a steady state. Meanwhile, as depicted in Fig. 7 (b), offsetting the power roller slightly in axial direction y will produce a difference in orientation between V_{id} and V_{pr} . The frictional force in the tilted direction produced by this difference will set the power roller into a tilting motion, transitioning it to a shifting state. When the power roller is returned from its offset, the tilting motion will stop and the power roller returns to the steady-state ratio. When looking to shift in the reverse direction, the power roller is offset in the opposite direction.

Each of the power rollers is offset by the hydraulic shifting piston mounted on the lower part of the trunnion (see Fig. 8). Hydraulic pressures P_{high} and P_{low} are exerted on the upper and lower parts of the pistons, respectively. A pressure difference is created by increasing P_{high} when shifting to the speed increasing side or P_{low} when shifting to the reduction side, moving the trunnion up or down.

2.4. Power Transmission Characteristics

Figure 9 depicts a concept drawing of the power transmission characteristics for the half-toroidal variator

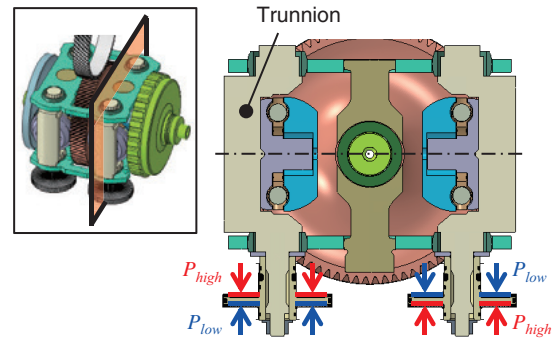


Fig. 8 Shifting piston

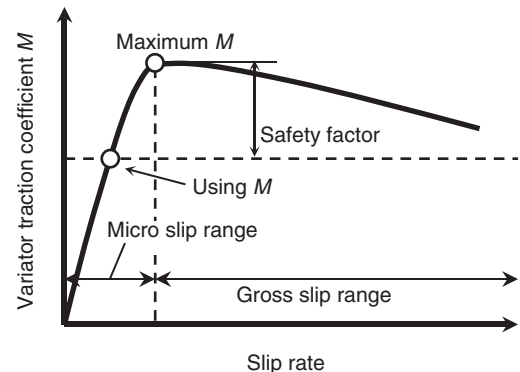


Fig. 9 Traction characteristic of variator

(M -Slip characteristics). Traction coefficient M of the half-toroidal variator is defined as given in Eq. (1):

$$M = \frac{T_{cav}}{n \cdot r_1 \cdot F_c} \quad (1)$$

Here, T_{cav} is the transmission torque per cavity, n is the number of power rollers per cavity, and F_c is the contact force between power roller and input disk. Given their relationship in Eq. (1), the greater the contact force is relative to transmission torque, the smaller M will be. This contact force F_c is reduced by the axial force from the hydraulic loading described in Section 2.2. If axial force is insufficient for the transmission torque, the variator will enter a gross slip and no torque will be transmitted. Conversely, it is known that applying axial force in excess reduces transmitting efficiency⁽¹⁰⁾. In order to establish traction drive and avoid reducing transmitting efficiency, the axial force created should be in step with the operation state in just proportion. Thus, the appropriate safety factor should be required upon pre-determining the limit value of M to operate the variator.

3. Goals and Targets

In order to predict the power transmission characteristics of the half-toroidal variator and use simulations to set the safety factor for gross slip, development proceeded with the following targets:

(1) Create a dynamic behavior analysis technique for

variators that reflects the shear characteristics of the traction fluid

(2) Quantify the relationship between traction coefficient M and transmitting efficiency

(3) Visualize the spin state of the traction surface

4. Dynamic Behavior Analysis Technique for Half-toroidal Variators

4.1. Identification of Traction Fluid Characteristics

The shear characteristics of the traction fluid were identified based on the actual measurement values to be reflected in the simulations. Figure 10 shows a sample relationship⁽¹¹⁾ between traction fluid limiting shear strength τ_L [MPa] and pressure p [GPa]. As shown, the shear strength of the traction fluid is characteristically known to quickly increase under high surface pressure. Also, the shear characteristics change depending on the fluid temperature and slip speed of the two surfaces the fluid flows between. Accordingly, shear force of the traction fluid to be used was measured with surface pressure, temperature, and slip speed as the parameters.

Traction fluid characteristics were measured using the four-roller machine⁽¹²⁾ shown in Fig. 11. Relative slip speed can be controlled by forcing a difference in rotational speed between the three outer rollers, which revolve at the same speed, and the center roller. However, as the rotation axes of the rollers on the four-roller machine parallel one another, no spin as seen in the toroidal variator is produced on the test apparatus traction surface.

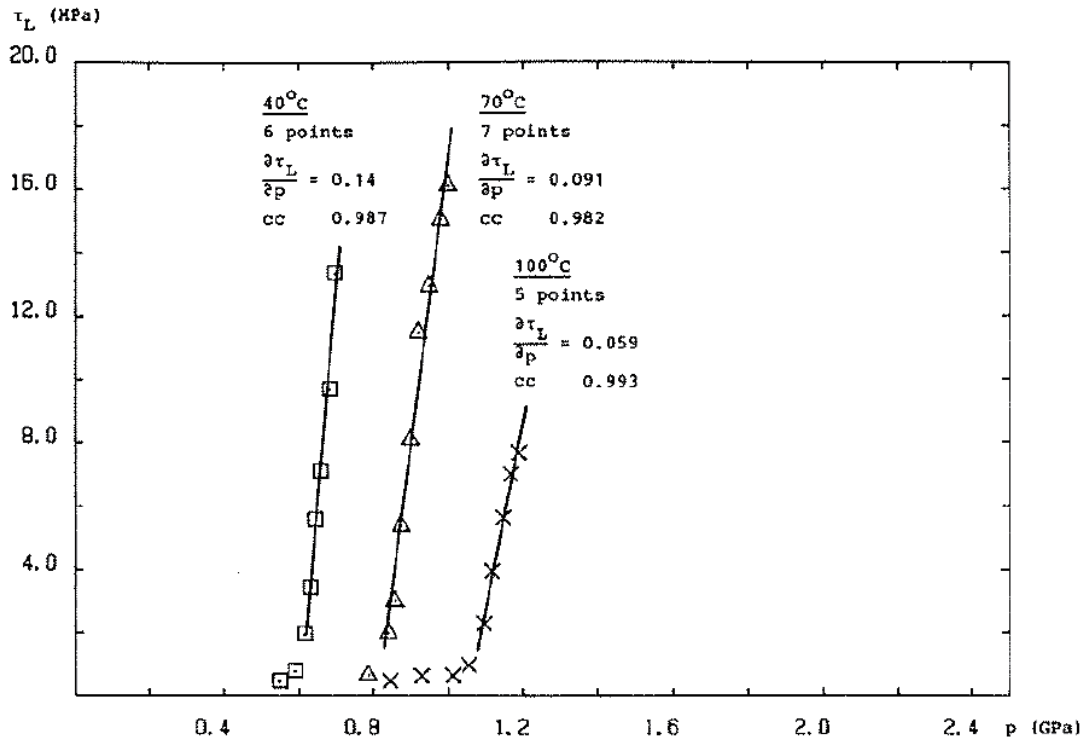


Fig. 10 Limiting shear strength for lubricant No. 9, traction fluid, as a function of pressure and temperature

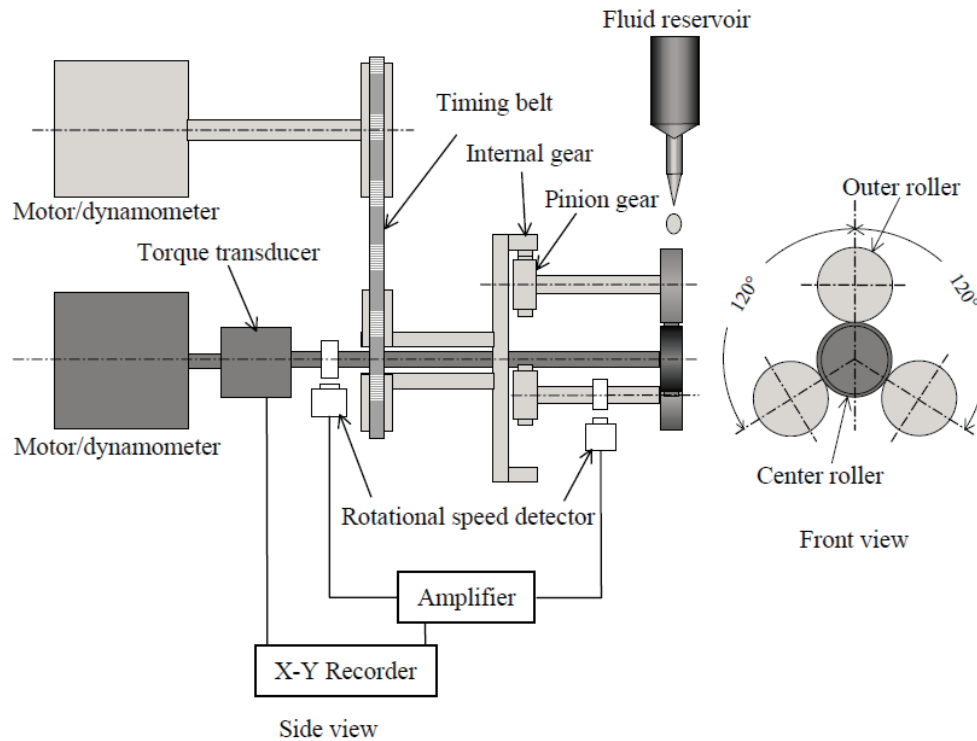


Fig. 11 Structure of the four-roller machine

As traction drive is treated as the model for friction transmission in the behavior analysis model, the shear force of the traction fluid should be replaced with the frictional force of the contact surface. A behavior analysis model was therefore created for the four-roller machine (see Fig. 12) to identify the frictional characteristics for achieving the same torque response as the measured results from Fig. 13. The created model was taken as the flexible body, and the mesh size was taken as the expressible size at which the rollers rolled smoothly.

As a sample friction coefficient identified by the four-roller test model, Fig. 14 gives the coefficient values for each contact pressure at 100°C. While measured traction coefficients are normally given as averaged for the entire contact surface, the friction coefficients given here are values for each nodal point measured on the contact surface.

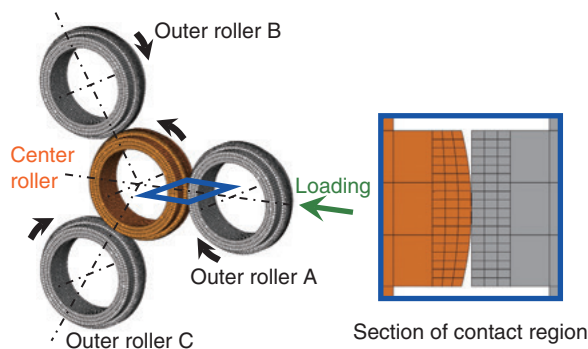


Fig. 12 Simulation model of four-roller machine

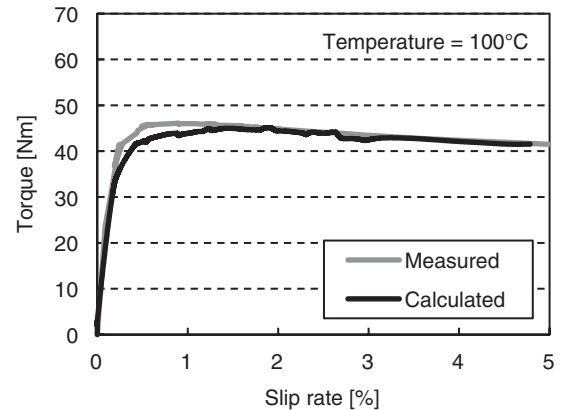


Fig. 13 Comparison between measured and calculated torque

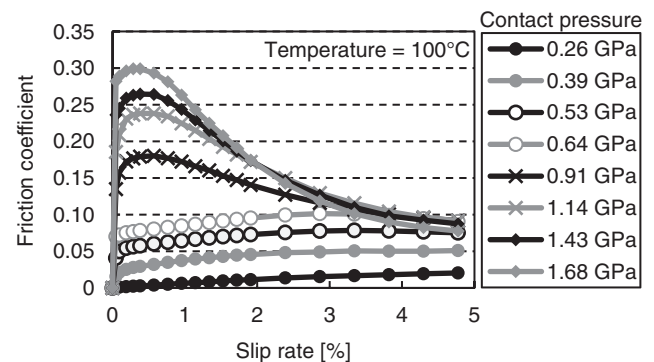


Fig. 14 Friction coefficient of traction surface for simulation model

The identified friction coefficients are high at high contact pressures, perceived to be characteristic of the traction fluid.

4.2. FEM Model for Half-toroidal Variator

Figure 15 shows a half-toroidal variator model using behavior analysis. The contact pressure required to establish traction drive reaches approximately 4 GPa at maximum. Therefore, in actual operation of the half-toroidal variator, the contacted surfaces and other parts operate in a state of elastic deformation. To account for this elastic deformation, FEM was used to model the parts. In order to utilize the traction fluid shear characteristics identified with the four-roller test model, the mesh size for the traction surface was made the same as that for the roller surfaces in the four-roller test model. A predetermined clamping force was applied to the power roller by hydraulically pressurizing the loading pistons. The shift force required for ratio control is generated by directly applying an axial force to the lower part of the trunnion in place of the shifting piston hydraulics. The shift force is controlled for feedback based on the operating state.

A block diagram using variator ratio control is given in

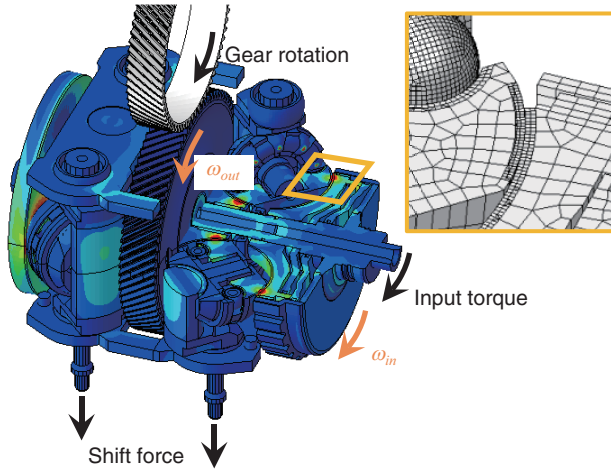


Fig. 15 Simulation model of half-toroidal variator

Table 1 Operating conditions

Input speed	1000 rpm
Input torque	300 Nm
Variator speed ratio i_V	0.430

Fig. 16. The feedback comprises that from ratio deviation and from offset deviation. Speed ratio i_V , representing the ratio of input disk rotation speed ω_{in} and output disk rotation speed ω_{out} , is expressed with the following equation:

$$i_V = \frac{\omega_{in}}{\omega_{out}} \quad (2)$$

For offset deviation, observational estimates were used with no sensors.

The M -Slip characteristics for the variator were predicted by simulation under the operating conditions in Table 1. As illustrated in Fig. 17, values for M were changed by decreasing loading piston pressure at a constant input torque. As shown in Fig. 18, test values generally reproduced the M values predicted in the simulations.

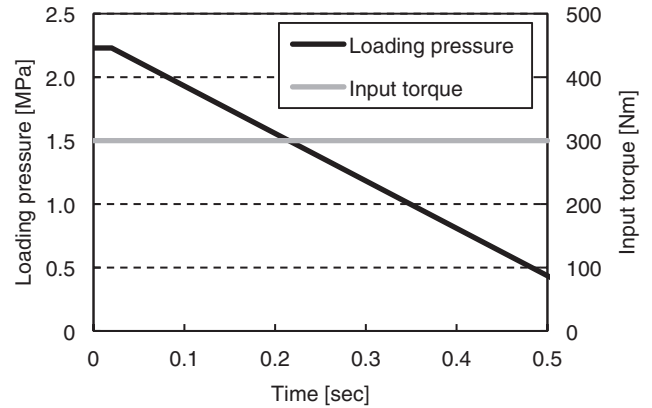


Fig. 17 Operation of loading pressure

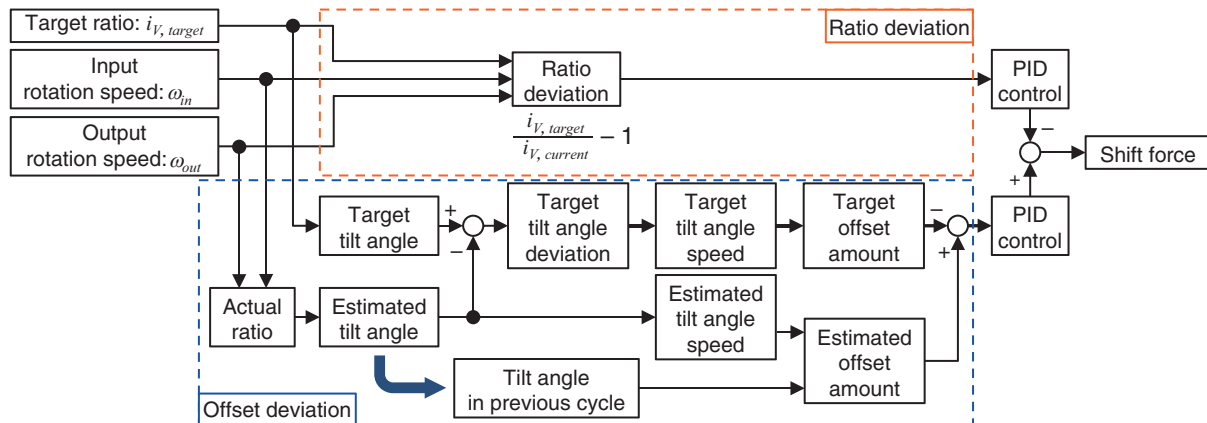
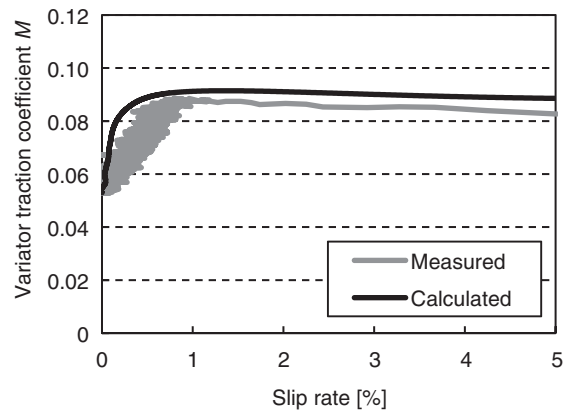


Fig. 16 Speed ratio controller

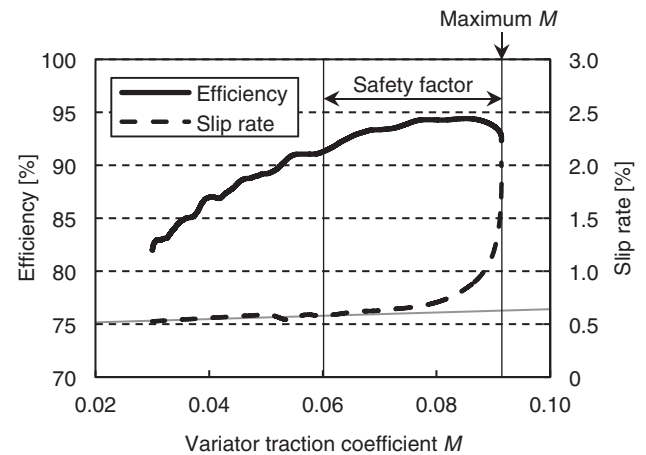
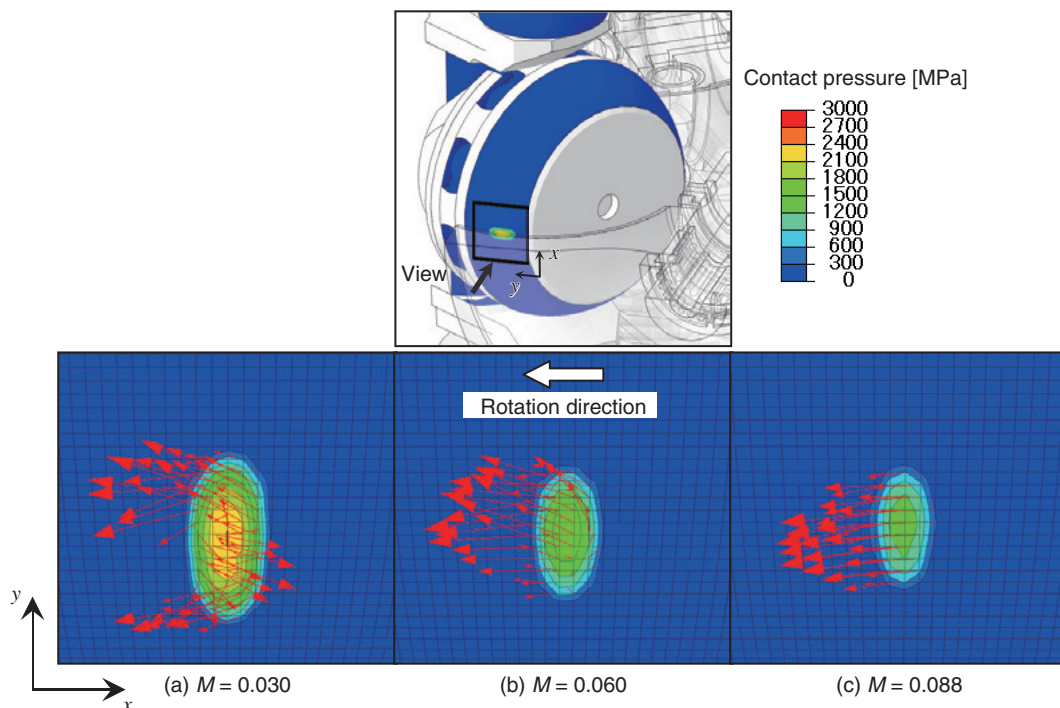
Fig. 18 M -Slip characteristics of variator

5. Safety Factor and Power Transmitting Efficiency

Figure 19 shows the relationship of transmitting efficiency and slip rate for M as obtained in the simulations. From Fig. 9, in the micro slip range, the safety factor for gross slip increases as M decreases. However, as seen in Fig. 19, transmitting efficiency for the variator is low when M is smaller. This is perceived to show that transmitting efficiency decreases the more excessive contact force is applied to the transmitted torque. One possible factor in this is increased spin on the traction surface. The distribution of frictional force on the power roller traction surfaces (see

Fig. 20) shows that spin increases as M decreases. In this way, spin on the traction surface will change according to the contact force, even at a constant torque and ratio. Also, when M is larger, transmitting efficiency is lower in Fig. 19 regardless of how small the traction surface spin. This may be due to the slip rate increasing sharply and more loss due to slippage near maximum M at the right edge of the graph.

The curve for slip rate change is linear at M values of near 0.06 and below and non-linear at M values greater than 0.06. The slip rate will sharply increase, with a tendency to reach gross slip, as M increases due to sudden changes in running resistance of the vehicle when M is larger.

Fig. 19 Relationship of efficiency and slip rate for M Fig. 20 Comparison of friction force vector of traction surface on power roller for difference of M

Thus, operating the variator at M values of approximately 0.06 — the point at which the slip rate curve is linear and transmitting efficiency is high — will retain a safety factor for gross slip with limited decrease in efficiency.

6. Conclusion

Simulations allowed for setting of the safety factor for gross slip of the half-toroidal variator by achieving the following:

- (1) In order to express traction drive with behavior analysis, frictional characteristics for the traction fluid were identified using a four-roller test model. The dynamic behavior, accounting for elastic deformation of the parts, was handled by using an FEM model of the variator on the identified frictional characteristics.
- (2) The power transmission characteristics for the half-toroidal variator (M -Slip characteristics) were predicted using dynamic behavior analysis. This allowed for quantification of the relationship between traction coefficient M and the transmitting efficiency.
- (3) Frictional force vectors for the traction surface at each operating condition were quantified, allowing visual representation of spin during actual operation.

References

- (1) Gaertner, L., Ebenhoch, M.: The ZF Automatic Transmission 9HP48 Transmission System, Design and Mechanical Parts, SAE Int. J. Passeng. Cars – Mech. Syst., 2013-01-1276, (2013)
- (2) Okumura, N.: Development of New Generation CVT with Auxiliary Gear Box, SAE Technical Paper, 2016-01-1109, (2016)
- (3) Imanishi, T.: Study on a half-toroidal infinitely variable power transmission for automobile use, Yokohama National University, Ph.D. thesis, (2008) (in Japanese)
- (4) Ishikawa, K., Inoue, E., Imanishi, T.: Research of CVT efficiency –Comparison of a Toroidal CVT and a Belt CVT–, Review of Automotive Engineering, Vol. 29, No. 4, p. 479-483, (2008)
- (5) Hata, H., Tsubouchi, T.: Development of Traction Fluid for Half-Toroidal CVT Cars, Journal of the Surface Science Society of Japan, Vol. 24, No. 6, p. 351-358, (2003) (in Japanese)
- (6) Sano, T., Tomita, M., Inoue, M., Takeuchi, Y., Yoninaga, M.: Study of the Prediction Method for Maximum Traction Coefficient, SAE Int. J. Passeng. Cars – Mech. Syst., 2013-01-0366, (2013)
- (7) Kishida, H., Imanishi, T.: Study of traction drive performance in Half-Toroidal CVT (First Report) -Measurement of maximum traction coefficient with actual prototype of Half-Toroidal CVT-, JSAE Annual Congress Proceedings, No. 133-09, p. 15-18, (2009) (in Japanese)
- (8) Nakano, M., Kumura, H., Sugihara, J., Mori, H., Maruyama, N.: Development of a Large Torque Capacity Half-Toroidal CVT, SAE Technical Paper, 2000-01-0825, (2000)
- (9) Oishi, Y., Inoue, E., Nishii, H.: Development of wide ratio coverage half-toroidal CVT, JSAE Annual Congress Proceedings, No. 26-12, p. 1-4, (2012) (in Japanese)
- (10) Gecim, B. A.: Modeling and Experimental Validation of Optimized Clamp Loading in a Toroidal-CVT, SAE Technical Paper, 2004-01-0351, (2004)
- (11) Höglund, E., Jacobson, B.: Experimental Investigation of the Shear Strength of Lubricants Subjected to High Pressure and Temperature, ASME, Journal of Tribology, Vol. 108, p. 571-578, (1986)
- (12) Matsuoka, T., Ishida, N., Komatsu, S., Matsuno, M.: Development of Toroidal Traction Drive CVTF for Automobile, SAE Technical Paper, 2002-01-1696, (2002)

■ Author ■



Masahiro TANAKA



Toshihiro SAITO



Koji MIYAMOTO