Development of IVT Variator Dynamic Model

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IVT (Infinitely Variable Transmission) continuously performs a vehicle forward, backward and geared neutral without any starting device by the torque split method. A torque controlled full-toroidal variator, transmitting power by traction drive, is the key component of the IVT that promises quick response and stable behavior of the entire driveline. Koyo, a variator parts supplier, presents in this paper its own variator dynamic model, simulation, an example of parametric study and the description of the contact point trajectory.

Key Words: infinitely variable transmission, geared neutral, torque control, full toroidal

1. Introduction

In the automotive field, recent concern about energy saving and environment conservation has oriented many research for the reduction of fuel consumption. CVT (Continuously Variable Transmission) using belt and traction drive types give some improvements compared to conventional AT (Automatic Transmission) by means of its stepless ratio change. However, these improvements are limited due to the essential use of such high-loss starting devices as a torque converter.

The IVT (Infinitely Variable Transmission), developed by Torotrak Ltd., uses a split torque method that combines a full toroidal variator and an epicyclic gear, in order to avoid any starting device. With IVT-equipped vehicles, it has been confirmed that fuel consumption can be improved by an average of 13% (10% in the city, 18% on the highway) in comparison with a 5-speed MT (manual transmission) vehicle based on the US CAFE (Corporate Averaged Fuel Economy)\(^1\).

The variator is the key component of the IVT allowing the transmission of power and a continuously variable-speed-ratio change. The power is transmitted by traction force produced by fluid shearing stress of highly loaded contacts between disks and rollers reaching a maximum contact pressure of \(1 \sim 3.5 \text{ GPa}\). State-of-the-art bearing technologies become therefore a key requirement regarding the contact point analysis, the development of the variator and especially regarding efficiency and durability issues. In that way, Koyo has achieved results by conducting EHL (Elasto-Hydrodynamic Lubrication) analysis of oil film and in developing materials\(^2\).

On the other hand, the variator is torque controlled. The speed change is therefore a consequent passive mechanism. Then, it is necessary to investigate a change in status (stress, oil film, etc.) of the contact point by accurate simulations of transient responses.

This paper introduces the development of a dynamic variator model and the results of the corresponding simulation conducted independently by Koyo and based on the background described above.

2. IVT Overview

2.1 IVT Basic Configuration

IVT is an automatic transmission capable of changing continuously the ratio in a wide range. As shown in Fig. 1, the IVT driveline consists of a torque split gear (G), a variator (V), a low and high regime clutches (L, H) and an epicyclic gear (E).

![Fig. 1 Example of IVT configuration](image1)

2.2 Full-Toroidal Variator

The IVT variator is a dual-cavity, full-toroidal type as shown in Fig. 2. The input torque is transmitted from the input disks to the output disks via the 6 rollers.

![Fig. 2 Full-toroidal variator](image2)
A hydraulic end load is applied to the back of a disk to clamp the rollers. With the high-pressure contact point between the disk and rollers, high-efficiency power transmission is performed due to the shearing forces of the traction oil in the EHL state.

In order to balance the traction forces and to transmit torque, the rollers are mounted on carriages and hydraulic pistons that provides the right reaction forces.

2.3 IVT Principle

The IVT has 2 modes called low regime and high regime defined by the state of the two clutches L and H. At the low regime, the clutch L is engaged and the clutch H is disengaged. The driveline is as shown in Fig. 3.

![Fig. 3 Low regime](image)

The engine power is split by the split gear. The two paths lead one portion of the power to the sun gear (S) of the epicyclic via the variator, and the other to the carrier (C) from the low regime clutch. These power flows are combined in accordance to the gear ratio \( q \) of the epicyclic gear, and is output to the annulus gear (A). The power flow directions depend indeed on the variator ratio and the hydraulic piston force direction. In Fig. 3, these directions correspond to a forward motion. Notice that power recirculation exists in that case.

The relation of the speeds in an epicyclic gear is given bellow:

\[
\omega_a = -\rho \cdot \omega_s + \omega_v \cdot (1 + \rho) \cdot \omega_c
\]

where:
- \( \omega_a \): Annulus gear angular velocity
- \( \omega_s \): Sun gear angular velocity
- \( \omega_v \): Carrier angular velocity

From the equation 1 and with a fixed engine speed (\( \omega_s = \) constant), the IVT can perform continuously forward motion (\( \omega_a > 0 \)), geared neutral (\( \omega_a = 0 \)) and backward motion (\( \omega_a < 0 \)) without any starting device. This is the origin of the name "Infinitely Variable Transmission".

In the high regime, the clutch L is disengaged and the clutch H is engaged, bypassing the epicyclic gear as shown in Fig. 4. This is the mode corresponding, in a MT, to the third speed to overdrive. Figure 5 shows the plot of the IVT’s speed ratio \( R_{ivt} \) (without the final gear and the differential gear) versus speed ratio \( R_v \) of the variator. Because regime change is carried out at the point S where sun, carrier and annulus gears are synchronized (\( \omega_v = \omega_s = \omega_a \)), no shock is observed when switching the clutches.

![Fig. 4 High regime](image)

![Fig. 5 IVT speed change ratio](image)

In the neutral or parking conditions where the vehicle completely stops, both clutches are disengaged. In this condition, the variator continues to turn, but power is not transmitted.

3. Variator Model

This section describes the variator modeling method developed by Koyo.

3.1 Traction Drive

The torque transmission in the variator is carried out by shearing force of thin oil film between disks and rollers which is generally called traction drive. The traction coefficient \( \mu \) on an ideal contact point (Fig. 6) is the ratio between the traction force \( F_t \) and the load force \( F_l \):

\[
\mu = \frac{F_t}{F_l}
\]

The two disk velocities \( V_s \) and \( V_v \) define the slip ratio:

\[
\frac{\Delta V}{V_v} = \frac{2(V_v - V_s)}{V_v + V_s}
\]

![Fig. 6 Traction drive](image)
A typical traction curve is shown in Fig. 7.

![Traction curve](image)

**Fig. 7 Traction curve**

Based on the Tevaarwerk and Johnson model\(^1\), the traction coefficient is calculated assuming no spin to simplify the process. The direction of shearing stress on that model was the same that the shear strain vector regardless of the elastic and the plastic regions. The sliding direction in the contact point between disks and rollers matches therefore the direction of the traction force.

### 3.2 Roller’s Degrees of Freedom

In Fig. 8, the rollers are supported by roller carriages and are driven by 2-way hydraulic pistons. The roller degrees of freedom are: roller rotation, linear motion of the hydraulic piston and, tilt, pitch and change in castor angle around the roller carriage allowed by a spherical bearing.

![Degree of freedom of a roller](image)

**Fig. 8 Degree of freedom of a roller**

The following assumptions allow the simplification of the roller motion to 3 degrees of freedom.

1) There is no axial movement of the disks. Therefore, the roller center moves along the toroid centerline of the disk.

2) The spherical bearing located at the end of the roller carriage is far enough from the roller center and the pitch and change in castor angle can be ignored.

Finally, the 3 degrees of freedom of the roller are the rotational velocity \(\omega_r\), the tilt angle \(\gamma\) around the roller carriage, and the transfer angle \(\theta\) of the roller center corresponding to the piston linear motion.

### 3.3 Description of Ratio Change Mechanism

**Figure 9** is a simplified diagram showing the balance of the forces acting on a roller in the steady state. These are the traction forces \(F_tr\) at both contact points and the piston reaction force \(F_p\). The balance is expressed as follows:

\[
F_p = \frac{F_tr \cdot \cos \beta}{2}
\]  

(4)

The input and output torques of the variator are defined by the reaction of the traction forces at the contact point position on these disks. This simple relation between variator torques and traction forces allows the control of torque via change in piston pressure (equation 4).

![Force balance in steady state](image)

**Fig. 9 Force balance in steady state**

![Surface speeds of disk and roller at contact point](image)

**Fig. 10 Surface speeds of disk and roller at contact point**

![Forces acting on roller](image)

**Fig. 11 Forces acting on roller**

When the roller unbalance appears due to a change in reaction oil pressure or change in traction \(F_tr\) caused by a change in input and output disk rotational speeds, the contact velocity vectors \(V_d\) and \(V_r\) of the disk and roller become unparallel as shown in **Fig. 10**. Because the direction of traction force matches the difference of the contact velocity vector \(\Delta V = V_d - V_r\), the traction force can be broken down into two components one parallel to the roller rotation \(F_{trr}\) and \(F_{trr}\) and the other perpendicular \(F_{trt}\) and \(F_{trt}\) (**Fig. 11**). This second component tilts the roller until the contact velocity vectors \(V_d\) and \(V_r\) of the disk and rollers once again become parallel. In this situation and with the effect of the castor angle
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4. Simulation Results

The results of simulation using the variator model are described below.

4.1 Ratio Step Response

Figure 14 shows the responses of the tilt angle $\gamma$, the roller center transfer angle $\theta$ and the input and output disk torques $T_i$ and $T_o$ for an input disk rotational velocity step change from 100 to 200 rad/s which corresponds to a ratio step 1:1 to 2:1. All responses are similar to a second order response with a rising time of about 0.02 s. This corresponds to approximately a half rotation of the rollers. The settling time is approximately 0.2 s. From these results, the variator can be considered as having a fast response and being stable in response to a quick ratio change. It also confirms the simulation results announced by Torotrak Ltd.

4.2 Piston Pressure Step Response

Figure 15 shows the variator response for a reaction pressure step of 2.5~3 MPa, whereas disk rotational velocities are fixed (ratio 1:1). The responses of tilt angle $\gamma$ and roller center position $\theta$ show very small amplitudes of movement, but the same mechanism as that in 4.1.
5. Conclusions

This paper discusses the modeling and the simulation of the IVT full-toroidal variator conducted by Koyo. The results are summarized as follows:

1) A dynamic variator model including traction drive was developed in a S-function for use with Simulink.
2) Simulation results of the variator response to a ratio step (1:1~2:1) show a behavior like a second order system with a rising time of 0.02 s and a settling time of 0.2 s. The stability and high-speed response of the variator were confirmed. These also confirm Torotrak Ltd. announcement.
3) Simulation results of the variator response to a piston pressure step (2.5~3.0 MPa) show an extremely fast response of the variator torque suggesting the high-speed response for torque control.
4) Trajectory of a contact point is a spiral like shape with a convergence point.
5) The dependence on the castor angle of the variator response was confirmed.

References


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