

# Development of Transient State Model for CVT HILS

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## ABSTRACT

HILS has been introduced to the CVT ECU development process in an attempt to increase efficiency. To create HILS, it was necessary to develop a CVT HILS model capable of simulating transient states in a CVT shift mechanism, including a hydraulic control system. The calculation burden of the model was required to be low enough to enable real-time simulation. The aim of this project was to develop an experimental model capable of calculating the transient characteristics of the pulley ratio on the basis of pulley ratio, the torque input to the pulley, and the speed of revolution of the pulley; in order to enable simulation of transient states, in particular in kick-down shift. Given the high calculation load of the hydraulic control system model, test data was employed to enable the higher order simultaneous equation model to be replaced by a lower order transfer function model. The newly developed HILS has been successfully introduced to the development process for CVT ECU, resulting in a 30% reduction in the number of procedures required for verification of functioning and setting parameters for programs.

## 1. Introduction

Honda has already applied the ECU development support technology Hardware In the Loop Simulation (HILS) to the development of AT ECU in order to increase efficiency, and the effectiveness of the technology has been reported<sup>(1)</sup>. The development project discussed in this paper attempted to introduce HILS to the CVT ECU development process. In the discussion below, the entire system is termed the CVT HILS system, while the CVT model employed to make the CVT HILS system function is termed the CVT HILS model.

The projected CVT HILS system was to be used in the development process to verify functioning in and set parameters for ECUs after alteration of their CVT control programs and electrical circuitry, prior to their being fitted in actual vehicles. This required a CVT HILS system possessing sufficient calculation accuracy to replace vehicle tests, which necessitated the development of a CVT HILS model capable of simulating transient states of ratio changing of the CVT. However, the calculation load for a model including a hydraulic control system is high<sup>(1)</sup>, making it unlikely to provide sufficient accuracy in real-time simulations using existing Digital Signal Processors (DSP). In addition, while

a report on a simulation of CVT transient state ratio changing does exist in the literature<sup>(2)</sup>, it was not possible to provide sufficient calculation accuracy with respect to the effect of differences in vehicle speed and engine output conditions using the pulley ratio factors employed in that study, in particular when calculating the rate of change of the pulley ratio during sudden shifts such as kick-down shift. It was therefore necessary to obtain factors for the pulley input torque and pulley input speed.

This paper will provide details of the development of a CVT HILS model which possesses sufficient calculation accuracy while enabling real-time simulation.

## 2. Configuration of CVT HILS System

As Fig. 1 shows, the CVT HILS system is composed of five elements: A host computer, a DSP system, an interface box, a relay box and an ECU for CVT control.

The host computer downloads the model to the DSP system and sets the parameters for simulation data, and displays and analyzes the results of the simulation.

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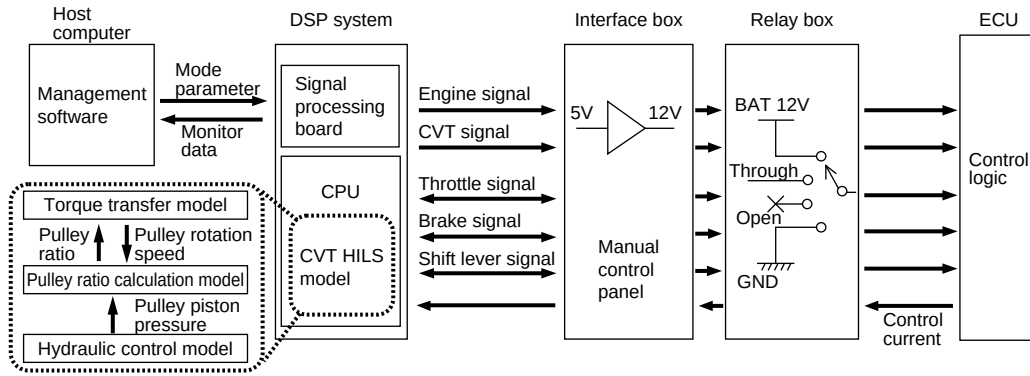


Fig. 1 HILS system configuration

The DSP system is made up of a CPU for model calculation and I/O processing, and a signal processing board, which conducts real-time simulations. The CVT HILS employs multi-rate modeling, with a minimum and maximum calculation frequency of 200  $\mu$ sec and 1 msec respectively.

The interface box conducts the signal processing required for electrical connection between the input and output signals of the DSP system and the signals from the ECU corresponding to conditions in an actual vehicle. The relay box shorts and disconnects the electrical circuits.

The ECU receives the signal from the DSP after it passes through the interface box and the relay box, and conducts shift control at a frequency of 10 msec, sending the results back to the DSP via the relay and interface boxes.

### 3. CVT HILS Model

#### 3.1. Configuration of the CVT HILS Model

The CVT HILS model discussed above in section 2, which is downloaded to the DSP, is basically made up of three models, as shown within the dotted lines in Fig. 1.

- (1) A torque transfer model; including the engine, CVT, and vehicle body
- (2) A hydraulic control model, which outputs the piston pressure for the two pulleys that control shifting
- (3) A pulley ratio calculation model, which changes the speed of the pulleys in response to the pulley piston pressure output by the hydraulic control model

These three models will be discussed in detail below.

#### 3.2. Torque Transfer Model

The torque transfer model is shown in Fig. 2. Engine torque is input via an input shaft, which is directly connected to the drive pulley, and is transferred to the driven pulley by means of a steel belt. The torque is then transferred to the drive shaft by the start clutch via the final gear<sup>(5)</sup>. Torque transfer and the speed characteristics of each shaft are solved by commonly used numerical integration methods, of which the Runge-Kutta method is a representative example<sup>(1)</sup>. The mathematical model is as follows:

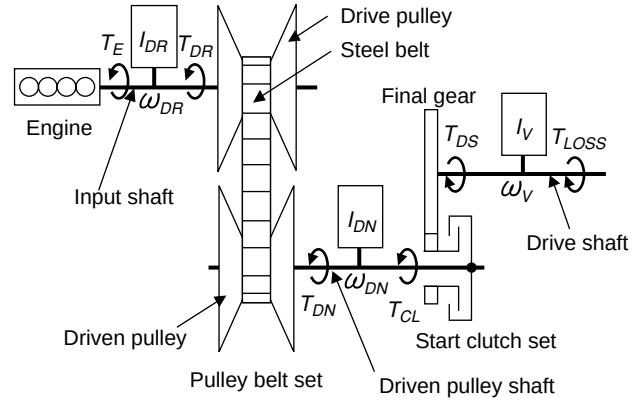


Fig. 2 Torque transfer model

$$\begin{aligned}
 \text{Input shaft} & T_E - T_{DR} = I_{DR} \cdot (d\omega_{DR}/dt) & (1) \\
 \text{Pulley belt set} & T_{DR} \cdot i = T_{DN} & (2) \\
 \text{Driven pulley shaft} & T_{DN} - T_{CL} = I_{DN} \cdot (d\omega_{DN}/dt) & (3) \\
 \text{Start clutch} & T_{CL} = \mu \cdot n \cdot r \cdot (P_{CL} \cdot A_{CL} - F) & (4) \\
 \text{Final gear} & T_{CL} \cdot GR_F = T_{DS} & (5) \\
 \text{Drive shaft} & T_{DS} - T_{LOSS} = I_V \cdot (d\omega_V/dt) & (6)
 \end{aligned}$$

#### Nomenclature

|          |                                                  |
|----------|--------------------------------------------------|
| $I$      | : Inertia                                        |
| $i$      | : Pulley ratio ( $i = \omega_{DR}/\omega_{DN}$ ) |
| $GR$     | : Gear ratio                                     |
| $\omega$ | : Speed                                          |
| $T$      | : Torque                                         |
| $\mu$    | : Clutch disk friction coefficient               |
| $n$      | : Number of clutch disks                         |
| $r$      | : Clutch disk effective radius                   |
| $P$      | : Hydraulic clutch pressure                      |
| $A$      | : Hydraulic clutch piston bearing area           |
| $F$      | : Clutch return spring load                      |

#### Subscripts

|        |                                     |
|--------|-------------------------------------|
| $E$    | : Engine                            |
| $DR$   | : Drive pulley set (Pulley input)   |
| $DN$   | : Driven pulley set (Pulley output) |
| $CL$   | : Start clutch                      |
| $F$    | : Final gear                        |
| $V$    | : Body                              |
| $LOSS$ | : Running resistance                |

### 3.3. Hydraulic Control Model

As shown in Fig. 3, the CVT hydraulic control model consists of a line pressure control section and a drive and driven pulley piston pressure control section<sup>(5)</sup>.

In the line pressure control section, the supply pressure generated by the oil pump is regulated to match the line pressure by the regulator valve. Line pressure is reduced by the linear solenoid valves operated by the control current of the electronic control unit (ECU) and the pulley control valves in the pulley piston pressure control section, thus enabling the pulley piston pressure on the drive and driven sides to be controlled.

Two issues were encountered in developing a hydraulic control model.

The first was the fact that the mechanism generating the time lag in pulley piston pressure response to ECU hydraulic pressure commands in transient states was unclear.

The second was that transient states necessitated a higher-order hydraulic control model with a high calculation load.

Figure 4 shows the results of a vehicle test in which the accelerator pedal was suddenly depressed from a fixed vehicle speed of 40 km/h (kick-down shift). The shaded sections in Fig. 4 show a considerable time lag in pulley piston pressure response to the ECU hydraulic pressure command.

An actual vehicle was employed in tests conducted using the setup shown in Fig. 5 in order to analyze the cause of this phenomenon for modeling.

These tests used a multipurpose ECU to provide patterned hydraulic control commands for pulley piston pressure. The ECU was programmed to increase the pressure command to the driven side in steps at specific engine speeds. The pressure command to the drive side was maintained at a fixed value. The multipurpose ECU was fitted in the vehicle, and the line pressure, pulley piston pressure, engine speed, and pulley ratio were measured at different engine speeds. Figures 6 and 7 show the results of measurements at 1 000 and 2 500 rpm respectively.

The change in pulley ratio between 1 and 2 seconds was focused on in both sets of results.

In the first frame of the graphs it can be seen that the pressure of the driven pulley piston, to which oil is flowing, is lower than the hydraulic pressure command. In the second frame of the graphs, it can be seen that the pressure of the drive pulley piston, from which oil is flowing, is higher than the hydraulic pressure command. Changes in the volume of the pulley piston are known to be a factor generating such changes in pressure with the flux and efflux of oil<sup>(2)</sup>.

The effect of the oil supply capacity of the oil pump was studied, with consideration of these changes in volume. A comparison of the first frames of the graphs shows that at 1 000 rpm (Fig. 6) the change in ratio is gentler than at 2 500 rpm (Fig. 7). Despite the minimal influence of a change in volume of the driven pulley piston, the piston pressure is significantly less than the hydraulic pressure command and the line pressure and piston pressure are equivalent,

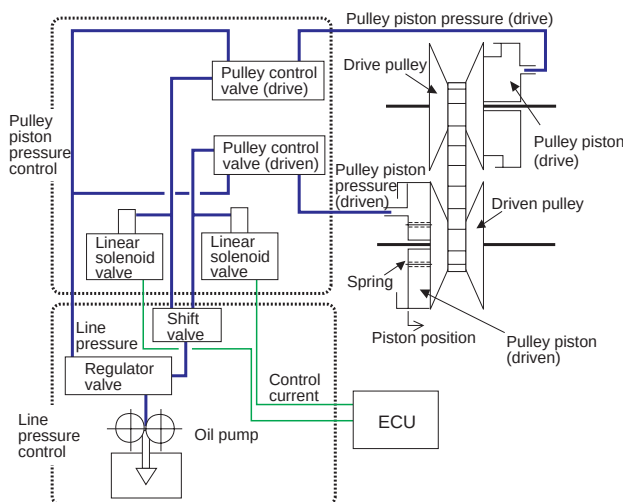


Fig. 3 Schematic drawing of CVT system

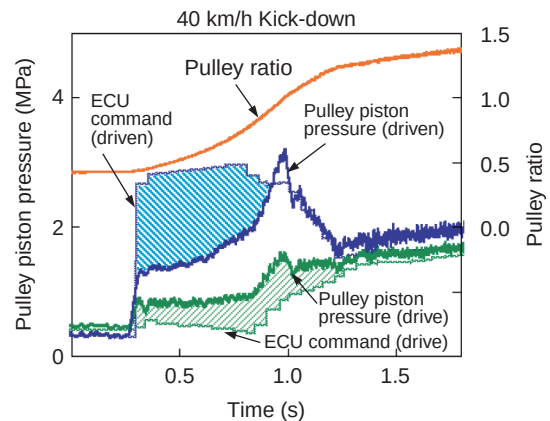


Fig. 4 Vehicle test results

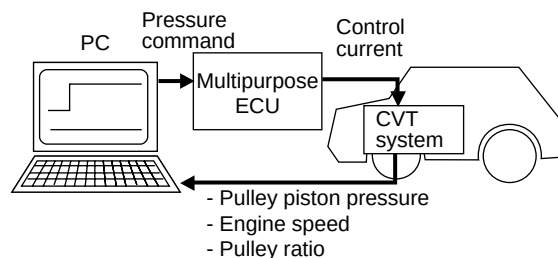


Fig. 5 Schematic drawing of test equipment

indicating that factors other than piston volume change are at work. This effect was believed to be due to the inability of the oil pump to respond to the ECU pressure command, which was increased in steps.

The hydraulic control model, as shown in Fig. 8, was constructed on the basis of these considerations. In addition to a hydraulic response paragraph, based on a standard oil inertia<sup>(3)</sup> derived from hydrodynamics, and a piston volume change paragraph which takes into account changes in the volume of the pulley piston, the hydraulic control model contains an amount of flow from oil pump paragraph, in which the oil supply capacity of the pump is

considered. These paragraphs determine the pulley piston pressure, but the method of calculation can be switched according to the type of shift being conducted by the CVT. For slow shifts the pulley piston pressure is determined by the hydraulic response and piston volume change paragraphs; for rapid shifts such as kick-down shift, when the pulley piston pressure generated by the amount of oil flow from oil pump paragraph drops below the pulley piston pressure generated by the sum of the piston volume change paragraph and hydraulic response paragraph, the amount of oil flow from oil pump paragraph is selected.

The method of modeling the three paragraphs will now be discussed in more detail.

The hydraulic response paragraph is a model which calculates the pressure drop due to oil inertia, assuming no change in the volume of the pulley piston. The model was created by identification from vehicle tests. Specifically, a frequency response test was conducted to identify the relationship of pulley piston pressure to the hydraulic pressure command in order to identify the effect of oil inertia. As a result, it was determined that it could be approximated by a secondary-order time-lag transfer function. The transfer function is shown as Eq. (7).

$$\frac{1}{\alpha Ts^2 + \alpha s + 1} \quad (7)$$

Here,  $\alpha$  represents the model parameter and  $T$  the time constant.

Figure 9 shows the results of pulley piston pressure calculated using the transfer function and the results of vehicle measurements for the same hydraulic pressure command. There is a good correlation between the two sets of results, which demonstrates the validity of the hydraulic response paragraph.

The piston volume change paragraph is a model which calculates changes in pressure with the flux and efflux of oil due to changes in the volume of the pulley pistons. Because hydraulic response could be approximated with a second-order time-lag transfer function, as discussed above, and the continuity equation expressing the relationship between the amount of oil flow from the control valve ( $Q_v$ ) and the pulley piston pressure is a first-order function<sup>(2)</sup>, linear transformation of the relationship between the spool operation power for the pulley control valve ( $F_v$ ) and  $Q_v$  is possible using a first-order time-lag function, as shown in Fig. 10.

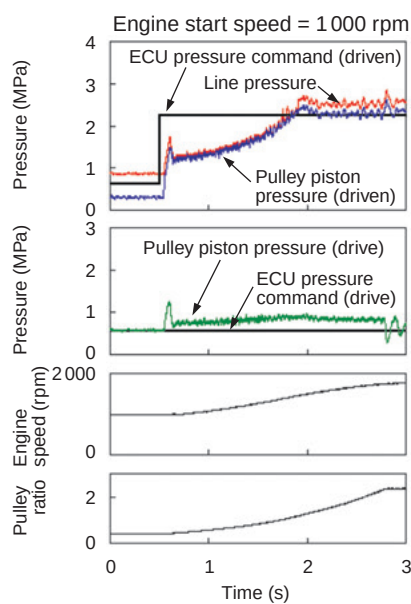


Fig. 6 Step shift test results at low engine speed

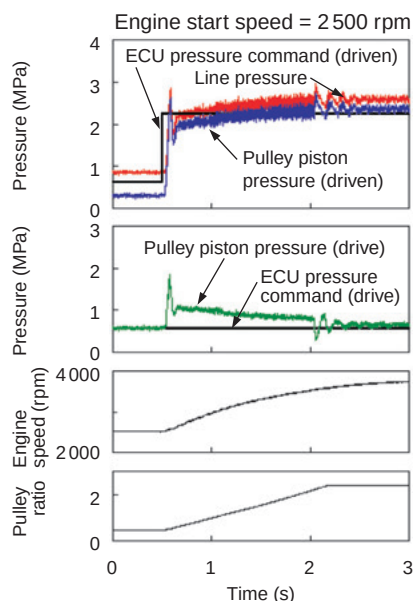


Fig. 7 Step shift test results at high engine speed

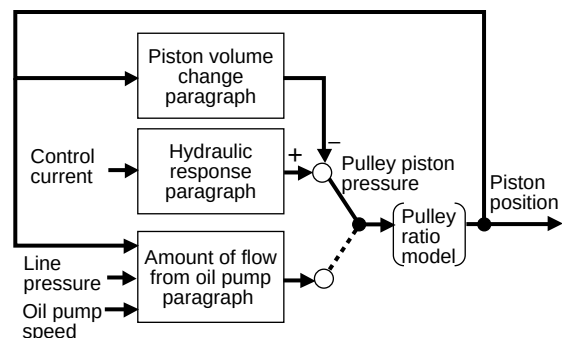


Fig. 8 Hydraulic control model

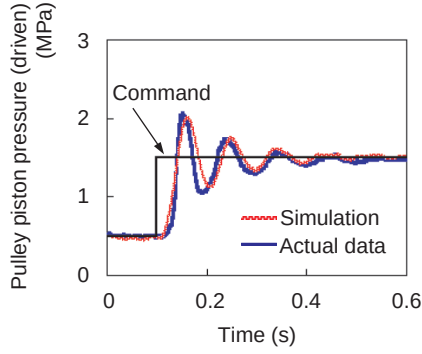


Fig. 9 Simulation results of hydraulic response paragraph

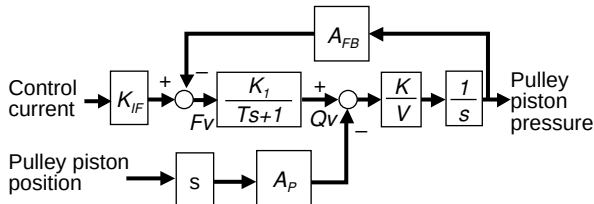


Fig. 10 Hydraulic response and piston volume response function

Here,

$V$  : Pulley piston volume

$A_p$  : Area of pulley receiving pressure

$A_{FB}$  : Area of pressure circuit feedback

$K_{IF}$  : Current-electromagnetic power gain

$K_i$  : Spool operation power-flowing quantity gain

$T$  : Spool operation power-flowing quantity time constant

$K$  : Modulus of elasticity of volume

$F_v$  : Spool operation power for pulley control valve

$Q_v$  : Quantity of flow through pulley control valve

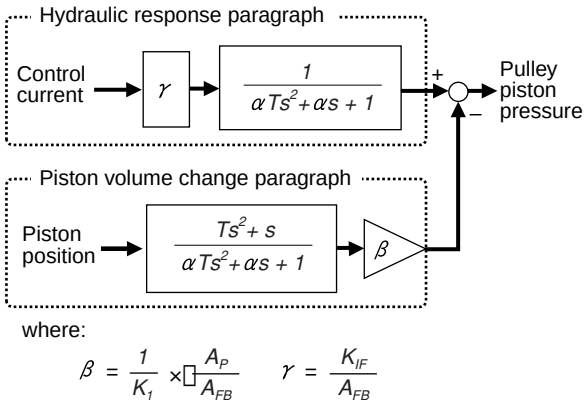


Fig. 11 Hydraulic response and piston volume response paragraph

Figure 11 shows the model generated by consolidating the model in Fig. 10 with model parameters  $\alpha$  and time constant  $T$ .

The piston volume change paragraph in the model shown in Fig. 11 can be modeled as the second-order transfer function shown below (Eq. 8).

$$\frac{Ts^2 + s}{\alpha Ts^2 + \alpha s + 1} \quad (8)$$

The results of vehicle tests and simulations employing conditions like those shown in Fig. 7, with high oil flow from the oil pump at high engine speed, demonstrated the validity of the piston volume change paragraph. To verify the validity of the hydraulic control model as a whole, results from vehicle tests were employed for the pulley ratio time series characteristics, one of the input parameters employed in the simulation. Figure 12 shows the results of the simulation and vehicle tests. The good correlation between the two sets of results demonstrates the validity of the piston volume change paragraph.

The amount of flow from oil pump paragraph is determined by the difference between the rate of flow from the oil pump (supply capacity) and the amount of pulley piston volume change against the hydraulic pressure command. When the rate of flow is less than the amount of volume change, the pulley piston pressure cannot change to match the hydraulic pressure command. In the model, when this is the case, pulley piston pressure is determined by the amount of flow from oil pump paragraph. The model is shown in Fig. 13.

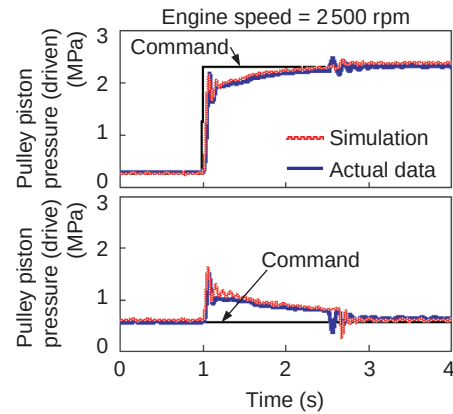


Fig. 12 Simulation results of piston volume change paragraph

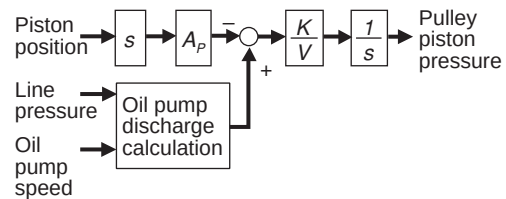


Fig. 13 Amount of flow from oil pump paragraph

The results of vehicle tests and simulations employing conditions of low oil flow from the oil pump at low engine speed, like those shown in Fig. 6, demonstrated the validity of the model used in the amount of flow from oil pump paragraph. As was the case in tests of the validity of the piston volume change paragraph, vehicle test results were used for the pulley ratio time series characteristics in the simulation. Figure 14 shows the results of the simulation and vehicle test. The good correlation between the results demonstrates the validity of the amount of flow from oil pump paragraph.

Having confirmed the validity of the hydraulic response paragraph, the piston volume change paragraph, and the amount of flow from oil pump paragraph, they were employed as a hydraulic control model.

### 3.4. Pulley Ratio Calculation Model

As Fig. 3 shows, in a CVT shift mechanism a steel belt is clamped by a drive and a driven pulley. The clamping force is the total of pulley piston pressure, centrifugal pressure force and the force generated by the springs in the pulley pistons<sup>(4)</sup>. The total

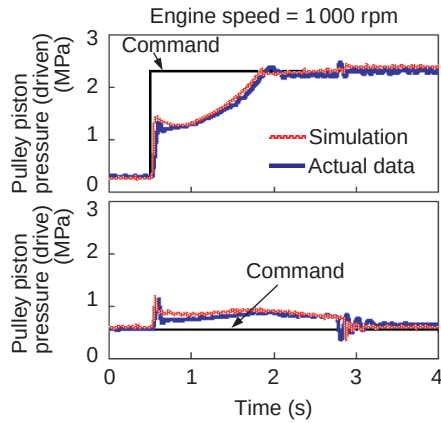


Fig. 14 Simulation results of amount of flow from oil pump paragraph

force clamping the belt is called total thrust, and the ratio of total thrust of the drive to the driven sides is called the total thrust ratio. The ratio between the maximum torque that can be applied without causing the steel belt to slip on the pulleys and the torque input to the pulleys is called the torque ratio<sup>(5)</sup>.

A method of calculating CVT pulley ratio has been discussed several times in the literature to date<sup>(2),(5)</sup>. In this method, the pulley ratio calculation model consists of a steady state part, which calculates conditions of equilibrium in which the pulley ratio does not change, and a transient part, which calculates the rate of change in the pulley ratio when disequilibrium occurs in the system. The remainder of this section will discuss the steady state and transient parts in more detail.

In the steady state part, the pulley ratio is in a state of equilibrium and does not change. Tests have indicated that the total thrust ratio that produces this state can be unambiguously defined by torque ratio and pulley ratio. Pulley piston pressure, pulley input speed and pulley input torque are entered in the model and the degree of discrepancy between the induced total thrust and the total thrust in a state of equilibrium is calculated. This is called the thrust for shift. The thrust for shift causes the pulley ratio to change until the next state of equilibrium is reached.

The factor that determines the rate of change in the pulley ratio generated by the thrust for shift is calculated in the transient part. A factor for pulley ratio that can function as an argument for calculation of this factor is known<sup>(2)</sup>. However, use of this factor for pulley ratio alone does not enable the rate of change of the pulley ratio to be calculated with a high degree of accuracy, particularly when analyzing sudden shift transient states such as kick-down shift in an actual vehicle. This was hypothesized to be due to changes in the friction between the pulley and the steel belt with torque input to the pulley and changes in the centrifugal load on the steel belt due to changes in the pulley input speed. Figure 15 shows the pulley ratio calculation model.

To test the hypothesis mentioned above, the test equipment shown in Fig. 16 was constructed to measure the torque input to the pulley in an actual vehicle. The measurement equipment employed the flywheel. The friction plate and filled grease, which

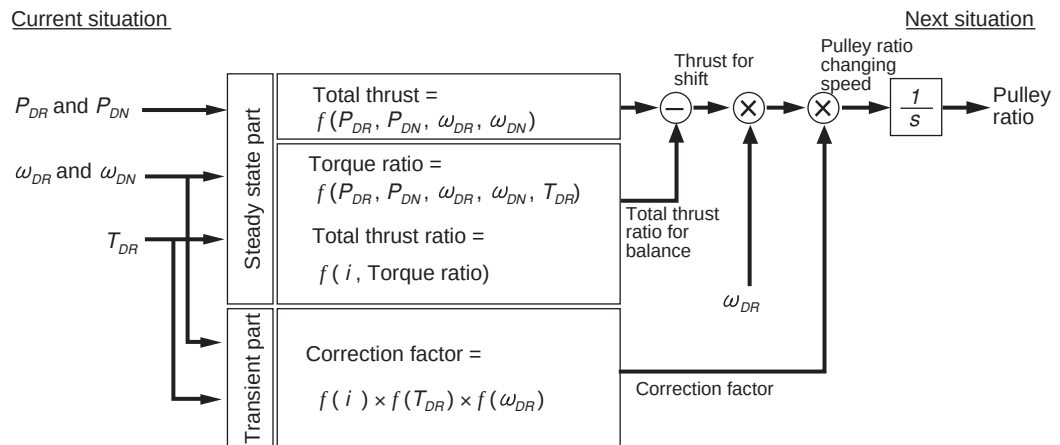


Fig. 15 Pulley ratio calculation model

acted as damping elements, were removed from the flywheel. And the flywheel was grooved on the input and output side to measure turn pulse. The phase difference between the input and output pulses was then determined. A torsional angle converter was employed to determine the torsion angle of the flywheel from this phase difference. The torsion angle was then converted to torque using

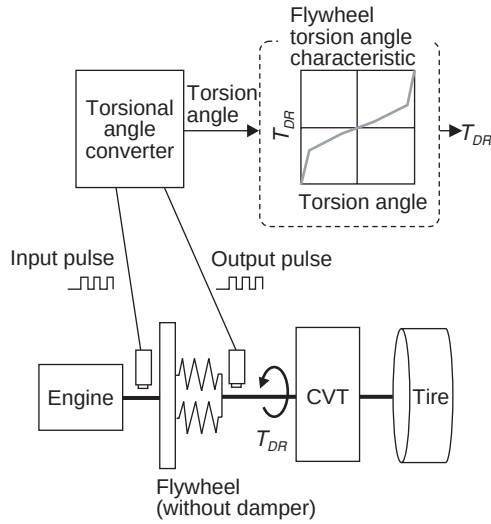


Fig. 16 Schematic drawing of test equipment

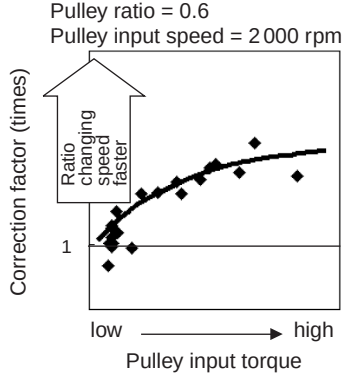


Fig. 17 Newly developed factor for pulley input torque

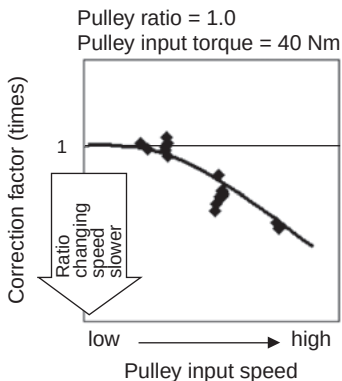


Fig. 18 Newly developed factor for pulley input speed

previously measured data on the flywheel torsion angle and torque characteristic. By this means, a factor was obtained enabling calculation of the rate of change of pulley ratio in sudden shift transient states such as kick-down shift.

The results of these measurements are shown in Figs. 17 and 18. Figure 17 shows a plot of the factor for pulley input torque and Fig. 18 a plot of the factor for pulley input speed. The factor which is shown on the vertical axis is an integrating factor for the rate of change of pulley ratio, as discussed above. When it is greater than one, it shows a high rate of change, and when it is less than one it shows a low rate of change. That is, the rate of change of the pulley ratio becomes higher as input torque increases, and becomes lower as the pulley input speed increases. This characteristic indicates the validity of the hypothesis discussed above.

### 3.5. Accuracy of the CVT HILS Model

Figure 19 shows vehicle test results and the results of simulations employing the CVT HILS model (Fig. 1) composed of the torque transfer model, hydraulic control model and pulley ratio calculation model discussed above.

Good correlation is displayed between vehicle test and simulation results for pulley piston pressure and rate of change of pulley ratio, demonstrating the achievement of a level of calculation accuracy in the simulation of CVT shift transient states enabling them to replace vehicle tests. The employment of the newly developed hydraulic control model and pulley ratio calculation model has enabled the creation of a CVT HILS model with a calculation burden low enough for real-time simulation with existing DSP.

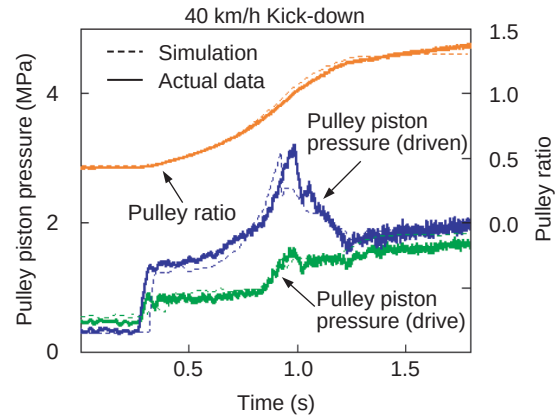


Fig. 19 Simulation results

## 4. Results of Application of Model

The time required for initial performance verification and parameter setting for CVT control programs in the newly developed ECU when employing the newly developed CVT HILS system was compared with the conventional process employing an actual vehicle.

Despite the fact that preparation of the CVT HILS system, including input of the design specifications of the target vehicle and connection of the ECU and interface box, required one day, performance verification and parameter-setting on the desktop required 30% less time than when a vehicle is employed.

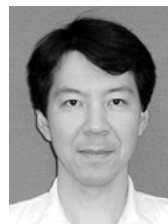
## 5. Conclusion

- (1) Test data has been employed to enable the higher-order simultaneous equation model conventionally used for the high calculation load hydraulic control model with a lower-order transfer function model, in order to allow real-time simulation using existing DSP. This has resulted in the creation of CVT HILS model with a calculation burden that does not exceed the processing capacities of existing DSP.
- (2) The employment of an experimental model that calculates the transient characteristics of the pulley ratio on the basis of pulley ratio, pulley input torque, and pulley input speed has enabled the creation of a CVT HILS model capable of simulating transient states in a CVT shift mechanism.
- (3) Initial performance verification and parameter-setting of new ECU can now be performed using the CVT HILS system prior to vehicle testing, reducing the time required by 30%.

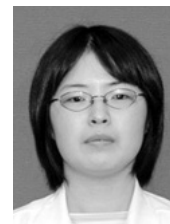
## References

- (1) Takeda, Y., Yoda, K., Terayama, S., Hagiwara, K., Saito, Y., Kamiyamaguchi, T.: Application and Utilization of HILS for the Development of an Automatic Transmission Control System, Honda R&D Technical Review, Vol. 14, No. 2, p. 99-106
- (2) Ide, T., Udagawa, A., Kataoka, R., Uchiyama, H.: A Study on a Response of a Metal V-Belt CVT, SUBARU TECHNICAL REVIEW, Vol. 22, p. 50-55 (1994)
- (3) Ishikawa, K., et al.: The Development and Utilization of ACTASS, AVEC'94, p. 224-229 (1994)
- (4) Osono, K., Aoki, T.: Development of Full-electronic Control Metal Belt CVT, HONDA R&D Technical Review, Vol. 8, p. 21-33
- (5) Ohta, S., Kunii, H., Sumi, M., Fujimoto, S., Yamaguchi, T.: Development of a Compact CVT for a Hybrid Car - Insight, HONDA R&D Technical Review, Vol. 12, No. 2, p. 61-70

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