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DYNAMICS OF A POWERTRAIN WITH HALF-TOROIDAL TYPE CVU DURING HARD ACERLERATIONS

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ABSTRACT

The paper presents an investigation into the effect of Half-Toroidal CVU dynamics on the transient behaviour of a completed powertrain under the 80% throttle condition. Description of the derivation of model of the H-T CVU as well as that of a complete powertrain is provided. Simulations were carried out to gain a good understanding of the transient behaviour of the powertrain as well as its components. For simplicity, a piecewise curved gear ratio map is assumed in the simulation. The obtained results show that significant transient exist in complete powertrain during a hard vehicle acceleration. The presented simplified H-T CVU model describes its effect to the dynamics of a completed powertrain effectively.

1. INTRODUCTION

Increase in fuel economy coupled with other advantages like smoothness of operations, easy driveability, powerful acceleration and an infinite range of gear ratios have made CVT an attractive proposition for automobile industry. Osumi et al [1] developed a dynamic model for the half toroidal variator and hydraulic subsystems that are mated to a powertrain. The results obtained were analyzed for geared neutral condition, launch and regime change characteristics and they showed transient for regime change and correlates well with the experimental data. James [2] carried out dynamic simulations for a Full Toroidal (FT) type infinitely variable transmission (IVT). The major thrust of his work is on dynamic simulation and experimental validation of the full toroidal variator integrated powertrain and hydraulic control systems for regime changes. Jones et al [3] developed a model for Perbury type FT-CVU and integrated it into a conventional powertrain model for simulation of the control system and fuel consumption studies. A modular approach has been used to model the CVT powertrain that consists of individual models for the engine, transmission, vehicle, control system and a driver model. The results obtained for the driving schedule, driver and powertrain control signals had been compared with data obtained from testing on an experimental vehicle equipped with a Perbury FT-CVT.

This paper attempts to investigate the effect of Half-Toroidal Continuously Variable Unit (HT-CVU) dynamics on the transient behaviour of a completed powertrain, the simulations were carried at 80% throttle condition as this condition has been identified as having the most unsatisfactory drive feel. The mechanism of the H-T CVU is described and its simplified dynamic model is proposed that effectively describes the dynamic interaction between rollers and toroids of the H-T CVU. The dynamic model of a complete powertrain is then assembled from parametric finite elements that are formulated using lumped mass and linearized torsional stiffness, damping. The kinematic relationship between gear elements has been modeled using gear ratios. A piecewise curved gear ratio map is assumed in the simulations. The inputs such as engine torque map, various resistant torques applied to other powertrain components that are almost the same as those used for real vehicle powertrain design analysis were use in the presented simulations. The obtained results have been discussed in details and concluding remarks have been provided.

2. SIMPLIFIED PHYSICAL AND MATHEMATICAL MODEL OF HT-CVU

The schematic geometry of a HT-CVU is as shown in Figure 1. A H-T CVU consists of an input toroid, power rollers and an output toroid. Both the toroids are in contact with two or three power rollers (disks). Power is transmitted by the shear action of elastohydrodynamic lubricant (EHL) between the toroids and rollers. The input of the CVU is directly linked to the output shaft of the engine. The output shaft of the CVU is linked to the input shaft of the planetary gear sets of the transmission. The speed ratio between the engine output shaft and the transmission output shaft changes continuously when the vehicle speed changes significantly within in a short time. The complete powertrain is therefore a parametric system of which dynamic characteristics changes with respect to the vehicle speed, and consequently its dynamic behaviour is significantly affected by dynamics of the CVU.

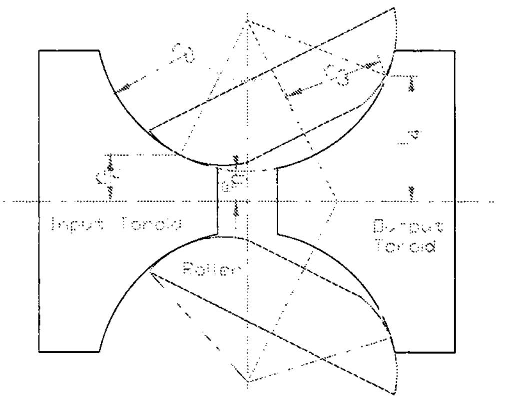

Figure 1: HT-CVU schematic

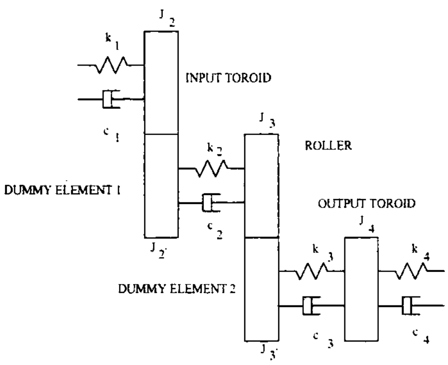

Figure 2: Discrete mass, stiffness and damping representation of HT-CVU

The contact between the toroid and the roller of the CVU is modelled by incorporating values for contact torsional stiffness and contact torsional damping in the model. A typical H-T CVU in a completed powertrain is then modelled using parameters of lumped mass moment of inertia, torsional stiffness and damping coefficients, and the speed ratio between input shaft and output shaft. The simplified physical model for the HT-CVU is shown in Figure 2. It is assumed that the rollers are synchronized and lumped mass model for the HT-CVU is constructed for a single roller.

The speed ratios between the input toroid and roller (R_1) and output toroid and roller (R_2) have been modeled using dummy elements $J_{2'}$ and $J_{3'}$ respectively. The dummy elements have zero inertia value. The contact between the input toroid and roller and the output toroid and roller has been modeled by using linear contact torsional stiffness (k_2, k_3) and linear contact torsional damping (c_2, c_3). The gear ratios (R_1, R_2) are

$$R_1 = \frac{\omega_{2'}}{\omega_2}, R_2 = \frac{\omega_{3'}}{\omega_3}.$$

The inertia, damping and stiffness element matrices and element co-ordinate vectors for dynamic analysis have been obtained by using the line finite element and are given as:

$$\mathbf{J}_{e2} = \begin{bmatrix} R_1^2 J_{2'} & 0 \\ 0 & J3 \end{bmatrix}, \quad \mathbf{C}_{e2} = \begin{bmatrix} R_1^2 c_2 & -R_1 c_2 \\ -R_1 c_2 & c_2 \end{bmatrix}, \quad \mathbf{K}_{e2} = \begin{bmatrix} R_1^2 k_2 & -R_1 k_2 \\ -R_1 k_2 & k_2 \end{bmatrix}, \quad \bar{\boldsymbol{\theta}}_{e2} = \begin{Bmatrix} \theta_2 \\ \theta_3 \end{Bmatrix} \tag{1}$$

$$\mathbf{J}_{e3}=\begin{bmatrix} R_2^2 J_{3'} & 0 \\ 0 & J_4 \end{bmatrix},\quad \mathbf{C}_{e3}=\begin{bmatrix} R_2^2 c_3 & -R_2 c_3 \\ -R_2 c_3 & c_3 \end{bmatrix},\quad \mathbf{K}_{e3}=\begin{bmatrix} R_2^2 k_3 & -R_2 k_3 \\ -R_2 k_3 & k_3 \end{bmatrix},\quad \bar{\boldsymbol{\theta}}_{e3}=\begin{Bmatrix} \theta_3 \\ \theta_4 \end{Bmatrix} \tag{2}$$

These element dynamic coefficient matrices vary when the speed ratio changes and therefore, the HT-CVU is a parametric sub-system. The mathematical representation of the HT-CVU in equations (1) and (2) will be integrated into the dynamic model of the completed powertrain.

Due to lack of experimental and published data the values of the contact torsional stiffness (k_2, k_3) for surface contact between the toroids and rollers have been assumed to be the same value for third clutch lock-up taken from Zhang et al [4]. The values for contact torsional damping (c_2, c_3) have been taken from Boedo and Freyburger [5] and Patterson [6].

3. POWERTRAIN MODEL

The powertrain consists of an engine, a HT-CVU, clutches and planetary gear sets which are used to expand the effective gear ratios available for all operating conditions encountered during normal running of the vehicle, driveline, components including wheels and tyres. The HT-CVU is located between an engine and clutches and planetary gear sets of a conventional powertrain. For torsional vibration analysis, the model of a completed powertrain can be represented by connected lumped masses, torsional springs and damping elements and is shown in Figure 3.

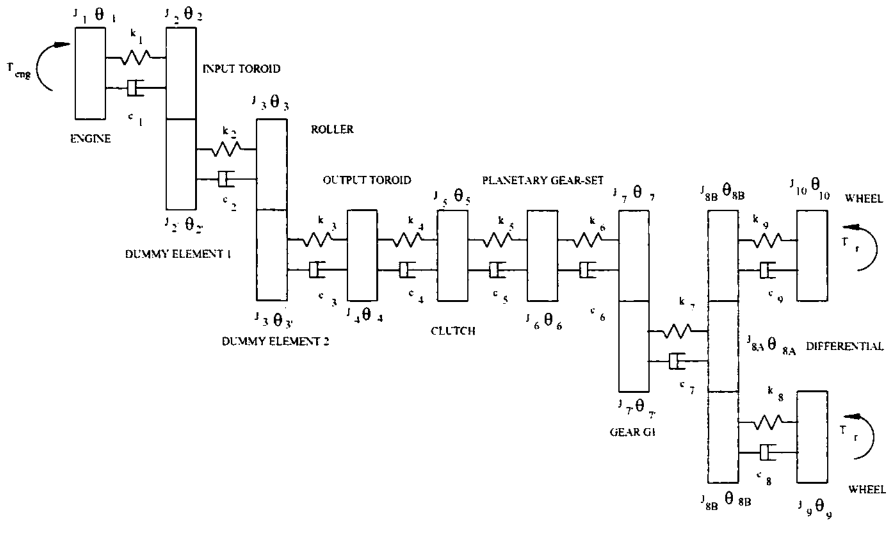

Figure 3: Lumped mass model of HT-CVU integrated powertrain

The differential is modelled with the independent co-ordinates of the pinion and the planetary gear train is modelled with the independent co-ordinates of the sun gear. The gear G_1 is modelled with the independent co-ordinates of the drive pinion. The gear ratios engaging gears, half-toroidal and rollers are determined as follows:

$$R_1=\frac{\omega_{2'}}{\omega_2}, R_2=\frac{\omega_{3'}}{\omega_3}, R_3=\frac{\omega_{6B}}{\omega_{6A}}, R_4=\frac{\omega_{7'}}{\omega_7}, R_5=\frac{\omega_{8B}}{\omega_{8A}}$$

Nine elements represented in terms of mass moment of inertia coefficient matrix, torsional stiffness coefficient matrix and damping coefficient matrix are obtained from individual element design specifications and gear ratios. There are ten independent coordinates in the powertrain. The equation of motion of the complete powertrain system is obtained by assembling these lumped mass and single stiffness elements into the global system.

$$\mathrm{J}\ddot{\boldsymbol{\theta}}+\mathrm{C}\dot{\boldsymbol{\theta}}+\mathrm{K}\boldsymbol{\theta}=\mathrm{T} \tag{3}$$

where the global co-ordinate and global torque vectors are as follows:

$$\boldsymbol{\theta}=\{\theta_1\ \theta_2\ \theta_3\ \theta_4\ \theta_5\ \theta_6\ \theta_7\ \theta_8\ \theta_9\ \theta_{10}\}^{\mathrm{T}}$$

$$\boldsymbol{T}=\{\mathrm{T}_{eng}\ 0\ 0\ 0\ 0\ 0\ 0\ 0\ {-\mathrm{T}_r}\ {-\mathrm{T}_r}\}^{\mathrm{T}}$$

Free vibration analysis of the powertrain system is based on equation (3) (let T=0) and was reported by Dutta-Roy and Zhang [7].

4. TRANSIENT ANALYSIS

For the transient analysis based on numerical simulations, the reduced first order coupled differential equation can be approximately solved using various numerical methods. The Runge-Kutta 4th Order method was used to solve the reduced first order equation. The inputs such as engine torque map, various resistance torque of a vehicle powertrain are the same or simular to those presented in reference [4].

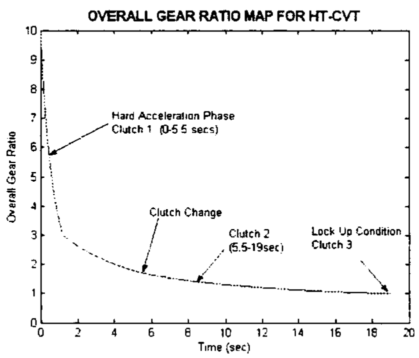

Figure 4: HT-CVU gear ratio map

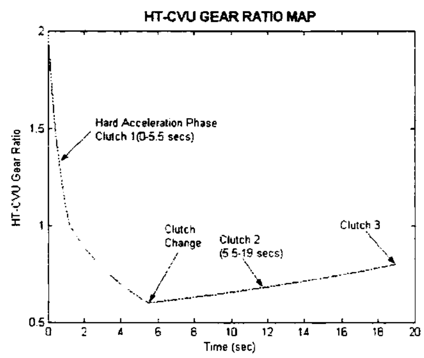

Figure 5: Overall gear ratio map of HT-CVT

A three piecewise curved gear ratio map within a large speed range, shown in Figure 4, has been used for simulating the HT-CVU ratio changes in all the simulations. Figure 4 shows that the HT-CVU ratio continuously change from 2 to 0.6 over the first 5.5 seconds of the acceleration period and at 5.5 second, clutch 1 of the transmission was changed over to clutch 2. Clutch 1 applies over a period of 0-5.5 seconds and Clutch 2 applies over a period of 5.5-19 seconds. It should be pointed out that the speed ratio between input and output of an actual HT-CVU changes smoothly over periods of 0-5.5 second and 5.5-19 second. The roughness at 1.2 seconds is due to the numerical approximation of the HT-CVU speed ratio using two piecewise curved map.

The overall gear ratio (product of R_1, R_2, and R_3) between engine and driving propeller of the powertrain is shown in Figure 5. It should be pointed out that the speed ratio between input and output of an actual HT-CVT changes smoothly over the whole of 0-19 seconds. The roughness at 1.2 seconds is due to the approximation of the HT-CVU speed ratio using two piecewise curved map. HT-CVT is the system that consists of a HT-CVU and planetary gear sets and clutches.

5. RESULTS AND DISCUSSIONS

Transient responses of a complete powertrain incorporated with a HT-CVT during a hard acceleration have been obtained numerically for various loading conditions. The simulation has been carried out for a vehicle starting from stand-still to a speed of 150 Km per hour. This simulation is to understand the transient behavior of the powertrain caused mainly by the gear ratios change and clutch change during vehicle acceleration. The result for transient responses of constant throttle condition is presented here.

The simulation is carried out for a vehicle starting from rest for a constant throttle position (at 80% throttle position). The forcing function (engine torque) is taken from the engine map for different engine speeds at 80% throttle condition. The resistive torques on the wheels is a function of the linear vehicle velocity and is numerically obtained at each step.

As it can be seen from Figure 6 during the initial 1.2 secs the CVT output torque to driveline steadily decreases from an initial high value with an increase in the engine output torque. This leads to good acceleration characteristics of the vehicle. The roughness at 1.2 secs is due to the numerical approximation of the HT-CVU speed ratio using two piecewise curved map. Between 1.2 to 5.5 secs the engine torque remains steady whereas the CVT output torque declines continuously. At 19 second the overall gear ratio is 1 as shown in Figure 5. As engine output torque and the CVT output torque are almost the same at 19 second it reflects the lock-up condition.

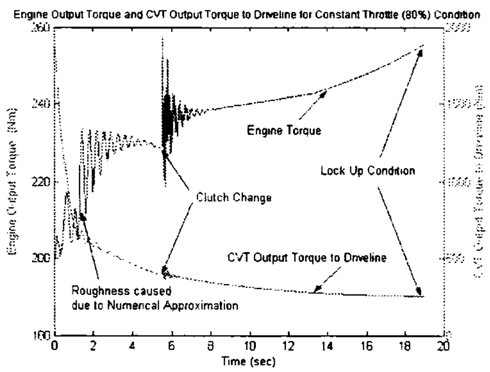

Figure 6: Output torque Engine and CVT

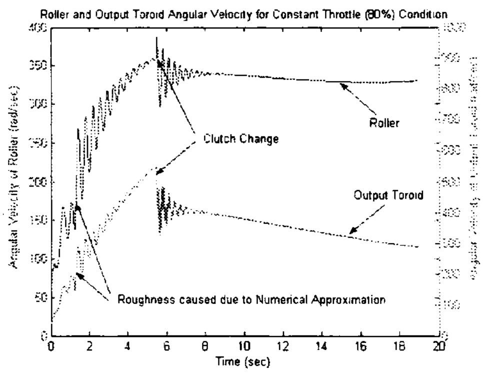

Figure 7: Angular velocities of roller and toroid

The angular velocities of the roller and output toroid are plotted in Figure 7. As can be seen from the figure during the initial 1.2 secs (hard acceleration phase) the roller and output toroid angular velocities increase. At 1.2 secs, due to the numerical approximation of the HT-CVU speed ratio using two piecewise curved map a roughness can be seen. Between 1.2 to 5.5 secs the roughness due to the numerical approximation gets damped out. At 5.5 secs the clutch change occurs. As a result a transient can be seen in both the roller and output toroid angular velocities. The point of interest in this simulation is the transient in the roller during clutch change. As it can be seen this would have an effect on the hydraulic ratio control mechanism of the roller.

In this simulation the hydraulics of the H-T CVU control unit is not included. However, the interaction between the mechanical and hydraulic sub-assemblies of the roller would have a tremendous influence in the ratio control of the HT-CVU. Another point of interest is the increase of the angular velocity of the roller and output toroid between 1.2 to 5.5 Secs. This reflects the possibility of instability creeping into the HT-CVU during the range of normal operation. The damping within the CVU and the overall damping of the system would definitely have an effect on the dynamics of the system.

6. DISCUSSION AND CONCLUSIONS

The damping level in the HT-CVU affects the dynamics of the system significantly and this needs to be investigated further. If the damping level in HT-CVU is low the input toroid and roller response builds up during high speed range and gets compounded by the fact that the damping level in other powertrain components is also low.

This paper presents a numerical investigation of the dynamics of a powertrain that uses a HT-CVU in its transmission component. Description of the derivation of the mathematical model of the powertrain system that consists of parametric elements due the continuous gear ratio change was provided. Simulations have been carried out to obtain the transient behaviour of the rollers and toroids of CVU during constant wide open throttle acceleration using an assumed piecewise speed ratio of the HT-CVU. The obtained results show that transient responses of input and output rollers of the HT-CVU exist when clutch changes during vehicle acceleration period starting from stand-still condition. The presented work also proves that the developed dynamic model, assembled from a number of parametric line elements in terms of mass, stiffness and damping matrices, of a conventional powertrain consisting of a HT-CVT is convenient and sufficient for dynamic analysis. Future work will focus on the investigation into the influence of varying dynamic characteristics of HT-CVU and damping on the dynamic behavior of the complete powertrain.

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