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A MODEL OF A CVT ADJUSTMENT DRIVE BASED ON TRANSIENT RESPONSE

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ABSTRACT
This paper deals with modeling and simulation of a CVT adjustment drive. The modeling approach is based on the transient response of the system. The transfer function of the system is developed based on some theoretical concepts and experimental data. For the verification of the model various experiments were done at different input voltages (-2V to 2V) for upshift and downshift of the CVT gear with applied forces (600N, 1000N) on driven side. In any case, error in position value is less than 10% with respect to measured value. The causes of small deviation of the results are stated from the model analysis and experimental results.

Keywords: Adjustment drive, Cascaded controller, Mechanical transmission, Power drive, Transfer function

1. INTRODUCTION
A Continuously Variable transmission (CVT) is a mechanical transmission, which provides a continuous range of speed ratios, unlike a normal transmission, which can provide only a few discrete ratios. It has also some advantages in comparison to other transmission systems due to its simple design. It is necessary to model a system and then to analyze the dynamic characteristics of the system for proper controlling. The mechanical model should be able to describe the mechanical behavior of the real system with an acceptable accuracy. Depending on the goals of analysis and the characteristics of the system, the model can be developed either with the concept of multibody system or the modeling principles of structural dynamics. A real system can be modeled as different elements depending on different views and circumstances. The aim of this work is to develop a computer model of the adjustment drive with which its controllers can be optimized.

2. DESCRIPTION OF TEST RIG
The CVT test rig (shown in Fig. 1) mainly has two parts: Prime mover and Adjustment drive. Prime mover is the power transmitting section and adjustment drive controls the speed ratio.

2.1 Prime Mover
Prime mover section includes Main power drives, shaft, pulley, V-belt, etc. Main power drives are two asynchronous motors which have a rated power of 33 kW and a rated torque of 200 N-m. Two parallel shafts are to transmit power from driver to driven set at a distance of 372 mm from each other and having a diameter of 60 mm.

The pulley can adjust its contact diameter from 96mm to 272mm by positioning the shipment of the movable sheave. The V-belt has two different wedge angle: 2^0 and 18^0 having some advantages [1] .

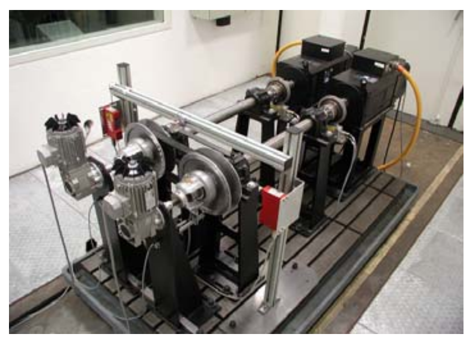

Fig 1. Total view of the test rig [2]

2.2 Adjustment Drive
The adjustment of the speed ratio is accomplished by an electro-mechanical adjustment mechanism which consists of geared motor, coupling, screwed spindle and adjustment housing (shown in Fig. 2). The adjustment has to be done for both the driver pulley and driven pulley simultaneously. The transmission ratio is altered with the change of contact diameter of pulley and belt. This is done by moving one sheave (movable) horizontally along the shaft. For getting the desired ratio, proper pressure has to be maintained to push the sheave to the correct position and to overcome the frictional force.

The adjustment actuator is an asynchronous squirrel-cage AC motor with a rated power of 0.37 kW. Screwed spindle transforms the rotational speed into axial shipment of sheave. The coupling is used to transmit the power required for adjustment from the actuator to the spindle.

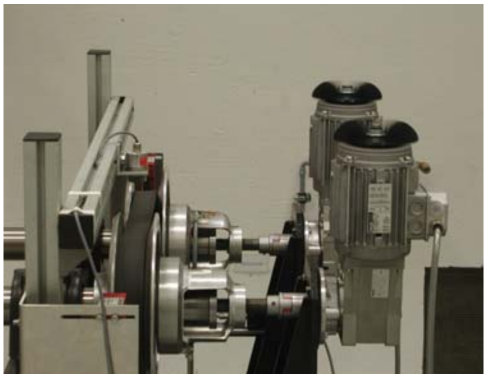

Fig 2. Total view of the adjustment drive [2]

2.3 Measurement and Control Electronics

For measuring the different parameters of the system different types of sensors are used like displacement, force, rpm, and Temperature. In the CVT test rig a MGCplus amplifier system is used to manipulate the sensor's signal by amplifying, signal conditioning and A/D-conversion to get the values of force, position, rpm etc. with their correct unit. Making the analogue signals continuously available for every measurement channel, the MGCplus provides high speed digital data acquisition at high resolution. The dSPACE (Digital Signal Processing and Control Engineering) / Simulink combination provides an integrated environment for modeling, simulation and real-time implementation. After creating the real-time code for the model, the ControlDesk is set up to acquire data, to watch and change the variables. The ControlDesk provides a graphical user interface (GUI). For adjustment drive the input from 0-10V is applied to the frequency converter (FC) via a dSPACE AutoBox. For the proper control of the system there are four controllers: Position, ratio, force and slip. All these controllers are conventional of PI or PID type [2].

3. DEVELOPMENT OF MODEL

The aim of modeling the adjustment drive is to test it with the present controller and then use it for developing an optimized controller. In case of CVT adjustment drive some experimental results are used to simplify the modeling efforts instead of pure theoretical calculations. For a linear system, step response provides all the information needed for developing a state-space representation or an s-domain Laplace representation of the system. Unfortunately, the given system has got some nonlinearities such as slip effects both in the case of motor and the belt. There are also some limitation problems like power limitation of the motor. Moreover, there is a distance limitation after which the sheaves cannot move. We have developed the total model of the system by elemental behavior analysis of actuator, coupling, screwed spindle, adjustment housing, etc.

3.1 Actuator

As the actuator system consists of some mass inertia, damping and stiffness element, the system can be considered as at least a second order system. The well-known closed-loop transfer function of a second order system is as follows [3]:

$$\frac{C(s)}{R(s)} = \frac{\omega_n^2}{s^2 + 2\omega_n \varsigma s + \omega_n^2} \tag{1}$$

where, C(s) is the output, R(s) is the input to the system, ω_n is the undamped natural frequency and ζ is the damping ratio.

By using a typical step response of a second order system, the parameters ω_n and ζ can be found out [3, 4]. With the use of step response and formulae [3, 4], we found the value of the natural frequency ω_n = 32.74 Hz, damping ratio ξ =0.5739 and the transfer function of the motor is obtained as in Fig. 3.

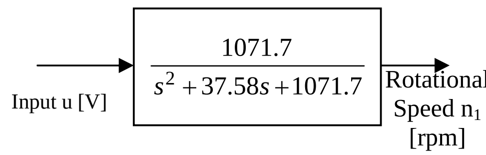

Fig 3. Transfer function for motor

3.2 Coupling

The coupling is a mechanical connection to transmit power from the actuator to the output shaft. It consists of a driver and a driven part and between them there is a toothed plastic connector. It causes a delay in the output, which causes a phase shift between the output and input. In the modeling the coupling is considered as a delay element whose transfer function is as follows (Fig. 4):

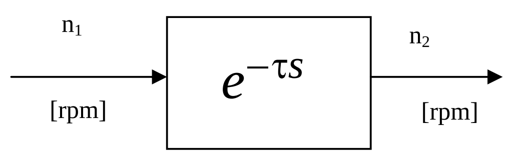

Fig 4. Transfer block for coupling

3.3 Screwed Spindle

The spindle gives 3 mm axial movement to the adjustment housing at each rotation. It is working as an integrator to transmit the rpm into axial distance. So the transfer function for the spindle should be as in Fig. 5.

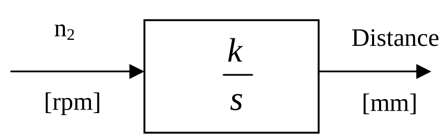

Fig 5. Transfer function for screwed spindle

Where, k is the pitch of the spindle and s is the distance traveled by the adjustment housing within a certain integration time.

3.4 Adjustment Housing

When the adjustment housing is modeled neglecting the elastic deformation and the material damping, it can be seen as a mass on which some forces are acting. Due to the resultant force, a displacement of the housing occurs. The resultant force is the sum of all forces applied by the belt [5], frictional force between the housing and the spindle thread and the axial force created by the spindle.

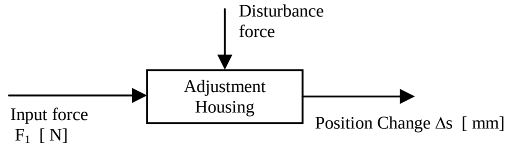

Fig 6. Block diagram of the Adjustment housing

3.5 Transformation from Axial Shipment to Radial Change

The total axial shipment Δs of the housing & sheave can be divided into two parts: One sheave has Δs_1 and the other has Δs_2. The wedge angles are γ_1 and γ_2. Let Δr be the radial displacement of the belt depending on the axial shipment Δs of the sheaves. With some geometrical calculation for the transformation of the axial displacement Δs to the radius change Δr, a proportional block or gain box is considered as follows [6]:

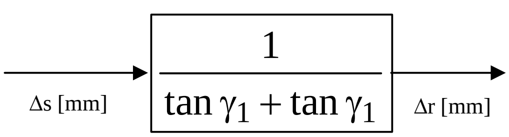

Fig 7. Transformation from displacement to radius change

3.6 Model of the Adjustment Drive

The transfer function that was determined before for the actuator did not consider the effect of mass inertia of the housing and the movable sheave. Here transient response is determined when it is connected with these two parts for including their effects. The undamped natural frequency ω_n is 7.75 Hz and the damping ratio ξ is 0.8311.

Table 1: Comparison of ω_n and ξ

Parameters	Case1	Case2
ω_n	32.74 Hz	7.75 Hz
ξ	0.5739	0.8311

The transfer function for the second case is as follows:

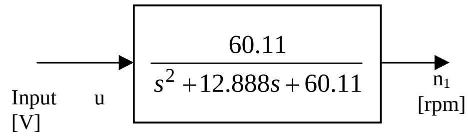

Fig 8. Transfer function for adjustment drive

4. SIMULATION RESULT

The simulation was done with the real time interface (RTI) model in the real field. For this, a control layout was made to give input to the system and to see the results of the model and measurement for making comparison. Experiment was performed both in the cases of increasing radius (upshift) and decreasing radius (downshift) during the change of speed ratio.

4.1 Upshift of the CVT Gear

When a positive voltage is given to the FC, the sheave moves inwards to the fixed sheave and as a result the contact radius increases. During the closing of the sheave the disturbance torque acts in the opposite direction. The equivalent voltage of the torque is subtracted from the input voltage and the net voltage is passed through the transfer function of the motor. The results obtained when the input voltage is +2V and the force at driven side is 600 N, are shown in Fig. 9, Fig. 10 and Fig. 11:

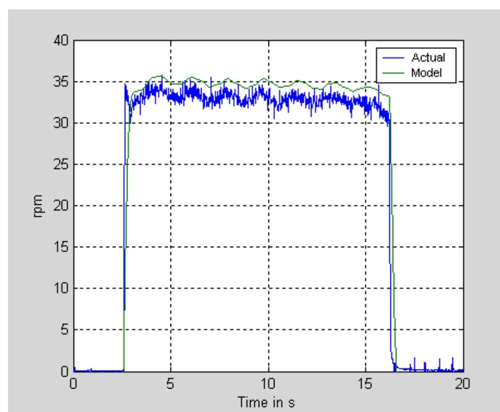

Fig 9. The rpm obtained from the model and motor.

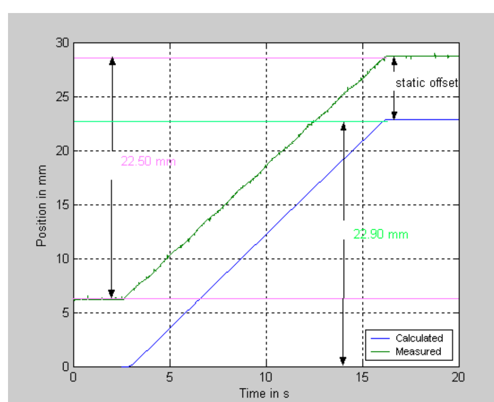

Fig 10. Distance traveled by the movable sheave due to spindle rotation

From the graphs it can be seen that there is a great similarity between calculation and measurement. The small difference between the calculated and measured rpm comes either from the sensor or from the transfer function of the model. The sensor was not fixed to a clamp, it was held by hand. At a high rpm, there is a little movement in the hand which causes an affect on the sensor readings. The time delay due to the coupling block is neglected in the RTI model, because the delay time of the coupling is very small so that it does not create a meaningful deviation to the speed response.

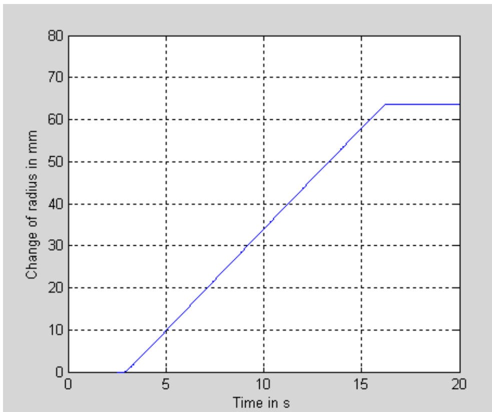

Fig 11. Radius change due to position change of movable shift

There is a swinging response for rpm (Fig. 9) both in the cases of calculated and the measured rpm. It is due to the transverse vibrational effect of the belt [1]. Like the string of a guitar or a violin, the belt can vibrate (as shown in Fig. 12). The frequency of vibration increases with decreasing length of the free span between the pulleys and with increased belt tension. This vibration is transmitted via sheave to the housing when it is measured.

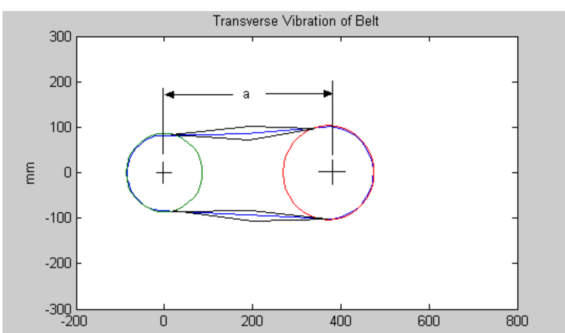

Fig 12. Transverse Vibration of the belt

In the Position vs. Time graph (Fig. 10) it can be seen that, for a certain time period the measured displacement of the sheave is 22.5 mm and the calculated value is 22.90 mm. So the error is as follows:

$$\Delta e = \frac{n_c - n_m}{n_m} \times 100\ \% \qquad (12)$$

= 1.78%

where, Δe is the error, n_c is the calculated value and n_m is the measured value of rpm.

This error may also come from the modeling error like as nonlinearities of the system which have not been considered in the model. A part of the error also may come from the inductance sensor due to elasticity and vibrational effects on the attachment part of the housing.

There is a static offset between the measured and calculated position. This is due to the fact that in the RTI model the Integrator subsystem is in *held state*. It holds the position value of the previous experiment. So this acts as a static offset for the next experiment. If a simple Integrator is used in lieu of Enable subsystem in that case also an offset occurs but caused by another reason. In that case, there is a constant negative offset at the entrance of the I-block caused by the disturbance torque. So the I-block's decrement occurs continuously and when an input voltage is given it starts to increase from that offset value. But in both cases, the net increment looks very similar.

There is a linear increment of the radius with respect to time. By adding this increment with a certain reference value, the contact radius of the pulley can be obtained. The radius obtained from the model is calculated as simply a function of the geometry of the pulley. But in reality it depends on other factors too like the speed of the prime mover, load torque, belt properties, etc.

Fig 13 is obtained from the experiment when input Voltage is +2V and the applied force on the driven side is 1000N.

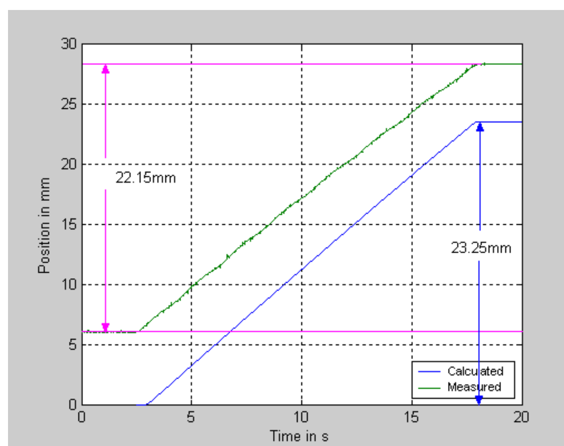

Fig 13. Displacement of sheave when input is +2V and force is 1000N

In this case the actual displacement measured by the sensor is 22.15 mm and the simulation result is 23.25 mm. The error increases to a value of 4.96%. So it can be concluded that there is a relation between the force of the driven side and the rotation of the driver. From the experiment it is observed that high force applied by belt on the driven side causes a decrement of revolution of the driver side. As a result a reduction of displacement occurs for the movable sheave.

4.2 Downshift of the CVT Gear

If a negative voltage is given to the FC, the motor rotates in the opposite direction. As a result the movable sheave moves away from the fixed one and the contact radius gradually decreases. In this case the force given by the belt helps the pulley to move faster. In the model, during the negative input voltage the equivalent voltage of the disturbance torque is summed up with the input. So the motor can rotate faster.

An experimental result is shown in Fig. 14 when a input voltage of –2V and the force on the driven side is 600N. There is a static offset in the rpm diagram due to the disturbance torque, which causes a continuous input voltage to the model of the system. In Fig. 14 the rpm is negative, which means the rotation is in opposite direction. Now it can be compared that when an input voltage is +2V and force is 600N then the sheave moves 22.90 mm in 13.5 s. On the other hand, when input voltage is –2V and force is 600N the sheave moves 23.5 mm in 11.6 s (as shown in Fig. 15). So it is clear that there is a first movement of the sheave during the opening (downshift of CVT gear).

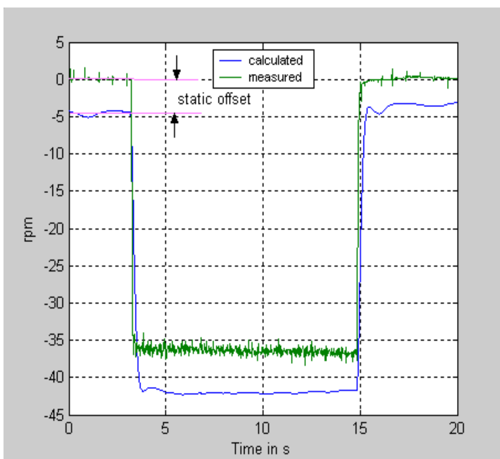

Fig 14. motor's rpm at input u = –2V and force is 600N

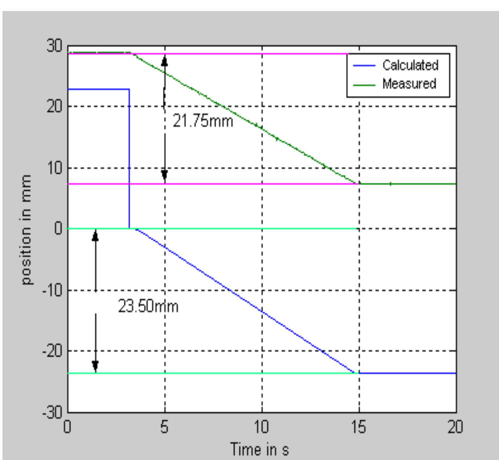

Fig 15. The displacement at u = –2V & force is 600N

5. CONCLUSION

In this paper the model of the CVT adjustment drive has been developed by assuming that the system is of 2[nd] order and simulation was performed. The model behaves like the real adjustment drive with a tolerable error at different input voltages in the cases of upshift and downshift of the CVT gear. It gives a good result when the force on the driven side is not so high. A high force affects the other side's rotation. At high force the driver motor has not enough power to overcome this force. By making an interaction between the driver and driven side considering the real behavior, the further development can be carried on. Actually, this small deviation of radius is not a dominant factor for controlling the speed ratio. The changing of the speed ratio depends mainly on the mechanical performance of the electro-mechanical adjustment system.

The error of the rpm sensor can be minimized by making an arrangement for holding it at a fixed position. If the input voltage and current or the torque at the output shaft can be measured, the model can be obtained by pure mathematical formulations. The PID controllers then could be optimized by using the developed model

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