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To cite this article: Ashish Ranjan *et al* 2017 *IOP Conf. Ser.: Mater. Sci. Eng.* **263** 062037

IOP Conf. Series: Materials Science and Engineering **263** (2017) 062037 doi:10.1088/1757-899X/263/6/062037

Design and control of electromagnetic clutch actuation system for automated manual transmission

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Abstract: There is a growing interest towards Automatic Transmission in India as it provides better comfort and drivability. But the high cost of this system is limiting itself to be successful in the Indian markets. Due to this, Automated Manual Transmission (AMT) is considered which provides a better solution towards automation as it enhances the drivability and fuel consumption characteristics of a manual transmission at lower costs. However, torque lag and comfort are major issues with AMT which can be addressed by reducing the shift time. In this paper we describe an Electromagnetic Linear Clutch Actuator as a replacement to current electrohydraulic and electromechanical actuator. A control system for the actuator is presented and a clutch engagement strategy is also implemented which reduces the engagement time to 0.78 seconds while reducing jerk and torque lag. The actuator and control system is simulated on a MATLAB Simulink and agreeable results have been obtained.

1. Introduction:

In India, the Manual type of transmission system in Automobile vehicles is prevalent. In manual transmission the gears are selected by the driver by disengaging the clutch, then by moving the gear selector knob and then engaging the clutch. These transmission have good fuel economy depends on the drivers skill. During stop and go pattern in heavy traffic, the driver has to engage and disengage the clutch frequently which results in faster wear of clutch disc and results in wastage of energy. On the other hand, the Automatic transmission and Automated Manual Transmission (AMT) are slowly making their presence in the Indian market [1]. In Automatic Transmission, the transmission control unit decides the gear ratio from the look up table based on accelerator pedal position and the gear change is controlled automatically with the aid of actuation and control mechanism. Hence, its known for smooth operation and high level of comfort [2,3] whereas Manual Transmission (MT) for its level of driver control. But the major drawback of the Automatic Transmission System is the cost factor due to which it is unable to create an impact in the Indian market where the demand for comfort with driver control and at an effective cost is prevailing. Thus this forecasts a Rapid growth of Automated Manual Transmission due to its efficiency and cost advantage. AMT systems are similar to the Manual Transmission, but the difference lying in the gear shifting mechanism. The gear shift in the AMT is achieved electronically with the aid of actuators and control systems [2]. This proves that, the system to have better shifting, to provide better fuel economy and light-weight compared to Automatic Systems. One of the major advantages is that a conventional MT can be transformed into an AMT with the implementation of a few components. AMT systems in Mid-sized cars offers around 5% better fuel economy than conventional AT [4].

2. Experimental methods:

2.1. Clutch Testing:

In this work, we have selected a MARUTI 800 car Clutch Assembly. This clutch Assembly is loaded with a diaphragm type spring in order to apply normal force on to the friction plate against the flywheel for the drive-engage state. Since the basic scope of the work is the clutch actuation, it is very important to determine the non-linearity of diaphragm spring characteristics for this selected clutch assembly. Hence, an experimental analysis was performed on a Spring Testing machine, where the clutch was loaded gradually and the corresponding bearing displacement was measured. The setup of the experiment is shown in fig 1 and explained as follows: On the Spring Testing machine sufficient arrangement and adjustments were made in order to acquire the needed parameters. The clutch is placed on the bottom disk and it is displaced vertically with the help of a lead screw-motor arrangement.

Figure 1. Experimental clutch testing Setup.

The release bearing was placed on the diaphragm of clutch against a pre-load setup. A Load cell is attached on the top surface which indicates the actual forces acting on the clutch. Now the bottom disk is elevated so that the top surface applies force on to the release bearing and the actual force is measured by the load cell. Along with this, the needle of the linear electronic scale is placed on the clutch surface which is used to measure bearing displacement. The data were recorded to obtain a graph of Force Vs Bearing travel as shown in fig 2. This gives the characteristics of the diaphragm spring in the clutch. From this graph, we can determine that, the spring needs a maximum axial force of 700 N at a bearing displacement of 3mm; and after which the force reduces and the complete disengagement occurs (with 1mm clearance between the friction plate and pressure plate) and it is observed at 8mm of bearing travel. Now with these data, it is possible to design a fork and lever arrangement in order to obtain same amount of force with mechanical advantage.

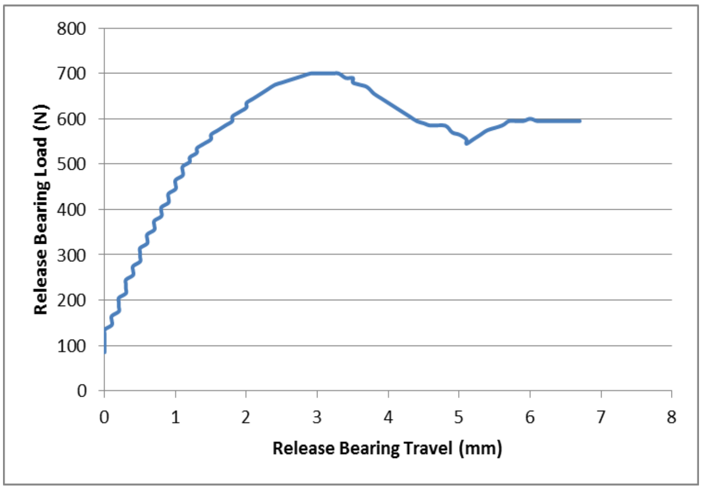

Figure 2. Diaphragm Spring characteristics

2.2 Designing of Fork and lever:

Designing of the Fork and lever is based on Torque and Leverage mechanism [4]. Our requirement is to deliver a maximum force of 700 N (considering a Factor of safety) on the diaphragm clutch to get a release bearing travel of 8mm. Now assume that the actuator that can deliver 200 N force, we can follow a set of calculations (trial and error method) in order to calculate the value of Mechanical Advantage and corresponding values of the stroke needed from the actuator.

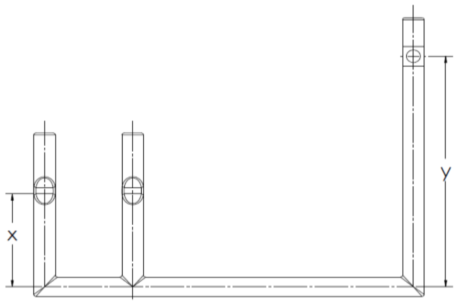

Figure 3. Design of fork and lever

Torque on bearing side (fork) = torque on actuator side (lever)

$$70 * x = 20 * y$$

$$y = 3.5 * x \qquad (1)$$

$$\text{Stroke} = y * \Theta$$

Based on this calculation, for various Actuator force the required Mechanical Advantage is calculated as shown in table 1:

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Table 1. Mechanical advantage for various Loads

Actuator Load (N)	Required Mechanical Advantage
200	3.5
250	2.8
300	2.3
350	2
400	1.75

2.3 Selection of actuator

From the calculated values of the required Mechanical Advantages against a few assumed Actuator forces, the amount of actuator stroke needed to provide a release bearing travel of 8mm for the mechanical advantage of 3.5 (Here we calculated for the first case in table 1) is determined as shown below.

$$\text{i.e} \quad x * \Theta = 8mm \qquad (2)$$

where Θ is the angle of rotation of the fork about its pivot axis.
Also,

$$\text{Stroke} = y * \Theta \qquad \text{(from 1)}$$
$$= (3.5 * x) * \Theta \qquad \text{(from 2)}$$
$$= 3.5 * 8$$
$$\text{Stroke} = 28 \text{ mm}$$

Table 2. Actuator Stroke based on various Mechanical Advantage.

Actuator Load (N)	Mechanical Advantage Needed	Actuator Stroke requirement for 8mm bearing travel (mm)
200	3.5	28
250	2.8	22.4
300	2.3	18.6
350	2	16
400	1.75	14

Fig 4. Shows the force and stroke relationship provided by the actuator manufacturer. As we require the maximum force at actuator stroke of 7mm, quarter duty cycle is selected.

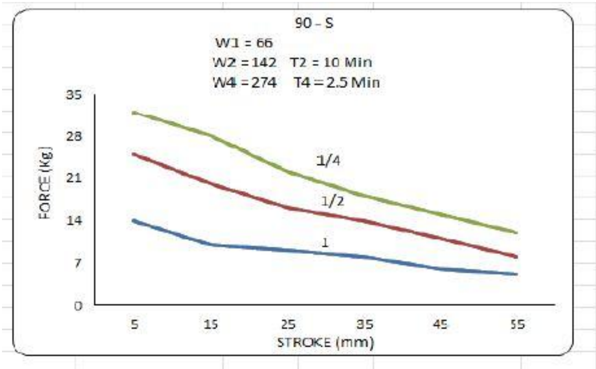

Figure 4. The Force Vs Stroke of the custom actuator

On evaluating the above results and conducting a market survey for the products meeting the requirements, the following actuator has been selected:

Selected mechanical advantage : 2.3
Maximum Force : 350 N
Maximum Stroke : 15mm

3. Control system:

3.1. Control system for Actuator:

In order to maintain a constant stroke length from the actuator, we provide a control system to achieve it. The governing equation for the actuator is given by [5],

$$F = \frac{dW_m}{dx} = \frac{SN^2 i^2 {\mu_r}^2 \mu_0}{2\left(l^2 + l_{eq} + (\mu_r - 1)x\right)^2} \tag{3}$$

Where,
S = cross section of the plunger
N =number of turns=1000
i =current
μ_r =Relative permeability of material (MS) =100
μ_0=Permeability constant = $4\pi \times 10^{-7}$ N·A^{-2}

$$\therefore F(x,i,t) = \frac{\kappa_{sol} i(t)^2}{\left(\kappa_\mu + x(t)\right)^2} \tag{4}$$

where

$$\kappa_{sol} = \frac{SN^2 {\mu_r}^2 \mu_0}{2(\mu_r - 1)};\ \kappa_\mu = \frac{l_2 + l_{eq}}{(\mu_r - 1)} \tag{5}$$

$$\therefore \frac{\kappa_{sol} i(t)^2}{\left(\kappa_\mu + x(t)\right)^2} = m_L \ddot{x}(t) + b_L \dot{x}(t) + k_L x(t)$$

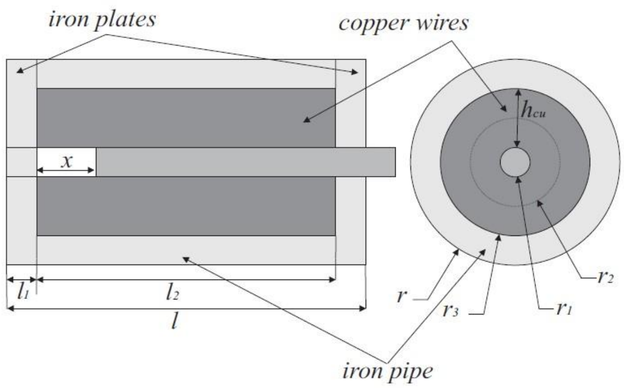

Figure 5. Schematic diagram of Electromagnetic Actuator

In the general equation 3, the actuator force F is the function of current (i) and displacement of plunger(x) and k_{sol} and k_μ are constants. From the above equation the force and position of the actuator can be controlled by varying the current. This relation is being put in the general spring equation in order to simulate load and this Equation 4 has been modeled using Simulink. A PI controller is used to control the current input to the system to get the actuator stroke value close to the reference stroke required [6].
The PI controller system as shown below,

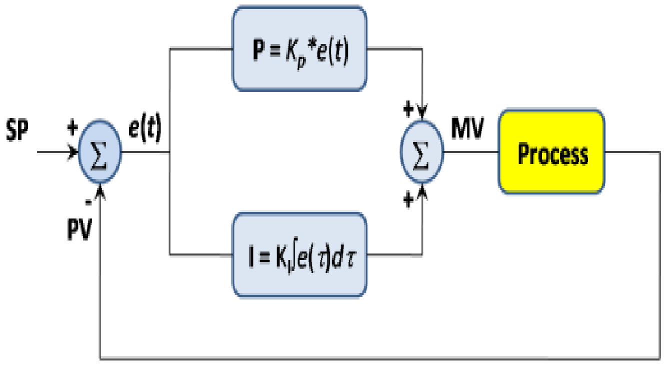

Figure 6. The PI Controller

The formula governing the PI controller is given as

$$i^*(t) = k_p\left(x(t) - x^*(t)\right) + k_i \int (x(t) - x^*(t)) \qquad (6)$$

Where k_p and k_i are constants acquired by tuning the PI controller for precise output. In the PI controller, the 'P' module accounts for the present values of error and the 'I' module accounts for the past values. It is a closed loop feedback system which gets the previous actuator position and compares with the reference position and generates an error signal. Once the error signal is generated, the PI controller will send a corrected value for processing till the error nullifies.

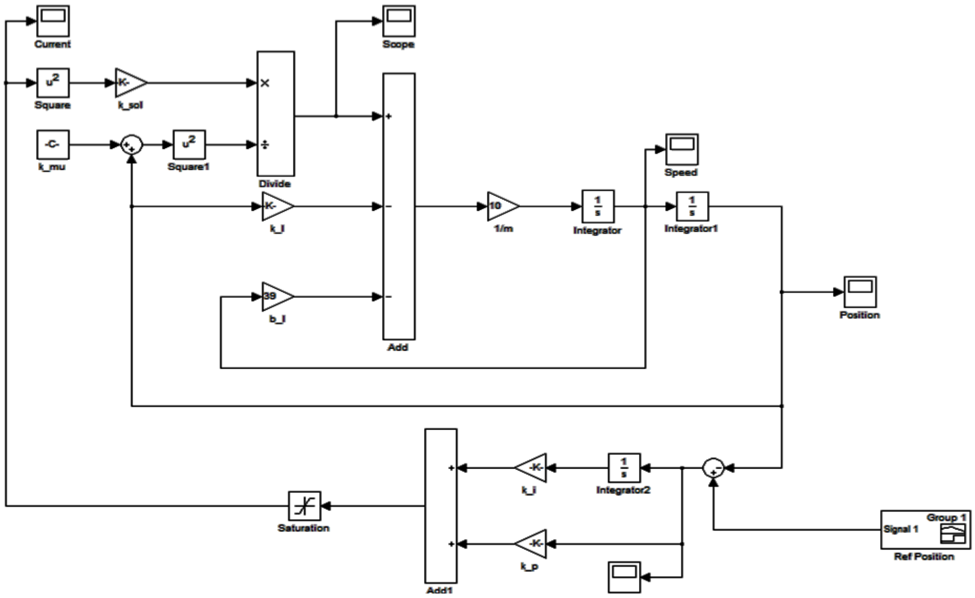

Figure 7. Implementation of the PI controller in MATLAB Simulink.

The model was tested for different input signal and the results were validated from published papers which helped us validate the system. With the input as the desired displacement, we acquired the amount of current needed to be supplied to the actuator and the corresponding amount of Force which the actuator would generate with respect to the equation.

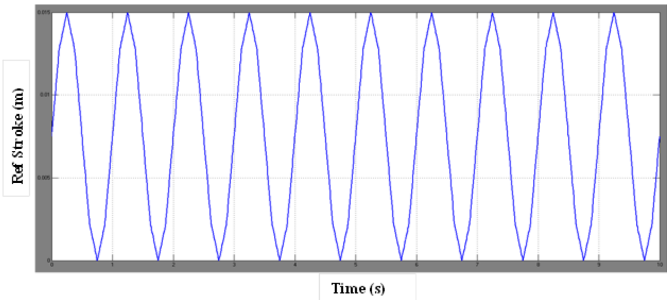

Figure 8. Reference Stroke (m) vs Time (s)

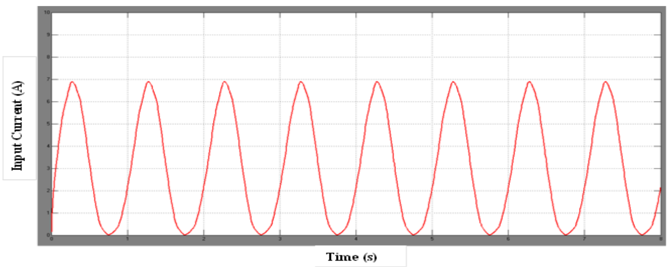

Figure 9. Input Current (A) vs Time (s)

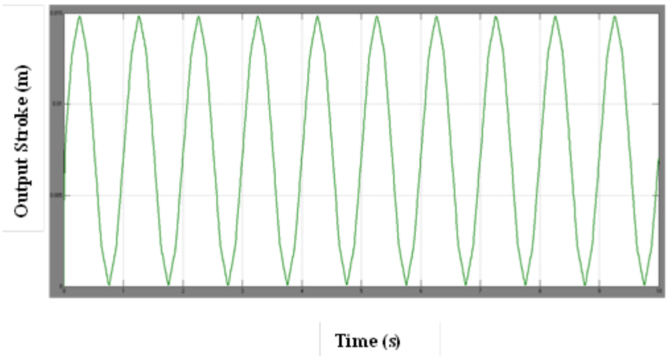

Figure 10. Output Stroke (m) vs Time (s)

In Figure 8 the reference stroke supplied to the system is given. The actual stroke when subtracted from the reference stroke generates an error signal which is then supplied to the PI controller. The PI controller then generates a feedback current signal as shown in Figure 9. This current signal is supplied to the system and a desired output stroke is achieved as shown in Figure 10.

3.2. Clutch Engagement Control:

The clutch engagement for an AMT should ensure that the engagement takes place in minimum amount of time while avoiding jerk. It is quite obvious that these objectives are conflicting in nature and an optimized control should be implemented to achieve the desired smooth engagement [7]. In order to implement the control system a simplified model of the clutch was used to simulate the two states during engagement i.e. slipping and locked.

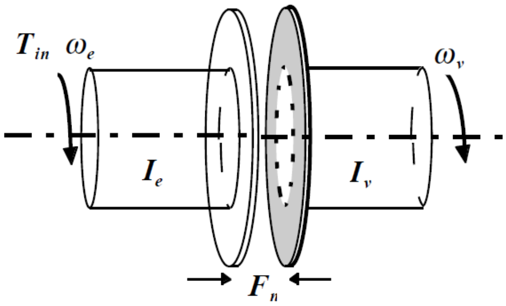

Figure 11. Schematic model of a dry friction plate clutch

The slipping condition will consist of the following two equations:

$$I_e\dot{\omega}_e = T_{in} - b_e\omega_e - T_{cl} \quad (7)$$

$$I_v\dot{\omega}_v = T_{cl} - b_v\omega_v - T_l \quad (8)$$

Where,
I_e : Engine Inertia
$\dot{\omega}_e$: Crankshaft rotor speed
T_{in} : Engine Torque
b_e : Damping rate at the engine side of clutch
T_{cl} : Torque transmitted by the clutch
I_v : Equivalent vehicle moment of inertia
ω_v : Clutch disc rotor speed
b_v : Damping rate at transmission side of clutch

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T_l : Equivalent load torque
The torque transmitted by the clutch can be given as

$$T_{cl} = kF_n \mathrm{sign}(\omega_e - \omega_v)$$

Where,

$$k = 4R\mu_d/3 \tag{9}$$

R : Equivalent Disk Ratio
μ_d : Dynamic Friction Coefficient
At locked state both the crankshaft rotor speed and the clutch disc speed synchronizes. Adding (5) and (6) we get,

$$(I_e + I_v)\dot{\omega} = T_{in} - (b_e + b_v)\omega - T_l$$

Where

$$\omega = \omega_e = \omega_v \tag{10}$$

This equation is used to simulate the engaged state of the clutch.

A control strategy is defined using state flow in MATLAB which switches between the slipping and locked stages based on the values of crankshaft speed and the clutch rotor speed from the slipping and locked sub-models created in Simulink [8,9].

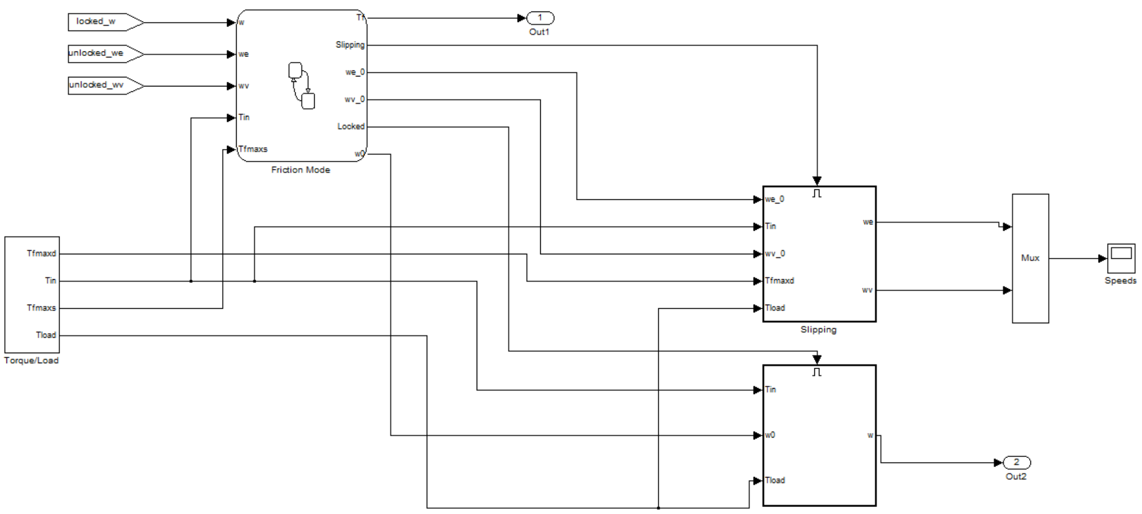

Figure 12. MALAB Simulink model of Clutch Engagement Control

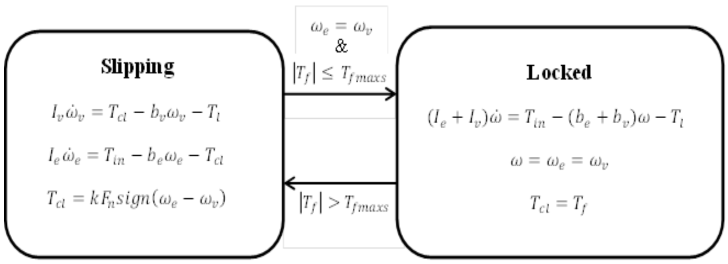

Figure 13. State diagram describing transition between Slipping and Locked stages

In Fig 14. the Stateflow scheme for clutch engagement is shown. As the torque transmitted during locked state ($\omega_e = \omega_v = \omega$) exceeds the maximum static torque value, the engagement

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model is switched to the slipping block. If the torque transmitted is less than or equal to the maximum static torque the stateflow switches back to the locked mode. In Fig 14. we can see the stateflow switching block gives an enable signal as an output to the slipping and locked blocks depending on the condition of the system

The Simulink model was validated from a predefined data for which the result was known.

The values of the input parameters were taken for a hatchback.

$$I_e = 0.2\ \mathrm{kgm}^2$$
$$I_v = 0.7753\ \mathrm{kgm}^2$$
$$b_e = b_v = 0.0185\ \mathrm{Nms}$$
$$\mu_d = 0.4$$
$$k = 0.098$$
$$F_{nmax} = 700\ \mathrm{N}$$

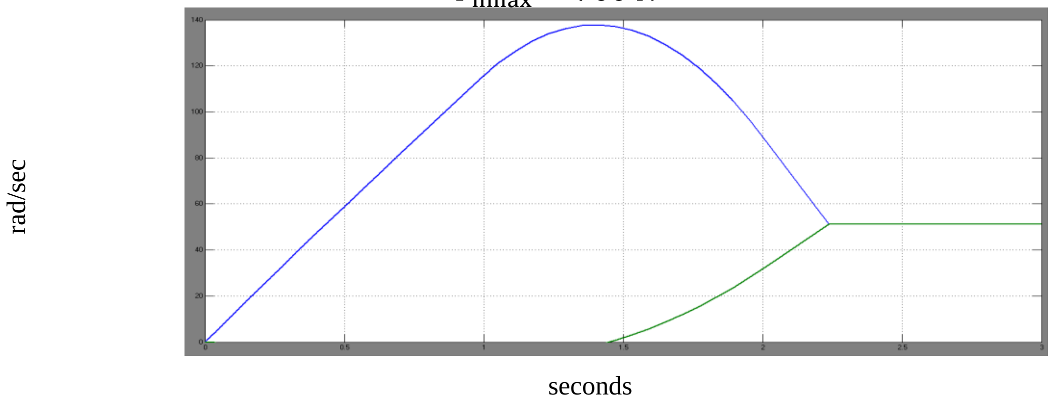

Figure 14. Variation of $\boldsymbol{\omega}_e$ and $\boldsymbol{\omega}_v$ with respect to time using State Flow model.

At engine torque of 30 Nm and load torque of 5 Nm the smooth engagement was achieved at 2.18 seconds. This value is comparable with manual transmission. Therefore, an engagement control was implemented successfully.

3.3 Linear Quadratic Controller:

To further improve the engagement time while maintaining the same level of comfort as that of a manual transmission a linear quadratic controller was implemented [10,11,12]. In order to implement this controller firstly the state-space model of the clutch system is made. A feedback gain is then derived which solves the Ricatti equation to minimize a cost function.

Based on the slipping and locked equations (5-8) a state space model is formulated as follows:

$$\begin{gathered}\dot{x}_1 = \frac{b_e}{I_e}x_1 - \frac{k}{I_e}x_3 + \frac{T_{in}}{I_e}\\ \dot{x}_2 = \left(-\frac{b_e}{I_e} + \frac{b_v}{I_v}\right)x_1 - \frac{b_v}{I_v}x_2 - \left(\frac{k}{I_e} - \frac{k}{I_v}\right)x_3 + \frac{T_{in}}{I_e} + \frac{T_l}{I_v}\\ \dot{x}_3 = u\end{gathered} \tag{11}$$

Where,

$$x_1 = \omega_e$$
$$x_2 = \omega_e - \omega_v$$
$$x_3 = F_n$$

The system of equations can be written in matrix form as follows:

$$\begin{pmatrix}\dot{x}_1\\ \dot{x}_2\\ \dot{x}_3\end{pmatrix}=\begin{pmatrix}-\frac{b_e}{I_e} & 0 & -\frac{k}{I_e}\\ -\frac{b_e}{I_e}+\frac{b_v}{I_v} & -\frac{b_v}{I_v} & -\left(\frac{k}{I_e}-\frac{k}{I_v}\right)\\ 0 & 0 & 0\end{pmatrix}\begin{pmatrix}x_1\\ x_2\\ x_3\end{pmatrix}+\begin{pmatrix}0\\ 0\\ 1\end{pmatrix}u+\begin{pmatrix}\frac{1}{I_e} & 0\\ \frac{1}{I_e} & \frac{1}{I_v}\\ 0 & 0\end{pmatrix}\begin{pmatrix}T_{in}\\ T_l\end{pmatrix} \tag{12}$$

This can be re-written as

$$\dot{x}=Ax+Bu$$

The cost function to be minimized is

$$J=\int_0^{t^*}\left(q{x_2}^2(t)+ru^2(t)\right)dt$$

The optimal feedback that minimizes the cost function is given as [13]:

$$u=-Kx$$

In order to compute the gain K a MATLAB program was written which optimizes the given finite horizon linear quadratic cost function by solving the Ricatti equation for a chosen time instant t*. As u is the derivative of Fn, its value should be integrated to obtain the force applied on the friction plate by the diaphragm spring for smooth engagement. The controller was validated for different engine torque and load torque values.

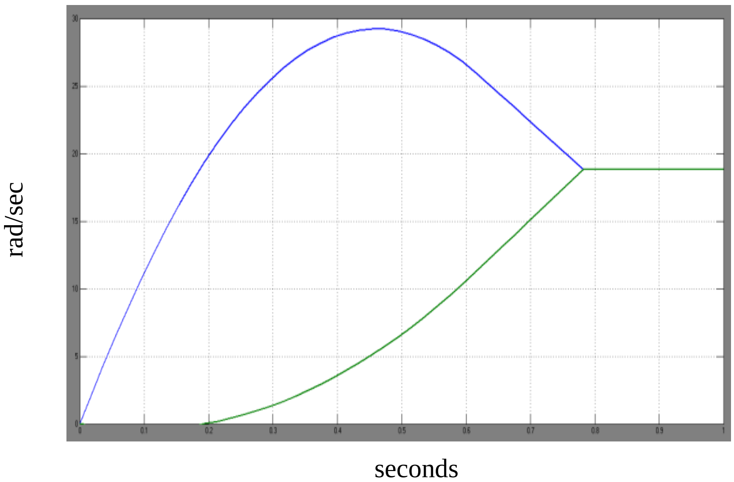

Figure 15. Variation of ω_e and ω_v wrt time using LQ controller

For a constant engine torque of 30Nm and load torque of 5 Nm the engagement time was found to be 0.78 seconds by implementing the LQ controller.

3.4 Integrated Actuator Control Strategy:

By performing curve fitting a relation between the actuation force F_n and the release bearing travel is found. The output of the LQ controller is Fn. A Simulink block is made which gives the output of the release bearing travel at a particular value of Fn. The release bearing travel is then converted to actuator stroke by multiplying with a suitable gain. The actuator stroke value is taken as a reference value for the actuator PI controller which controls the current value to achieve the desired actuator stroke for smooth engagement.

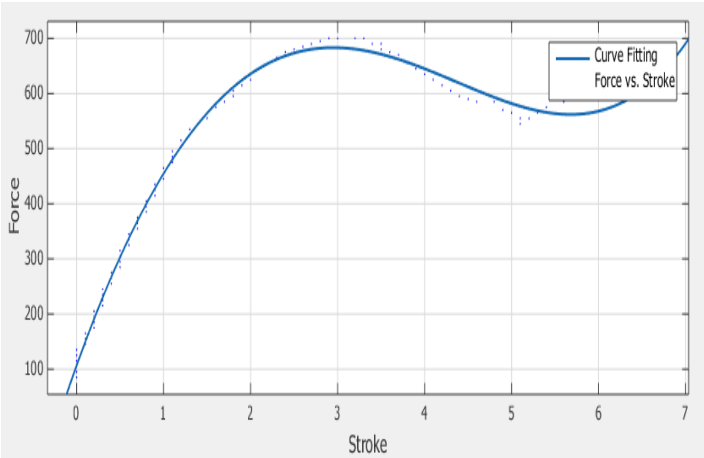

Figure 16. Curve fitting of diaphragm spring characteristics

The relation obtained from curve fitting was

$$F_n(x) = 0.5535x^4 + 2.382x^3 - 94.73x^2 + 440x + 106.7$$

The clutch engagement control and the actuator control were coupled together as shown in the Simulink block diagram

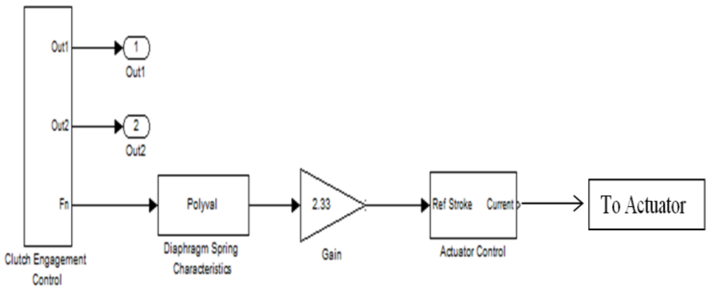

Figure 17. Integration of clutch engagement control in MATLAB

4. Conclusion

In this study, an electromagnetic linear actuator was selected based on the data obtained from clutch testing. Further the mathematical model of the actuator was made and a control system was proposed for the same. To ensure smooth engagement while reducing the engagement time, two clutch engagement controls were implemented. The engagement time was reduced from 2.18 seconds to 0.78 seconds while reducing the jerk using a Linear Quadratic Controller. The clutch engagement control was coupled with the actuator control to obtain the desired stroke to ensure smooth engagement within the specified time.

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