Intelligent Control Electromagnetic Actuated Continuously Variable Transmission System for Passenger Car

Ataur Rahman*, Sazzad Sharif, Mohiuddin AKM, Ahmed Faris Ismail, Sany Ihsan Izan
Department of Mechanical Engineering, Faculty of Engineering,
International Islamic University Malaysia,
50728 KL, Malaysia
*Corresponding author: arat@iium.edu.my

Abstract. Continuously variable transmission (CVT) system transmits the engine /battery power to the car driving wheel smoothly and efficiently. Cars with CVT produces some noise and slow acceleration to meet the car power demand on initial start-ups and slow speed. The car noise is produced as a result of CVT adjustment the engine speed with the hydraulic pressure. The current CVT problems incurred due to the slow response of hydraulic pressure and CVT fluid viscosity due to the development of heat. The aim of this study is to develop electromagnetic actuated CVT (EMA-CVT) with intelligent switching controlling system (ICS). The experimental results of ¼ scale EMA shows that it make the acceleration time of the car in 3.5-5 sec which is 40% less than the hydraulic CVT in the market. The EMA develops the electromagnetic force in the ranged of 350 –1200 N for the supply current in the range of 10-15 amp. This study introduced fuzzy intelligent system (FIS) to predict the EMA system dynamic behaviour in order to identify the current control for the EMA actuation during operation of the CVT. It is expecting that the up scale EMA-CVT would reduce the 75% of vehicle power transmission loss by accelerating vehicle in 5 sec and save the IC engine power consumption about 20% which will makes the vehicle energy efficient (EEV) and reduction of green house gas reduction.

1. Introduction

Acceleration stays in a “sweet spot” to minimise wasted power, thus improving a vehicle’s fuel efficiency. In the most advanced form, the CVT and its control forms an actuation system for the transmission which determines the best operating speed and torque of the engine given a set of driver inputs and car operating conditions. The CVT transmission has 15% less fuel consumption than that of a conventional gearbox along with a reduction in harmful emissions of about 30%. Comparison has been made with the simulation study based on different CVT concepts with a manual transmission for gasoline and diesel engines and show up to 10% less fuel consumption for gasoline engines and up to 19% for diesel engines [1, 2]. The author [3] has conducted a study on the performance for a car of 3000cc by simulation on the manual transmission, automatic transmission and the continuously variable transmission and he reported that time taken to accelerate to 100 km/h is 10.20 s for manual transmission, 10.76 s for automatic transmission, 7.85 s for CVT. CVT is preferred over other automatic transmission because of the comfortability, reliability, durability as well as the efficiency [4]. The power flow through the power train does not need to be interrupted during acceleration as in conventional transmissions and this makes it possible to gain a smooth, rapid and steeples response to
drivers’ demand without disturbing jerk. Technically, oil hydraulic system is preferred over the pneumatic due to leakage and compressible fluid when high accuracy and great amount of pressure are needed [5]. Hydraulic control high gear and low gear mechanism for CVT is quite effective to develop the sufficient pressure but the problem is to hold the movable sheave of the pulley as it is desired. Though spring can be used, owing to spring oscillation there will be slip, which results torque losses. High pressures and flow rates are required in the hydraulic control system to maintain the high torque at start-ups and slow speed. However, fluid pumping losses are the major causes of torque loss in modern CVT [6]. The half toroidal CVT could not replace the belt-push drive CVT because of its bulk deformation of contact component resulting imperfect contact angle. This problem was exacerbated by the comparative delay controller strategies which caused the engine speed to go to its maximum speed range following a full pedal input and it was only after a delay that the car speed increased as well [7].

Clamping force on the pulley produces a normal force, which results in traction force between a V-belt’s element and the pulley (Tarutani et al. 2005, Nishizawa et al., 2005). Insufficient clamping force causes an excessive slipping of the V-belt and causes the engine to develop more power for the hydraulic pressure of the CVT to maintain the clamping force [8,9,10]. Therefore, to address these issues, some researchers have explored a novel idea of replacing the application of the existing electro-hydro-mechanical (EHM) system for the clamping force with an electro-mechanical (EM) system in a pulley-based CVT (EM CVT). The author [11] has studied on the disk spring based EM-CVT and reported that the gear can be achieved 3.0 for reduction and 0.6 for overdrive. The researcher have not reported about the actuation time for the movable sheave towards the fixed sheave. The actuation time is the core objective for this study. It could be mentioned that if the actuation time is more the vehicle will be in no traction force, which is the main reason for the customers of CVT to feel ‘No-Traction” on hilly track.

The market for belt type CVT systems is growing rapidly. Since belt production began at Van Doorne’s Transmissie (VDT) in 1985, more than 5 million vehicles have been equipped with a push-belt CVT. Today, approximately 1 million push-belt CVTs are manufactured annually for the Japanese, USA, European and Korean markets. More than 45 different vehicle models are currently available with push-belt CVT systems. The push-belt performance is illustrated by the CVT system installed in the Nissan Murano, which has a ratio coverage of 5.4 and operates with a 3.5 liter V6 180 kW/350 Nm engine and a torque convertor, applying drive side torque levels on the belt over 500 Nm. Further extension of the CVT application range can be achieved, among other things, by continuously increasing the push-belt fatigue strength [2].

The current state-of-the-art of push-belt performance can be illustrated by using the EMA-CVT, which could have fast actuation to accelerate the vehicle in 5-7 sec without incurring fatigue strength and transfer the power to the tires in any road conditions with targeted CVT ratio and without making noise and jarking, good drivability, and better fuel economy.

The objective of this paper is to present a intelligent control electromagnetic actuated CVT system for the vehicle to make the vehicle energy efficient by reducing the acceleration time, ensuring optimal traction in hilly track, reducing transmission losses and noises.

2. Methodology
The EMA-CVT is developed with a small battery pack of 12V with 2 sub module of lithium ion battery (each module is built with 4 cells 2 in series and 2 in parallel; nominal cell voltage 3.5 V and capacity 43 Ah), 1.6 mm diameter copper coil of 900 m length and a high carbon steel plunger. The alternator always keeps battery full charging mode to power the EMA as required. This EMA-CVT system’s development is conducted with incorporating the car traction torque dynamic analysis. The EMA design has been performed in such a way that the drive mechanism of the actuator be able to develop the electromagnetic force to actuate the CVT for the maximum gear ratio at start-ups or slow speed (reduction) and lowest gear ratio for maximum speed (overdrive or cruising speed).
2.1 Kinematics of EMA-CVT

Kinematics of the EMA-CVT is developed based on the traction torque of the car and the clamping force of the CVT. The electromagnetic force analysis of the EMA has been made based on the clamping force of the pulley for the different number of windings and the length of the EMAs. The electromagnetic force would be greater than and equals to the clamping force of the pulley to develop the highest gear ratio for start-ups or low speeds and the lowest gear ratio for the highest speed. The mathematical model has been developed for estimating the driving torque of the primary pulley based on the engine of the full-scale car. The transmission dynamics of the EMA-CVT could be modeled as:

\[
T_{inw} = [T_e - J_e \alpha_e]GR
\]

with

\[
\alpha_e = \frac{1}{J_e} [T_e - T_{in}]
\]

where, \(T_{inw}\) is the torque applied to the wheels by the EMA-CVT, \(J_t\) is the inertia of the transmission unit and \(GR\) is the gear ratio, \(\alpha_e\) engine angular acceleration, \(T_e\) is the torque generated by the engine and \(T_{in}\) is the torque applied by the engine to the CVT and \(J_e\) is the inertia of the engine. For electromagnetic actuated CVT system, CVT speed ratio as well as instantaneous torque is changed with the axial (back and forth) movement of pulley movable sheave through attraction and repulsion of solenoids. The instantaneous torque incremental for the primary pulley of the EMA-CVT is calculated by using the equation:

\[
\Delta T_{ins} = \frac{T_{out(max)} - GR_{ins}T_{in(min)}}{GR_{ins} + 1}
\]

with

\[
GR_{ins} = 1 + \frac{\tan \theta (L_s - 2ds)}{r + \tan \theta ds}
\]

where, \(T_{out(max)}\) is the maximum output torque in the secondary pulley and \(T_{in(min)}\) is the minimum input torque in the primary pulley from the engine and \(GR_{ins}\) represents the instantaneous gear ratio, \(L_s\) is the stroke length in m, and \(ds\) is the instantaneous displacement of the movable sheave in m, \(r\) is the radius of pulley in m and \(\theta\) is the pulley angle in deg. The traction torque for initial condition of the car is computed [12]:

\[
T_{initial} = T_l g \left( \frac{l_f f h}{r} \right) \left( \frac{L_w}{1 + \frac{h}{L_w} (r_{wheel})} \right)
\]

where, \(\mu_R\) is the adhesion coefficient of the road, \(m_c\) is the mass of car in kg, \(f_i\) is the rolling motion resistance coefficient, \(h\) is the height of the centre of gravity in m, \(L_w\) is the wheel base in m, \(l_f\) is distance of the CG from the front wheel in m, \(\theta_s\) is the slope angle of the road in deg, \(g\) is the acceleration due to gravity in m/s\(^2\), \(r_{wheel}\) is for radius of the drive wheel in m. The power transferred from the engine to the primary pulley and from the primary to secondary pulley can be computed as:
\[ P_{in} = \left[ \frac{\Delta v_p + v}{R_p} \right] (T_{in}) \quad \text{and} \quad P_{out} = \left[ \frac{v - \Delta v_s}{R_s} \right] (T_{out}) \]  \tag{4}

where, \( P_{in} \) is the power generated by the engine or input power of the primary pulley in Watt and \( P_{out} \) is the output power of the primary pulley in Watt and \( \Delta v \) is slip velocity in m/s. The slip velocity for the primary pulley and the secondary pulley can be calculate as, \( \Delta v_p = \omega_p R_p - v \) and \( \Delta v_s = v - \omega_s R_s \), where \( \omega_p R_p \) and \( \omega_s R_s \) is the speed of the primary and secondary pulley, \( v \) is the belt speed, \( \Delta v_p \) and \( \Delta v_s \) are the slip velocity respectively.

### 2.2 Clamping Force

The EMA gets advantage to pull the movable sheave whereas the difficulties for pushing the sheave against the rotating belt which is considered as the clamping force (Figure 1). Two sets of solenoids with common plunger are used in this study to overcome the clamping force with developing the electromagnetic force. In each sets of solenoids one develops the pulling force and other one develops pushing force.

\[ F_p = \frac{T_{in} \cos(90 - \theta_b)}{2\mu_p R_p} \]  \tag{5}

\[ F_s = \frac{T_{out} \cos(90 - \theta_b)}{2\mu_s R_s} \]  \tag{6}

where, \( T_{in} \) and \( T_{out} \) is transmission torque of primary and secondary pulley in Nm, \( \theta_b \) is the belt angle, \( \mu_p \) and \( \mu_s \) belt frictional coefficient of primary and secondary pulley. \( R_p \) and \( R_s \) are radiuses in m of primary and secondary pulley respectively.
The developed electromagnetic actuator (EMA) produces magnetic force must be higher than that of clamping force to push the movable sheave of the fully towards the fixed sheave to maintain the gear ratio of the CVT in start-ups or on the hilly traction. The acceleration time of EMA-CVT system for ¼ scale car Nm is estimated by using the equation,

\[ t = \frac{m_c r_{\text{wheel}} R_p}{R_s T_e (v_c)} \]

where, \( m_c \) is the mass of the car in kg, \( r_{\text{wheel}} \) is wheel radius in m, \( v_c \) is car velocity in m/s and \( t \) is the acceleration time.

### 2.3 Electromagnetic Force

The pushing and pulling the movable sheaves of the pulleys by EMA to meet the load demands of the car is indispensable. The actuator solenoid has been designed to develop the maximum electromagnetic force to overcome the maximum clamping force. Each set of the solenoids has been equipped with individual pulley which one has been used for pushing and the other one for pulling along with a common plunger.

The mathematical models are developed for the EMA by considering the dynamic behavior of the magnetic flux, density, strength, electromagnetic force and energy according to the Faraday’s Law, Ampere’s Law and Lenz’s law, Maxwell’s dynamic condition, and the modified equations [15] (Hayt & Buck, 2006). The magnetic concentration is gathered in the centre of the solenoid \( P \). Therefore, overall magnetic flux density can be presented [13]:

\[
\mathbf{B} = 2 \sum_{\alpha} \sum_{\beta} \left\{ \frac{\mu_0}{4\pi} \left( \int_{S_{\text{wire}}} \mathbf{J} \cdot d\mathbf{S}_{\text{wire}} \right) \right\} \frac{a}{2} \left[ \sum_{\alpha} \sum_{\beta} \right] \left( \frac{\mu_0}{4\pi} \left( \int_{S_{\text{wire}}} \mathbf{J} \cdot d\mathbf{S}_{\text{wire}} \right) \right)
\]

\[ = 2 \sum_{\alpha} \sum_{\beta} \left\{ \frac{\mu_0}{4\pi} \left( \int_{S_{\text{wire}}} \mathbf{J} \cdot d\mathbf{S}_{\text{wire}} \right) \right\} \frac{a^2}{2} \left[ \sum_{\alpha} \sum_{\beta} \right] \left( \frac{\mu_0}{4\pi} \left( \int_{S_{\text{wire}}} \mathbf{J} \cdot d\mathbf{S}_{\text{wire}} \right) \right)
\]
where, $\mu$ is magnetic permeability, $dH$ magnetic flux intensity along the $z$ direction $\alpha$ is the angle between resultant magnetic flux intensity and $z$ axis. $J$ is the current density across wire segment cross section $S_{wire}$, $a$ is the radius of the wire segment from the centre of the solenoid and $dl$ is the length of the wire segment split. $\mu$ is magnetic permeability, $dH$ magnetic flux intensity along the $z$ direction $\alpha$ is the angle between resultant magnetic flux intensity and $z$ axis. $J$ is the current density across wire segment cross section $S_{wire}$, $a$ is the radius of the wire segment from the centre of the solenoid and $dl$ is the length of the wire segment split. $h_2$, $h_1$ is the outer and inner radius of the solenoid from the centre $P$. $L_{solenoid}$ is the length of the solenoid. It is noted that the development of $F_{em}$ is the function of supplied current (i.e. $F_{em} = f(a)\left(\int JdS_{wire}\right)$). The supplied current of the solenoid is controlled with the controlling the voltage for the desired $F_{em}$. The main purpose of controlling current is to prevent the actuator from temperature spike. Figure 2 shows the development of electromagnetic force. The magnetic flux (B) develops with supplying current to each loop of the EMA solenoid. The magnetic flux squeezes to the solenoid force. The magnetic flux (B) develops with supplying current to each loop of the EMA solenoid. The magnetic flux squeezes to the solenoid force. The magnetic flux (B) develops with supplying current to each loop of the EMA solenoid. The magnetic flux squeezes to the solenoid force. The magnetic flux (B) develops with supplying current to each loop of the EMA solenoid. The magnetic flux squeezes to the solenoid force. The magnetic flux (B) develops with supplying current to each loop of the EMA solenoid. The magnetic flux squeezes to the solenoid force. 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$$\ddot{F}_{em} = \frac{\mu N_{wind\ per\ length} I_{single winding}}{2L_{solenoid}} (\sin \alpha_2 - \sin \alpha_1)$$

Simplifying Eq. 7, the total $B$ at point $P$ can be estimated by using the equation [13]:

$$B = \hat{z} \sum_{a=n}^{m} L_{solenoid} \sum_{z=n-L_{solenoid}}^{z=2L_{solenoid}} \left( \frac{\mu \left( \int JdS_{wire} \right) a^2}{2 \sqrt{a^2 + z^2}} \right)$$

$$= \hat{z} N_{loop} \frac{\mu N_{wind\ per\ length}}{2L_{solenoid}} (\sin \alpha_2 - \sin \alpha_1)$$

with radius ($a$) for the first loop of the solenoid depends on solenoid inner dimension ($h_3$). Therefore, the electromagnetic force:

$$F_{em} = N \frac{I^2}{L_{solenoid}} (\sin \alpha_2 - \sin \alpha_1) L_{single\ winding}$$

If the solenoid length is much larger than its radius, then $\alpha_2 \simeq 90^\circ$ and $\alpha_1 \simeq (-)90^\circ$ in which Eq.9 reduced to:

$$F_{em} = N \frac{I^2}{L_{solenoid}} L_{single\ winding} = N \frac{I^2}{L_{solenoid}} 2a$$
with $m = a \cdot r$

where $a$ is the radius of the wire segment from the centre of the solenoid and $L_{solenoid}$ is the length of the solenoid and number of windings $N$ are defined for satisfying the desired magnetic force.

![Figure 3. Assembly of Developed EMA and plunger](image)

![Figure 4. Electromagnetic force for the variation of current and coil windings](image)

Table 1. Specification of EMA

| Parameter                                                                 | unit | Value   |
|----------------------------------------------------------------------------|------|---------|
| Coil diameter ($d_w$)                                                      | mm   | 1.6     |
| Cross sectional area of wire                                              | mm$^2$ | 2.01   |
| Length of solenoid ($L_{solenoid}$)                                       | mm   | 200     |
| Solenoid Minimum diameter ($h_1$)                                        | mm   | 31      |
| Solenoid Maximum diameter ($h_2$)                                        | mm   | 171     |
| Cross-sectional surface of the Solenoid                                   | mm$^2$ | 755    |
| Packing factor, $f_p = \frac{105(\text{Actual windings})}{200(\text{Theoretical windings})}$ |      | 0.525   |
| Number of turns ($N$)                                                     | turns | 2843    |
| Length of coil ($L_w$)                                                    | mm   | 90232   |
| Volume of the Solenoid coil ($Vol^m$)                                     | mm$^3$ | 2.3×10$^6$ |
| Mass of EMA                                                               | kg   | 20.65   |
3. Development Of EMA

The EMA-CVT in this study is designed and developed for a car of mass 633 kg. It is developed with different size of solenoid for pushing and pulling the movable sheave towards and away from the fixed sheave is shown in Figure 3. The bigger solenoid has been used to push the movable sheave towards the fixed sheave to develop the maximum gear ratio (GR) of 3.6 to develop the maximum torque (reduction) or the vehicle on uphill gradient of 10%. While, the smaller solenoid has been used to take away (pull) the movable sheave from the fixed sheave to develop the minimum GR of 0.65 for the car is in overdrive. EMA for generation of electromagnetic force is designed in such a way that it can push and pull the rotating pulleys. Figure 4 shows the magnetic force produces by the EMA for current. The dynamic analysis of EMA with rear spring and plunger by applying the motion equation,

\[ - \sum F = m \ddot{x} \]

\[ F_{em} - F_{clampingforce} = m \ddot{x} + kx \]  \hspace{1cm} \text{(11)}

where, \( m \) is total mass of pulley and plunger which close to 1.5kg, \( k \) is the spring stiffness, \( x \) is the displacement of pulley or plunger here it is known as stroke length, \( \dot{x} = \frac{T_{out}}{2F_{em} \tan(90 - \theta_b)} \) with

\[ dx = \frac{dr}{\tan} \quad \text{and} \quad \dot{x} = \frac{1}{2F_{em} \tan \theta} \frac{d^2T_{out}}{dt^2} \]

The equation (11) can be rewritten as,

\[ \mu NI^2 - \frac{T_{out} \cos \theta_b}{2\mu_s R_s} = m \ddot{x} + kx \]

\[ \frac{NI^2}{2} \frac{T_{out} \cos \theta_b}{R_s} = m \int \frac{d^2T_{out}}{dt^2} + k \int \frac{T_{out}}{F_{em} \tan} \]

\[ \frac{m}{2F_{em} \tan} \frac{d^2T_{out}}{dt^2} + \left( \frac{k}{2F_{em} \tan} + \frac{\cos \theta_b R_s}{2\mu_s} \right) T_{out} = NI^2 \]  \hspace{1cm} \text{(12)}

where, \( \mu \) is magnetic permeability, \( N \) is number of windings, \( I \) is the supplied or consumed current, \( T_{out} \) is the torque output, \( \theta_b \) is the belt angel, \( \mu_s \) is the frictional coefficient, and \( R_s \) is the pulley radius.

The analytical mathematical model of the EMA-CVT can be fitted with the Laplace transfer function for getting the gain on vehicle traction torque (\( T_{out} \) with the current control) and can be modeled as,

\[ aS^2T_{out}(S) + bT_{out}(S) = \frac{2c}{S^3} I(S) \]  \hspace{1cm} \text{(13)}

where, \( a = \left( \frac{m}{2F_{em} \tan \theta} \right) \), \( b = \left( \frac{k}{2F_{em} \tan \theta} + \frac{\cos \theta_b R_s}{2\mu_s} \right) \), \( c = \mu N \)

Therefore, \( \text{Gain} = \frac{T_{out}(S)}{I(S)} = \frac{2c}{S^3(aS^2 + b)} \)
where, $T_{\text{out}}$ is the input torque which defines the traction torque of the vehicle and can be modeled as,

$$T_w = n_d T_{\text{out}},$$

where, $T_w$ is the vehicle traction torque and $n_d$ is the differential constant.

3. RESULT AND DISCUSSION

A $\frac{1}{4}$ scale car is considered for this study both for the theoretical and experimental performance investigation of EMA. Figure 3 shows the EMA with plunger for controlling the primary pulleys operating diameter.

3.1 Theoretical Performance

The EMA-CVT is able to develop 274 N force equivalent to clamping force on the 10% with current supplying 6 A. Figure 5 shows the $\frac{1}{4}$ scale car model with EMA-CVT. Figure 6 shows the performance of the CVT movable sheave operation with applying 100 N load to the shaft of the secondary pulley’s fixed sheave. The larger EMA develops the electromagnetic force to move the movable sheave towards toe fixed sheave to start-ups, which is called reduction. Total force push the movable sheave, $F_{\text{pushing}} = F_{\text{em}} + k_s \delta_s$, where, $F_{\text{em}}$ is the electromagnetic force in N, $k_s$ is the spring stiffness in N/m, and $\delta_s$ is the spring deflection in m. While, the smaller EMA develops the electromagnetic force to pull the movable sheave towards the EMA and against the spring force, $F_s = k_s \delta_s$ for the vehicle overdrive. Gear ratio of the CVT has made 1.0 by controlling the same diameter for both of the pulleys.

Result (Figure 6a) shows that the pushing time is greater than the pulling time. However, both of the actions of the pulley is equalized with current supplying 4 A and more. This happens because of setting-up the gear ratio where the pulley does not require any force to oppose the clamping force and releasing force. The responding time for static condition is less compared to with load and without load (Figure 6b). It is obvious that during pulling EMA exercises some benefits due to mechanical advantage. In general, the travelling time (responding time) of the plunger of the EMA for pulling mechanism lower than the travelling time of pushing mechanism. Therefore, with a proper development of EMA, a typical car alternator with 40 Ah gives optimal gear ratio in any load condition within the range.

![Figure 5a. CAD drawing of proposed quarter scale car with EMA-CVT system [12].](image-url)
The EMA of the CVT is mainly contributes on the actuation with developing electromagnetic force. The electromagnetic force develops due to the development of magnetic field for the supplying to the EMA. The electromagnetic force increases with increasing the current supply to the solenoid as shown in Figure 7. The electromagnetic force is the function of current supply and number of windings, which can be presented as, \( F_{em} = f(I, N) \).

![Figure 5b. Developed ¼ scale car with EMA-CVT system [12].](image)

![Figure 6. EMA performance on moveable sheave operation.](image)
3.2 EMAs’ Power Control

A movable sheave position control system with fuzzy logic controller design is proposed to realize the electromagnetic force targets and thus minimize the total power consumption and eliminate unnecessary power loss based on the rpm of drive wheel only on simulation. It has the advantage of fuzzy controller being simple (relations between input and output variables can be explained in a linguistic-based rule base), robust (performance is not depending on training and new input variables and rules can be easily added) and not requiring precise mathematical model (Carman, 2008). In the control system of the solenoid, wheel torque (T) is selected as controlled variable and power supply (P) as regulated variable through the change in variable resister knob position. The Fuzzy-Proportional-Derivative-Integrator (FPID) controller acts as a self PID tuner. The structure of the FPID controller is shown in Figure 8. The fuzzy logic controller (FLC) with FPID is preferred to control the EMA of the proposed study to control the EMA-CVT to meet the load demands of the car. The FLC is used in this study to control power supply to the EMA for maintaining the desired traction torque of the car in different road conditions.

![Figure 7. Performance of EMA for different current.](image)

![Figure 8. Block diagram of the control system.](image)
Based on the difference between measured value of torque \((Y_a)\) and reference value \((Y_r)\), the plunger position i.e. movable sheave displacement is controlled by a regulation of variable, i.e., power supply \((P)\). The reference torque is calculated based on the maximum allowable (maximum boundary) generated torque by the motor and then is compared with the measured torque numeric values using dynamometer (rpm sensor) and slope sensor. Hence, the resultant deviation, i.e., torque error \((TE)\), and differential torque or rate of torque error \((RTE)\) are continuously calculated in operation (Figure 9).

The controlled variable is considered as Torque Error \((TE)\), Rate of change of Torque Error \((RTE)\) and regulated variable as Power consumption \((P)\) by EMA actuator. It is noted that the values of membership functions \((TE, RTE)\) change their values with respect to time. For instance, the degree of \(TE\) changes in Nm from 5 to 90, \(RTE\) is fluctuates in Nm/s from -5 to 5, and \(P\) is measured in watt from 50 to 200, respectively. It is noticed that different membership functions acquire on zero and nonzero values indicating the degree to which the linguistic value suitably illustrates the present value of \(TE\). For example, at \(TE = 5\); it is certain that the torque error is “very low (Vlow)”, and as the value of \(TE\) moves toward 26.25 it is become less certain that it is “Vlow” and more certain that it is “Low” as shown in Fig.9. The membership function of the fuzzy inputs from fuzzy logic expert system where \(i\) stand for input from into Eq. (14), (15) and (16).

\[
\mu_{Vlow}(TE(i)) = \begin{cases} 
1; & (TE) \leq 5 \\
\frac{26.25 - (TE)}{21.25}; & 5 < (TE) < 26.25 \\
0; & (TE) \geq 26.25 
\end{cases} 
\]  

(14)
The fuzzy inference system (FIS) seeks to determine which rules fire, to find out which rules are relevant to the current situation. For crisp input $TE (i_1) = 20$ Nm, and $RTE (i_2) = 2.5$ Nm/s, the rules 3 and 6 are satisfied to be fired. The firing strength of truth values of membership function for individual rules is obtained as:

$$\alpha_3 = \min \{\mu_{\text{low}}(TE), \mu_{\text{pos}}(RTE)\} = \min \left(\frac{26.25 - 20}{21.25}, \frac{2.5 - 1}{2}\right) = 0.294$$

$$\alpha_6 = \min \{\mu_{\text{low}}(TE), \mu_{\text{pos}}(RTE)\} = \min \left(\frac{20 - 5}{21.25}, \frac{2.5 - 1}{2}\right) = 0.705$$

### 3.3 Experimental Result

Electromagnetic force is the heart of this current CVT system, which offers desired gear ratio by placing the pulley shave to its exact position. Figure 10 shows the behavior of electromagnetic force response of the system with varying the current supply. It supposed to be linear change with respect to power followed gear ratio. However, due to slip in belt and pulley during the change of gear ratio the electromagnetic force trend is showing its fluctuating behavior. Sufficient Electromagnetic force is indispensible for perfect gripping the belt and pulley.

Point ‘A’ (Figure10) shows the initial start-ups of the car, which is called reduction, the ¼ scale car needs gear ratio 2.33 for the supply current to the EMA of primary pulley’s EMA. While, point ‘B’ indicates the overdrive of the car at gear ratio 1.1 and the current required 2.85A to EMA to push the primary pulley’s movable shave towards the fixed pulley. Figure 11 shows the electromagnetic force developed by the EMA by the supplying current. The electromagnetic force 144 N develops by the EMA for the current 1.35 A for the initial start-ups while electromagnetic force develops 248 N for the current supply 2.89 N. It is noted that the secondary pulley movable shave’s initially placed at maximum position for the reduction or initial start-ups. Beyond the point ‘B’ the movable shave needs to push towards the fixed to get the reduction gear ratio for developing the maximum traction.
force for the vehicle. In this case the EMA needs current supply 3.37 A to develops the electromagnetic force of 320 N. The develops traction force for the car by the engine is estimated as,

\[ F_t = n_d \cdot \frac{T_e}{r_w} \]

where, \( n_d \) is the differential speed ratio, \( \eta \) is the transmission efficiency in percentage, \( T_e \) is the engine torque in Nm, and \( r_w \) is the wheel radius in m. By referring to the current supplied to the coil, the variety of responding (travelling) time of the movable sheave is recorded. It is difficult to adjust infinite number of gear ratio by manual control. In order to ensure this intelligent control system would be a better solution. Fuzzy logic intelligent system has been considered for this study to maintain the gear ratio as required for the car to maintain its traction power and cruising speed.

![Figure 10. Current requirement for CVT gear ratio.](image)

![Figure 11. Electromagnetic force, Gear Ratio VS Supply current.](image)
Figure 12. Travelling time vs Power Supply.

Figure 12 illustrates the gear ratio with respect to power where higher power was observed during higher gear ratio under any pushing condition of pulley sheaves. It seems to be that pushing and pulling time is equal. However, in practice the travelling time (responding time) of the plunger of the EMA for pulling mechanism was found significantly lower than that of pushing mechanism. The basic reason was the load on matching area of the sheaves is higher and the pulling of the sheave was found easier than the pushing. Furthermore, clamping releasing force always tends to pull the movable sheave of the pulley towards the solenoid of the EMA while, clamping force push the movable sheave against the rotating belt. Spring could be enhancing the pulling time even further but it would consume more power during pushing. It should be noted that the movable sheave of the pulley during push needs to press against the rotating belt, and spring to create the higher GR for the higher torque of the car in starting and inclined road. Hence, the electromagnetic force required to push the movable sheave, \( F_{\text{push}} = F_{\text{em}} = \frac{n_g T}{r_w} k_s \delta \) while the pulling force, \( F_{\text{pull}} = F_{\text{em}} = k_s \frac{n_g T}{r_w} x \), where, \( k_s \) is the spring constant in N/m, \( \delta \) is the spring deflection in m, and \( x \) is the fraction of traction force. It is noticed during experiment that for low rpm (in start-ups or climbing slope) acceleration time is higher compared with the other kinds of rpm. High torque means high clamping force of the CVT, which needs to overcome by the solenoid. Once the car get its full inertia need less time to be accelerate. This minimum acceleration time or travelling time otherwise; is the consequence of fast response characteristics of solenoid.

3.3.1 Fuzzy Simulation Verification
Dynamic torque behavior in between 30-90 Nm is observed in both experimental and fuzzy simulation method shows in Figure 13. The fuzzy simulation can be verify by experimental data taken form the \( \frac{1}{4} \) scale model. Fuzzy data has been extracted for the MATLAB workspace. Fig.13 represents the torque characteristics of the final drive. Transmitted torque to the final drive has only 2.74% of average deviation with respect to gear ratio because of belt slippage as well as electromagnetic force. Enhanced electromagnetic force could be a better solution for avoiding belt slip. It is noticed that 108 N electromagnetic forces is required in the primary pulley solenoid to transmit the initial torque when
Figure 13. Torque verification with experimental and fuzzy simulation.

\[ T_{\text{fuzzy}} = -0.0072 F_{\text{em}}^2 - 1.7723 F_{\text{em}} + 88.65 \]
\[ R^2_{\text{fuzzy}} = 0.9986 \]

\[ T_{\text{exp}} = 0.0561 F_{\text{em}}^2 - 3.653 F_{\text{em}} + 92.479 \]
\[ R^2_{\text{exp}} = 0.9508 \]

Figure 14. Correlation between actual and predicted value of Torque.

\[ R^2_{\text{goodness of fit}} = 0.9099 \]

Figure 15. EMA-CVT performance over manual transmission.

\[ T_{\text{wheel for CVT}} = -0.144 GR^2 + 12.802 GR + 27.467 \]
\[ R^2_{\text{CVT}} = 0.931 \]

\[ T_{\text{wheel for MT}} = 7.1479 x^2 - 18.812 x + 42.248 \]
\[ R^2_{\text{MT}} = 0.9959 \]
the gear ratio is 1.8. It is also observed that to keep the car in motion the minimum torque needed 32.6Nm for this current set up. The correlations between experimental values and predicted (FLES) values of traction torque have been illustrated in Figure 14. The correlation coefficient of traction torque or goodness of fit is found as 0.9099. It indicates that the predicted data over the measured data have a closed agreement and thus, validity of the fuzzy simulation results. However, the mean relative error of measured and predicted values from the FLES model on traction torque is found as 9.01% since this study neglecting some transmission losses. Relative error of predicted values is in the acceptable limits of 10%. The goodness of fit gives the ability of the developed system and its highest value is 1 according to statistical method [18,19].

The final torque in the wheel is compared with the down scaled conventional VIVA manual car in Figure 15. The available five transmission ratio is considered as it is discrete transmission unit for same final gear ratio 3. Similar gear ration has been chosen from a number of gear ratio in CVT for EMA in order to observe torque transmission performance of CVT compared with manual. CVT perform better in the region of low torque of gear ratio. Afterward, it became stable at gear ratio 2 and leveled off. It represents the up to certain limit CVT performed much better to its counterpart manual transmission. However, for high torque this system is not applicable. This is why the concern of this study to implement this system in small car not in heavy vehicle.

Figure 16 shows after an hour operation of the solenoid with maximum current flow the temperature jumps to 44°C from the ambient temperature. To some extend in thermal management point of view this heat generation is a matter of deep concern. Heat sink could be better solution; However, current experiment overlook this constrain of the system which could be the further scope of research.

4. Conclusion
i. The EMA able to develop the electromagnetic forces 101.22 N to 274.82N equivalent to the clamping forces by supplying current in the range of 1-4 amps with 36 volts.
ii. The EMA is able to develop the electromagnetic forces 101.22–274.82 N and dynamic torque 31.38 - 85.19Nm at transmission gear ration 0.820 - 2.369.
iii. The solenoid able to pull and push the plunger in the desired distance when current supply is 3 amp in almost same time. While, it is different when the current supply to the EMA is less than 3 amp.

iv. The responding time of the plunger for kerb weight of the ¼ car in static condition is less compared to with-load of 15kg.

v. The fuzzy simulation represents the time required reaching the desired maximum torque 85.19 Nm is 3.75sec.and

vi. The corresponding current is 4.5 or the maximum desired torque 85.19 Nm.

This virtual control of the EMA operated CVT with fuzzy controller helps the system to operate manually when the real fuzzy controller is not attached with EMA in practice.

Experimentally, it is observed that:

i. The EMA develops the electromagnetic force in the ranged of 108–301 N which has been maintained with supply current maximum 3.37 amp.

ii. The traction torque has been estimated maximum 90 Nm by using the electromagnetic force 301N which has been found with the corresponding supplied current 3.37 amp for the maximum gear of 1.8.

iii. Travelling time of the plunger is in the range of 1.8– 3.1sec without loading. It shows higher with the incremental loads. However, prime mover was unable to transmit torque when the applied load increased more than 3kg. This could be due to the lower rating of prime mover.

iv. The correlations between measured (experimental) and predicted (FLES) values of traction torque has found 90.99%, which closely verify the fuzzy simulation model.

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