Development of simulation approach for CVT tuning using dual level genetic algorithm

Deepinder Jot Singh Aulakh*

Abstract: Presented work aims to develop Genetic Algorithm (GA) based simulation approach for tuning of Continuously Variable Transmission (CVT). This study uses force balance to model the behaviour of CVT in MATLAB and employs dual level GA to optimize the tuning variables for desired output from CVT i.e. engagement of belt and sheaves at peak of engine torque curve, start of shifting at peak of engine power curve and keeping constant engine RPM (peak of power curve) during shifting. The variables for tuning are flyweight mass, primary and secondary spring stiffness and profile of primary and secondary cam. The results obtained from simulation are validated through experimental testing. The simulation results show good coherence with experiments in terms of engagement and shift starting RPM and also most of the shifting occurs at constant RPM. Also, the behaviour of CVT tuned by simulation is compared with the traditional method of experimental tuning and results obtained show that the simulation method is comparable to the traditional method in terms of accuracy. This study concludes with strong confidence in the potential of GA simulations for tuning.

Subjects: Automotive Design; Automotive Technology & Engineering; Computer Aided Design (CAD)

Keywords: CVT; genetic algorithm; tuning; simulation; force balance; fitness function; V belt

1. Introduction

Continuously variable transmissions (CVT) are the popular mode of power transmission in automobiles. The operation of V belt CVT is controlled by components such as flyweight mass, primary and secondary cam, spring stiffness, and the belt and sheaves profile. The CVT is designed to provide a smooth and efficient transmission of power from the engine to the wheels, allowing for continuous adjustment of the gear ratio. This allows for a wide range of speed and torque for different driving conditions, improving the overall performance and fuel efficiency of the vehicle.

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Deepinder Jot Singh Aulakh is currently working as a quality engineer at Heromotocorp Ltd. He has done his Bachelor of Technology in Mechanical engineering from NIT Jalandhar. His key area of interest is dynamics of road vehicles. His work include design of retractable steering column and variable length wishbones for suspension of off road vehicles and converting 3,100 hp diesel engine to 3,300 hp in a project with Indian railways. CVT are vastly used as mode of power transmission in automobiles. The research reported in this paper presents an alternate approach of computer simulation as compared to previous CVT tuning techniques which involved experimental testing to tune the transmission. The results obtained are quite encouraging as the output of CVT is very close to expected results. Thus opening a further prospective in CVT tuning.

PUBLIC INTEREST STATEMENT

Traditional method tuning of CVT is presents a challenge for engineers as they involve large number of experimental iteration and involve rigorous task of changing the components of the CVT physically. Study presented in this paper focus on providing an alternative to this by developing a computer simulation that could eliminate the task of experimental tuning. The user just need to know the desired output from the CVT and its measurements to obtain the set of parameters that would tune the CVT. The approach proposed in this paper is validated by experiments and the results are quite encouraging.
secondary cam profile, and primary and secondary spring stiffness. CVT tuning is the process of deciding on which configuration of these components we want our CVT to operate, to obtain the desired output. Aaen Olav worked extensively on CVT tuning. In his work, the focus is on experimental methods to tune the CVT. By extensive testing, the parameters of CVT are decided upon for optimal operation (Aaen, 2007). But these methods are time-consuming and costly, and also errors associated with data acquisition can cause improper tuning (Aaen, 2007). The computer simulation approach can be used to overcome this problem as it is faster and did not require resources as in experimental methods.

There is the good amount of work in the field of computer simulation of CVT, as Seigars, developed a conceptual operation of a CVT and formulated a model of a CVT system using Adams View (Seigars, 1996). The disadvantage of this model was the inability to have dynamically changing diameters on pulleys during motion. This made it impossible to build a conventional model for the full range of motion of a CVT shifting.

Ariyono et al. developed a nonlinear system by using Adaptive Neural Network Optimization Control (ANOC) that indirectly control the engine speed by adjusting the CVT pulley ratio (Ariyono et al., 2007). Schulte, also developed a actuated control system using Takagi- Sugeno (TS) fuzzy observer based LMI for CVT to have an optimized engine performance (Schulte, 2010). The problem with this approach is that an additional control system for CVT needs to be mounted on the vehicle for keeping the engine in the satisfactory RPM range.

Timothy R. developed simulation technique for tuning of CVT using force balance equations. His technique involved calculation of flyweight mass, spring stiffness (Timothy, 2013). Willis C. R. also developed a simulation-based technique for calculation of flyweight mass and spring stiffness using energy based equations (Willis, 2006). Although Timothy and Willis provided a strong basis for simulation-based approaches. But the emphasis was only laid on shift starting and engagement RPMs while straight shift and backshift characters were not considered. Also, both of these studies did not consider the torque feedback ramp, cam profiles and belt characteristics as variables (Timothy, 2013; Willis, 2006). While in actual tuning ramp and cams are also changed and belt affects the axial forces produced in CVT (Aaen, 2007).

Presented study focuses on filling these gaps by developing new genetic algorithm based tuning technique. For this dual level Genetic algorithm in MATLAB program based on force balance model of CVT is used to simulate the behavior of the CVT. The use of GA helps the governing equation to be optimized with considering all the factors as variables which were not the case in previous studies. As in work of Timothy (2013) and Willis (2006), the governing equations are solved for one or two variables with respect to the constant value of other variables thus limiting the optimization range. The profiles of cam and torque feedback ramp (secondary cam) are also one of the variables and straight shift (constant RPM) is given primary importance in GA optimization. Since the force balance equations are developed from V belt of CVT, the belt characteristics also get consideration (Beccari & Cammalleri, 2001; Sorge, 1996a, 1996b; Sorge, Beccari, & Cammalleri, 2001). Also, the analysis is done for entire shifting range of the CVT (Seigars, 1996) and there is no need to mount any additional control system on the CVT as in thus simplicity of the system is maintained (Ariyono et al., 2007; Schulte, 2010).

1.1. Genetic algorithm
GAs are optimization algorithms for computational problems by maximizing or minimizing fitness functions (Kinnear, 1994). For this, they imitate the natural processes such as of reproduction, evolution, mutation and natural selection to obtain ‘fittest’ solutions (Goldberg, 1989). These algorithms are more powerful and efficient than random search and exhaustive search algorithms without the need of any extra information about the given problem (Mitchell, 2007). GA has remarkable abilities which include being able to solve non-smooth, non-continuous, non-differentiable cost functions, to escape the local optima and to move towards the global optimum. Due to these inherent
advantages, the genetic algorithm (GA) is employed in this study as the solution space is vast and several local optima should be possible (Goldberg, 1989; Kinnear, 1994; Mitchell, 2007).

The basic components common of genetic algorithms are a fitness function which we need to optimize, a population of chromosomes i.e. Candidate solution, selection of which chromosomes will reproduce crossover to produce next generation of chromosomes, random mutation of chromosomes in a new generation (Goldberg, 1989).

2. CVT and its characteristics

2.1. CVT operation

Ideally, the whole operation of CVT is divided into four phases Idle, Engagement, Straight shift and Shiftout as shown in Figure 1. In Idle phase engine speed is below CVT engagement speed because flyweight does not create enough centrifugal force for primary pulley sheave to shift and engage with the belt. The belt seats in a low ratio (lower gear) setting (Aaen, 2007).

In engagement phase engine speed is sufficient to cause enough flyweight centrifugal force to compress the primary spring and engage the belt. This phase starts with initial slipping between belt and sheaves as engine speed is not high enough to clamp the belt properly but as the engine speed increases the centrifugal force of flyweight increases and causes enough clamping force so that the slipping is eliminated and engine speed increases and vehicle accelerates along the low ratio (Aaen, 2007).

Straight Shift starts when the engine speed reaches an optimal power output rpm (usually peak power rpm) the primary pulley start to push belt out of its seating area and thus increasing its diameter and decreasing secondary diameter starting the shift. Ideally, the vehicle accelerates consistently through its range of speed while engine speed remains constant and CVT shift from lower gear ratio to higher. The vehicle accelerates and the engagement of both pulleys allows for sensitive torque feedback (Aaen, 2007).

Shift out phase starts when the pulleys have been fully shifted to high range gear ratio and last from optimal engine speed to engine peak at which it cannot operate any faster (Aaen, 2007).

2.2. Tuning aim

The CVT under study is Gaged GX9 model (Figure 2) and is mounted on the Mini BAJA (BAJA SAE) vehicle (130 kg) designed for off-road racing. The CVT works with Briggs and Stratton engine Model 19. Figure 3 shows power torque curve of the engine measured using Eddy current dynamometer.
As the intended use is in off-road racing, the aim of CVT tuning is to get the higher initial torque in order to provide it a strong push during launch, slopes or muddy patches. So the engagement RPM is needed to be at peak of the engine torque curve. On another hand, shift start RPM is needed to be at the peak of the power curve of the engine so that the entire shift range can be placed in peak power of the engine (constant RPM) (Aaen, 2007). This is due to higher power requirement in CVT shifting phase to help vehicle in gaining top speed more quickly while coming out of a corner. Because shifting range is the most prevalent case in real driving conditions, if higher load condition (turns or obstacles) occur in this range and higher torque from the transmission is needed. The engine rpm falls during this (if CVT is not properly tuned) and it takes time for the engine to regain its RPM. Thus reducing vehicle acceleration. So, CVT needs to be tuned such that in shifting phase higher torque demand can be just met by downshifting CVT while keeping engine speed constant (Aaen, 2007). Keeping these points in consideration engagement and shifting rpm were decided as 2900 rpm and 3600 rpm respectively as they are the peak torque and power rpm of the engine (Figure 3).

2.3. CVT Geometry

The parameters of the CVT geometry are shown in Figure 4. The subscripts a and b refer to the primary and secondary pulleys respectively. The speed ratio \( T_v \) is given by \( R_b/R_a \) with the range of 3.9 to 0.9. For simulation, this range is divided into 31 speed ratios with the difference of 0.1 each and each speed ratio is solved individually in loops by GA for obtaining parameters that would produce desired output at that ratio. Similarly, the \( R_b \) is divided into 31 steps from maximum to minimum of 0.11–0.028 m and corresponding \( R_a \) is calculated using \( T_v = R_b/R_a \). The wrap angles are determined by relations:
\[ \theta_a = 180 + (2 \times \beta) \quad \& \quad \theta_b = 180 - (2 \times \beta) \]

where \( \beta = \tan^{-1}\left(\frac{R_a - R_b}{l_h}\right) \)

Centre distance of two pulleys \( (l_h) \) is 0.3 m. The axial displacements of the two movable sheaves are:

\[ x_a = 2 \times (R_a - R_{a_{\text{min}}}) \times \tan(\alpha) \]
\[ x_b = 2 \times (R_{b_{\text{max}}} - R_b) \times \tan(\alpha) \]
\[ \alpha = 20^\circ \] is the half wedge angle of the pulleys.

**2.4. Force balance equations**

To obtain the fitness function required for execution of genetic algorithm force balance equations are developed for CVT. Euler-Grashof theory of V belt is used to determine the amount of axial thrust required to operate V belt without any slip (Beccari & Cammalleri, 2001). The belt elastic penetration into the groove side compression of the pulley walls and the consequent radial components of the frictional forces play an important role in the V-belts mechanics (Gerbert, 1999; Sorge, 1996a). Considering this effect, functional dependence \( Z(\theta_h, T, V) \) is imposed along the arcs of contact on expression depicting of the axial thrust (Sorge, 1996a, 1996b; Sorge et al., 2001).

The accuracy of the equations has been tested in research by Beccari, A. and Cammalleri, M where they used these equations to model the behavior of variator by substituting the spring in place of the hydraulic actuator to actuate the CVT (Beccari & Cammalleri, 2001).

\[
F_{xa} = \frac{(T_1 - qV^2) \times Z_a(\theta_{a_h}, T, V)}{2 \tan(\alpha + \phi)} \tag{1}
\]
\[
F_{xb} = \frac{(T_2 - qV^2) \times Z_b(\theta_{b_h}, T, V)}{2 \tan(\alpha + \phi)} \tag{2}
\]

The rotation equilibrium results in Equations (3) and (4):

\[ Ca = (T_1 - T_2) \times R_a \tag{3} \]
\[Cb = (T_1 - T_2) \times R_b \tag{4} \]
Ca & Cb is torque produced on primary and secondary pulley respectively. T1 & T2 are the tensions on tight and slack sides of the V belt, $\phi = 26.5^\circ$ is the friction angle (Bushan, 2001), q is mass per unit length and V is the speed of the belt. The function dependence $Z(th, Tv)$ can be determined for both primary and secondary pulley ($Za$ and $Zb$). Sorge et al. simplified $Za$ and $Zb$ equations in their work to develop axial thrust model of the automotive CVT as following Equations (5–11) (Sorge, 1996a; Sorge et al., 2001).

\[ Za \approx th_a \]  
(5)

\[ Zb(th_b, Tv) = th_b + \frac{k1 - k2}{p - k2} \times \frac{1}{2 \left( \frac{\sinh(th_b)}{\Omega} - th_b \right)} \]  
(6)

\[ k_1 = k \times \frac{\tan(\alpha + \phi)}{\tan(\alpha)} \]  
(7)

\[ k_2 = k \times \frac{\tan(\alpha - \phi)}{\tan(\alpha)} \]  
(8)

\[ k = 0.8 \times \left( \frac{Rb}{Rb \ 1:1} \right)^2 \text{\textcolor{red}{Rb 1:1 is Rb at Tv = 1}} \]  
(9)

\[ p = \left( (1.5 - f \tan \alpha)^2 + 2k \left( 1 + \frac{f}{\tan \alpha} \right) - (1.5 - f \tan \alpha) \right) \]  
(10)

\[ \Omega = \sqrt{\frac{(k1 - p) \times (p - k2) \times \cos(\alpha + \phi) \times \cos(\alpha - \phi)}{p \times \cos \phi}} \]  
(11)

Figure 5(a) shows the free body diagram of the primary cam. The calculation for the net axial force exerted on V belt in primary sheave is produced by flyweights and primary spring as given in Equation (12). The minus sign signifies that flyweights have to work against primary spring to exert the axial force on movable primary sheave to start upshift.

Figure 5(b) shows the free body diagram of the torque feedback secondary cam. It exerts the axial force on the sheaves directly proportional to the twist (angle turned by the cam) in the secondary torsional spring and tends to oppose the upshift (Aaen, 2007). The axial force on the secondary sheaves exerted by the cam is given by Equation (13). Where $k_b$ is the torsional stiffness of the spring and $\delta b$ is the twist angle of the secondary spring.

\[ Fx = m \times (\omega^2) \times r \times \tan(\theta) - \int_0^{\frac{\pi}{2}} k_a(xa) \times dx \]  
(12)

\[ Fx' = |k_b \times \delta b| \times \tan(\delta \pm \phi) \]  
(13)

After rearranging Equations (1–13) and eliminating belt tensions the following relation (Equation 14) is obtained for primary pulley rotational speed. This equation serves the purpose of monitoring the Engine rpm in terms of the parameters of the primary and secondary cam, stiffness & mass and provides the basis for the further analysis using the genetic algorithm.

\[ \omega = \sqrt{\frac{\left\{ \frac{k_b \times \delta b \times \tan(\delta) \times 2 \times \tan \alpha \times \frac{\gamma}{2} + Ca \times \frac{\beta a}{2}}{2 \times \tan \alpha} \right\}}{m \times r \times \tan(\theta)}} \]  
(14)
3. Methodology

The resulting equation (Equation 14) from the force balance is simulated on MATLAB using the dual level genetic algorithm for optimizing the flyweight mass \( m \), primary and secondary spring rate \( k_a \) & \( k_b \), primary and secondary cam profile. The simulation process flow is shown in Figure 6. At first level, GA parameters required for shift starting are calculated. After that, calculations are done for constant RPM shift for entire shifting range. At last values for desired engaging RPM engaging CVT at required RPM are calculated.

The entire simulation is based on the principle that the Equation (14) is derived from the axial force balance and the variable values that would satisfy Equation (14) will also satisfy the condition of axial force balance. As in Olav Aean (2007) force balance is also the condition for the primary and secondary pulley to displace and start the shift along with shifting the pulley through entire shifting range. However, this does not apply for the engagement of the primary pulley which only engages when the flyweight force overcomes the force of the primary spring and produces side force required to clamp the belt as in Section 3.3. Although engagement is the first step in during practical operation of CVT, the calculation for this is done in last so that adequate amount of flexibility is available for more prevalent operating conditions i.e. Shift start and constant RPM shifting (Aaen, 2007).
The primary spring is of variable stiffness while the secondary spring is torsional type constant stiffness spring. For simplification initial compression of the primary spring is kept as 1.5 mm. For determining the profile of the primary cam the center distance ($r$ as in Figure 5(a)) between flyweight and primary pulley axis is calculated with respect to the displacement of the primary sheave ($x_a$). While for the secondary cam the profile is measured by plotting the movable sheave rotation ($delb$) w.r.t to the displacement of the secondary sheave ($x_b$).

The GA used in this study is single objective GA as the only objective is to optimize the RPM of the engine. For the convenient implementation of the basic genetic operators, the parameters are encoded into a binary string (Goldberg, 1989). The binary string is commonly referred to as a chromosome, while its features are known as genes. The precision of each of the design parameters included in a chromosome is based on the desired range of the parameter (i.e. its minimum and maximum values) and how many bits are used in its representation. Also to achieve faster convergence elitism is used as a special selection mechanism.

Figure 6. Process flow diagram.
3.1. Shift start (Genetic algorithm level 1)

The first level GA is used to calculate condition required to achieve shift start at 3,600 RPM. Equation (14) is used as fitness function and the fitness limit is set to the 3,600 thus GA tends to optimize parameters so that fitness function can be minimized at 3,600 RPM. Thus aligning the purpose of tuning for shift start at 3600 RPM.

For realistic results initial population range supplied for these parameters is kept between the maximum and minimum of parameter values of spare parts supplied by the OEM (Table 1). The number of generations is set to 200 for accuracy and minimum processing time. As after 200 generations, no significant change in solution was observed. By similar logic, the population size was kept as 50. Uniform crossover is used and crossover parameter is kept as 0.6 (Serafino, 2007).

The mass and secondary spring stiffness calculated in first level GA are kept constant for subsequent steps of analysis; because secondary spring is of constant stiffness and it is impractical to change the value of mass for each speed ratio.

3.2. Constant RPM shifting (Genetic algorithm level 2)

The second level GA is used to tune CVT for the constant RPM shifting from low to the high ratio. The variables in this step are primary spring stiffness($ka$), primary cam profile(r) and secondary cam profile ($\Theta$). To obtain the fitness function for second level GA Equation (14) is differentiated w.r.t speed ratio ($Tv$) resulting in the equation for $d\omega /dTv$. The fitness limit of this GA is set to 0 so that it optimizes the solution to achieve $d\omega /dTv$ as 0 thus aligning with the purpose of constant shift with respect to speed ratio.

The differentiation is done to achieve the simplicity and accuracy of executing the GA as in non-differential form to define different population range at each speed ratio analyst has to rely on assumptions and solve a different GA for each of the speed ratio manually rendering erroneous results and cumbersome processing (Mitchell, 2007). By differentiation for all speed ratio only one population range of rate of change of these parameters w.r.t $Tv$ needs to be provided. Thus, for each speed ratio, a GA run through the loop and optimum rates of changes are determined to result in an optimum value of parameters for each speed ratio along with providing the basis for next speed ratio. The population range is given in Table 2. The number of optimum generations is set as 150 while the population is 70 along with crossover probability of 0.6 with uniform crossover (Serafino, 2007).

3.3. Engagement RPM

After all the values are determined for each shift starting and shifting range. The primary cam profile for engaging the CVT at 2,900 RPM is calculated by Equation (15).

$$th_{eng} = \tan^{-1}\left(\frac{ka \times x_{a_initial}}{r \times m \times \left(\omega^2\right)}\right)$$  (15)
4. Results

4.1. Profile of primary cam
The function of the primary cam is to transfer force created by rotation of the flyweights into movable sheave. This force has to first counter the pretention of the primary spring and as engagement start, it has to counter loads of primary spring and the axial thrust from driven sheaves (Aaen, 2007). The profile of primary cam obtained from simulation is in form of the axial displacement of primary sheave vs. radial position of flyweight(r) w.r.t axis of the primary clutch is shown in Figure 7.

From the profile obtained it can be inferred that as the primary sheave shifts, the angle of force transfer (θ) decreases. Thus decreasing the amount of force transferred as pulley shifts towards higher speed ratio. This aligns with the lesser requirement of the axial force in higher speed ratios because same amount of torque (because the engine is running at constant RPM) is transferred at the higher radius of the primary pulley thus decreasing the tensions in the belt resulting in lesser demand for clamping force (axial thrust). Also in the secondary pulley, the amount of axial thrust produced by the secondary cam decreases (as in next section) thus further lowering the demand of force from the flyweight for shifting.

The profile for engagement is obtained is of constant angle because only purpose this field is to provide enough clamping force for engagement thus a constant angle is enough to meet the increasing need of clamping force (axial) by the increase of Engine RPM.

4.2. Secondary cam
The torque feedback cam provides the axial force according to Equation (6) w.r.t the torque produced at the secondary pulley. More the torque more is the side force provided. As explained in Section 2.4, the profile of the torque feedback cam is given in Figure 8 in terms of angle rotated by the movable secondary sheave vs. the axial displacement of the Secondary movable sheave (Aaen, 2007). From the profile, it can be inferred that angle of the side force transfer(δ) decreases as the cam rotates thus supplying lesser side force in greater twists. This justifies the lesser side force requirement by the V belt in secondary sheave at higher speed ratios because of lesser torque produced at the secondary sheave due to decreasing radius of belt rotation. On the other hand, the gradually decreasing angle of force transfer improve the back shift characteristics of the CVT because in case the vehicle demands more torque lesser angle of force transfer renders quick back shifting characteristics. Thus quickly changing the speed ratio instead of loading engine and lowering its speed.

4.3. Primary spring
Primary spring stiffness is obtained w.r.t sheave displacement as shown in Figure 9. The spring constant gradually increases thus increasingly countering the force produced by the flyweights. Thus assisting primary cam to supply the adequate amount of axial force. The increasing spring constant also helps in good back shifting characteristics. Because it will provide increased back force to the sheaves thus quickly back shifting the CVT (Aaen, 2007).

| Parameter change per \((0.1 \times T_v)^{\dagger}\) | Lower value | Upper value | Units |
|-----------------------------------------------|--------------|--------------|-------|
| Primary spring stiffness                      | 70           | 150          | N/m \((0.1 \times T_v)\) |
| Axial distance                                | 0.001        | 0.0015       | m \((0.1 \times T_v)\) |
| Secondary cam rotation                        | 1            | 2            | ° \((0.1 \times T_v)\) |

\(\dagger\)For simplification the rate of change is set as \(0.1 \times T_v\) instead of \(1 \times T_v\) because the speed ratio is divided in to steps of 0.1.
4.4. Secondary spring
The torsion type spring constant stiffness spring is used in the secondary clutch and the spring con-
stant obtained from the first level GA is 24,300 N-mm/°. The spring rate is kept constant because of
the manufacturing limitations.

Mass of the flyweight is obtained as 195.9 gm.

5. Validation

5.1. Comparison to expected results
To validate the results obtained from the simulation; experimental testing was done for resulting
configuration of CVT by mounting it on BAJA SAE vehicle (138 kg). The testing setup is shown in
Figure 10. The goal of the testing is to measure the upshift and downshift curve of the CVT. For this
purpose, a digital tachometer is mounted on the vehicle to measure the RPM of the primary clutch
and RPM of the secondary clutch are measured from the speed of the vehicle. After 10 readings and taking the mean of the rpm at corresponding speed ratios. The results obtained are shown in Figure 12 for both upshift and downshift. Both the engaging rpm and shifting rpm obtained in the experiment are comparable to the simulation with 2830 rpm compared to 2900 in case of engaging and 3660 rpm compared to 3600 in case of shift starting.

The shift graph also shows coherency with expected behavior for most of the shifting phase. But during upshift, the rpm of primary pulley gradually increases as we approach the high gear ratio. On the other hand during the downshift, shifting starts at comparatively lower RPM and the RPM approaches the 3600 RPM as the CVT shifts further downwards. This can be attributed to the fact that the current simulation model is static and does not take in to account the transient effects such as radial speed of the belt (i.e. rate of change of radius). Thus using only linear relation for determining the friction coefficient according to coulombs model of friction and not taking in to account the variable friction which tends to increase with respect to relative sliding speed between the belt and sheave (radial speed of belt) (Kong & Parker, 2006). Figure 11 shows total sliding speed per speed ratio of the belt in both secondary and primary pulley w.r.t to the speed ratio. The increased belt radial speed in higher gear ratios increases the friction force that acts on the belt. Thus in upshift CVT
needed more force from the flyweights to push the belt radially, in turn increasing the RPM of the engine. Also in downshifting this effect comes in to play but in opposite direction, because now friction is opposing the decrease in radius at primary pulley and increase in radius at secondary pulley thus working against the springs resulting in demand of lesser force from the flyweights so engine rpm lowers resulting in downshifting to start at lower RPM.

The reason of coherency between experiment and simulation in lower gear ratios can be attributed to the lesser radial speed of belt in that shift range thus having the lesser effect on the friction coefficient resulting in more accurate results.

5.2 Comparison to results of experimental tuning
To check the performance of the simulation compared to other tuning techniques. The behavior of CVT tuned by simulation tuning was compared to the results of the experimental technique of tuning. This technique involved using heuristic approach for manually adjusting the parameters of the CVT and then documenting the behaviors of CVT and repeating this process until the result obtained is satisfactory. The values of the parameters that were obtained using this approach are shown in
Table 3. The behavior of the CVT is shown in Figure 13. These results are very much comparable to the results obtained after simulation (Figure 12).

The engagement RPM is same for both tuning methods as 2830 for simulation and 2850 for experimental tuning for both upshift and downshift. The shift start RPM is 3660 for simulation and 3620 for experimental. For the rest of the shifting range, the RPM almost remain constant in experimental tuning but the simulative approach has shown a difference from the expected constant RPM approach. Similarly, for the downshift the RPM shows a decrease in the shift start rpm (3480 RPM) while for simulation approach this is 3650 RPM. The percentage errors for both approaches are shown in Table 4.

6. Conclusion

The new approach of simulation tuning using GA is studied in this paper. The study is validated through two tests. First one is by testing the simulation results against expected results. This test showed good coherency between actual behavior and expected results. Deviation from expected values that resulted in higher gear ratio has further opened the perspective for future studies that can account for the transient conditions that occur in practical applications.

| Table 3. Parameters obtained through experimental approach |
|----------------------------------------------------------|
| Parameter                  | Value          |
|----------------------------|----------------|
| Mass                       | 150 gm         |
| Primary spring stiffness   | 30,000 N/m     |
| Secondary spring stiffness | 25,000 N-mm°   |
| Primary cam                | 20–60–30°      |
| Secondary cam              | 45°            |

| Table 4. Percentage error in the RPMs of both approaches |
|---------------------------------------------------------|
| Ideal | Upshift RPM Experimental | % error | Downshift RPM Experimental | % error | Upshift simulation | % error | Downshift Simulation | % error |
|-------|--------------------------|----------|----------------------------|----------|-------------------|----------|----------------------|----------|
| Engagement | 2,900 | 2,850 | 1.7 | 2,850 | 1.7 | 2,830 | 2.4 | 2,830 | 2.4 |
| Shift start | 3,600 | 3,620 | 0.5 | 3,480 | 3.3 | 3,660 | 1.6 | 3,650 | 1.3 |
| Shift end | 3,600 | 3,580 | 0.5 | 3,580 | 0.5 | 3,770 | 4.7 | 3,570 | 0.8 |

Table 3. The behavior of the CVT is shown in Figure 13. These results are very much comparable to the results obtained after simulation (Figure 12).
For the second test, the results of simulation tuning are compared to experimental tuning. Though results of both tuning approaches are comparable in many aspects but still simulation approach is not as accurate as the experimental one. The transient model and more involved friction model may be able to match the accuracy of experimental results.

Also, this study only considered slip characters of the belt, not properties such as flexural, longitudinal or axial stiffness that can significantly affect the performance the CVT. Future study can be conducted in this field to also take account of these parameters.

Also, the future work will include integration of finite element model of CVT with GA so that deformation of pulleys and belt can be taken in to account for more accurate results.

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