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PERFORMANCE STUDIES OF CUSTOM CONTINUOUSLY VARIABLE TRANSMISSION FOR ALL-TERRAIN VEHICLE APPLICATIONS

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Abstract

The off-road vehicles are a necessary in today's world for rescuing operations, military, racing and farming applications creating a huge demand for the All-Terrain Vehicle in the automobile market. The market size is estimated to be worth about $ 9.2bn (₹594Cr) by 2020. The drive-train of an All-Terrain Vehicle(ATV) is one of the major component in propelling the vehicle. Continuously Variable Transmission(CVT) coupled to a constant reduction gearbox, provides ease of driveability and the required high torque and top speed. In case of an ATV, high torque and high-top speed is the requirement, since it should be able to negotiate various terrains such as bumps, hill-climb, etc. Several authors have discussed, methods in designing CVT components of commercial vehicles which run on tarmac, and only few researchers have discussed methods to develop a CVT for an ATV requiring high velocity and high torque. The aim is to develop a lightweight, compact CVT system with high torque and top speed without any compromise on reliability. The force balance method is used to develop the analytical model. This model is used to design the various components of the CVT operating with gear ratios between 4:1 to 0.7:1. The structural analysis of all the CVT components are carried out. The designed and fabricated CVT components were then assembled on the ATV and performance studies were done by testing in various terrain conditions such as bumps, drops, slush and pulling weight from zero speed. All the components performed well, with no failures in all these conditions.

Keywords: All-Terrain Vehicle, Centrifugal Force, Continuously Variable Transmission, Data Acquisition System, Drive-Train, Light Drivetrain

1. Introduction

A Continuously Variable Transmission is a type of an automatic transmission

Nomenclatures

A	Ramp angle, deg
F_1	Force on arm, N
F_{2H}	Horizontal force, N
F_C	Centrifugal force, N
G	Shift ratio
M	Mass of weights in CVT, kg
R	Radius from centre of the pulley, m
R	Radius of ramp, m
S	Side force (Fig. 4), N.
T	Torque, Nm.
T_1, T_2	Tension in belt, N.

Greek Symbols

θ_1, θ_2	Angles made by the centrifugal force and horizontal force (Fig. 3), deg.
θ	Belt wrap angle, deg.
μ	Coefficient of friction between belt and CVT sheaves.
$\boldsymbol{\omega}$	Angular velocity, rad-deg.

Abbreviations

ATV	All-Terrain Vehicle.
CVT	Continuously Variable Transmission.
RPM	Revolutions per minute.

system, that changes gear ratio with a change in the input RPM and the torque feedback at the wheels. It involves a drive pulley attached to the output shaft of the engine, and a driven pulley connected to the input of a fixed reduction gear box. When compared with a manual transmission, there is no power loss in the system due to the dip in power when the clutch is engaged during changing of gears. The main design principle for the functioning CVT is the centrifugal force generated. In the drive pulley, as the engine RPM increases, the centrifugal force goes on increasing and thus applies more force on the sheaves and shifts the gear ratios.

The centrifugal force [1] is given by Eq. (1)

$$F_c = m\omega^2 r \tag{1}$$

The driven pulley consists of a torsional spring that is controlled by a cam mechanism. It also contains a torque ramp, which helps the back shifting of the CVT because of the resistance to torque at the wheels translated to torsional moment at this ramp. The sheave angles, sheave diameters, and the ramp angles are the parameters, which determine the initial and final ratios of the CVT.

In an all-terrain conditions, it is extremely unpredictable as to what type of terrain the vehicle might encounter. In a manual transmission, the driver is put under more pressure to select the right gear to get across the various terrains. Hence, by using a CVT, this issue is taken care of as the CVT shifts, depending only on the throttle response from the driver and the torque feedback from the terrain. However, the novelty of this design is that, a gear ratio of 4:1 to 0.7:1 is obtained without the help of a centrifugal clutch, which is more suitable to gain advantage over the various terrains the ATV is subjected to. Another novelty present is the actuation mechanism of the driven pulley, as compared to OEM CVT.

In the past, many authors have discussed about mainly 2 different methods to analyses the CVT system. They are the energy balance method and the force balance method. Cammalleri [1] talks about deriving the various forces involved in the drive and driven pulley of the CVT and hence, discusses the force balance method in detail. Bertini et al. [2] talks about the various losses that are to be considered in the analytical model.

Yusuke [3] discusses the performance parameters, of a CVT, as a function of belt tension. Christopher [4] talks about the various energies, like potential energy of the spring, kinetic energy of the sheaves, kinetic energy of the rollers etc. These energies are then balanced to calculate the parameters of the CVT, thus showcasing energy balance method. Sorge [5] has proposed a closed form solution, for the belts that are stiffer in the longitudinal direction rather than the transverse direction. Oliver et.al [6] in brief discusses the design equations for a CVT controlled based on speed and torque. Deep [7] discusses the various features and specifications of a general All-Terrain Vehicle. Khan et. al. [8], focuses on the various types of design required for a four-stroke engine model, helps to formulate a model that can predict the transmission ratio time response to changing conditions of the drive pulley axial force, load torque and engine RPM. Forces on the system are determined, and Newton's law of motion are applied to a distinct drive belt, piloting to a system of ordinary differential equations, which is solved numerically. Model outputs are contrasted with investigation results, and acceptable agreement is witnessed mutually for the steady-state and transient cases under high loads of up to 150 Nm.

Olav [9], in his book on clutch tuning handbook, talks about the various types of CVT s available in the market and the types of cams used in them to change the gear ratio. The author also mentions about the various available tuning methods and emphasizes that the trial and error method to tune a CVT to the desired engine be the most suitable and the most accurate method. However, one must be careful while doing and mentions about designing a CVT that is tune-able is required. David et.al. [10] discusses the wear characteristics of coated and uncoated pulley sheaves in a CVT. Ichiro et. al, [11] discuss the slip velocity analysis based on the evaluation of the pulley displacement by FEA. Jian et. al [12] provide an insight into the methods used to analyse the CVT components, for both static structural as well as fatigue strength.

The available CVT deal with changing weights and springs for better tunability, but in this paper, aims at developing an extremely easy to tune system. Another novelty in this design is the actuation of the driven pulley and the compared to the existing OEM CVT. This CVT design ensures improvement with respect to the tune-ability and consequently decrease the overall rotational inertia.

The objectives of this paper are to develop a mathematical model of a CVT using force balance method. The outcomes of this model is studied and then implemented into the designing of the various CVT components. Static structural analysis of the components is carried out and then, fabricated. The CVT is then assembled on the ATV, and the testing is carried out in all-terrain conditions to determine the initial and the final gear ratios of the CVT.

2. Data Acquisition of Existing System

The available CVT, was tested to acquire its shifting characteristics and its maximum shifting ratio. A Data acquisition system consisting of proximity sensors and an Arduino circuit board was developed and fabricated in-house. It consists of a proximity sensor placed near the hub of the wheels and another one near the flywheel of the engine. This was used to measure the RPM of the wheel and the Engine RPM. From the data obtained, a plot of the wheel RPM, engine RPM is plotted and from these two values, the gear ratio of the CVT is calculated. This test was carried out, by riding the ATV on rugged terrains, which consisted of bumps, pot holes, tight corners and flat out stretches, for around 150 kilometres, in 3.5 hours' time. From the data obtained, we could infer that, for the given arrangement in that CVT, it is designed for low-end torque conditions, and thus would not allow the buggy to reach a higher top speed. The graph plotted from the data collected is shown in Fig. 1. Figure 2 shows the change in gear ratio of the CVT with time, while running the ATV on rugged conditions.

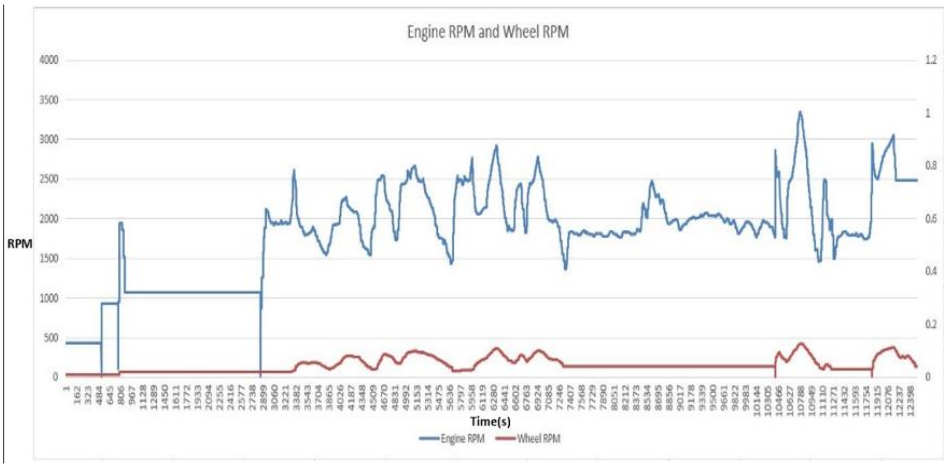


Fig. 1. The plot of wheel RPM and engine RPM vs. time in seconds.

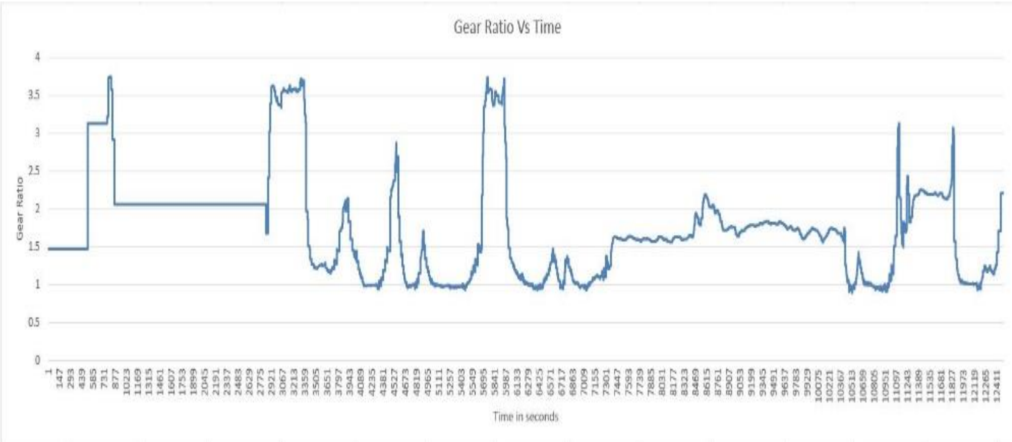


Fig. 2. The plot of gear ratio vs. time in seconds.

3. Analytical model: Force balance method

Before designing the CVT, an analytical model is developed to analyse the various forces acting in the system. This is performed using developing equations using simple free body diagrams on the various parts of the CVT.

3.1. Input parameters

For the analytical model to provide a solution, certain parameters must be input. These parameters involve the engine characteristics and the drive and the driven sheave diameters. The engine characteristics are important, since the transmission must match with the engine seamlessly. The sheave diameters play a major role in fixing the final and initial drive ratio of the CVT. The sheave diameter selected, ensures that either the belt does not slip out of the CVT system (because of smaller diameters of the sheaves), or the belt does not move quickly to shift out or shift down (because of larger diameters of the sheaves).

The first ratio is determined based on the maximum torque condition required to move the ATV on a hill climb with a gradient of up to 40 degrees, whereas the final drive ratio is determined from the back calculations with a top target speed of 60Kmph.This speed was selected as the required top speed, according to the driver's requirement and also a unanimous survey conducted amongst a large group of people who like to commute in an ATV in conditions not suitable for normal car to drive in. The stresses acting on the components due to the various fabrication process, that give rise to both plastic and elastic stresses, have been considered during the design calculations.

3.2. Drive pulley equilibrium condition

To convert the radial force (Centrifugal force) acting on the flyweights in the CVT, into linear forces, we use a set of ramps which is as shown below. The distance of the ramps from the axis of rotation is determined based on the centrifugal force required to clamp the belt in the low ratio. At this condition, the force balance is performed and the equation for the horizontal clamping force at the driver pulley is arrived at in Eq. 2. The free body diagram is represented in Fig. 3.

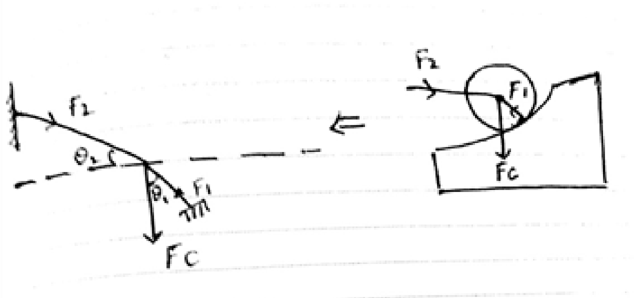


Fig. 3. Freebody diagram of the pulley.

The horizontal force is given by,

$$F_{2H} = \frac{F_C \, Cos\,\theta_2}{cos\,\theta_2\;cot\,\theta_1 - sin\,\theta_2} \tag{2}$$

3.3. Driven pulley equilibrium condition

The driven pulley consists of a torsional spring to hold the belt in position and to respond to the torque feedback. Figure 4 shows the free body diagram of the driven pulley assembly.

$$S = \frac{T\,.\,R\,.\,12}{2\,.\,R\,.\,tan\,\alpha} \quad (3)$$

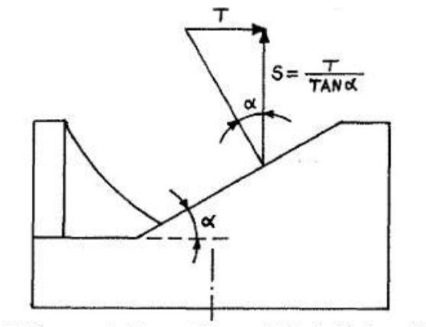


Fig. 4. Free body diagram in driven pulley.

4. Design

The analytical model is used to determine the various forces acting on the CVT. These forces are taken as an input for the design of the various components. These forces are then considered to design the various components of the CVT with the help of CATIA V5. The exploded view of the CVT system designed is shown in Figs. 5.

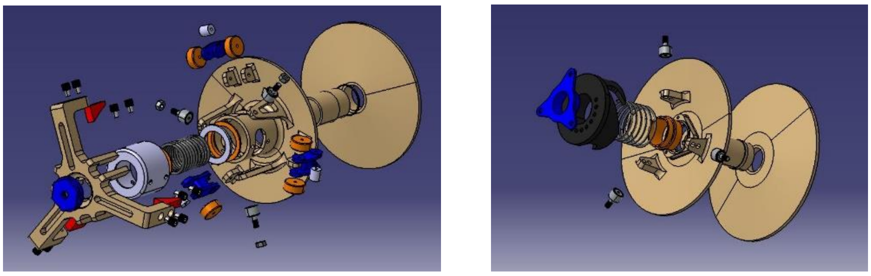


Fig. 5. (a) and (b) depicts the exploded view of the drive and driven pulley of the designed continuously variable transmission respectively.

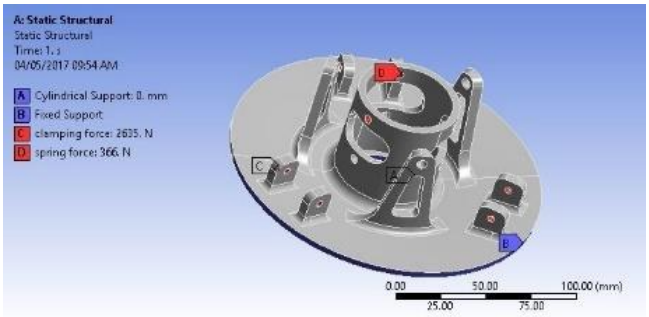
5. Finite Element Analysis of the Components

Once the design was completed, the structural analysis of the various CVT components is carried out. The stresses acting on the components due to the various fabrication process, that give rise to both plastic and elastic stresses, have been considered as well. The mesh type was selected as tetrahedral elements, and the size was selected as 1.5 mm. This was selected after conducting a through convergence test of different mesh elements. The boundary conditions and the forces acting on the CVT parts have been determined from the whole assembly itself. Hence the static analysis consists of the parts under the assembly as such. Figures 6 to 10 show the stress concentration and deformation of the components.

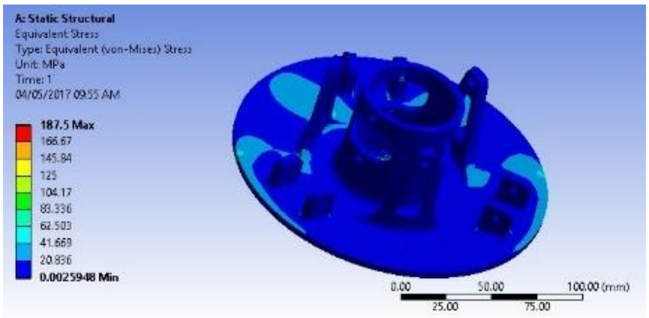
5.1. Drive pulley FEA analysis.

The drive pulley is a part of the CVT that is attached to the output shaft on the engine side and to the ramps, ramp holder and many other components. Hence, it needs to be analysed for all these loads.

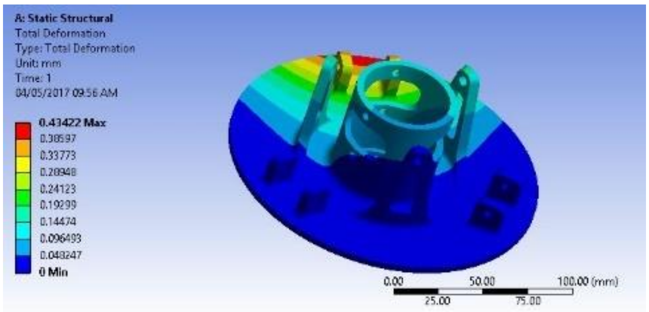
As shown in Fig. 6(a), a cylindrical support provided in the place where the sheave slides over the post the pulley is fixed at the point where the belt sits on the pulley, and a clamping force equal to 2600 N along with the spring force of 366 N was applied to the projections to which the spider arms are bolted. This was then analysed. As shown in Figs. 6(b) and 6(c), a maximum stress of 187 MPa was observed near the bolt holes. The maximum deformation was found to be 0.43 mm.



(a)



(b)

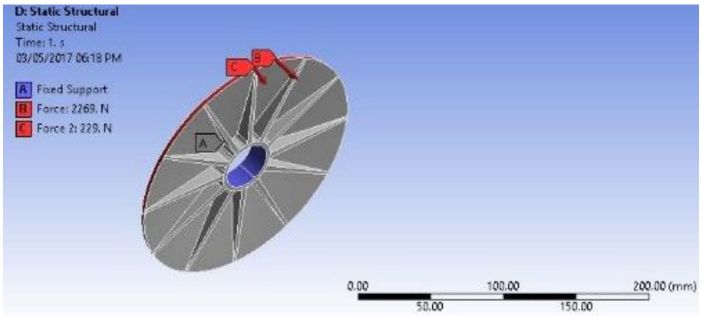


(c)

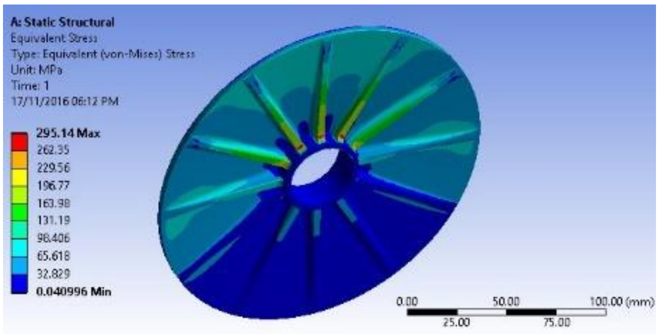
Fig. 6. The loading condition (a), stress concentration (b) and deformation (c) of the drive pulley moveable sheave.

As shown in Fig. 7 (a), the fixed support is provided in the threaded region of the pulley present at the centre and two forces act on the pulley. A clamping force of is applied on the sheave cone angle, and the belt tension is applied in a direction into the sheave. The clamping force is calculated to be 229 N, and the belt tension was found to be 2269 N. This was applied on the pulley.

As shown in Figs. 7(b) and (c), a maximum stress of 295 MPa was found to be acting on the sheave. A maximum deformation of 0.84 mm was observed. In this iteration, we find that the maximum stress concentration is found at the points where the ribs meet the shaft hole. Considering the maximum deformation case, it is observed at the tip of the sheave because the sheave, acts as a cantilever beam, with the thickness being the least at the edges. Next, the Ramp holder was analysed, and the results obtained were a maximum stress of 152 MPa, with a deformation of 0.96 mm.



(a)



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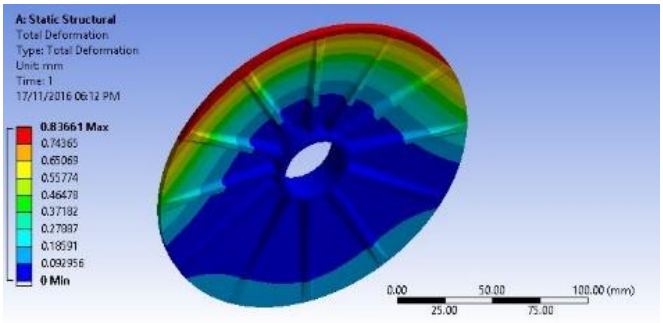
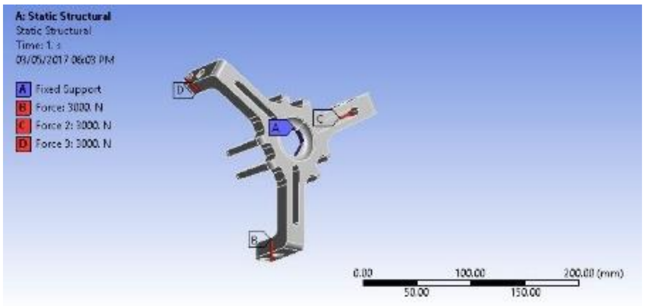


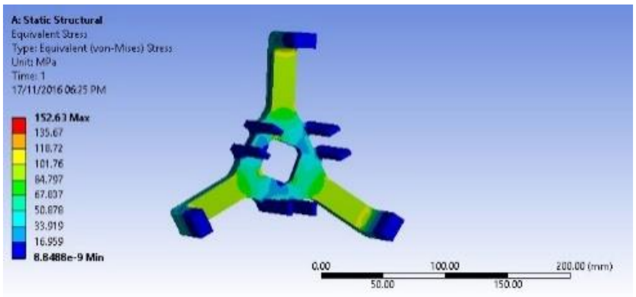
(c)

Fig. 7. The loading condition (a), stress concentration (b) and deformation(c) of the drive pulley fixed sheave.

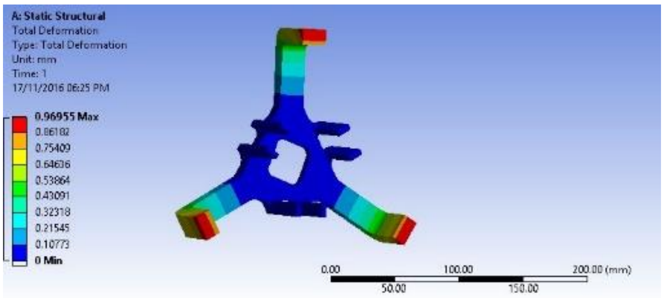
Figure 8(a) shows the ramp holder is subjected to a centrifugal force of 3000 N on each of the three ramp holders. This was applied radially outward, and the analysis was carried out. Figures 8(b) and 8(c) shows a maximum stress of 152 MPa, was acting on the projections that house the ramps and a maximum deformation of 0.969 mm was observed.



(a)



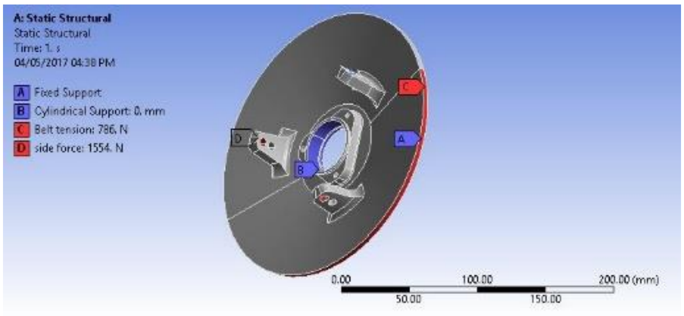
(b)



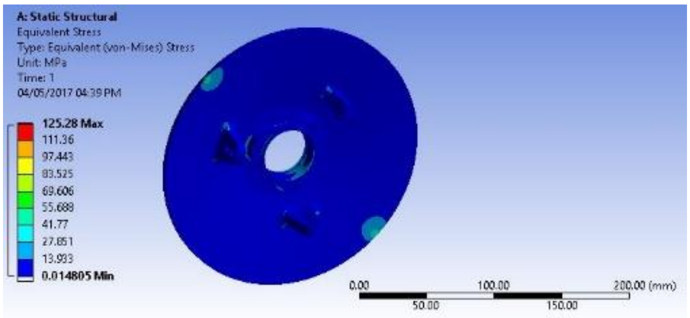
(c)

Fig. 8. The loading condition (a), stress concentration (b) and deformation (c) of the ramp holder.

As shown in Fig. 9(a) the moveable sheave has two forces acting on it. The first being the belt tension and then the side force. The belt tension,786 N, acts on the sheave along the wrap angle, and the side force acts on the bolt holes provided for fixing the spring cover, with a magnitude of 1554N.As shown Fig. 9(b) and (c) the maximum stress developed in the sheave was found to be 126 MPa, which was observed near the bolt holes. This gave rise to a maximum deflection of around 0.079 mm at the edge of the sheave from the periphery to the shaft.



(a)



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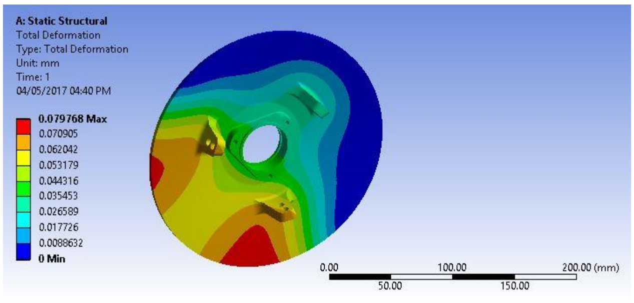


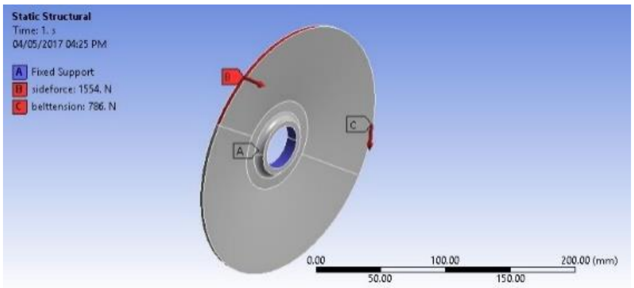
(c)

Fig. 9. The loading condition (a), stress concentration (b) and deformation(c) of the driven pulley moveable sheave.

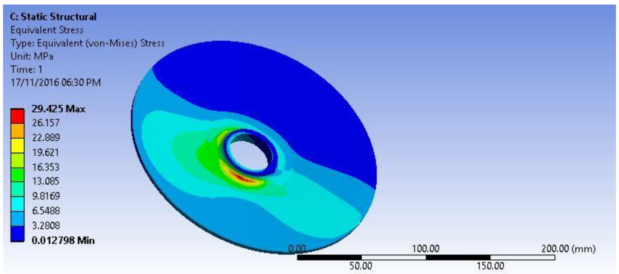
5.2. Driven pulley FEA analysis

The driven pulley is a part of the CVT that is attached to the output shaft on the wheel side and contains the torque groove for easy back shifting. Hence, it needs to be analysed for all these loads.

Figure 10(a) shows the sheave undergoes two types of loading conditions, a side force from the belt on the sheave acting along the wrap perimeter of the belt on the sheave. Belt tension is another force that acts on the perimeter of the sheave. As per calculations, a side force of 1554N is obtained and a belt tension of 786N is obtained. This is applied on the sheave, and then the analysis is carried out. A maximum stress of 29.425 MPa (shown in Figs. 10(a) and 10(b)), was found to act on the back side of the sheave near the fillet of along the hole diameter. This analysis also resulted in a deformation of 0.19 mm, at the periphery of the sheave



(a)



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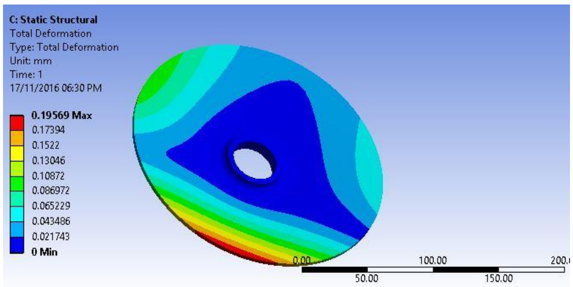
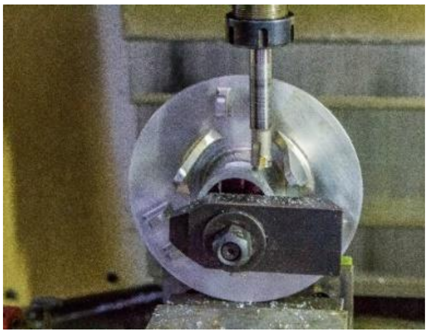


(c)

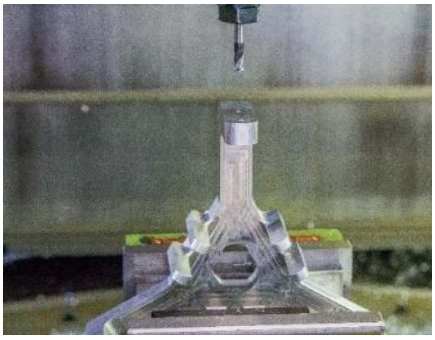
Fig. 10. The loading condition (a), stress concentration (b) and deformation (c) of the driven pulley fixed sheave.

6. Fabrication

The CVT was fabricated using Aluminium 7075-T6, due to its good thermal conductivity of 130W/m-K and good strength to weight ratio of 179Nm/kg. The DFM principles used were mainly to reduce the set-up time in the CNC machine, such that many operations could be completed in a single setup. The tolerance of the sheaves angle was mentioned to be very critical as these would decide the force acting on the belt. The CVT has many complex profiles, and hence there was a need for a three axis CNC Vertical Milling Machin (VMC). This was carried out to meet the accurate tolerances required for the CVT to function without any difficulties. The end mill tool of 10 mm diameter was used for the roughing cycle and was later changed to 5 mm end mill tool for the finishing cycle. The dial gauge with a least count of 1µm was used to check for circularity and flatness of the component as per the drawing. At last, the various holes required were drilled with the respective drill bit sizes and then a tapping tool was used to tap holes where ever necessary. Figures 11(a), (b) and (c), shows the various processes carried out to fabricate the CVT.



(a)



(b)

(c)

Fig. 11. The various machining processes involved in fabrication of the CVT. (a) shows roughing cycle, (b) shows the drilling cycle and (c) shows the verification of tolerances using a dial gauge.

7. Testing of the Fabricated CVT

A small off-road testing track was constructed at the back of the college to test our various components. The track involved a lot of off-roading trails like bumps, slush, weight pull, drops and many more. The car was driven on the trails that consisted of bumps, slush, weight pull, drops and many more obstacles to record the shifting of CVT dynamically with the help of a Data Acquisition system (DAQ).

The CVT's gear ratios were dynamically tested with the help of electronics. A system consisting of a pair of proximity sensors to measure the RPM of the engine and the wheel. This was then connected to a Data Acquisition system (DAQ), as shown in Fig. 12, which recorded the various values when the vehicle was tested dynamically. The following graph shows the same. According to the data obtained, the initial gear ratio was found to be 4.1:1 and the overdrive ratio was found to be 0.7:1 as shown in Fig. 13, which is the same derived from analytical model.

Fig. 12. The DAQ system installed on the ATV.

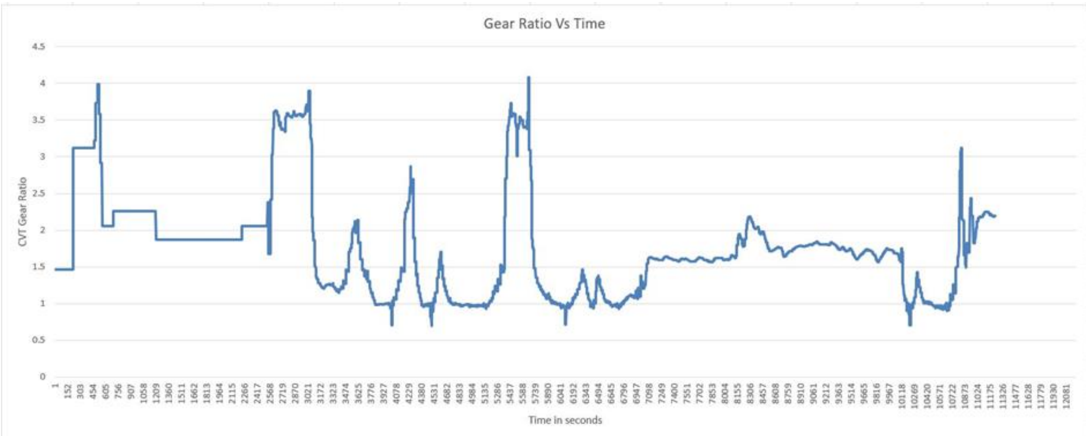


Fig. 13. The DAQ system installed on the ATV.

8. Conclusions

A CVT suitable to the required needs of an ATV is designed and fabricated, with the help of an analytical model. The following conclusions are drawn upon through testing:

- The analytical model was developed for the CVT using force balance method to determine the various forces acting on the CVT.
- This analytical model was the basis on which the design of the components of the CVT was carried out.4
- The CVT was designed to obtain an initial gear ratio of 4:1 and an overdrive gear ratio of 0.7:1.
- All the components are analysed in static structural analysis and the maximum deflections was less than 1 mm.
- The components are assembled on an ATV and was tested on various terrain conditions.

- On testing, the design and fabricated CVT conveniently shifted between a high gear ratio (4.1:1) and a low gear ratio (0.7:1).
- There was no deviation observed in the overall shifting ratio for the CVT as compared to the analytical model.

9. Future Work

In future, with the help of better electronic sensors and DAQ, the CVT components can be studied further for the actual stress and deformation values, dynamically. Thus, this can help with better optimization of results, as we can compare the experimental data to the analytical data.

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