Design and Optimization of Continuous Variable Transmission System for an ATV

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Abstract – *The CVT has components whose motion results in the change of the radius of the sheaves and thus changing the gear ratio. It is necessary to acquire the data of the changing gear ratio with respect to changing RPM of the engine. CVT, use infinitely adjustable drive ratio instead of discrete gears to attain optimal engine performance.*

The project aims at designing and manufacturing of the CVT. The CVT is designed considering the requirement of the SAE INDIA BAJA event and engine used in the event i.e. Briggs and Stratton 10HP engine. This CVT provides better acceleration and ease in handling as compared to the manual transmission and cost effective as compared to the other transmission systems available in the market.

***Key Words*: CVT, SAE BAJA,**

1. INTRODUCTION

A Continuously Variable Transmission (CVT) is an unconventional type of transmission system which can change stepless through continuous and infinite number of effective gear ratio between maximum and minimum values. This contrasts with other mechanical transmission system that only allow a few different distinct gear ratios.

The flexibility of the CVT permits the driving shaft to maintain a particular angular velocity over a specified range of output velocity. This can provide a better fuel economy than other transmission system by enabling the engine to run at its most efficient RPM for a range of speeds.

A CVT has three major parts which are drive pulley, drive belt and the driven pulley. Front movable drive and is mounted on the drive shaft. Centrifugal clutch with frictional material adhered to its periphery and clutch drum which is connected to driven shaft forms a driven pulley. When engine is running at idle condition, clutch will rotate but drum will be stationary. Hence no jerks are observed during the working as there are no steps in transition during the gear ratios. This also results in better fuel economy of the vehicle.

2. PROBLEM DEFINITION

To design, optimize and manufacture a CVT used for a student made single seater All-Terrain Vehicle in SAE BAJA racing series. As the restriction of common engine is imposed by SAE BAJA, not allowing any modifications made in the engine. Also the component life is an important factor which needs to be considered. Thus it is necessary to emphasize on the power delivered to the wheels and frequent servicing of the CVT. So the following parameters are considered while designing:

1. **Improved gear ratio range:**

   Gear ratio range is what decides the top speed and acceleration of the vehicle. Hence an optimum gear ratio range is calculated.

2. **Design for Manufacturing and Assembly:**

   Design for manufacture is the ease in the manufacturing of the parts i.e. the parts should be easily manufactured. Similarly, ease in assembly of the parts is Design for Assembly.

3. OPERATION OF CVT

The CVT operates in following four phases:

1. Idle

In Idle phase, engine speed is below CVT engagement speed because flyweight does not create enough centrifugal force for the primary sheave to shift and engage with the belt. The belt seats in a low ratio (low gear) setting.

2. Engagement

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 the flyweight increases and causes enough clamping force so that the slipping is eliminated and engine speed increases and vehicle accelerates along the low ratio.

3. Straight Shift
Straight shift starts when the engine speed reaches an optimal power output (usually peak power RPM) the primary pulley starts 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 the pulleys allows for a sensitive torque feedback.

4. Shift-out
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.

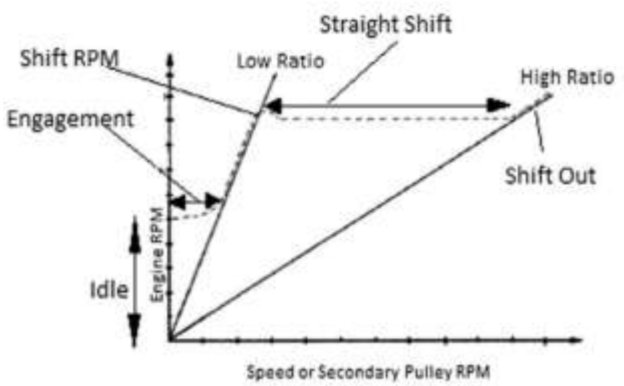

Fig-1: Operation of CVT

4. ANALYTICAL DESIGN OF CVT COMPONENTS

Data Available:

Engine Power = 10 HP = 7.45Kw

Max. Engine RPM = 3800 rpm

Min. Engine RPM = 1750±100 rpm

Wheel Diameter = 21 inch = 0.533 m

Considering max speed = 60 Kmph = 16.66 m/sec

Weight of the vehicle = 250 Kg (With driver)

Gearbox Reduction = 11

Design of CVT -:

1. Gear ratio calculation as per the requirement of the vehicle

$V_{max} = \frac{\pi DN}{60}$

$16.66 = \frac{\pi * 0.533 * N}{60}$

N = 597.26 rpm

Therefore, the higher ratio is calculated,

High ratio = $\frac{speed\ of\ driver}{speed\ of\ driven}$

$= \frac{3800}{11*597.26}$

= 0.58

Determination of lower ratio –

Considering angle of inclination as 40° and vehicle weight as 160 Kg with driver weight as 90 Kg.

Mass of vehicle with driver = 250 Kg

Angle of Inclination = 40°

Maximum torque = 20 Nm

Gearbox Reduction = 11

Wheel Diameter = 21 inch = 0.533 m

The force due to slope in opposite direction of motion will be:

$F_{slope} = \text{mg} \sin \alpha$

= 250×9.81× sin 40

= 2123.9 N

And Force available in the tires in direction of motion will be:

$F_{Tyre} = \frac{(Torque \times low\ ratio \times gearbox\ reduction \times 2)}{(wheel\ diameter \times 0.0254)}$

$= \frac{(20 \times low\ ratio \times 8.3 \times 2)}{(24 \times 0.0254)}$

= 544.619×low ratio

But to allow motion in uphill direction, $F_{tyres} \geq F_{slope}$

Therefore, $F_{tyre} = F_{slope}$

544.619×low ratio = 2123.9

∴ Low ratio = 3.9

Calculation of diameter of primary and secondary pulleys as per gear ratios:

Higher gear ratio = 0.58

Lower gear ratio = 3.9

Diameter of the primary pulley is considered as per the packaging space available at the vehicle.

∴ $D_p = 152.5$ mm

& $d_p = 32.5$ mm

Where, D_p is primary pulley diameter for high ratio and d_p is primary pulley diameter for low ratio.

Now,

$$LR = \frac{(Higher\ diameter\ of\ secondary\ pulley)}{(Lower\ diameter\ of\ primary\ pulley)}$$

$$3.9 = \frac{(Ds-14)}{(dp+14)}$$

$$3.9 = \frac{(Ds-14)}{(32.5+14)}$$

∴ $D_s = 195.35$ mm

Also,

$$HR = \frac{(Lower\ diameter\ of\ secondary\ pulley)}{(Higher\ diameter\ of\ primary\ pulley)}$$

$$0.58 = \frac{(ds+14)}{(Dp-14)}$$

$$0.58 = \frac{(ds+14)}{(152.5-14)}$$

∴ $d_s = 55.25$ mm

Diameters of primary and secondary pulley are:(Pitch diameter)

$d_p = 46.5$ mm

$D_p = 138.5$ mm

$d_s = 69.25$ mm

$D_s = 181.35$ mm

Diameters of the primary and secondary pulley are:(Edge to Edge diameter)

$d_p = 32.5$ mm

$D_p = 152.5$ mm

$d_s = 55.25$ mm

$D_s = 195.35$ mm

Belt Length Calculation:

Considering lower diameter of primary pulley and higher diameter of secondary pulley

And center to center distance between pulleys = 214 mm

$L = 2C + [\pi(D+d)/2] + [(D-d)^2/4C]$

Where,

D = 195.35 mm

D = 32.5 mm

C = 214 mm

∴ $L=2\times214+[\pi(195.35-32.5)/2]+[(195.35-32.5)^2/(4\times214)]$

∴ L = 816.7 mm

Now, for higher diameter of primary pulley and lower diameter of secondary pulley

$L = 2C + [\pi(D+d)/2] + [(D-d)^2/4C]$

Where,

D = 152.5 mm

d = 55.25 mm

∴ $L=2\times214+[\pi(152.5+55.25)/2]+[(152.5-55.25)/(4\times214)]$

∴ L = 765.20 mm

As, Maximum length is considered

Therefore, Length of Belt = 816.7 mm

Axial displacement of sheaves of pulleys

1. Axial displacement of movable primary pulley:

Diameter of primary pulley (During lower ratio) = 32.5mm

Diameter of primary pulley (During high ratio) = 152.5mm

Now,

V-belt angle = angle between the sheaves

$\therefore 2\beta = 24°$

$\therefore \beta = 12°$

Where, β = half wedge angle

$S_p = 2*[(D_p\text{-}d_p)/2] *\tan \beta$

$S_p = 2*[(138.5\text{-}46.5)/2]* \tan(12)$

$S_p = 19.05$ mm

Axial displacement of the primary pulley is 19.05 mm

2. Axial displacement of secondary pulley:

Diameter of secondary pulley (low ratio) = 55.2mm

Diameter of secondary pulley (high ratio) = 192.35mm

Now,

$S_s = 2*[(D_s\text{-}d_s)/2]* \tan \beta$

$S_s = 2*[(181.35\text{-}69.25)/2] *\tan(12)$

$S_s = 23.76$ mm

Axial displacement of the secondary pulley is 23.76 mm.

Calculation of Centrifugal force and flyweight mass:

Centrifugal Force = $mr\omega^2$

$N_{max} = 3800$ rpm

$r_{max} = 65$ mm

$\omega_{max} = \frac{2*\pi*N}{60} = \frac{2*\pi*3800}{60} = 397.93$ rad/sec

Considering, mass of all flyweights = 300 gm

Centrifugal Force = $\frac{0.3*65*(397.93)*(397.93)}{1000}$

= 3087.79 N

Therefore, Maximum centrifugal force is 3087.79 N

And,

As there are total 6 flyweights i.e. 3 arms each considering 2 flyweights respectively,

Mass of one flyweight = 300/6 = 50 gm

5. DESIGN OF CVT

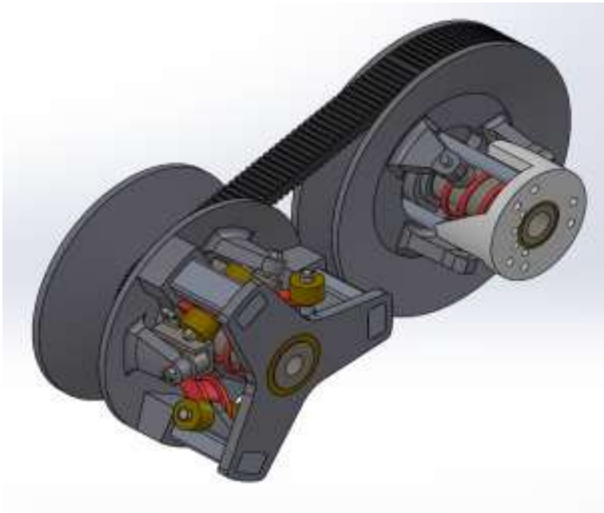

Fig-2: Assembly

6. FINITE ELEMENT ANALYSIS OF CVT

1. Primary Pulley: (Movable Sheave)
The primary pulley is mounted on the engine shaft which consists of ramp, roller mechanism. Hence it is analyzed for various load conditions.

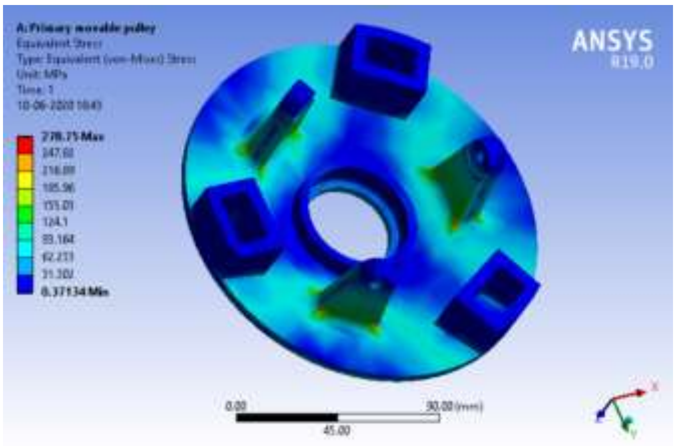

Fig-3: Stress Concentration

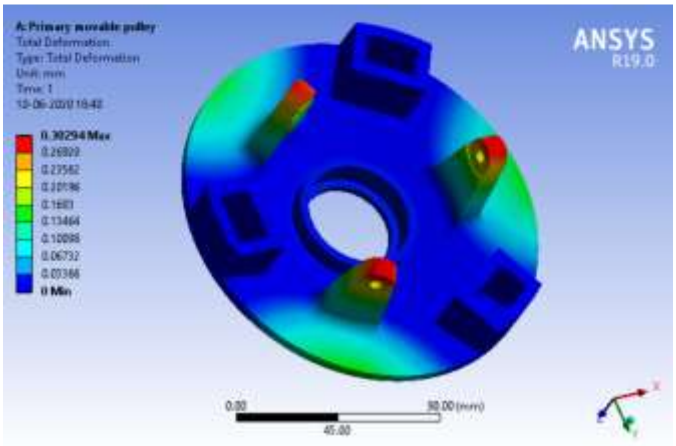

Fig-4: Total Deformation

As per the results, a cylindrical support is provided in the place where sheaves slide over the post and pulley is fixed at the point. The clamping force of 3000N is applied keeping in mind the factor of safety. Hence this result was obtained i.e. a maximum stress of 278 MPa and total deformation of 0.30mm was observed.

2. Primary Pulley: (Fixed Sheave)

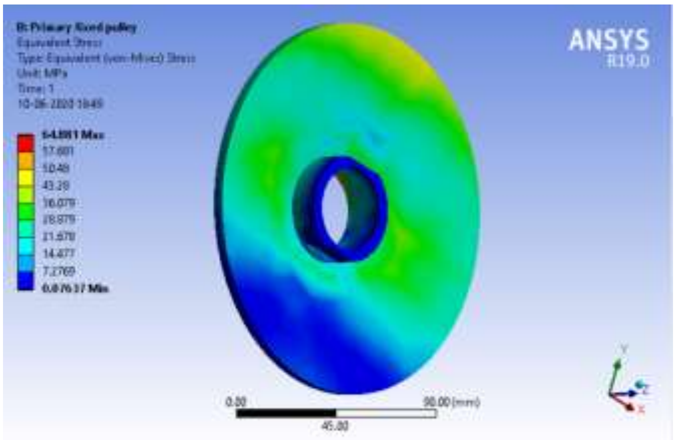

Fig-5: Stress Concentration

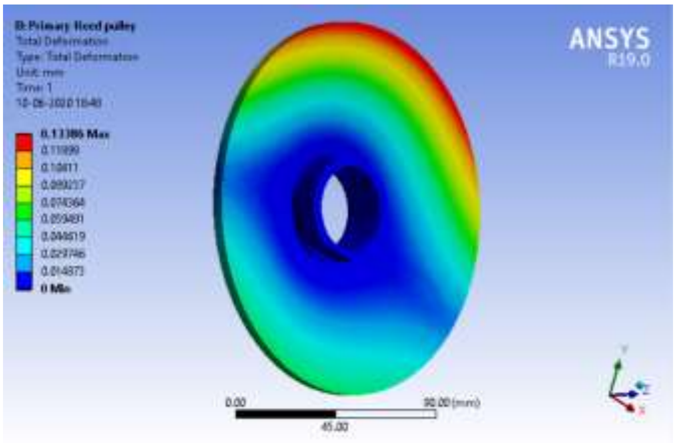

Fig-6: Total Deformation

In this case, while analyzing the fixed support is provided in the threaded region of the pulley at the center. Here a clamping force over sheave cone angle and belt tension is applied in direction of sheave.
A maximum stress of 65 MPa and total deformation of 0.13 mm was observed.

3. Secondary Pulley: (Movable Sheave)

The Secondary pulley is part of CVT which is mounted on the output shaft on the wheel side and contains the torque groove for easy back shifting. Hence, it is analyzed for all the loads of this condition.

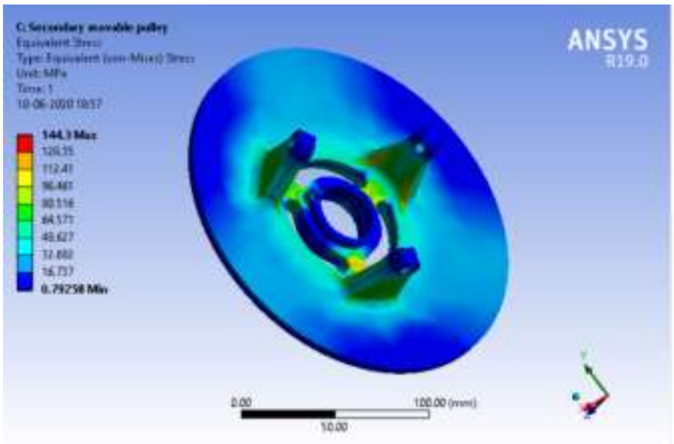

Fig-7: Stress Concentration

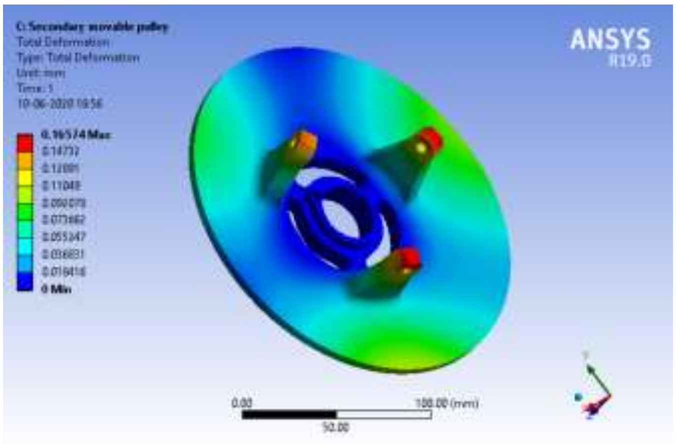

Fig-8: Total Deformation

This sheave undergoes two types of loading, i.e. a side force from the belt on the sheave acting along the perimeter of the belt and belt tension is another force acting on the perimeter of the sheave. Thus, following results are obtained where a maximum stress of 144 MPa was observed. Total deformation of0.16 mm was observed.

7. CONCLUSIONS

- Assuming various set of data and dimensions, the CVT has been theoretically designed as well as its assembly has been done. Accordingly, a CAD model has been designed to show the various parts and functioning of the CVT.
- The design was carried out based on the analytical model.
- The CVT was designed to obtain the initial gear ratio of 3.9:1 and an overdrive gear ratio of 0.58:1.
- All the components are analyzed in static structural analysis and maximum deformation was less than 1mm.
- CVT results in increased acceleration and torque by tuning and iterating various flyweights and spring iteration.
- Assembly and Disassembly timing was reduced and easy replacements of part can be done.

8. REFERENCES

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