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DESIGN & DEVELOPMENT OF A POWERTRAIN SYSTEM IN SCOPE OF BAJA VEHICLE.

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Abstract — *The goal of designing a BAJA vehicle is to facilitate the driver with an ATV that guarantees safety, dynamic stability, precise response to driver's efforts and sustainability in long term. The main objective of the drivetrain of a baja buggy is to provide consistent and reliable power to the wheels to tackle harsh terrain. This paper focuses on the selection, design and fabrication of transmission system of Baja Buggy. The design and FEA of component are inseparable aspects. The analysis provided clear understanding on projected stresses and possible failure locations. This allowed team to consider new techniques and adjustments in design to lower the possibility of failure. This marks the efforts put in CAE of multiple components on vehicle. The report also focuses on optimisation done to ensure proper use of resources and removing any cost adding wastage.*

Keywords* - Ansys, Creo Parametric 2.0, CFD analysis, Gaged CVT, Custome Gear Box, Gradability.*

1. INTRODUCTION

Designing the drivetrain of BAJA vahicle is a challenging task. The primary target was to minimize as much weight as possible in the drivetrain system. We focused on cutting off unnecessary masses from components that had no help whatsoever in strengthening the component. Also we focused on packaging of the transmission system to make it more compact and serviceable. A standard Briggs and Stratton, model-19 engine, power the drivetrain of BAJA vehicle. The engine empowers system with theoretical max power of 10 HP at 3600 RPM and max torque of 19.65 N-m at 2800 RPM[1]. The engine is engaged with the transmission gearbox with help of a CVT, Gaged make. A two-stage, customised gearbox, with reduction of 7.8:1, achieves the secondary reduction. Finally, customised driveshafts connect the wheels to output of gearbox, via arrangement of constant velocity joints.

II. SELECTION OF TRANSMISSION SYSTEM

The main objective of the drivetrain system is to provide consistent power to the rear axle to propel the car with minimum loss of power[2]. The power transferred must be able to move the vehicle up steep grades and propel it at top speed of 60kmph on level terrain. Acceleration is also an important characteristic controlled by the drive train. Drivetrain should be compact, lightweight and having a low height. Drivetrain should be properly mounted on chassis so that the power transmitting elements should be properly aligned with each other so as to reduce the power losses[3]. System should have enough serviceability for easy and quick maintenance.

The drivetrain of vehicle is designed keeping these objectives in mind. The first step was to chose the type of transmission system for the buggy. The basic types of transmission system used commercial vehicles are Manual Transmission, Continuously Variable Transmission and Automatic Transmission[4].

With a study over these systems it was concluded that the Continuously Variable Transmission dominated over the other systems in following aspects[4];

1. Ability to operate the engine at peak power throughout the shift.
2. Light weight.
3. Change the engine speed to access required torque and speed according to terrain
4. Quicker acceleration and faster response to changing terrain.
5. Ability to tune according to terrain.

The initial speed reduction is achieved by a continuous variable transmission of Gaged Engineering. The CVT was selected on basis of adaptability, smoother performance, and availability of reduction ratio in desired range. The CVT provides variable primary speed reduction in range of 3.9-0.9[5].

2.1 CVT and its component
Primary Spring:- Pushes clutch open.
Flyweight: - Pushes clutch close.
Helix Angle: - Provides upshift rate/back shifting rate.
Belt: - Link between engine and secondary shaft

III. SECONDARY SPEED REDUCTION GEARBOX

A secondary speed reduction gearbox is required to achieve required torque and required top speed. Secondary reduction should be chosen properly considering following factors :

a. Efficiency of CVT,
b. Maximum speed of vehicle
c. Gradability.

Calculation for secondary reduction

$$V = \frac{\pi * D * N}{60 * G}$$

Where,

V = Required maximum velocity m/s,

D = Tire diameter m,
N = Engine speed rpm,
G= Reduction ratio at full shift

$$\therefore V = \frac{3.142 * 584.2 * 10^{-3} * 3800}{60 * G}$$

G = 6.99

But CVT reduction achieved at full shift= 0.9

G = 0.9 x Fixed reduction

Fixed reduction = 7.75

The Main Objectives in designing the gearbox are as follow:

1. Required reduction ratio is 7.75
2. Reduction in weight and volume.
3. Ease of serviceability.
4. Required centre to centre distance is less than 200mm.
5. The jackshaft should coupled with CVT and output shaft with Mahindra Alfa Differential

Fig 1: Customized Two Stage Reduction Gearbox

Fig 2: Exploded view of Gearbox

Efficiency of CVT reduces as the transmission goes into the overdrive[6]. Considering the overdrive ratio of 0.9 of Gaged GX9 CVT, the secondary reduction required to achieve the top speed of 60 km/hr. comes to be 7.74. However, a taller gear ratio of 7.84 is selected for the secondary reduction since taller gearing achieves the top at a ration before overdrive. Thereby increasing the efficiency of CVT in addition secondary reduction of 7.84 ensures enough torque in the start of motion.

3.1 GEAR DESIGN

With the reduction being fixed the design of gearbox begins with the design of the gear pairs. Spur gear pair was selected for its ease of manufacturing. The gear ratio was divided into two stages considering the size of the final gear to be accommodated on the Mahindra Differential. Hence it was decided to achieve the reduction of 2 in first stage and further reduction in second stage thus ensuring proper packaging of gearbox.

Then the gear pair was designed according to AGMA standard for failure against bending, wear and fatigue[7]. The objective was to obtain minimum standard module for gears without compromising the strength of the gears.

Calculations for the gears led to the final module of 2.25mm for first stage and 2.5mm for second stage reduction.

All the calculations were done for high strength alloy steel 20MnCr5. This material was selected for feasibility of heat treatment for surface hardness of 45 HRC.

The following table illustrates the parameters of gears after multiple iterations.

Table 1: Gearbox Specification

First Stage	
Teeth on Pinion	18
Teeth on Gear	41
Module	2.25
Reduction Ratio	2.27
Second Stage	
Teeth on Pinion	18
Teeth on Gear	62
Module	2.5
Reduction Ratio	3.44
Overall Reduction Ratio	7.84

The calculations for final gears can be found in Appendix B.

3.2 GEARBOX SHAFT DESIGN

With the design of gear pairs the tangential and radial forces arising due to meshing of gears were considered for determining the diameter of jackshaft and intermediate shaft of the gearbox. The shafts were designed using ASME code for shaft design[7].

The gearbox and CVT are supposed to be coupled with the help of jack shaft. The CVT being an OEM part the diameter of the jackshaft was fixed to be 19.05 mm. The diameter of intermediate shaft was calculated to be 20 mm.

3.3 BEARING SELECTION

Single row deep groove ball bearings were selected to hold the jackshaft and intermediate shaft rigidly in the gearbox. The bearing selection was done on the basis of dynamic load carrying capacity of the bearings[7].

With the calculation of required dynamic load carrying capacity of the bearings, Bearing 6004[8] was selected for the jackshaft and intermediate shaft from the catalogue. Whereas the differential being an OEM part it was mounted with its original Bearing 16010[8] into the gearbox.

3.4 GEARBOX HOUSING

After the complete design of gear pairs, the shafts and selection of the bearings the next task was to design the housing of the gearbox and carry out the packaging of the gearbox for optimum weight and volume. Since the CVT was supposed to be coupled with the gearbox using the jackshaft it was necessary to maintain enough clearance between the CVT and the roll cage members. This was sorted by tilting the second gear pair at an angle of 110 from the horizontal. Keeping all these constraints in mind the housing material was chosen to be Aluminium 6061-T6. Aluminium being light in weight also has good heat dissipation capacity for better cooling of the gearbox.

The housing was designed with optimal clearance around the gears to allow enough quantity of oil flow for proper lubrication and heat dissipation[8]. Static structural analysis was performed on the CAD model of the housing of with various thickness of cross section starting from 3mm upto 5mm. The results were compared for balance between strength and weight. From the results it was concluded that the housing with a cross section thickness of 4.5mm was structurally sound with optimum weight.

The load cases that were considered for the analysis are the shaft loads and the weight of the gearbox acting on the mounting points with the maximum acceleration of 0.75g.

Fig3: Geartrain Layout

3.5 GEARBOX HOUSING MACHINING

Survey was carried out for machining process of the aluminium gearbox housing. Aluminium casting is the most feasible machining process for commercially available gearbox housings because of their mass production rates. Since the gearbox housing of Zephyr 6.0 was a single prototype casting would have increased the project cost considerably. To cut off this unnecessary cost machining the housing from single aluminium billet on a Vertical Machine Centre (VMC) was the most feasible solution.

IV. CVT TUNING

The objective of the CVT Transmission is to hold the engine at a constant rpm throughout the shift of the gear ratio. The engine should produce maximum power at this shift rpm. Also the CVT should engage at a rpm near the peak torque to ensure that enough torque is delivered to the wheels at initial start of the vehicle. The engine produces peak torque at in range of 2600-2800 rpm.

The engaging and shift rpm were recorded when the CVT was received from the manufacturer. With the primary flyweights of 129gms in each arm of CVT the CVT happened to engage at 2350rpm. To achieve the target engagement rpm of 2600 the flyweight mass is reduced by 24 gms. Thus allowing the CVT to engage at 2600 rpm.

After achieving the required engaging rpm the start line conditions were improved by changing the preload of torsion spring in the driven pulley[6]. A high preload on the driven spring increases the shift rpm of the CVT since higher preload provides more resistance for the belt to drive into the secondary pulley. It was found that keeping the torsion spring in Hole #6 provided the best start line conditions giving aggressive take off's. Also a higher preload ensures quick back shifting into higher ratios when the vehicle sets out into turns and bumps where the momentum of vehicle reduces considerably. Quick back shifting makes the vehicle to get out of the corners without losing speed and momentum[6].

4.1 CVT COOLING

Friction between the belt and the sheaves of the CVT Pulley is inevitable. Tough friction is required for power transmission from driving to driven pulley it also causes heating issues thereby reducing the belt life[9]. The recommended operating temperature of the belt must be within 180^0 C for best results. To maintain the temperature well within the limits its necessary to keep the casing of the CVT well ventilated. Following the rules of BAJA SAEINDIA 2017 its required to enclose the entire operating area of CVT with a casing either of Aluminium 6061 or AISI 1010. Considering the weight and heat dissipation capacities Aluminium sheet 6061-T6 was utilized to manufacture the casing. After the modeling of the casing CFD analysis was performed to determine the air flow rate during operating conditions of the CVT. With the results of CFD analysis ducts were provided for air intake into the casing. Thereby ensuring effective cooling of CVT. Again these results were practically verified by smoke test as shown in Figure.

Fig 4: Smoke test of CVT casing.

V. DRIVESHAFTS

The drive shaft connect the output from the gearbox to the wheels .In order to avoid output shaft failure, a series of test were performed on a shaft used by commercial ATV's.

The first test recollected the superficial hardness of shaft.

The second recollected the hardness in the transversal section for different radii. This second test suggested that a thermal treatment are performed on the shaft. This study suggests that after machining, the piece must pass through thermal treatment to harden the surface and to avoid break due to torsional stresses.

Thus maintaining a hard case ensures safety against torsional stress and a soft core gives better toughness to the shaft to handle fatigue loads.

For output shaft, EN24 alloy steel was selected for its high strength and feasibility for heat treatment. OEM constant velocity joints from Maruti 800 were utilized at the wheel ends to provide enough angle of articulation for high wheel rates to be tackled during tough terrain.

VI. CAE ANALYSIS

7.1 MODAL ANALYSIS OF GEARBOX CASING

Modal analysis or vibration analysis has been done to ensure that the natural frequency of the gearbox casing does not match the frequency of vibration of the rotating gears in its working range[9].
In the working range, the frequency of rotating gears of gearbox has the value from 426.66 Hz to 1093.33 Hz. After modal analysis, the following results were obtained.

Table 2: Modes of vibration of casing

MODE	FREQUENCY
1	0
2	0
3	0
4	0.00289
5	0.00308
6	0.00510
7	1704.2
8	2043.7
9	2631.9
10	2732.0

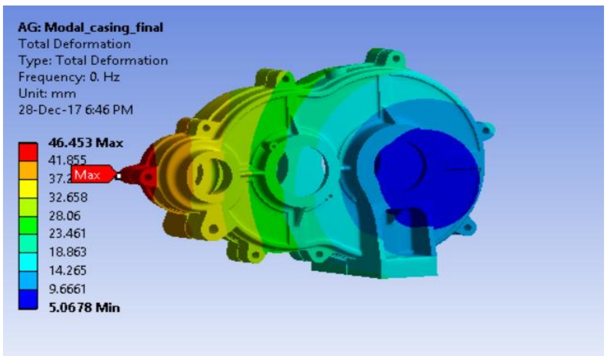

Fig5: Modal analysis of casing

VII. RESULTS

- Weight reduction of transmission system=27%
- Bulk reduction of transmission system=5 Litres
- Reduction in C.G height = 1.5 inches

Above result values are in comparison the erstwhile years BAJA vehicle.

- Maximum speed of the vehicle =56 kmph
- Gradability = 57%
- Acceleration =4.96 m/s^2

VIII. CONCLUSION

The foremost intent to minimize the weight and bulk of the transmission system of a BAJA vehicle thereby enhancing the performance of the vehicle and cutting off unnecessary cost appeared to be accomplished depriving compromised safety, durability and serviceability of the integrants. Further this report can be employed to design transmission systems other types of vehicle.

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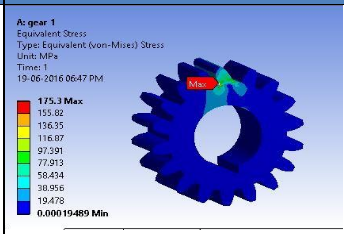
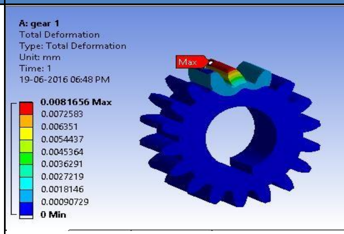
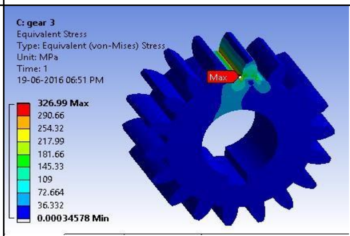
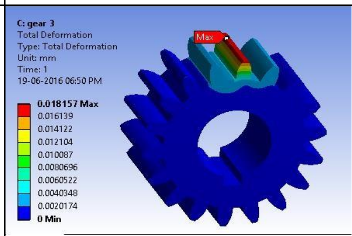
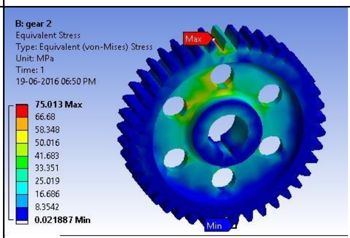
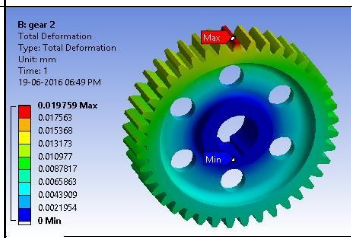
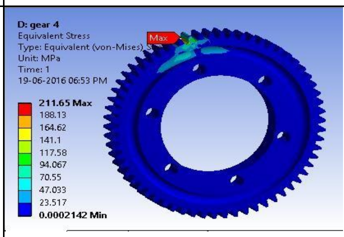
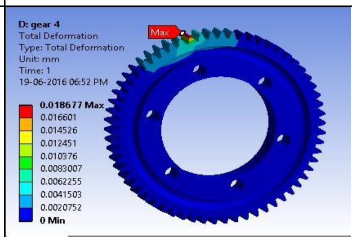
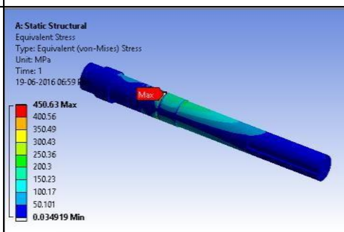
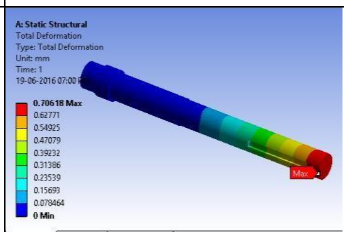
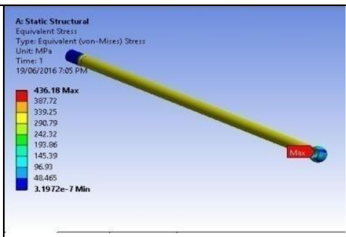
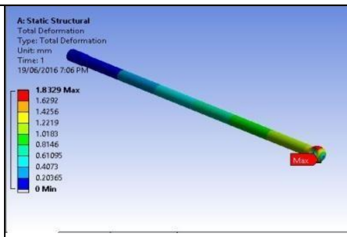
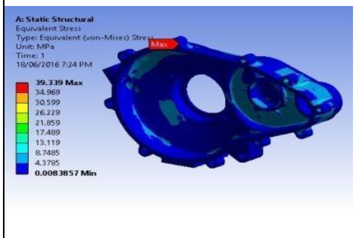
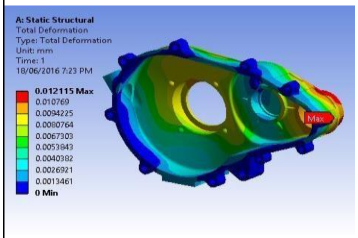
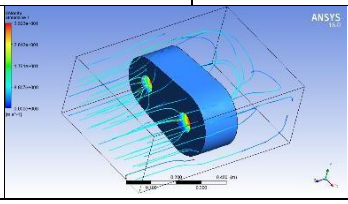
[9] Nitin S Gokhale, "Practical Finite Element Analysis" Published by Finite to Infinite

APPENDIX A

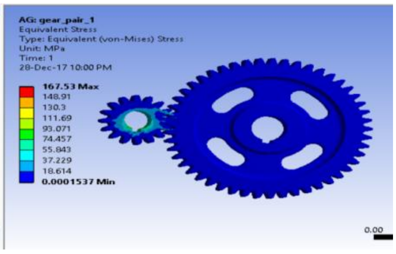
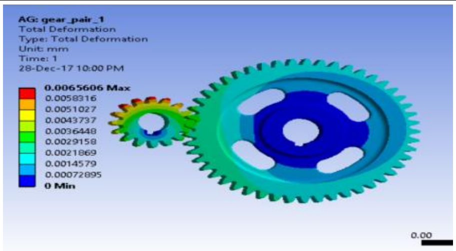
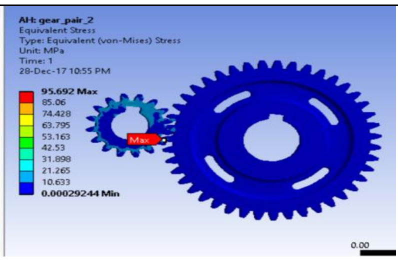
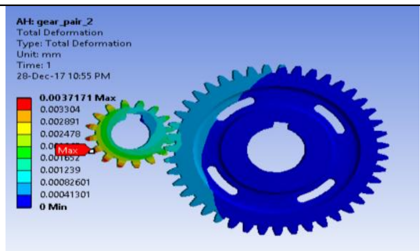
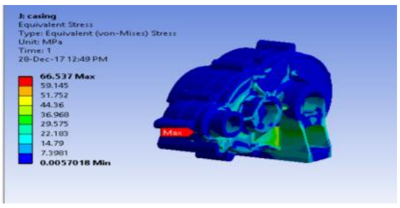
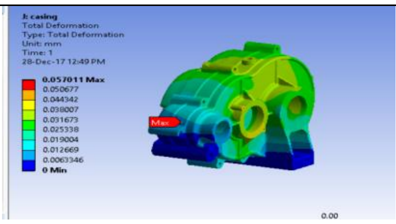
ANALYSIS OF DRIVE TRAIN COMPONENTS

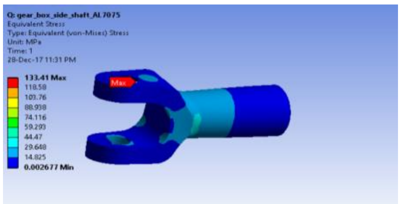
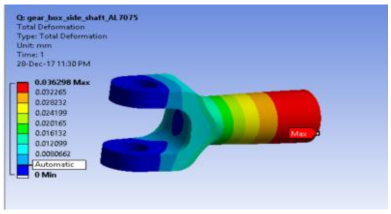
<table>
<tr><th>Sr No</th><th>Component</th><th>Stress</th><th>Deformation</th></tr>
<tr><td rowspan="2">1</td><td rowspan="2">Stage 1 Pinion</td><td>
</td><td>
</td></tr>
<tr><td>175.3 N/mm^2</td><td>0.00816 mm</td></tr>
<tr><td rowspan="2">2</td><td rowspan="2">Stage 1 Gear</td><td>
</td><td>
</td></tr>
<tr><td>326.99 N/mm^2</td><td>0.01815 mm</td></tr>
<tr><td rowspan="2">3</td><td rowspan="2">Stage 2 Pinion</td><td>
</td><td>
</td></tr>
<tr><td>75.013 N/mm^2</td><td>0.01975 mm</td></tr>
<tr><td rowspan="2">4</td><td rowspan="2">Stage 2 Gear</td><td>
</td><td>
</td></tr>
<tr><td>211.65 N/mm^2</td><td>0.01867 mm</td></tr>
<tr><td rowspan="2">5</td><td rowspan="2">Jack Shaft</td><td>
</td><td>
</td></tr>
<tr><td>450.63 N/mm^2</td><td>0.70618mm</td></tr>
</table>

6	Drive Shaft		
		436.18N/mm^2	1.8329 mm

7	Gearbox Housing		
		39.339 N/mm^2	0.0121 mm
8	CFD of CVT Cover		

ANALYSIS OF DRIVE TRAIN COMPONENET

Sr.No.	Component	Stress	Deformation
1.	Gear pair 1	 167.73MPa	 0.0065606mm
2.	Gear pair 2	 95.692Mpa	 0.0037171mm
3.	Casing	 66.537Mpa	 0.057011mm

4.	UV	 133.41Mpa	 0.036298mm

APPENDIX B

Final calculations

Zp = Number of teeth on pinion = 18

Zg = Number of teeth on gear = 41

G = Gear Ratio = 2.27

According to Lewis equation bending force acting on single tooth of the gear is given by [7]

$$Fb = \sigma b * b * m * y \text{ ....(i)}$$

where,

σb = bending stress

b = face width of gear

m = module

Y = lewis form factor

$$\therefore \sigma b = \frac{sut}{3}$$

where,

S_{ut} = Ultimate Tensile Strength

For 20MnCr5 we Ultimate Tensile Strength of 950 N/mm2

σb = 316.66 N/mm2

b = 9.5 × m

For 200 full depth involute tooth profile Lewis Form Factor is given,

Y = [0.484 - G/Zp]

Y = [0.484 - 2.27/18]

Y = 0.357

from equation (i) we have

$Fb = 1073.95\ m^2$.......................(a)

Now we calculate the wear strength of the gear tooth by

$F_w = dp \times b \times Q \times k$..........................(ii)

where,

dp = PCD of the pinion

Q = Ratio Factor

K = Load Stress Factor

$Q = (2xZ_g)/(Z_g+Z_p)$

Q = (2x41)/(18+41)

Q = 1.3898

$k = 0.16(BHN/100)^2$

$K = 0.16(270/100)^2$

K = 5.1984

b = 9.5m

dp = 18 m

From equation (ii) we have

$F_w = 18\ m \times 9.5\ m \times 1.3898 \times 5.1984$

$F_w = 1235.42\ m^2$

Since Fw>Fb

Gear pair is weaker in bending

Gear pair is designed against bending failure.

From power curve of Briggs & Straton engine we find that engine produces power of 6.7Kw at 3800rpm.

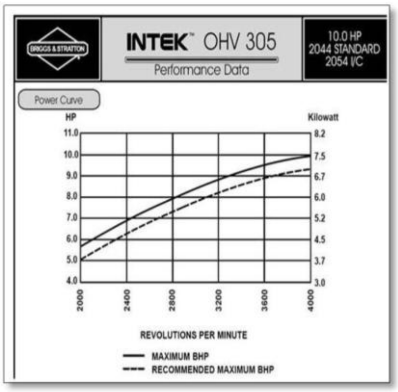


Fig 5: Power curve of Briggs & Stratton OHV engine

Te = P/W

$$\therefore Te = \frac{6.7\times10^3}{\left(\frac{2\pi N}{60}\right)}$$

$$\therefore Te = \frac{6.7\times10^3}{\left(\frac{2\pi\times3800}{60}\right)}$$

Te = 16.83 N-m

Considering the efficiency of the belt drive CVT to be 80%.

Therefore the torque acting on the gear is given by,

$$\tau = \eta_{CVT} \times Te \times CVT\ redn$$

$= 0.8 \times 16.83 \times 0.92$

$= 12.1176$ N-m

Now,

Tangential force acting on the gear is given by,

$$Ft = \frac{\tau}{r}$$

Ft = [12.1176 / (d/2)]

Ft = [12.1176 / (mt/2)]

Ft = 1346.4/m

Pitch Line velocity of the gear is given by,

$$V = \frac{\pi * d_{p*N_p}}{60}$$

Where,
d_p = mt = 18m
N_p = 4222.2 rpm
V = 3.97m m/s

Take,
Ka = 1 & Km = 1.2

$$Kv = \frac{5.6}{5.6+\sqrt{3.97m}}$$

$$Feff = \frac{\left(\frac{1\times1.2\times1346.4}{m}\right)}{\left(\frac{5.6}{5.6+\sqrt{3.970m}}\right)} = \frac{288.51+\left(5.6+\sqrt{3.97m}\right)}{m}$$

Now,

Fb = N × Feff
Assume a factor of safety of 2
N = 2
Thus solving for m we get,

$$1073.95\ m^2 = \frac{2\times288.51}{m}\left(5.6 + \sqrt{3.97m}\right)$$

$$\therefore\ m^3 = 0.5372\left(5.6 + \sqrt{3.97m}\right)$$

$$\therefore\ m^3 - 1.04\sqrt{m} - 3.008 = 0$$

m = 1.6307 mm

Taking next higher standard module we have,
m = 2.25 mm
Now, the dynamic load acting on the gear is calculated by Buckingham's Formula given by,

$$Fd = \frac{21v(bc+ F_{tmax})}{21v+\sqrt{bc+Ft_{max}}}$$

dp = 2.25 × 18 = 40.5 mm
dg = 2.25 × 41 = 92.25 mm
b = 19.5m
V = 8.953 m/s
ep = 8+0.63(2.25+0.25√40.5)
=10.41μm
eg = 8+0.63(2.25+0.25√92.25)
= 10.93 μm
e = ep + eg
= 21.96 x -10000 mm
c = 11500e = 245.41 N/mm

Fmax = Ka × Km × Ft
Fmax = 715.08 N

bc = 9.5 x 2.25 x 245.41
= 5245.63

$$Fd = \frac{21\times 8.953(5245.63+718.08)}{21\times 8.953+\sqrt{5245.63+718.08}}$$

Fd = 4227.35 N
Feff = Fd + Fmax
Feff = 4227.35 + 718.08
Feff = 4945.43N
Now comparing the available FOS with target FOS we get,
Fb = N × Feff
N = (1073.95 × 2.25^2) / 4945.43
N = 1.099

Since available Factor of safety if greater than 1 design is safe.
Following the same procedure the next gear pair is designed for the second stage reduction.

APPENDIX C

DESIGN OF JACK SHAFT

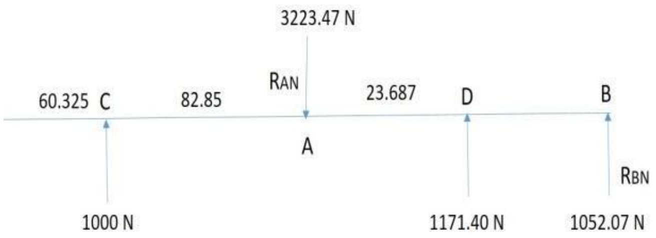


Fig 8 : Horizontal FBD of Jackshaft

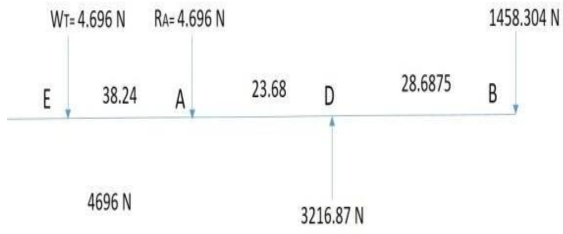


Fig 9 : Vertical FBD of Jackshaft

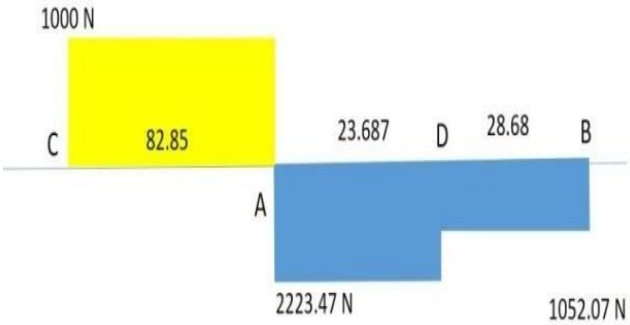


Fig 10: Horizontal Shear Force Diagram of Jackshaft

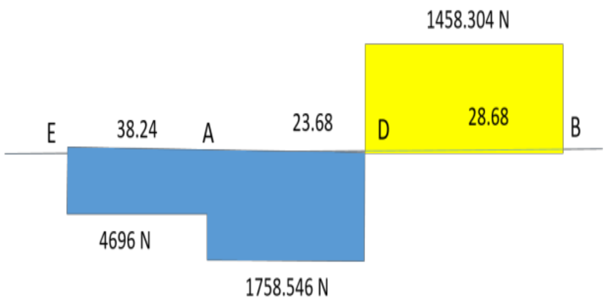


Fig 11: Vertical Shear Force Diagram of Jackshaft

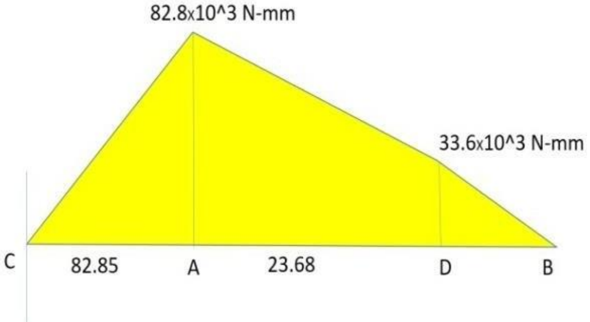


Fig 12: Horizontal Bending Moment Diagram of Jackshaft

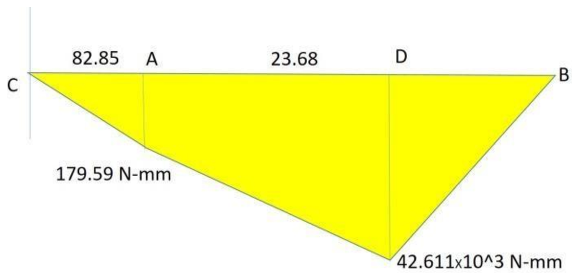


Fig 13: Vertical Bending Moment Diagram of Jackshaft

From Horizontal and Vertical Bending Moment Diagram

Resutant bending moment at pt A is,

$BM_A = \sqrt{(82.8*10^3)^2 + (179.59 * 10^3)^2}$

$= 197.75 *10^3$ N-mm

Resultant Bending moment at pt D is,

$BM_D = \sqrt{(33.6*10^3)^2 + (42.611*10^3)^2}$

$= 54.26*10^3$N-mm

& maximum Bending Moment occurs at pt A

$BM_A = 197.75$ N-mm

Equivalent twisting moment (T_e),

$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau \times d^3$$

Torque acting on jackshaft be T

T_{engine} = engine torque

= 16.83 N-m

T = 0.8 * 16.83 * 0.9

T = 12.1176 N-m

$T_e = \sqrt{(197.75*10^3)^2 + (12.1776*10^3)^2}$

T_e = 198.12 N-mm

According to ASME code,

$\tau_{allowable}$ = 0.18Sut or 0.3Syt............whichever is less

$\tau_{allowable}$ = 0.18*950 or 0.3*680

= 171 N/mm^2 or 204 N/mm^2

$\tau_{allowable}$ = 171 N/mm^2

Now,

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

$$198.12 \times 10^3 = \frac{\pi}{16} \times 171 \times d^3$$

Te = 18.07 mm

Taking the next standard diameter we have,

d = 20 mm

Thus diameter of jackshaft is determined to be 20mm.

Similarly the diameter of intermediate shaft and final drive shaft is calculated.