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DESIGN AND DEVELOPMENT OF TWO SPEED GEAR BOX FOR AGRO MACHINERY

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ABSTRACT

Conventional gearboxes are capable of varying a given input speed. It is achieved by meshing of gears in various gear ratios. The torque values are different during different gear ratios. Hybrid gearboxes are capable of transmitting various torque levels at the same gear ratio. They have a high torque producing capacity compared to a conventional gearbox. These gearboxes have provisions for several inputs and several outputs, unlike one input and one output of a conventional gearbox. It allows the choice of varied speeds to the inputs.

These gearboxes can be used in a lot of practical applications. As they have very high loading capacities, they can be used in off-road, commercial vehicles, military vehicles and other specialty vehicles. They can also be used in cranes, pumps, tractors, lawn mowers etc. Its most important application is that it can be used in a hybrid car.

KEYWORDS: Conventional gearboxes, Commercial vehicles, High Loading Capacities Speed, Torque.

I. INTRODUCTION

A transmission is a machine in a power transmission system, which provides controlled application of the power. Often the term transmission refers simply to the gearbox that uses gears and gear trains to provide speed and torque conversions from a rotating power source to another device.

The most common use is in motor vehicles, where the transmission adapts the output of the internal combustion engine to the drive wheels. Such engines need to operate at a relatively high rotational speed, which is inappropriate for starting, stopping, and slower travel. The transmission reduces the higher engine speed to the slower wheel speed, increasing torque in the process. Transmissions are also used on pedal bicycles, fixed machines, and where different rotational speeds and torques are adapted.

In motor vehicles, the transmission generally is connected to the engine crankshaft via a flywheel or clutch or fluid coupling, partly because internal combustion engines cannot run below a particular speed. The output of the transmission is transmitted via the driveshaft to one or more differentials, which drives the wheels. While a differential may also provide gear reduction, its primary purpose is to permit the wheels at either end of an axle to rotate at different speeds as it changes the direction of rotation.

Conventional gear/belt transmissions are not the only mechanism for speed/torque adaptation. Alternative mechanisms include torque converters and power transformation. Hybrid configurations also exist. Automatic transmissions use a valve body to shift gears using fluid pressures in response to speed and throttle input.

Most modern gearboxes are used to increase torque while reducing the speed of a prime mover output shaft (e.g. a motor crankshaft). This means that the output shaft of a gearbox rotates at a slower rate than the input shaft, and this reduction in speed produces a mechanical advantage, increasing torque. A gearbox can be set up to do the opposite and provide an increase in shaft speed with a reduction of torque. Some of the simplest gearboxes merely change the physical rotational direction of power transmission.

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II. SURVEY AND NEED OF 2 SPEED GEAR BOX

1) Sprayers gears are commonly used on farm to spray pesticides, herbicides there are many kind of machine operated Sprayers.
2) The tractors mounted Sprayers most commonly used in farm. The Sprayers may have boom 12 to 50 Ft.
3) So fan require to rotate with high speed about 3000 to 5000 rpm.
4) For Sprayers 180-25 HP tractor mounted machine but to tractor has maximum speed from 900 to 1000 rpm.
5) So far increase the rpm required to use gear box which change the speed from 1000 to 5000 rpm.
6) Also require to vary speed.

III. LITERATURE REVIEW

Maruti Patil, P Ramkumar, K Shankar (30/01/2019) [1] Multi objective minimization of power loss and volume of a two-stage helical gearbox with additional novel tribological constraints was carried out. The results were compared with a single objective optimization with and without tribological constraints. The single objective problems minimize the volume only following the traditional approach in gear optimization literature. The simulation was done for a variety of oils (ISD VG 68,150,360 and 580) and at 1000 and 1500 rpm with three different gear profiles.

Diego Cabrera Fernando Sancho Chuan Li MarielaCerradaRen´e-VinicioS´anchezFannia Pacheco Jos´e Valente de Oliveira (**2016**) [2] The case study is the identification of fault severity in helical gearboxes from a vibration signal, where the design and extraction of condition parameters is a non-trivial task when performed by the conventional classical methods. Moreover, the results obtained by the latter methods are hardly applicable to other480 real-world systems as the features containing the most representative information are highly dependent on the specific mechanism.

IV. INPUT DATA FOR DESIGN

Maximum Engine RPM(Input)	980
Ratio	1:4 & 1:5
Speed (rpm)	3900 & 4900
Input rotation	Anticlockwise
Output rotation	Depends upon actuating lever position
Oil specification	SAE 30
Input Torque	181.63 Nm
Operation	Lever Mechanisms

V. CALCULATIONS

DESIGN OF GEAR:

The first criteria in designing the gears are to keep them simple, less weight and at the same time to keep the cost as low as possible. So, the weight and cost have their respective weightage during the design such that both the parameters could be worth enough. The machinability is another important consideration.

Gear Profile Parameters:

Type of gears used: Spur Gear & stub involute. The following the parameters required to obtain the required gear profile.

Input Data:

P= 18 -25 HP

N_i= 980 rpm (Input speed)

N_1= 3900 rpm

N_2= 4900 rpm

Material Specification	Value
Surface Hardness	HRC45
Brinell Hardness Number	420
Ultimate Tensile Strength (S_{ut})	700 N/mm^2

Gear ratio is $\frac{Ni}{N1} = 1{:}\ 4$

Assume minimum number of teeth of pinion to avoid interference is,

Z_p=10

α =20°

ψ=15°

$$d_p = \frac{z \times m_n}{cos\psi}$$

$$=\frac{10 \times mn}{72+18}$$

d_p=10.352mn mm

Motor Torque:

$$M_t = \frac{60 \times 10^6 \times P}{2\,\pi \times Np}$$

P= 25 Hp

= 18.64 kw

Np=980

$$M_t = \frac{60 \times 10^6 \times 18.64}{2\pi \times 980}$$

M_t =181631.51 $N.mm$

Tangential Force:

$$Pt = \frac{2 \times Mt}{dP \times m_n}$$

$$Pt = \frac{2 \times 181631.51}{10.35 \times m_n}$$

$$Pt = \frac{35097.87}{m_n}$$

$$Peff = \frac{Cs \times Pt}{Cv}$$

$$= \frac{1.5X35097.87}{cos10.635 \times m_n}$$

$$= \frac{1.5X35097.87823889.36}{m_n}$$

$$Y = 0.320$$

Beam strength:

$$Sb = m_n \times b \times \sigma b \times Y$$

$$= m_n \times 10m_n \times 233.33 \times 0.320$$

$$746.6560mn^2 = \frac{8389.36}{m_n}$$

$$m_n^3 = 110.34$$

$$m_n = 4.75$$

Due to higher ratio the module also is more gear box is bulky,so as per company suggestion availability of hob of 3 module .

So we take,

$$m_n = 3$$

So for ratio 1:4

980	I/P (G_1)	O/P(G_2)

As bigger module and ratio gear box may become bulky so we take minimum no. of teeth 10.

But to avoid interference we make center distance more than calculation (suggestion of design Engineer.)

$$G_{2=}\ Pinion$$

$$Z_2 = 10$$

$$m_n = 3$$

$$\psi = 15°$$

$$\alpha = 20°$$

$$d_p = \frac{z_2 \times m_n}{cos\psi}$$

$$d_p = \frac{10 \times 3}{\cos(15)}$$

$$d_p = 31.05 \;\; m$$

$$G_{1=}Input\ gear$$

$$Z_1 = 50$$

$$m_n = 3$$

$$\psi = 15°$$

$$\alpha = 20°$$

$$d_p = \frac{z_1 \times m_n}{cos\psi}$$

$$d_p = \frac{50 \times 3}{\cos(15)}$$

$$d_p = 155.29\ mm$$

For the ratio 1:5

980	I/P (G_1)	4900	O/P (G_2)

$$G_{2=}As\ pinion$$

$$Z_2 = 12$$

$$m_n = 3$$

$$\psi = 15°$$

$$\alpha = 20°$$

$$d_p = \frac{z_2 \times m_n}{cos\psi}$$

$$d_p = \frac{12 \times 3}{\cos(15)}$$

$$d_p = 37.26 \;\; mm$$

$$G_{1=}I/P$$

$$Z_1 = 48$$

$$m_n = 3$$

$$\psi = 15°$$

$$\alpha = 20°$$

$$d_p = \frac{z_1 \times m_n}{cos\psi}$$

$$d_p = \frac{60 \times 3}{\cos(15)}$$

$$d_p = 149.07 \ mm$$

DESIGN OF SHAFT:

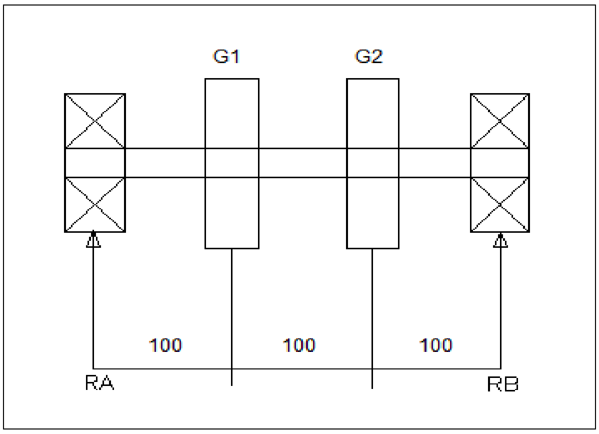

Fig-1: Design of Shaft

Tangential Force:

Gear 1

$$F_{G1} = \frac{2 \times T}{D_1}$$

$$F_{G1} = \frac{2 \times 181.631}{0.155}$$

$$F_{G1} = 2343.62 \ N$$

Gear 2

$$F_{G2} = \frac{2 \times T}{D_2}$$

$$F_{G2} = \frac{2 \times 181.631}{0.149}$$

$$F_{G2} = 2438 \ N$$

Normal load acting on shaft:

$$W_1 = \frac{F_{G1}}{cos \propto}$$

$$= \frac{2343.62}{\cos 20°}$$

$$W_1 = 2494.02\, N$$

$$W_2 = 2594.46\, N$$

Shaft considering twisting and bending moment:

$$M = W1 \times 0.1$$

$$M = 2594.46 \times 0.1$$

$$M = 259.44\ \ Nm$$

Equivalent twisting moment:

$$T_e = \sqrt{M^2 + Mt^2}$$

$$T_e = \sqrt{(259.44)^2 + (181.631)^2}$$

$$T_e = 316.700 \times 10^3 \quad Nmm$$

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

$$316.700 \times 10^3 = \frac{\pi}{16} \times 233.33 \times d^3$$

$$d = 19.04\, mm$$

Therefore we take maximum diameter d=25mm

Due to high power transmission we make a splines on shaft .

so design a shaft by considering spline on shaft.

$$M_t = \frac{1}{8} P_m \ln(D^2 - d^2)$$

Where,

D = major diameter of splines (mm)

d = minor diameter of splines (mm)

l = length of hub (mm)

n = number of splines

M_t = transmited torque (N-mm)

P_m = permissible pressure on spline (N-mm^2)

A = total area of splines (mm^2)

R_m = mean radius of splines (mm)

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$$181631.51 = \frac{1}{8} \times 6.5 \times 20 \times 8(D^2 - 0.8600)$$

$$D^2 - 0.8600 - 1397.16 = 0$$

$$D = 38 \text{ approx}$$

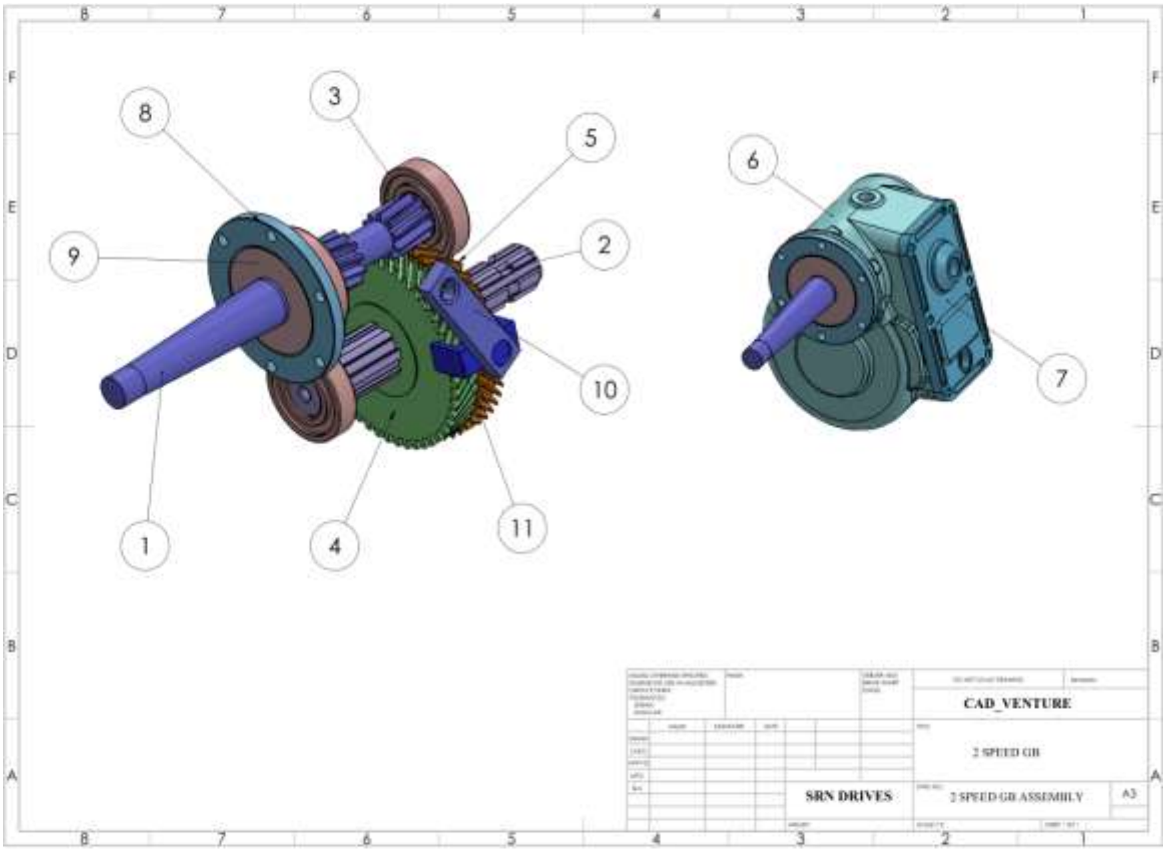
$$D \cong 40\ mm$$

$$d = 34.4$$

$$d \cong 35mm$$

So Minimum diameter of shaft we take Ø35mm.

VI. DRAWINGS



SR. NO.	PART NAME	QTY	MTL	REMARK
1	OUTPUT SHAFT	1	C45	
2	INPUT SHAFT	1	C45	
3	BEARING	4	STD	BEARING NO. 207
4	DRIVE GEAR 1	1	C45	
5	DRIVE GEAR 2	1	C45	
6	CASING	1	CAST IRON	

7	CASING COVER	1	CAST IRON	
8	CIRCULAR PLATE	1	MS	
9	OIL SEAL HOUSING	1	STD	ID-Ø35, OD-Ø80, 20THK
10	SHIFTING LINKAGE	1	MS	
11	GEAR SHIFTER	1	C45	
12	OIL SEAL HOUSING	1	STD	ID-Ø35, OD-Ø72, 20THK
13	M8X30	6	STD	

VII. TESTING RESULTS

SR NO	INPUT POWER(HP)	INPUT RPM	RATIO	OUTPUT RPM	OBSERVATION
1	25	1000	1:04	4000	SMOTH WORKING
2			1:05	5000	NOISE
3	22	850	1:04	3400	SMOTH WORKING
4			1:05	4250	NOISE
5	20	800	1:04	3200	SMOTH WORKING
6			1:05	4000	SMOTH WORKING
7	18	750	1:04	3000	SMOTH WORKING
8			1:05	3750	SMOTH WORKING

VIII. CONCLUSION

Today's Gearboxes in All Terrain Vehicles occupy more space, heavy and have limited life based on the operation. Operating these Gearboxes for continuously will produce heat that may affect the structural integrity. The high speed gear box designed to be coupled with engine PTO that can vary the transmission ratios. Increase RPM from Engine PTO to Fan Out-put. Based on the analytical calculations and Finite Element Analysis we can conclude that all the components are not bound to failure within the given working parameters. The Gearbox is lightweight, compact and has increased life over the conventional Manual Gearboxes.

IX. REFERENCES

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