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DESIGN AND MANUFACTURING OF 2-STAGE SPEED REDUCER FOR A BAJA ALL TERRAIN VEHICLE

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ABSTRACT

The gearbox designed and fabricated should be reliable, safe and cost effective. It should be able to transmit sufficient power and torque to generate the required traction at the wheels at a particular rpm. The vehicle should be able to complete all the dynamics events and the endurance race satisfactorily. The reduction ratio of the gearbox was finalized after considering various road resistances, maximum acceleration and grad ability performance. Standard procedures have been adopted for design of gears, shafts and casing of the gearbox.

Bearing selection is also done according to standards specified. The analytical calculations are evaluated using Hypermesh for finite element analysis. Machining of components is done after exploring various methods and further heat treated to improve their wear strength. After the completion of the ATV prototype, the 2-stage speed reducer has undergone rigorous testing to check its durability.

Key words: Design, Speed Reducer, Gearbox, Baja, Transmission, Traction, Driving resistances, Reduction ratio, Gears, Shaft, Casing, Bearing, Torque.

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1. INTRODUCTION

The main goal when developing a vehicle transmission is to convert the power from the engine into vehicle traction as efficiently as possible, over a wide range of road speeds. This has to be done ensuring a good compromise between the number of speeds, climbing performance, acceleration and fuel consumption of the vehicle. Technical and technological advances have to be taken into account, as do operational reliability and adequate service life. [1]

Gisbert Lechner, Harald Naunheimer (1999) [1] have given full account of the development process for automotive transmission, detailed study of driving resistances, various characteristics for transmission variables and drivetrain power losses. Selection of optimum gearbox ratio considering maximum acceleration and gradability is also analyzed.

Budynas, Nisbett (2008) [3] have explained the basics of machine design, including design process, engineering mechanics and materials and failure prevention under static and variable loading conditions. Analysis of spur gears to resist bending failure of teeth as well as pitting failure of tooth surfaces using AGMA standard 2001: D04 is done.

Reza N. Zajar (2013) [2] has explained vehicle dynamics concepts in detail. Effect of CG on vehicle dynamics as well forward vehicle dynamics, tire dynamics, and driveline dynamics. Forward dynamics refers to weight transfer, accelerating, braking, engine performance, and gear ratio design.

Tushar N. Khobragade, P. Priadarshani (2008) [7] have studied static analysis of gearbox casing using FEA softwares Hypermesh/Optistruct and Ansys 11.0. The optimum design is achieved using iterative design and analysis procedure. The various loading and boundary conditions are given for static analysis of gearbox casing**.**

2. CALCULATING GEARBOX RATIO ACCORDING TO TRACTION REQUIREMENT

2.1. Assumptions

Weight Distribution: 40% - Front; 60% - Rear

Total Weight of the Vehicle (with driver): 220kg

Static coefficient of friction: 0.85

2.2. Driving Resistance

The anticipated driving resistance is an important variable when designing vehicle transmission. [1] Driving resistance is made up of:

- Wheel resistance or Rolling resistance F_R
- Air resistance F_L
- Gradient resistance F_{st}
- Acceleration resistance F_a

2.2.1. Rolling Resistance F_R:

Wheel resistance comprises the resisting forces acting on the rolling wheel. It is made up of rolling resistance, road surface resistance and slip resistance. [1]

$$F_R = f_R \times G_R \qquad f_R = \frac{e}{r_{dyn}} \tag{1}$$

Where, f_R = rolling resistance coefficient,

G_R = wheel load,

e = eccentricity,

r_{dyn} = dynamic wheel radius.

Table 1 Values of Rolling Resistances f_R. [1]

Road surface	f_R
Very good earth tracks	0.045
Bad earth tracks	0.16
Loose sand	0.15-0.13
Smooth tarmac road	0.01
Bad worn road surface	0.035
Smooth Concrete Road	0.011

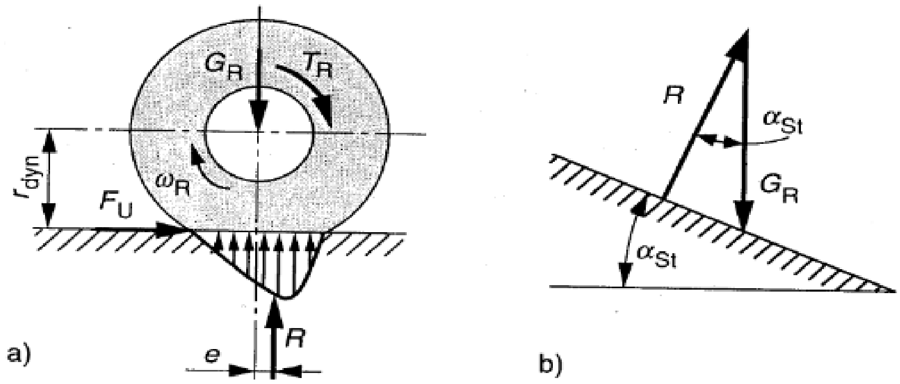


Figure 1 Rolling Resistance [1]

2.2.2 Air Resistance F_L:

The air resistance is made up of the pressure drag including induced drag (turbulences induced by differences in pressures), surface resistance and internal (through-flow) resistance. [1]

$$F_L = \frac{1}{2} \rho_L \, c_W \, A \, v^2 \tag{2}$$

Where, ρ_L = air density= 1.199kg/m^3,

c_W = coefficient of drag=1.2,

A = projected frontal area in m^2,

v = speed of vehicle.

2.2.4. Acceleration Resistance F_a:

In addition to the driving resistance occurring in steady state motion (v = constant), inertial forces also occur during acceleration and braking. The total mass of the vehicle m_F and the inertial mass of the rotating parts of the drive acceleration or brakes are the factors influencing the resistance to acceleration [1]

$$F_a = \lambda \, m_F \, a \tag{3}$$

λ = Rotational inertia coefficient which represents proportion of total mass that is rotary

2.2.3. Gradient Resistance F_{st}:

The gradient resistance or downhill force relates to the slope descending force and is calculated from the weight acting at the centre of gravity. [1]

$$F_{St} = m_F \, g \, \sin(\alpha_{St}) \tag{4}$$

Where, m_F = mass of vehicle,

g = acceleration due to gravity (m/s^2),

α_{St} = grade angle =45deg

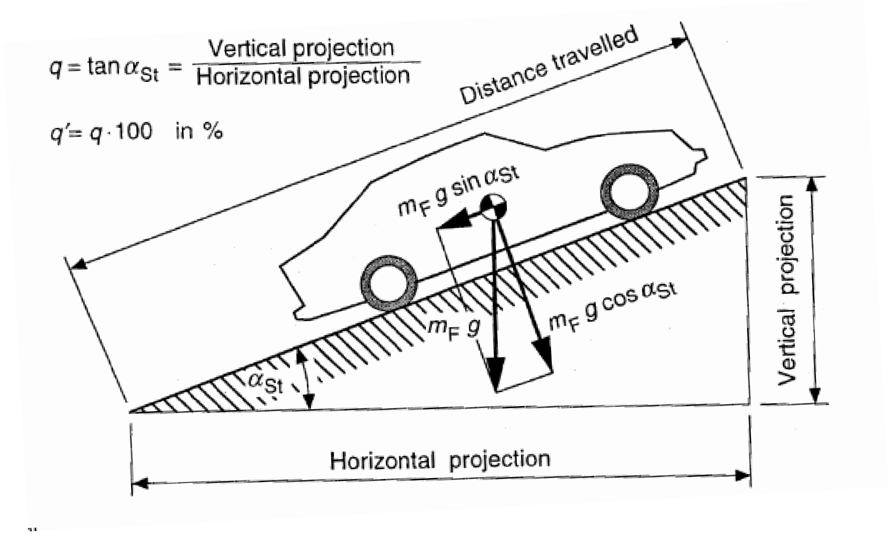


Figure 2 Free Body Diagram of a Vehicle on an Inclined Plane [1]

2.2.5. Total Driving Resistance

$$F = F_R + F_L + F_{ST} + F_a \tag{5}$$

$$F = m_F\, g\, (f_R \cos(\alpha_{St}) + \sin(\alpha_{St})) + \frac{1}{2}\rho_L\, c_W\, A\, v^2 + \lambda\, m_F\, a \tag{6}$$

2.3. Traction Available

The traction available at wheels can be found out from engine characteristics. [1]

$$\text{Traction} = \frac{T_M\, G_r\, \eta_{all}}{r_{dyn}} \tag{7}$$

Where T_M is the max torque=18.9N-m

G_r is the overall gear ratio

η_{all} is the total transmission efficiency=0.85(assumed)

r_{dyn} is the dynamic radius of the wheel.(22x7-10)=0.276m

2.4. Gearbox Reduction Ratio

2.4.1. Selecting the Largest Power-Train Ratio

At the drive wheels, a balance must be struck between the required acceleration and gradient road surface, trailer load. The gearbox reduction ratio is calculated by equating traction available at wheels to the traction required. [1]

Maximum traction available = Maximum traction required

$$\frac{T_M\, G_r\, \eta_{all}}{r_{dyn}} = m_F\, g\, (f_R \cos(\alpha_{St}) + \sin(\alpha_{St})) + \frac{1}{2}\rho_L\, c_W\, A\, v^2 + \lambda\, m_F\, a \tag{8}$$

3. DESIGN OF TWO STAGE SPEED REDUCER

The overall reduction ratio obtained in previous step is split into two stages using the formula:

$$i_1 = 0.76 \times i_T^{0.65} \qquad i_2 = \frac{iT}{i1} \tag{9}$$

3.1. Design of Gears According to AGMA 2001-D04

The intent of the AGMA strength rating formula is to determine the load which can be transmitted for the design life of the gear drive without causing root fillet cracking.

The module of gears for each stage is obtained from the following formula [6]

$$d = \frac{2C}{G+1}\,;\ m = \frac{d}{z} \tag{10}$$

C is Center distance between input to output gear and G is the overall gear ratio. The gears are designed using AGMA 2001-D04 standard procedure considering factor of safety for spur gear bending and wear with stress cycles 10^7 and reliability of 0.99. The gears were manufactured using hobbing cutter with pressure angle of 20^0 with minimum number of teeth on pinion as 18.

3.1.1. Bending Strength Equation

The fundamental formula for bending stress number in a gear tooth is: [3]

$$\sigma = w_t{*}k_o\, k_v\, k_s{*}\frac{P_d}{F}{*}\frac{Km{*}Kb}{J} \tag{11}$$

where, W_t = tangential transmitted load, lbf (N),

K_o = overload factor,

K_v = dynamic factor,

K_s = size factor,

P_d = transverse diametral pitch,

F = face width of the narrower member, in (mm),

K_m = load-distribution factor,

K_b = rim-thickness factor,

J = geometry factor for bending strength (which includes root fillet stress-concentration factor K_f).

Gear bending endurance strength: [3]

$$\sigma_{all} = \left(\frac{s_t}{s_f}\right){*}\frac{Y_N}{K_{T*K_R}} \tag{12}$$

Where, K_T = Temperature factor,

K_R = Reliability factor,

Y_N = Stress cycle factor.

Bending Factor of Safety: [3]

$$S_f = \frac{\frac{s_t Y_N}{K_{T*}K_R}}{\sigma} \tag{13}$$

3.1.2. Wear Strength Equation: [3]

$$\sigma_c = C_p \left(W_t K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I}\right)^{1/2} \tag{14}$$

Where, σ_c = contact stress number, lb/in^2,

C_p = elastic coefficient, [lb/in^2]$^{0.5}$,

W_t = transmitted tangential load lb,

K_o = is overload factor,

K_v = is dynamic factor,

K_s = size factor,

K_m = load distribution factor,

C_f = surface condition factor for pitting resistance,

F = net face width of narrowest member, inch,

I = geometry factor for pitting resistance,

d = operating pitch diameter of pinion, inch.

Gear contact endurance strength: [3]

$$\sigma_{c,all} = \frac{S_c Z_n C_h}{S_t K_t K_R} \tag{15}$$

Where, S_c = allowable contact stress number lb/in^2,

Z_N = stress cycle factor for pitting resistance,

C_H = hardness ratio factor for pitting resistance,

St = safety factor for pitting,

K_T = temperature factor,

K_R = reliability factor.

Wear factor of safety: [3]

$$S_H = \frac{S_{c*} Z_{n*} C_H}{\frac{K_{T*}K_R}{\sigma c}} \tag{16}$$

4. DESIGN OF SHAFT

The forces acting on the gears were calculated using the free body diagram. These forces are then considered for the design of shafts and the selection of bearings. [4]

4.1. ASME Code for Shaft Design

$$\tau_{max} = 0.30\, S_{yt} \tag{17}$$

$\tau_{max} = 0.18\ S_{ut}$ (18)

If keyways are present the above values are reduced by 25 per cent.

$$\tau_{max} = \frac{16}{\pi d^3}\sqrt{(k_b M_b)^2 + (k_t M_t)^2} \quad (19)$$

$$\sigma_1 = \frac{16}{\pi d^3}\left[k_b M_b + \sqrt{(k_b M_b)^2 + (k_t M_t)^2}\right] \quad (20)$$

The diameters of the shafts were calculated using the above formulae, the shear force and bending moment diagrams were drawn considering the forces acting on the gears and the self-weight of the gears. [4]

5. BEARING SELECTION:

5.1. Equivalent Dynamic Load

The equivalent dynamic load is defined as the constant radial load in radial bearings (or thrust load in thrust bearings), which if applied to the bearing would give same life as that which the bearing will attain under actual condition of forces. [8]

The expression for the equivalent dynamic load is written as,

$$P = XVF_r + YF_a \quad (21)$$

For Spur gears axial loading is zero, $P=F_r$

Where, P = equivalent dynamic load(N)

F_r = radial load(N)

V = race rotation factor=1 (inner race rotating)

X and Y = radial and thrust load factors=1

5.2. Load-Life Relationship

The life expressed in operating hours using [8]

$$L_{h10} = \frac{(60 * n * L_{10})}{10^6} \quad (22)$$

The basic rating life of a bearing in accordance with ISO 281 is

$$L_{h10} = \left(\frac{C}{P}\right)^a \quad (23)$$

Where, L_{10} = basic rating life (at 90% reliability) [million revolutions]

L_{h10} = basic rating life (at 90% reliability) [operating hours]

C = basic dynamic load rating [kN]

P = equivalent dynamic bearing load [kN]

n = rotational speed [r/min]

a = exponent of the life equation

for ball bearings, a = 3

for roller bearings, a = 10/3

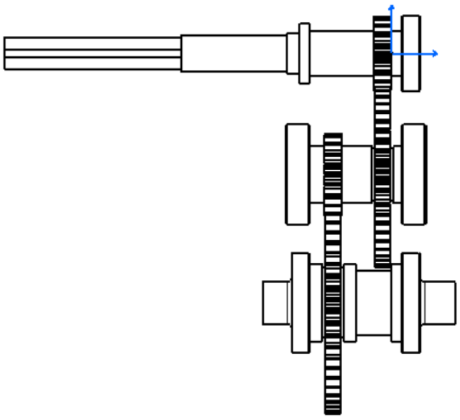


Figure 3 Layout of 2 stage speed reducer

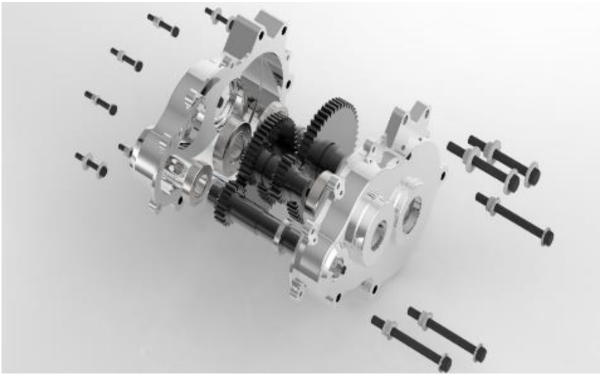


Figure 4 Exploded view of assembly

6. FINITE ELEMENT ANALYSIS

Finite element analysis of gearbox shaft, gears and casing is done using Hypermesh software. Tangential load is applied on single gear tooth face of integrated pinion shaft and gears. The static structural analysis was done considering standstill position and sudden application of load. For analysis of casing the reactions obtained from freebody diagram at bearing position were applied and the gearbox mounting points were fixed. Load values and stress induced are mentioned in following tables. [7]

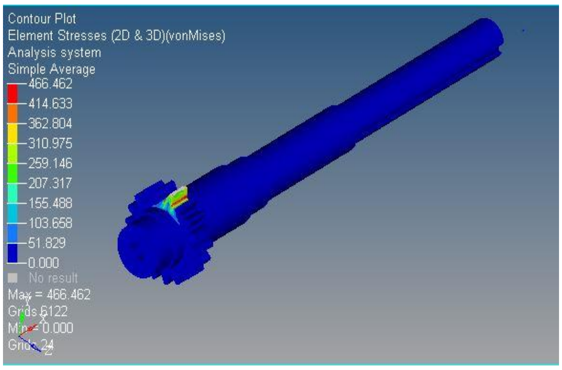


Figure 5

Table 2 FEA of Input Shaft

INPUT SHAFT	
Static structural analysis	
S_{yt}	880 N/mm^2
Maximum stress	466.462 N/mm^2
FOS	1.8

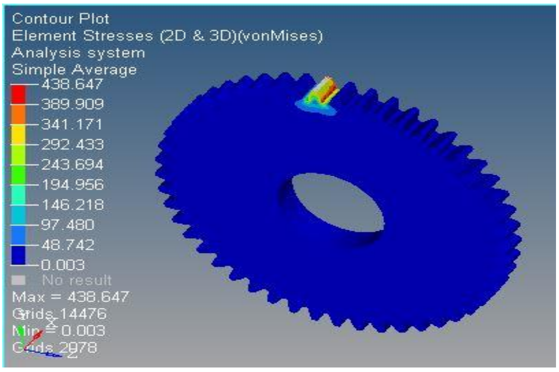


Figure 5

Table 3 FEA of First Stage Gear

FIRST STAGE GEAR	
Static Structural Analysis	
S_{yt}	880 N/mm^2
Maximum stress	438.647 N/mm^2
FOS	2

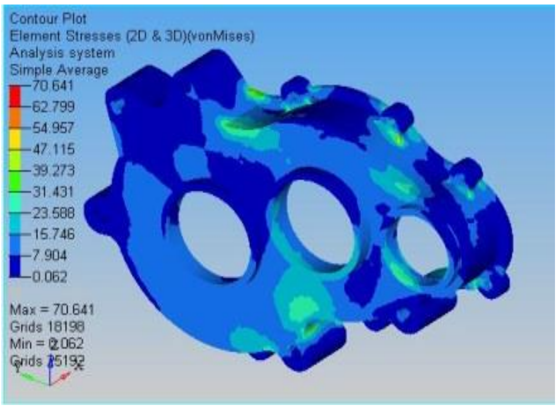


Figure 6

Table 4 FEA of Gearbox Casing

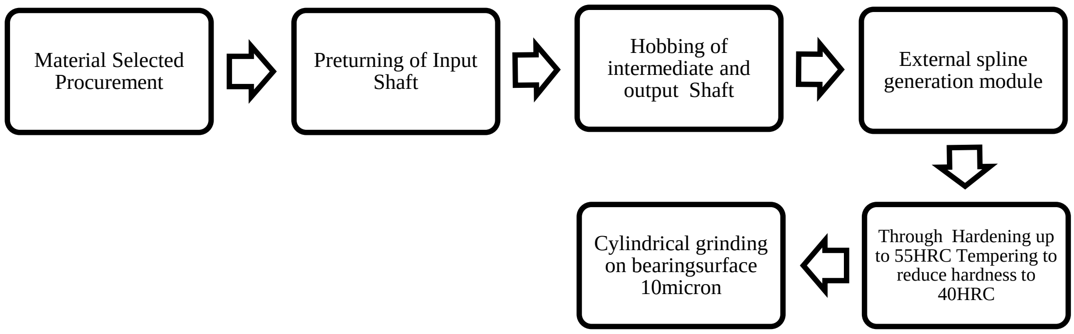
GEARBOX CASING	
Static Structural Analysis	
S_{yt}	270N/mm^2
Maximum stress	70.641 N/mm^2
FOS	4.28

7. MANUFACTURING OF 2-STAGE SPEED REDUCER

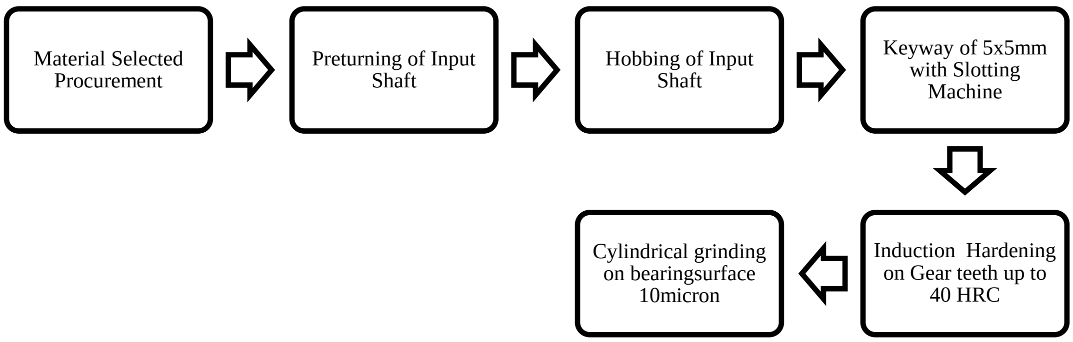
In order for gears to achieve their intended performance, durability and reliability, the selection of a suitable gear material is very important. High load capacity requires a tough, hard material that is difficult to machine; whereas high precision favors materials that are easy to machine and therefore have lower strength and hardness ratings. Gears are made of variety of materials depending on the requirement of the machine. They are made of plastic, steel, wood, cast iron, aluminum, brass, powdered metal, magnetic alloys and many others. The final selection should be based upon an understanding of material properties and application requirements. Materials such as SAE8620, 20MnCr5, 16MnCr5, Nylon, Aluminum, etc. used for automobile gears. Taking into account hardness and machinability of material selected gear tooth were obtained using hobbing cutter of specified module.To improve wear resistance case hardening was done to maintain case hardness of 40HRC and 3mm depth. Bearing Tolerances were achieved by grinding on shafts. [5]

7.1. The Manufacturing Process Flow For Various Components Of Gearbox Is As Follows

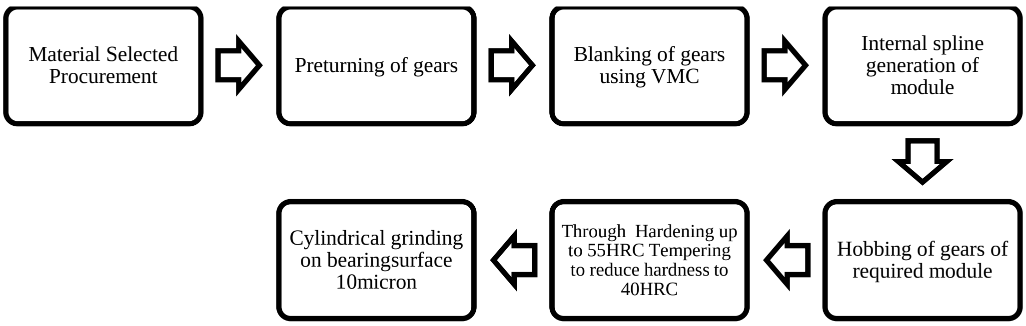
7.1.1. Input shaft



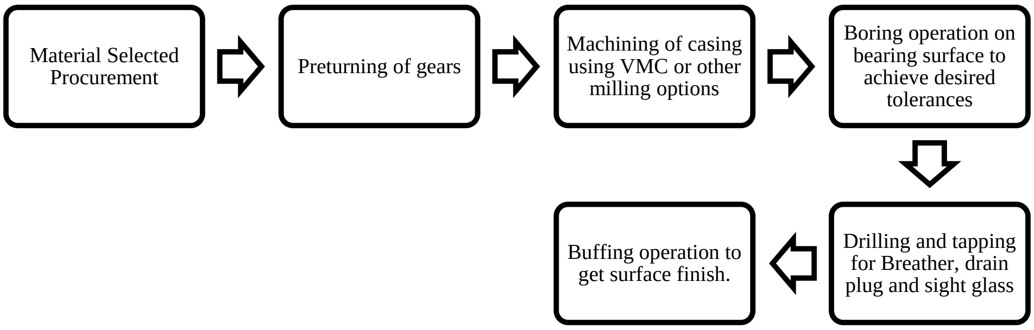
7.1.2. Intermediate and output shaft



7.1.3. Gears



7.1.4. Casing



7.2. Assembly of Gearbox

The assembly of the gearbox was done on the manual press machine available in the workshop. The load was applied gradually using the hand wheel. Initially all the bearings were fitted in both the casings. The shafts with gears mounted on them were press fitted in the bearings on one side. The other side of the casing was then fitted on. A rubber gasket with a gasket maker was used to seal the gearbox and prevent any leakage. Both the sides of the casing were then fixed with the help of Allen bolts of size M6.

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