Modelling and Analysis of a Two-Stage Gearbox

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Abstract The world today as we know it is advancing in many factors as per manufacturing and design. One of the key factors during this process is to reduce weight and volume without changing the performance of the required product. This can be achieved by the selection of materials and manufacturing processes. In this study, a two-stage reduction gearbox was designed and analysed for its use on an all-terrain vehicle (ATV). Computer-aided design (CAD) and computer-aided engineering (CAE) software packages are used for designing and analysis of gears and shafts. A similar approach was adopted for designing of gears and shafts, whereas sequential approach thereafter for the design of casing geometry. The product was designed in accordance with the American Gear Manufacturers Association (AGMA) standards and is meant for off-road application.

Keywords Powertrain · Gears · All-terrain vehicle (ATV) · GearTraxPRO · SolidWorks · ANSYS

1 Introduction

For the all-terrain vehicles (ATV) that are configured for an automatic transmission drive using a continuously variable transmission (CVT), a gearbox has to be coupled with the CVT to get desired speed and torque output. Since the CVT is providing variable transmission ratios, a forward–neutral–reverse (FNR), two-step reduction gearbox can be used for this purpose. The work proposed here is to design a two-stage gearbox consisting of both helical and spur gears. This gearbox is designed for an ATV, made for the BAJA competitions. Here weight reduction is focused upon as the main goal to achieve maximum acceleration and also to get the centre of gravity as low as possible. Thus, factors such as weight, strength and size are optimized

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Table 1 Input parameters

Input parameters	Sign	Value
Coefficient of friction	fr	0.16
Mass of vehicle + driver	mf	225 kg
Gradeability angle		30 degree
Air Density	PI	1.199 kg/m^3
Coefficient of drag	Cw	0.5
Velocity of car	V = Vf	25 km/hr
Acceleration of car	a	2.83 m/s^2
Efficiency of CVT	E1	0.88

to achieve the best possible gearbox. The analysis operations conducted during the research were done according to Navneet et al. [1], in their study in 'Analysis and Simulation of Gearless Transmission Mechanism' and also by Timir Patel et al. [2], in their study 'Design and Analysis of an Epicyclic Gearbox for an Electric Drivetrain'.

2 Vehicle Parameters

Engine power = 10 hp = 7.456 kW @ Max RPM = 3800 rpm
Engine torque = 19.2 Nm @ 2700 rpm
Transmission type = CVT + Gearbox drive
Gearbox type = forward spool two-stage helical gearbox
Overall reduction = 9
Stage 1 reduction = 3, stage 2 reduction = 3
Gear material = 20 MnCr5, Yield strength: 350–550 N/mm^2, Tensile strength: 650–880 N/mm^2.

The Table 1 shows the parameters taken into account of an ATV for which the gearbox is being designed.

3 Torque Requirements

To determine the required torque, the most important factor that turns in is the total resistance experienced by the vehicle. The driving resistance is an important variable taken into consideration while designing transmission systems. Driving resistance is made up of mainly four resistances:

- Wheel Resistance
- Air Resistance
- Gear Resistance

Table 2 Resistance calculations

Engine RPM	Total reduction	Engine (N-m)	Torque (N-m)	Traction (N)	Velocity (m/s)	Gradient	Acceleration (m/s^2)	Total resistance (N)
1800	27	18	487	1498	5.8	32	1.9	932
2000	26.4	18.7	494	1520	6.6	32.1	1.9	943
2200	25.9	18.8	487	1499	7.4	32.1	19	932
2400	24.7	18.9	467	1438	8.5	29.1	2	901
2600	23.4	19	445	1368	9.7	25.8	1.7	867
2800	22.5	19.1	430	1322	10.9	23.6	1.6	844
3000	19.8	18.9	374	1151	15.3	17.4	1.3	759
3200	15.3	18.8	288	887	23.6	12.7	0.9	626
3400	12.6	18.3	231	711	32.5	4.2	0.6	537
3600	11.7	18.2	213	655	40.8	1.5	0.6	510
3800	6.75	17.6	119	366	49.3	1	0.5	502

- Acceleration Resistance.

The Table 2 comprises of the resistance values experienced by the ATV at different RPMs. As the engine increases its RPM, the resistance value changes [3–5].

Taking reference from the Table 2, we determined what is the exact reduction required for our BAJA vehicle [4, 6, 7].

4 Design of Gearbox

Design calculations of various parts of gearbox are as follows [4, 6–9].

4.1 Design of Gears

Lewis bending equation is.

$$\sigma = \frac{K_v W^t}{F m Y} \tag{1}$$

where

σ is the bending stress on gear teeth (not considering dynamic loading)
F is the face width
m is the module
W^t is the tangential load

Y is the Lewis form factor
K_v is the velocity factor

AGMA bending stress equation is

$$\sigma = \frac{K_v W^t K_s K_o K_H K_B}{b m_t Y_J} \tag{2}$$

where

σ is the bending stress-induced
K_v is the dynamic factor
W^t is the tangential transmitted load
b is the face width of the narrower member
K_o is the overload factor
K_s is the size factor
K_B is the rim thickness factor
K_H is the load distribution factor
Y_J is the geometry factor for bending strength
m_t is the transverse module

AGMA pitting stress equation is

$$\sigma_c = Z_E \sqrt{W^t K_o K_v K_s \frac{K_H}{d_{wl} b} \frac{Z_R}{Z_I}} \tag{3}$$

where

σ_C is the pitting stress-induced
Z_E is the elastic coefficient
d_{wl} is the pitch diameter of the pinion
Z_I is the geometry factor of the pitting resistance
Z_R is the surface condition factor

AGMA allowable bending stress equation is

$$\sigma_{\text{all}} = \frac{S_t Y_N}{S_F Y_\theta Y_Z} \tag{4}$$

where

S_t is the allowable bending stress
Y_N is the stress cycle factor for bending stress
Y_Z is the reliability factor
Y_θ is the temperature factor
S_F is the AGMA factor of safety.

Table 3 For reduction stage 1 pinion and gear

Parameter	Value
Gear type	Spur
Helix angle	Zero
Pressure angle	20 degree
No. of teeth on pinion	18
No. of teeth on gear	54
Max. RPM	1267
Module	2
Face width	15 mm

AGMA allowable pitting stress equation is

$$\sigma_{c,\text{all}} = \frac{S_c Z_N Z_W}{S_H Y_\theta Y_Z} \tag{5}$$

where

$\sigma_{c,\text{all}}$ is the allowable pitting stress
Z_N is the stress cycle factor
S_c is the allowable contact stress
S_H is the AGMA factor of safety
Z_W is the hardness ratio factors for pitting resistance.

To avoid failure in gearing, the following condition must be satisfied:
$\boldsymbol{\sigma}$ (both of Lewis bending equation and AGMA strength equation) $\ll \boldsymbol{\sigma}_{\textbf{all}}$, $\boldsymbol{\sigma}_{\boldsymbol{c}} \ll \boldsymbol{\sigma}_{\boldsymbol{c},\textbf{ all}}$.

Solving the above equations by inserting the specified vehicle parameters, the above conditions are satisfied. Therefore, the design is safe.

The values obtained by solving the equations in GearTraxPRO (Camnetics Suite) are shown in Table 3 for stage 1 pinion and gear and Table 4 for stage 2 pinion and gear.

The following tables show the parameters taken as for both pinion gears for design [3] (Table 5).

4.2 Design of Shafts

Parameters:

Material: AISI/SAE 4340
UTS: 925 MPa
Yield strength: 680 MPa
Allowable shear stress for shaft: 462.5 MPa

Table 4 For reduction stage 2 pinion and gear

Parameter	Value
Gear type	Helical
Helix angle	20 degree
Pressure angle	20 degree
No. of teeth on pinion	18
No. of teeth on gear	54
Max. RPM	423
Module	2
Face width	25 mm

Table 5 Factors for spur and helical gears

Factors	Symbols	Value
Overload factor	Ko	1
Dynamic effect factor	Kv	1.178
Size factor	Ks	1.0029
Load distribution factor	Km	1.149
Rim thickness factor	Kb	1.139
Bending strength factor	Yj	0.317
Geometry factor	I	0.12
Temperature factor	Kt	1
Reliability factor	Kr	1
Hardness factor	Ch	1

Torque transmitted by pinion:

$$T = \frac{P \times 60}{2\pi N_P} \tag{6}$$

Equivalent no. of teeth:

$$T_E = \frac{T_P}{\cos^3 \alpha} \tag{7}$$

Tooth factor:

$$y' = 0.154 - \frac{0.912}{T_E} \tag{8}$$

Tangential tooth load:

$$W_T = \frac{2T}{m \times T_P} \tag{9}$$

$$\frac{1}{n} = \frac{16}{\pi d^3}\left\{\frac{1}{S_e}\left[4\left(K_f M_a\right)^2 + 3\left(K_{fs} T_a\right)^2\right]^{1/2} + \frac{1}{S_{ut}}\left[4\left(K_f M_m\right)^2 + 3\left(K_{fs} T_m\right)^2\right]^{1/2}\right\} \tag{10}$$

where

Factor of safety (FOS) $= n$
Diameter of the shaft $= d$
Endurance limit at critical location $S_e = k_a\ k_b\ k_c\ k_d\ k_e\ k_f\ S'_e$
k_a is the surface condition modification factor
k_b is the size modification factor
k_c is the load modification factor
k_d is the temperature modification factor
k_e is the reliability factor
k_f is the miscellaneous effects modification factor
S'_e = rotary beam test specimen endurance limit
Stress concentration factor for Bending, $K_f = 1 + q(K_t - 1)$.
Stress concentration factor for torsion, $K_{fs} = 1 + \text{qshear}(K_t - 1)$.
q is the notch sensitivity
M_e is the midrange bending moment
M_a is the maximum bending moment
T_a is the alternating torque
T_e is the maximum torque
S_{ut} is the tensile strength.

By drawing shear force and bending moment diagrams, we are able to find the values of M_a and T_e.

Resultant bending moment: $Ma = \sqrt{(M_1)^2 + (M_2)^2}$.

Putting the values in Eq. 10, we get $n = 1.37$.

From the above equation, we get $d_P = 17$ mm (standardized according to the availability of support bearing sizes and oil seals).

The value of principal shear stress is equated which is significantly less than the permissible shear stress of the shaft material. Hence, the design is safe.

Repeating the above calculations for the intermediate shaft and the output shaft, we get shaft diameters as **22 mm** and **27 mm**.

4.3 Design of Casing

The casing design was aimed at achieving the lowest possible volume, while constraining the shafts with gears mounted on them. The geometry is kept simple for lowering the machining cost.

5 CAD Modelling of Gears, Shafts and Casing

SolidWorks 2018–19 modelling software was used to design the various components of the gearbox. Based on the input design parameters from GearTraxPRO software, modelling was done in SolidWorks. The images of the components are shown in the figures below. Gear material was selected as 20MnCr5 [10]. Figures 1 and 2 show the gears of the first stage reduction of the gearbox. Figures 3 and 4 show the gears of the second stage reduction of the gearbox. Figures 5, 6 and 7 show the CAD model of the input shaft, intermediate shaft and output shaft, respectively. Figures 1, 2, 3, 4, 5, 6 and 7 are the gears and shafts designed as per parameters in SolidWorks [11, 12, 17].

Fig. 1 Spur pinion

Fig. 2 Spur gear

Fig. 3 Helical pinion

Fig. 4 Helical gear

Fig. 5 Input shaft

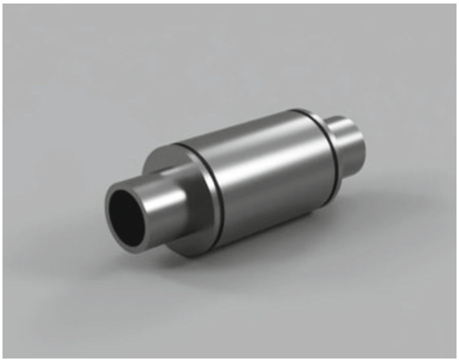
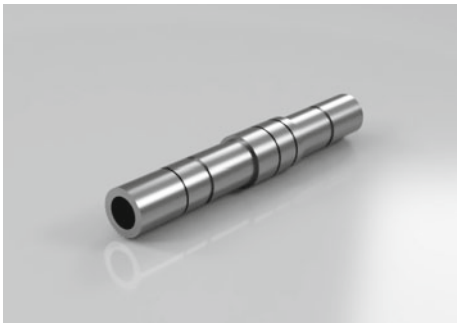


Fig. 6 Intermediate shaft

Fig. 7 Output shaft



The CVT is coupled with the input shaft with the help of the keyway in the input shaft. The casing design was optimized for minimum machining cost by reducing the number of contours that were made keeping in mind the weight of the gearbox. Following Figs. 8 and 9 show the final geometry including the left and right casing sides. Aluminium 6061-T6 was chosen as casing material [11].

The final assembly was done using SolidWorks, which included all the gears, shafts, bearings and the shifter. Figures 9 and 10 show the assembled view of the gearbox.

Fig. 8 Exploded view of complete gearbox

Fig. 9 Assembled view of gearbox without casing

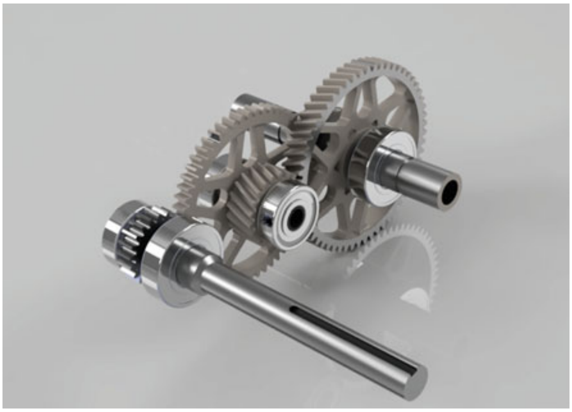


Fig. 10 Complete assembly of gearbox

6 Static Analysis

6.1 Gears

Maximum loading conditions were assumed for checking the gears for maximum safety. It was assumed that the worst loading condition on the gearbox would be when the vehicle is stuck and the engine is working at full power to get out of the obstacle. At this point of time, maximum torque would fall on the intermediate shaft gear 3. In the result, maximum stress-induced under full-loading condition was less than maximum allowable stress for the selected material [1]. The following pictures are of the analysis done on gears on ANSYS software [5, 12–14, 16].

Model (A4) > Static Structural (A5) > Solution (A6) > Total Deformation

Time [s]	Minimum [mm]	Maximum [mm]
1.	0.	5.897e-002

Gear teeth are subjected to both bending and wear. The section where it experiences the maximum stress is the root of the tooth. It is considered that there are three teeth in contact with the mating gear [12]. Tangential force (Ft) because of the torque which the gear is transmitting is applied on these three teeth. The total tangential force is distributed amongst the three teeth considering one tooth takes 50% of force and others take 25% each. The radial forces (Fr) generated on the gear are acting towards the centre of the gear [12, 13].

Magnitudes of the tangential and radial forces for the application in the analysis are taken from the GearTraxPRO output values in the design of the gears. The analysis result shows that the design is safe, even when the worst loading condition is considered. Figures 11, 12 and 13 show the dimensioning and force calculation outputs for reduction set 1 (for gear 1 and gear 2). The results were obtained by simulation in ANSYS software. Based on the results and analysis, we chose EN24 as our material for gear manufacturing [10] (Fig. 14).

6.2 Shafts

Shafts are mainly subjected to bending and torsion. It is considered that the shafts are subjected to maximum torsion at the location of gear through spline. Hence, the torque is applied at that position. Bearing portion where taken as the frictionless support and the fixed support will be the output of the shaft at both ends where the drive shaft through the joints will attach [5, 14, 15] (Figs. 15, 16 and 17).

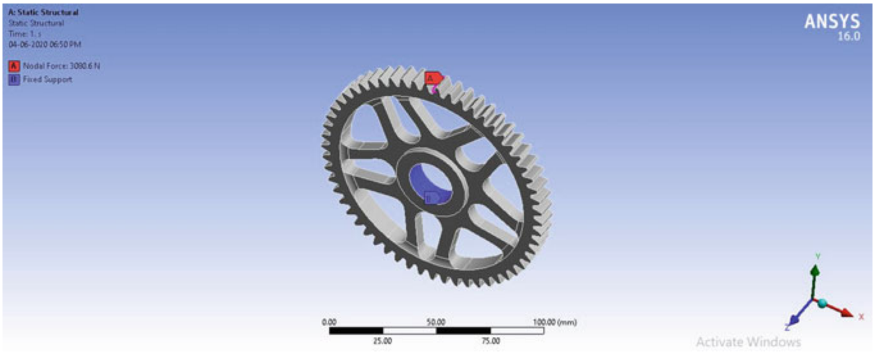


Fig. 11 Force model of gear

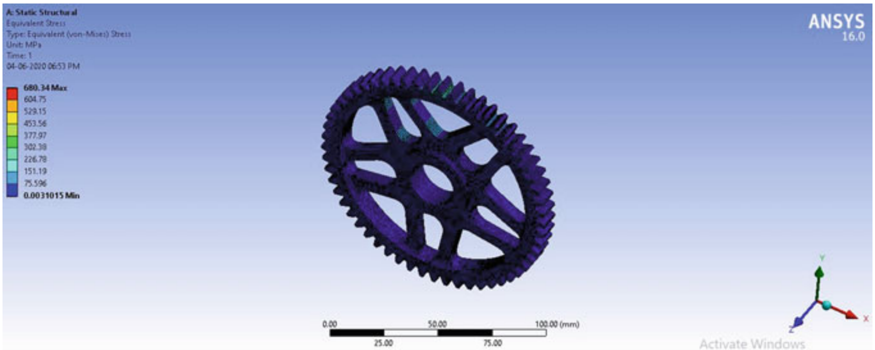


Fig. 12 Equivalent stress of gear

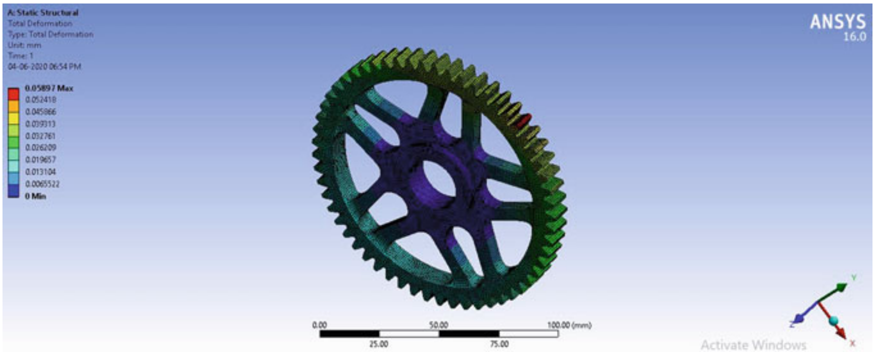


Fig. 13 Total deformation model

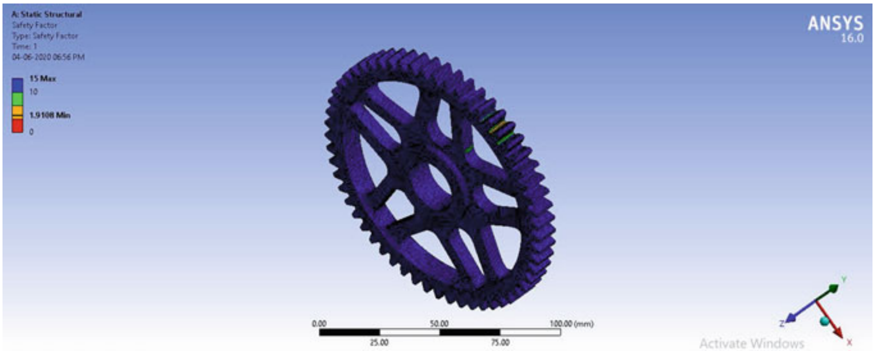


Fig. 14 Factor of safety of gear

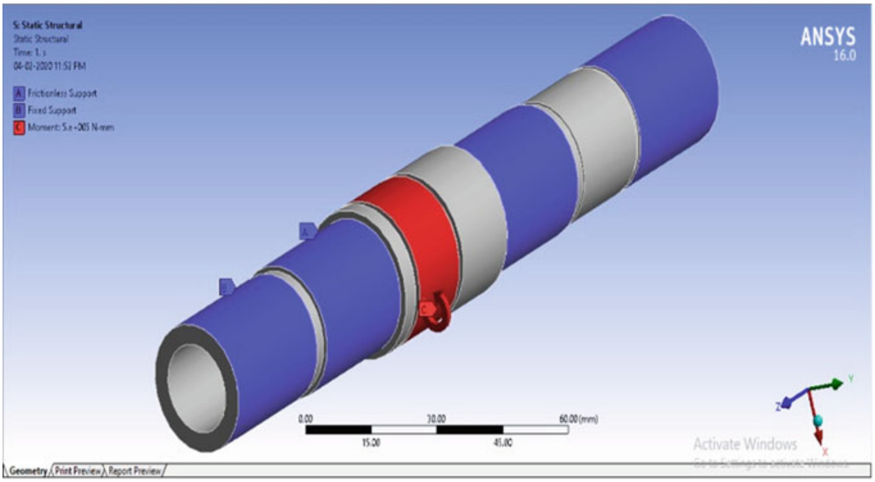


Fig. 15 Force model of output shaft

6.3 Casing

See Figs. 18, 19, 20 and 21.

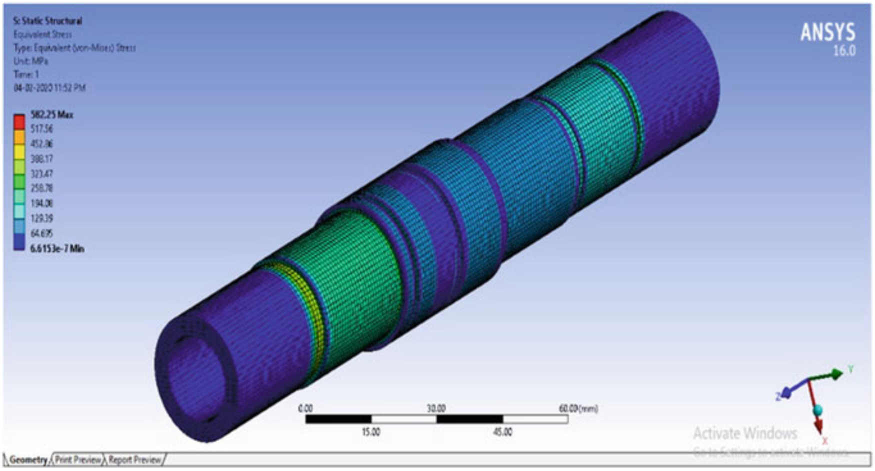


Fig. 16 Equivalent stress of output shaft

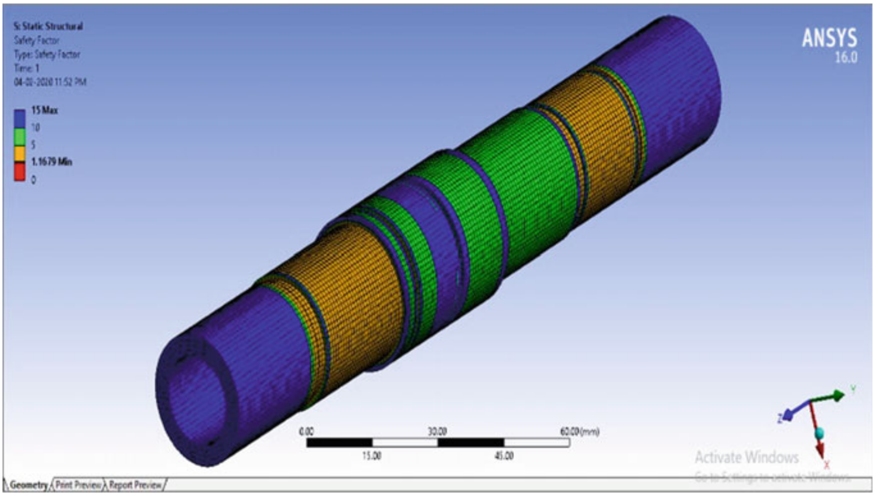


Fig. 17 Factor of safety of output shaft

7 Software Results for Gear Design (GearTRAXPRO)

The following pictures are screenshots of the GearTRAXPRO software while obtaining a SolidWorks part file of the gear. It also includes all parameters taken into consideration for designing the gear.

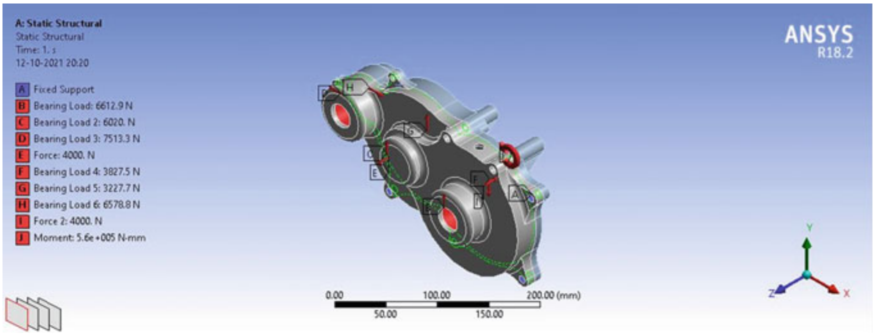


Fig. 18 Force model of casing

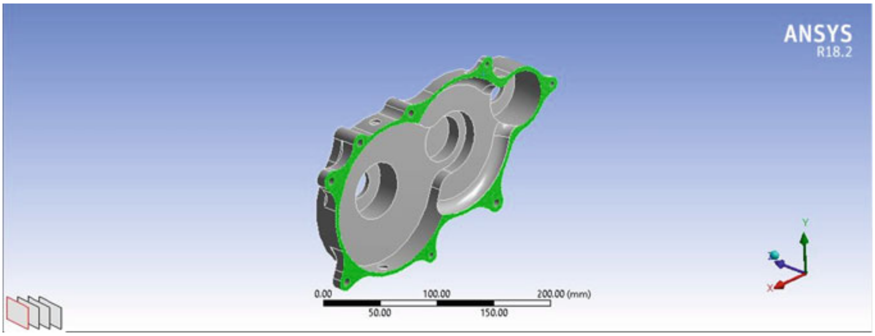


Fig. 19 Contact region

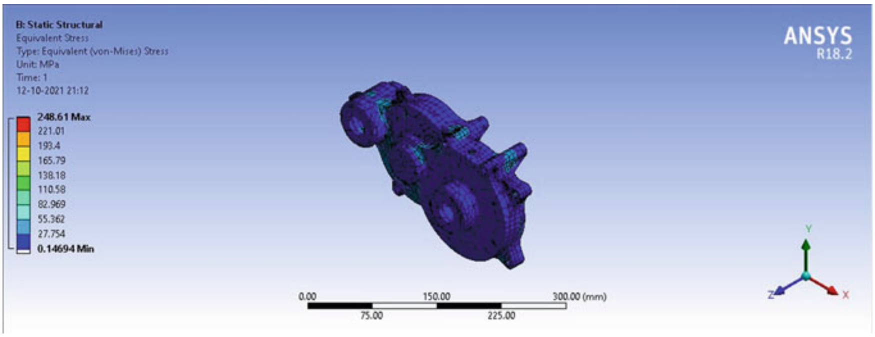


Fig. 20 Equivalent stress of casing

Figures 22 and 23 show the dimensioning and Figs. 24, 25, 26, 27, 28 and 29 force calculation outputs and other parameter calculations for reduction set 1 (for gear 1 and gear 2). The results were obtained by simulation in GearTraxPRO software.

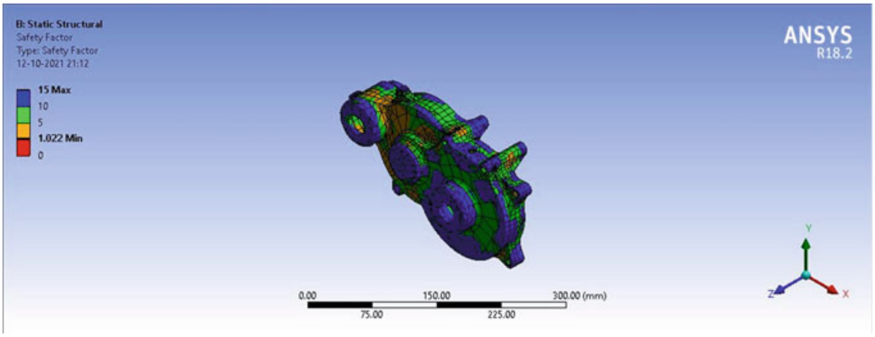


Fig. 21 Factor of safety of casing

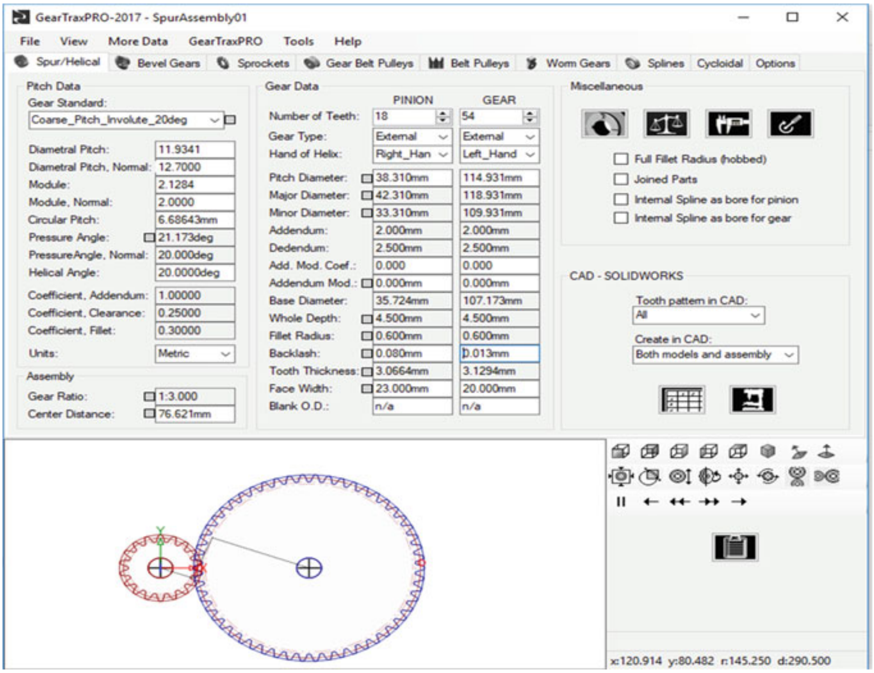


Fig. 22 Gear set 1 dimensioning

8 Conclusion

Comparison of existing alternative options in the market and this new design of the gearbox show that there is a significant reduction in weight of the gearbox. This is achieved while keeping the performance parameters as required by the potential

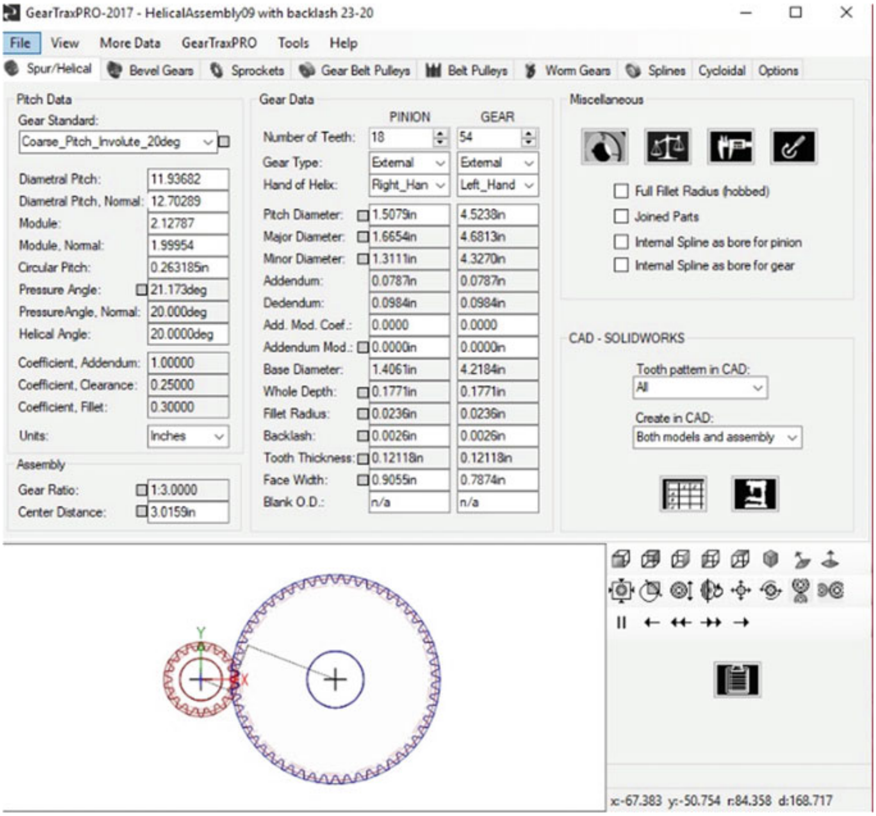


Fig. 23 Gear set 2 dimensioning

customers. The gearbox design was finalized in geometry and material (in accordance with AGMA standards) and was forwarded to manufacturers [3] for getting a prototype model for testing. Once manufactured, the model will be tested on a BAJA ATV, by replacing the old gearbox with this gearbox. An overall weight reduction of 9.5 kg is expected from the design, when compared with its alternative market option.

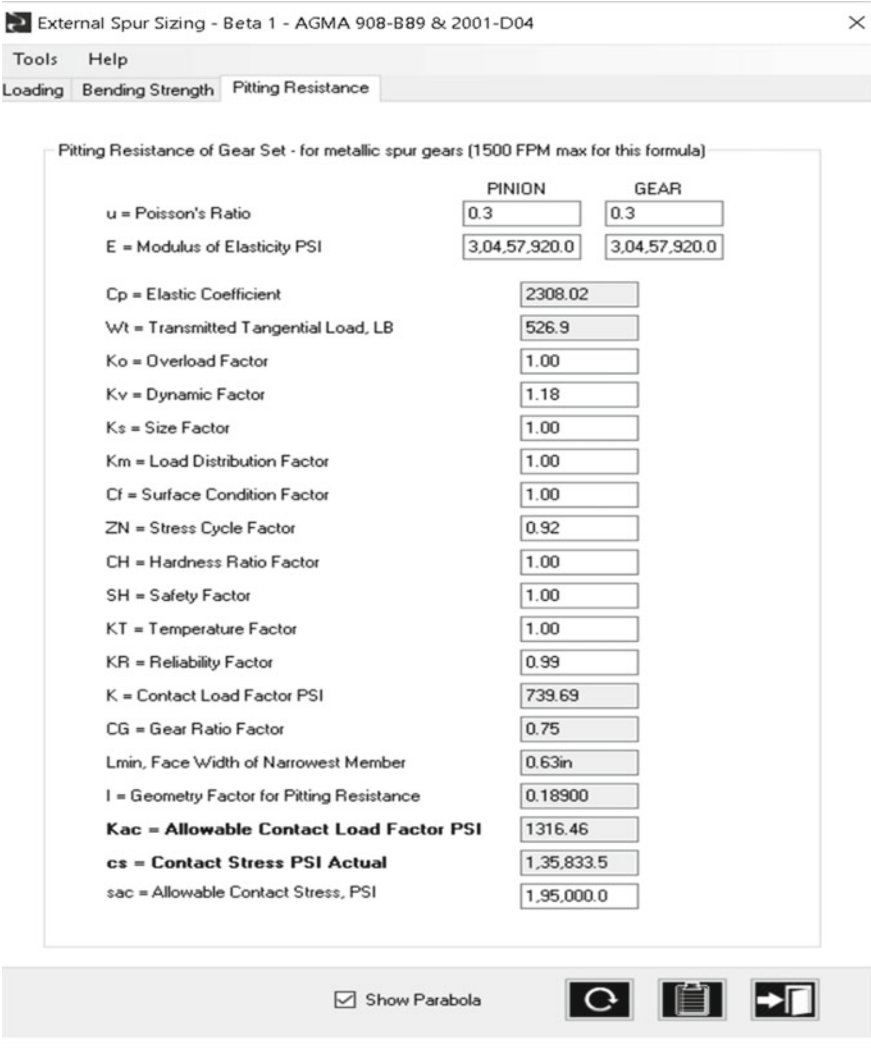


Fig. 24 Pitting resistance for gear set 1

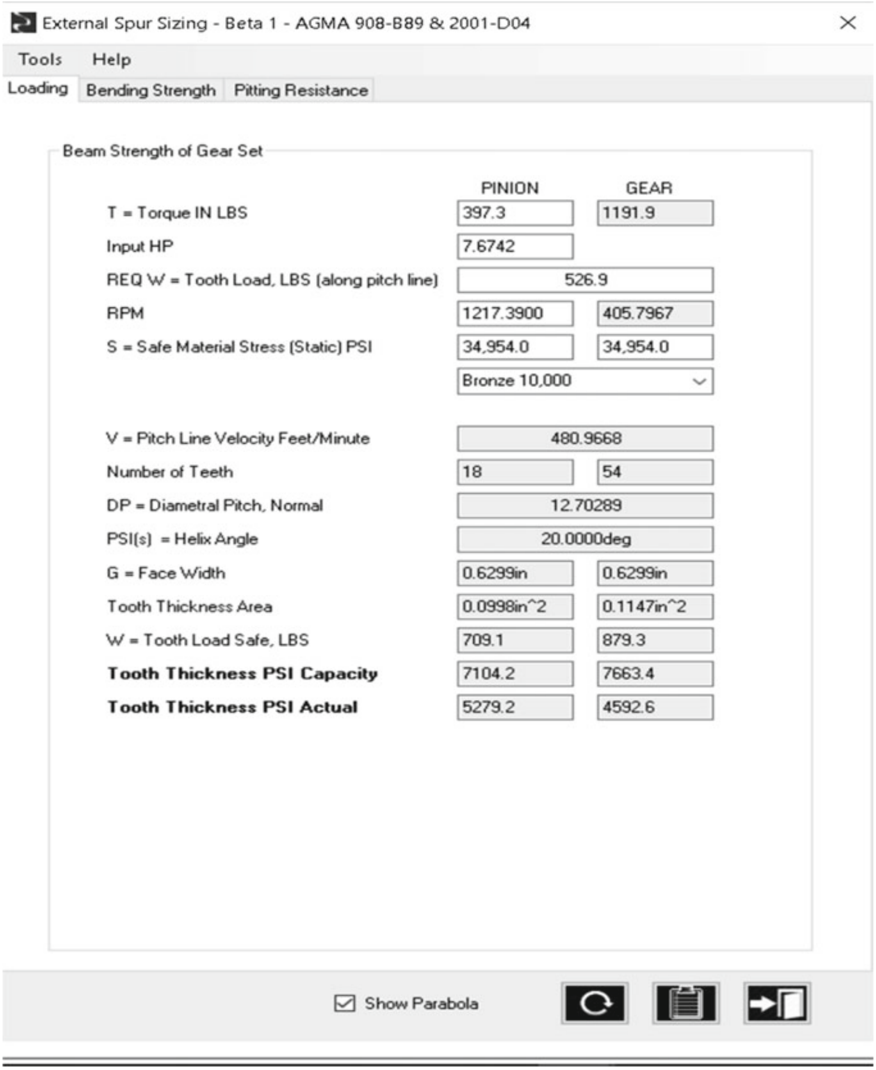


Fig. 25 Force calculations for gear set 1

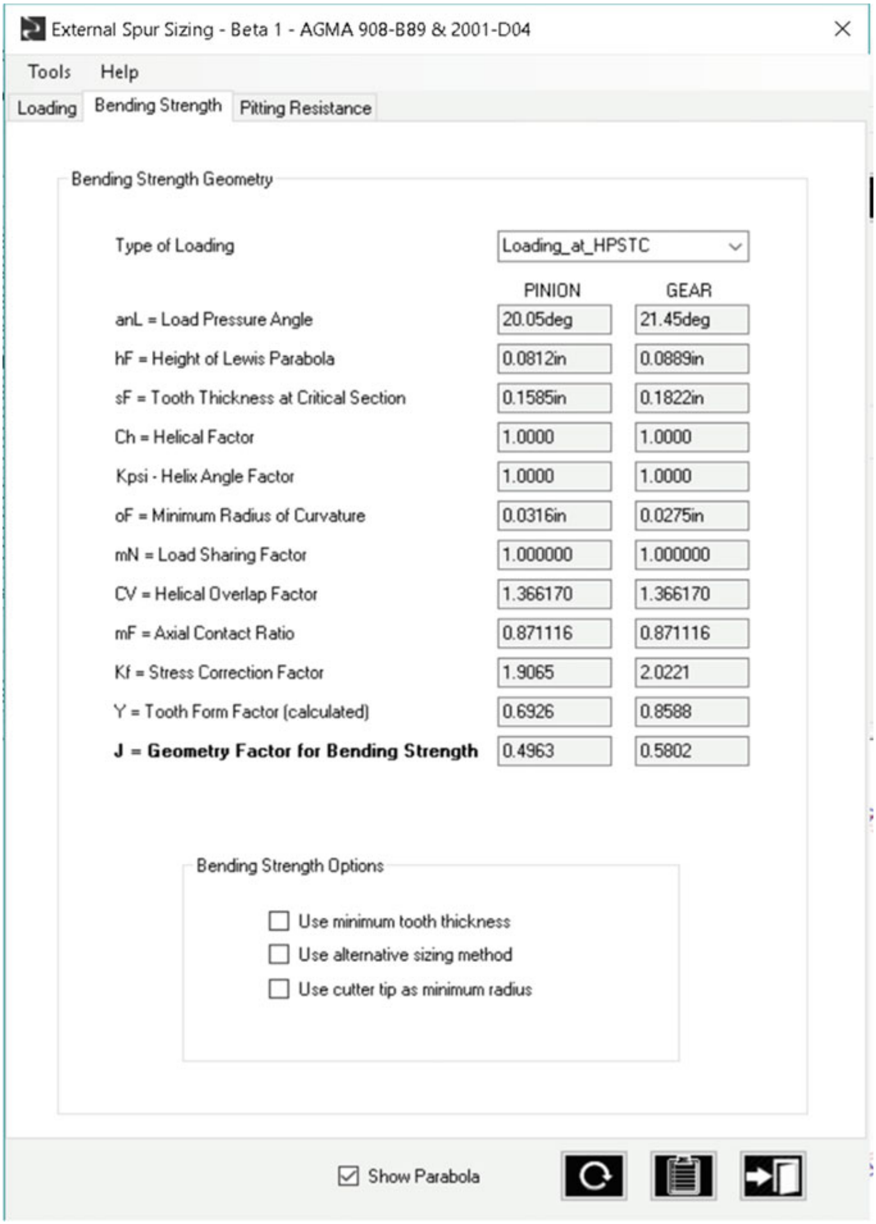


Fig. 26 Bending strength for gear set 1

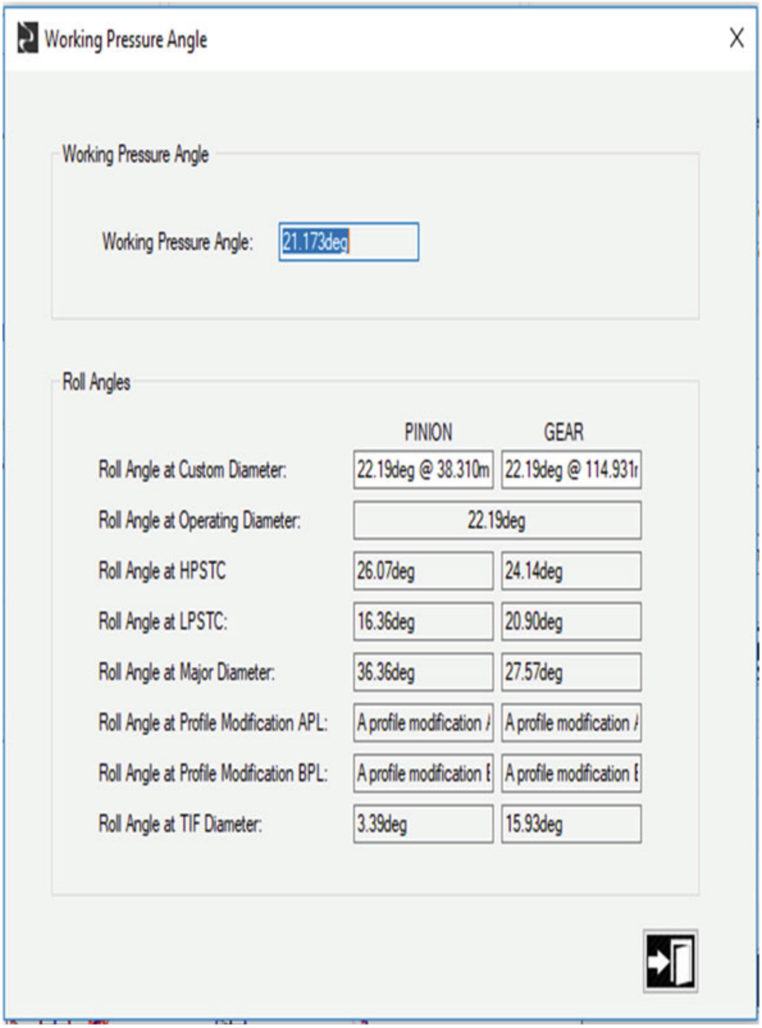


Fig. 27 Pressure angle for first stage reduction

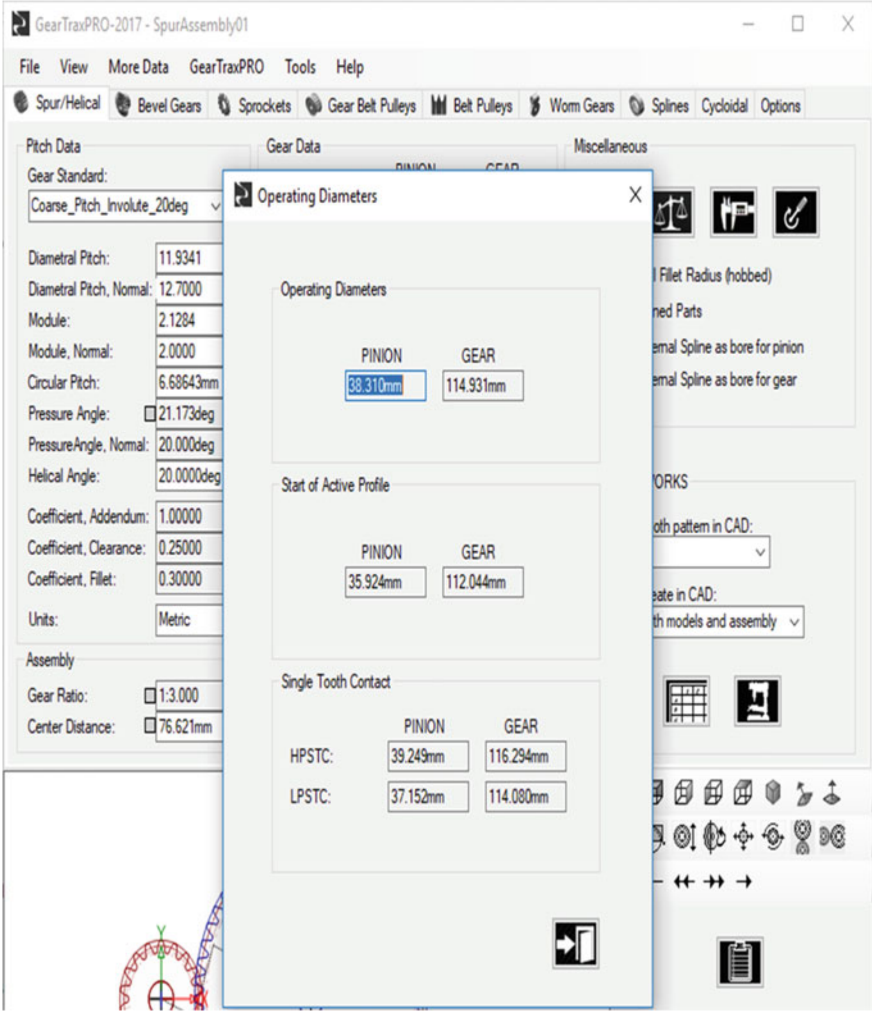


Fig. 28 Operating parameters for reduction gear 1

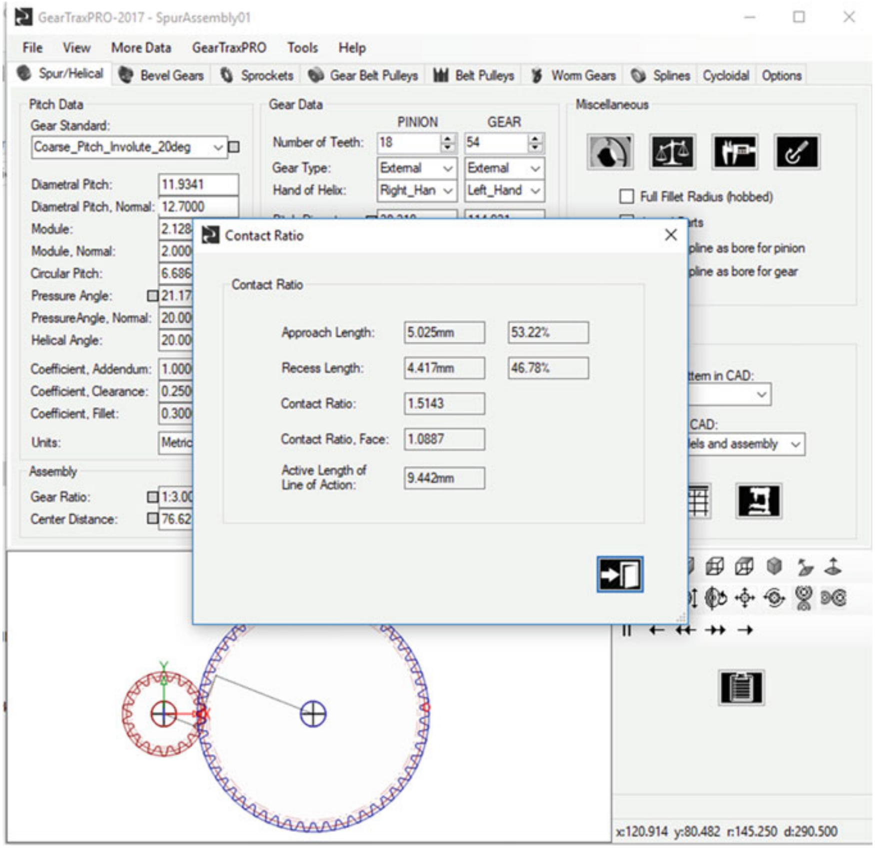


Fig. 29 Contact ratios for gear set 1

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References

1. Bardiya N, Karthik T, Rao LB (Sep 2014) Analysis and simulation of gearless transmission mechanism. Int J Core Eng Manag (IJCEM) 1(6)
2. Patel T, Dubey A, Rao LB (Sep 2019) Design and analysis of an epicyclic gearbox for an electric drivetrain. Int J Recent Technol Eng (IJRTE) 8(3). ISSN: 2277-3878
3. Herring DH, Otto FJ, Specht FR (2013) Gear materials and their heat treatment. BNP Media
4. Payne ET (2015) Design of an SAE baja racing off-road vehicle powertrain. The University of Akron in Akron, Ohio, USA, Spring

5. Sekhar K, Dharmadhikari P, Panchal S, Rao LB (2020) Design and optimization of a two stage gearbox using GEARTRAX. Recent Trends Mech Eng, Lect Notes Mech Eng 445–477
6. Ashby M (2001) Materials and process selection for engineering design, 3rd ed. Pergamon Press, Ch. 4, pp 110–144
7. Sharma A, Singh J, Kuma A (June 2015) Optimum design and material selection of baja vehicle. Int J Curr Eng Technol 5(3)
8. PSG (2006) PSG design data book, PSG college of technology. Kalaikathir Achchagam
9. Budynas R, Nisbett K (2006) Shigley's mechanical engineering design, 8th edn, Ch 3, sec 14. pp 711–762
10. Gligorijević R (1976) Materials for gears manufacturing their dynamic and structural characteristics and heat-treatment. J Technol 5(6):367–379
11. Gupta A, Yashvanth VP, Rao LB (2020) Design of gears using aluminium 6061-T6 alloy for formula SAE steering system. Recent trends in mechanical engineering, lecture notes in mechanical engineering. pp 489–505
12. Vara Prasad V, Satish G, Ashok Kumar K (2013) Trade of analysis for helical gear reduction unit. Int J Eng Res Technol 2(11):54–63
13. Aru S, Jadhav P, Gajabhiye S, Jadhav V, Angane P (2014) Design and analysis of centrally suspended cage-less differential. Int J Mech Prod Eng Res Develop 4(4):49–60
14. Anand S, Srikeshav AD, Sharran B, Rao LB (2020) Design and analysis of helical teeth harmonic drive. Recent trends in mechanical engineering, lecture notes in mechanical engineering. pp 507–519
15. Gujaba KP, Parashar S, Rao LB (2020) Design and analysis of permanent magnetic gears. Recent trends in mechanical engineering, lecture notes in mechanical engineering. pp 521–530
16. ANSYS user manual, Version 14
17. Solidworks user manual (2020)