© FEB 2019 | IRE Journals | Volume 2 Issue 8 | ISSN: 2456-8880

Design of Synchromesh Mechanism and Stress Analysis of Gear for Hijet

MYO ZAW [1], AUNG KO LATT [2]
[1, 2] *Dept. of Mechanical Engineering, Technological University (Myitkyina), Kachin State, Myanmar*

Abstract -- The gears are generally used to transmit power and torque. Gears are one of the most critical components in mechanical power transmission systems. The efficiency of power transmission is very high when compared to other kind of transmission. In the gear design the bending stress and surface strength of the gear tooth are considered to be one of the main contributors for failure of the gears in gear set. Thus, analysis of stresses has become popular as an area of research on gears to minimize or to reduce the failures and for optimal design of gears. In this paper bending stress at the root of the helical gear tooth can be calculated by using analytical method which is calculated by the Lewis stress formula. In this project, helical gear is designed by using solid works 2016 and the numerical solution is done by ANSYS, which is a finite element analysis package. The main objective of this research has to reduce the stresses induced in gear tooth profile by changing five different helix angles (13˚, 18˚, 23˚, 28˚ and 33˚). In this paper, 23˚ of helix angle is selected for helical gear used in synchromesh gearbox according to the ranges of helix angles and AISI 5160 OQT 400 is chosen for helical gear the maximum von-Mises stress and strain values of AISI 5160 OQT 400 are less than other two materials. The results are then compared with both the Lewis equation and ANSYS procedure.

Indexed Terms: Bending stress, helical gear, Helix angle, Lewis equation, surface strength

I. INTRODUCTION

The power developed inside the engine cylinder is ultimately aimed to turn the wheels so that the motor vehicle can move on the road. The reciprocating motion of the piston turns a crankshaft which rotates the flywheel through the connecting rod. The circular motion of the crankshaft is now to be transmitted to the rear wheels. It is transmitted through the clutch, gearbox, universal joints, propeller shaft or drive shaft, differential and axles extending to the wheels. The application of engine power to the driving wheels through all these parts is called power transmission. The power transmission system is usually the same on all modern passenger cars and trucks, but its arrangement may vary according to the method of drive and type of the transmission units. Figure 1 shows the power transmission of an automobile. The motion of the crankshaft is transmitted through the clutch to the gear box or transmission, which consists of a set of gears to change the speed. From gear box, the motion is transmitted to the propeller shaft through the universal joint and then to the differential through another universal joint. The differential provides the relative motion to the two rear wheels while the vehicle is taking a turn.

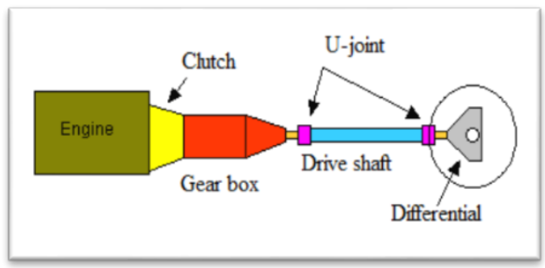

Fig. 1: Automobile power transmission system [6]

Thus the power developed in the engine is transmitted to the rear wheels through a system of transmission. The gear box is necessary in the transmission system to maintain engine speed at the most economical value under all conditions of vehicle movement [3]. A gearbox as usually used in the transmission system is also called a speed reducer, gear head, gear reducer etc., which consists of a set of gears, shaft and bearings that are factory mounted in an enclosed lubricated housing. Speed reducers are available in a broad range of sizes, capacities and speed ratios. Their job is to convert the input power into the output power with lower speed and correspondingly higher torque [8]. There are two types of transmission, namely manual transmission and automatic transmission. Manual transmission type is used so many cars in various models and locations. If the car has a manual gear box, the driver moves the shift lever to shift gears. Automatic gearbox does the job automatically without any effort by the driver. Manual transmission consists of a cast iron or aluminium housing, shafts, bearings, gears, synchronizing devices and shifting mechanisms. Automatic transmission includes a torque converter, compound planetary gear set, two or more disc clutches and one or more bands. In an automatic transmission model, the clutch and gear shifting operation is done automatically to the best suit with the driving condition and torque is converted continuously. These transmission systems are efficient, convenient, easy to operate, durable and reliable, but they are relatively expensive to manufacture and service compared to standard transmission system.

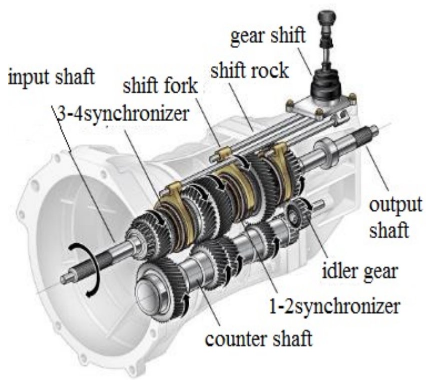

Fig. 2: Synchromesh gear box [7]

Synchromesh gear box is an extension of constant mesh gear box where synchronizers are used instead of dog clutches to avoid noise and jerk. Figure 2 shows the Synchromesh gearbox [5]. In this types of gearbox, all the gears of the main shaft are in constant mesh with corresponding gears of the countershaft. The gears on the main shaft which are bushed are free to rotate. The synchronizers are provided on main shaft. The gears on the lay shaft are, however, fixed. When the left synchronizer is slid to the left by means of the selector mechanism, its teeth are engaged with those on the clutch gear and direct gear can be obtained. The same synchronizer, however, when slid to right makes contact with the third gear and third gear is obtained. Generally two types of gear are used in synchromesh gear box include are spur gear and helical gear. In spur gears, the teeth are parallel to the axis whereas in helical gears the teeth are inclined to the axis. Both the gears transmit power between two parallel shafts. However in this paper, helical gear is used for synchromesh gear box because their contact ratio is higher than spur gears and they operate smoother and quieter than spur gears [7].

Fig. 3: Spur and helical gear [3]

a) Pitch circle diameter:
The pitch circle diameter is the diameter of pitch circle. Normally, the size of the gear is usually specified by pitch circle diameter.

b) Addendum:
The Addendum is the radial distance between the pitch and addendum circles. Addendum indicates the height of tooth above the pitch circle.

c) Dedendum:
The dedendum is the radial distance between pitch and the dedendum circles. Dedendum indicates the depth of the tooth below the pitch circle.

d) Module:
It is the ratio of pitch circle diameter to the number of teeth.

e) Pressure angle:
It is the angle that the line of action makes with the common tangent to the pitch circles.

f) Face width:
The width of the tooth measured parallel to the axis of the gear.

g) Number of teeth:
It indicates the number of teeth on the gear.

h) Helix angle:
It is a constant angle made by helices with the axis of rotation.Fig.4 shows the terminology of helical gear [4].

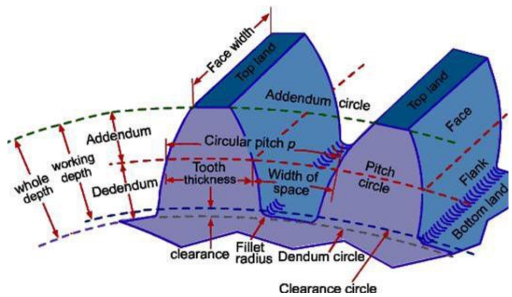

Fig. 4: Gear terminology [9]

II. DESIGN CALCULATION OF HELICAL GEAR PAIR

The design calculation of the helical gear pair has the following steps. The material for the gear pairs is taken as AISI 5160 OQT400. TABLE I shows the parameters considered for design a helical gear.

Design Parameter	Specification
Power (P)	20 kW
R.P.M (Np)	6500
Helix angle (ψ)	23°
Pressure angle (ϕ)	20°
Modulus of Elasticity (E)	207 GPa
Ultimate Strength	2220 MPa
Yield Strength	1790 MPa
Brinell Hardness (BHN)	627
Number of teeth of pinion	11 mm
Number of teeth of gear	34 mm
Poisson ratio (ν)	0.3

Table 1: Parameters considered for design a helical gear

Pinion and gear are same material and so pinion is weaker. So based design on pinion.

1. Unknown diameter case
The actual induced stress can be calculated by using Lewis equation.

$$S_{ind} = \frac{2M_t}{m^3 k \pi^2 y_p n_p \cos\psi} \quad (1)$$

2. Calculation of Torque (M_t)

$$M_t = \frac{9550 \times P}{N_p} \quad (2)$$

3. Calculation of pitch line velocity (V)

$$V = \frac{\pi \times D_p \times N_p}{60} \quad (3)$$

$$n_f = \frac{n}{\cos^3\psi} = y_p = 0.088 \text{ (from the table)}$$

K = 6 (for helical)

4. Calculation of allowable stress (S_{all})

$$S_{all} = S_0 \times \left[\frac{5.6}{5.6 + \sqrt{V}}\right] \quad (4)$$

5. Calculation of endurance stress (S_o)

$$S_o = \frac{S_u}{3} \quad (5)$$

6. Calculation of number of teeth

$$n = \frac{D}{m} \quad (6)$$

7. Strength Check

Compare S_{all} and S_{ind} (7)

If $S_{all} > S_{ind}$, Design is satisfied. If not so, keeping on calculating by increasing the module until it is satisfied need to be done.

8. Calculation of the face width of helical gear

$$b_{min} = k_{red} \times \pi \times m \quad (8)$$

$$b_{max} = k \times \pi \times m \quad (9)$$

$$k_{red} = k_{max} \times \frac{S_{ind}}{S_{all}} \quad (10)$$

After determining the design from strength point of view, it is necessary to check the dynamic effect.

9. Dynamic Check

$$F_t = \frac{2M_t}{D_p} \quad (11)$$

Calculation of dynamic load (F_d)

$$F_d = F_t + \frac{21V(bC\cos^2\psi + F_t)\cos\psi}{21V + \sqrt{(bC\cos^2\psi + F_t)}} \tag{12}$$

Calculation of limiting endurance load (F_0)

$$F_0 = S_0 b y_p \pi m \cos(\psi) \tag{13}$$

Calculation of limiting wear load (F_w)

$$F_w = \frac{D_p \times b \times K \times Q}{\cos^2\psi} \tag{14}$$

$$K = \frac{S_{es}^2 \times \sin\phi_n}{1.4} \times \left[\frac{2}{E}\right], \quad S_{es} = (2.75BHN - 70)\ \text{MN/m}^2$$

$$Q = \frac{2 \times D_g}{D_g + D_p}$$

$$\tan\phi_n = \tan\phi \cos\psi$$

	Symbol	Pinion	Gear	Unit
No. of teeth	n	11	34	-
Pitch circle diameter	D	27.5	85	mm
Outside diameter	D_0	32	89	mm
Root diameter	D_R	21	78	mm
Face width	b	21	21	mm
Module	m	2.5	2.5	mm
Speed	N	6500	5000	rpm

Table 2: Design result data for helical gear pair

The required condition to satisfy the dynamic check is F_0, F_w > Fd. If not, keeping on calculating by increasing the module until it is satisfied need to be done [6].

III. MODELLING OF HELICAL GEAR

In this paper, AISI 5160 OQT 400 is used as the helical gear materials. The material properties of AISI 5160 OQT 400 is given in the TABLE III.

Material Properties	Value
Young modulus	207 GPa
Poisson ratio	0.3
Density	7850 kg/m^3
Coefficient of Thermal Expansion	1.15e-05
Tensile Yield Strength	2220 MPa
Tensile Ultimate Strength	1790 MPa

Table 3: Properties of AISI 5160 OQT 400 helical gear

The procedure to model the gear of 34 number of teeth with the combination of the all above mentioned parameters in the Solid Works software, other set of gears are modelled in the similar way [8]. Figure 6 shows solid model of helical gear generated by Solid Works software.

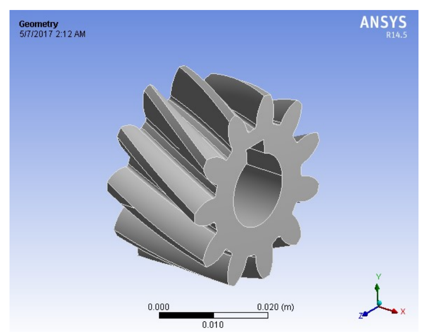

Fig. 5: Solid model of helical gear

Finite Element Method is a numerical technique for finding approximate solutions to boundary value problems. A boundary value problem is a differential equation together with a set of additional restraints, called boundary conditions. FEM uses various methods to minimize an error function and produce a stable solution [2].

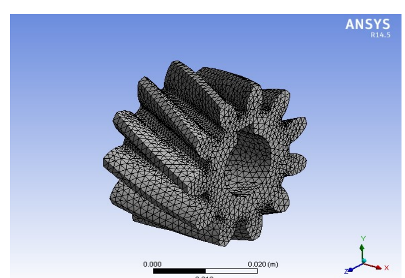

Fig. 6: Meshed 3-D model of helical gear

IV. LEWIS EQUATION USED TO CALCULATE BENDING STRESS

The bending stress is one of the crucial parameters during the analysis of helical gears. When the total repetitive load acting on the gear tooth is greater than its strength then the gear tooth will fail in bending. According to Lewis equation, the Beam Strength of helical gear tooth is given by

$$\sigma_b = \frac{F_t}{b y_p p_c \cos(\psi)} \tag{15}$$

Helix angle (Degree)	Bending Stress (MPa)
13	214.98
18	206.98
23	201.70
28	196.86
33	194.82

Table 4: Theory Result for Bending Stress by Using Lewis Equation

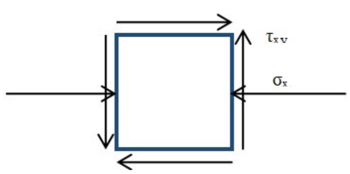

Fig. 7: Stresses in the x-y plane

$$\tau_{xy} = \frac{T \times R}{J} \tag{16}$$

$$\sigma_{1,2} = \frac{1}{2}(\sigma_x + \sigma_y) \pm \frac{1}{2}\left[(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2\right]^{\frac{1}{2}} \tag{17}$$

$$\bar{\sigma} = \frac{1}{\sqrt{2}}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]^{\frac{1}{2}} \tag{18}$$

Helix angle (degree)	von-Mises stress for Lewis (MPa)
13	215.56
18	207.58
23	202.32
28	197.49
33	195.45

Table 5: Theory Result for Bending Stress by Using Lewis Equation

V. FEM ANALYSIS USED TO CALCULATE THE BENDING STRESS

Helical gear assembly was imported in Ansys 14.5 and the boundary conditions were applied to the gear model. The fixed support was applied to the inner rim of the pinion and torque which the value is 37.054Nm is applied to the helical pinion in anticlockwise direction. The model was analysed for the root bending stress for the applied tangential force (2694.8364N). Only 3-D analysis was performed because of the helical profile of its teeth.

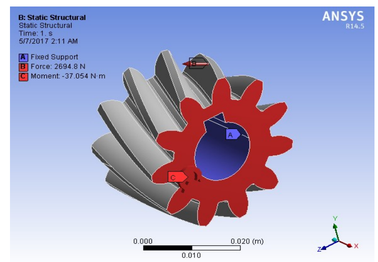

Fig. 8: Boundary condition of helical gear

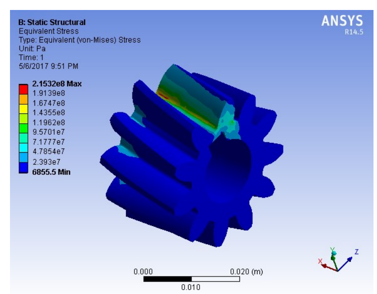

Fig. 9: von-Mises stress for helix angle 13˚

Figure 9 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 215.32 MPa.

Figure 10 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 208.17 MPa.

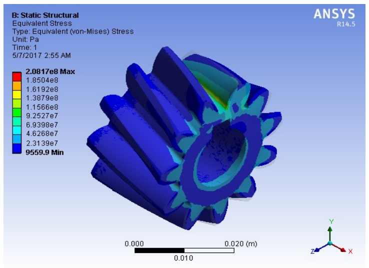

Fig. 10: von-Mises stress for helix angle 18˚

Figure 11 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 201.63 MPa. Figure 12 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 200.11 MPa. The maximum von-Mises stress developed by FEA is almost nearer to the calculated von-Mises stress.

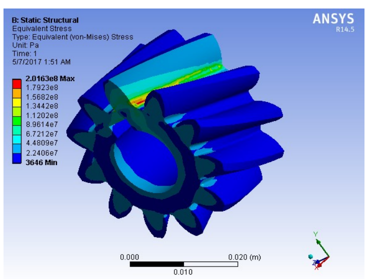

Fig. 11: von-Mises Stress for Helix Angle 23˚

Figure 13 shows that the maximum von-Mises stress is developed at the root of the pinion tooth of 193.28 MPa.

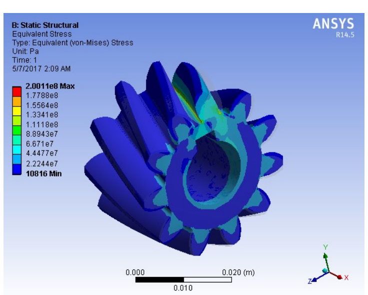

Fig. 12: von-Mises Stress for Helix Angle 28˚

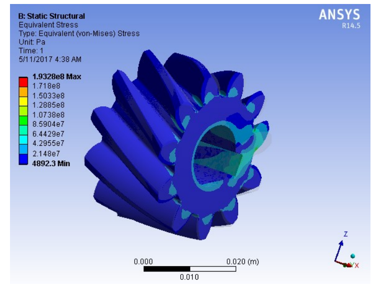

Fig. 13: von-Mises Stress for Helix Angle 33˚

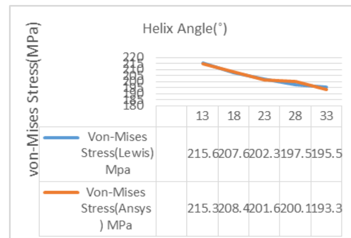

Fig. 14: Graphical representations for von-Mises stress comparison

VI. RESULTS AND DISCUSSIONS

In this section, the modelled helical gear is analyzed to study the effect of helix angle on bending stress under static load with different parameters. Throughout the analysis, each gear is studied for five different helix angle (ψ=13°, 18°, 23°, 28°, 33°). Face width, gear ratio, speed, module and the applied load are kept constant.[2] From the Figure 14, there is a variation in the maximum bending stresses with the change in helix angle. The maximum bending stress value decreases with the increase of helix angle. When comparing, Lewis values are little lower than the ANSYS values. Error percentages are around 1%.

VII. CONCLUSION

Helix angle is important geometrical parameter in determining the state of stresses during the design of gears. The helix angle ψ, is always measured on the cylindrical pitch surface. It ranges between 10˚ and 45˚. Commonly used values are 15, 23, 30 or 45˚. Lower values give less end thrust. Higher values result in smoother operation and more end thrust. In single helical gears, the helix angle ranges from 20° to 35°, while for double helical gears(herringbone gears), it may be made up to 45°[5]. Keeping the face width, gear ratio, speed, module constant and for variation of helix angle, the von-Mises stress decreases linearly. The helix angle 23˚, corresponding to the range of helix angle is taken for further optimization.

REFERENCES

[1] Dadi vijay et.al, 'Design And Structural Analysis Of High Speed Helical Gear Using ANSYS' Department of Mechanical Engineering Kakinada Institute of Technology And Science, Divili(2016).

[2] Parth J. Bhatt et.al, 'Material optimization of high speed single helical gear by using fea approach 'PG student, department of mechanical engineering Noble group of institute Junagadh - 362 001, India(2016)

[3] K.C.Lathiya et.al, 'Design modification and Failure Analysis of Damper Spring of 310 D.P.A of Single Plate Clutch System'Mechanical, Noble Group of Institution Junagadh (2015)

[4] B.Venkatesh et.al, 'Parametric investigation of the combined effect of Gear parameters on Tangential Force and Dynamic Tooth Load of 40 Ni2 Cr1 Mo 28 Steel Helical Gear' Vardhaman College of Engineering, Hyderabad, India((2014)

[5] B. Venkatesh et.al, 'Effect of Bending Stress on Steel Alloy Of Helical Gear For High Speed Applications' Department of Mechanical Engineering, Vardhaman College of Engineering, India(2014).

[6] J. Venkatesh, P. B. G. S. N. Murthy, "Design & Structural analysis of High Speed Helical Gear by using Ansys ," IJERA., India, vol. 4,Issue 3, pp. 01–05, (2014)

[7] A. Sathyanarayana Achari et.al,'A Comparison of Bending Stress and Contact Stress of a Helical Gear as Calculated by AGMA standard and FEA' M.Tech Scholar, , Department of Mechanical Engineering, NMAMIT, Nitte, Karnataka, India(2014)

[8] V.Rajaprabakaran et.al,'Spur Gear Tooth Stress Analysis And Stress Reduction', Student of M.E.in Mechanical dept., JJ College of Engineering and Technology, Trichy (2014)