# Design and Strength Analysis of Reverse Speed Spur Gear of Gear Box for Toyota Light Truck

THIN THIN SOE[1], SI THU THWIN[2], EI MON KYAW[3]
[1, 3] Lecturer, Mechanical Department, Pyay Technological University
[2] Associate Professor, Mechanical Department, Pyay Technological University

***Abstract-* This paper presents the design of the spur gear on different geometric condition and finding of the effect of these on tooth load by changing the module and face width. 3C diesel engine, which has the power rating 69 kW at 4200 rpm is used as the engine of Light Truck. In this gear box, the forward speed gears are helical type whereas reverse speed gears are spur type. In an automobile gear box, the reverse speed gears have a minimum speed of revolution that means maximum torque will be exerted. So in this paper, the design of spur gear for 3C diesel engine is described. The selected material for in the spur gear pair is AISI 5160 OQT-400 Alloy Steel. Tooth load is calculated with help of Lewis equation. Dynamic tooth load and wear tooth load are calculated with help of Buckingham equation.**

***Indexed Terms-* Design of spur gear, module, face width, Lewis equation, and Buckingham equation.**

## I. INTRODUCTION

There are so many types of transmission in automotive. There are hydraulic transmission, electric transmission and mechanical transmission [6]. Mechanical transmission is used to contract the transmission line of Light Truck. Power train of Light Truck is front mounted engine with rear wheel drive. This gear box is synchromesh type and has just five forward speeds and one reverse [3]. It consists of housing, shafts, bearing, gears, synchronizing devices and shifting mechanisms. Transmissions are generally equipped with spur and helical gears to assure longer life, quieter operation and improving efficiency. Second, third, fourth and fifth speed gears may be constant mesh design but first and reverse speed gears are sliding mesh design. The transmission consists of the input (clutch) and output (main) shaft assemblies, the counter gear and reverse Idler assembly [5].

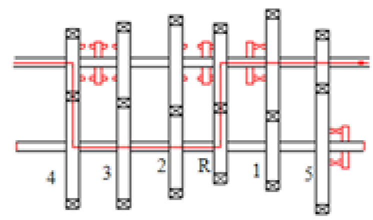



Fig 1. Reverse Speed Gear Train [4]

The transmission of reverse gear is shown in Fig.1. The clutch sleeve is in the neutral position. The sliding gear has been moved into mesh with the reverse idler gear. The input shaft turns the cluster; the cluster drives the reverse idler; the idler drives the sliding gear in a direction opposite to engine rotation. This will cause the output to drive the car backward [4].

## II. LITERATUREREVIEW OF GEAR TRAIN

Gear is defined as a machine element used to transmit motion and power between rotating shafts by means of progressive engagement of projections called teeth. Among them, spur is used in this reverse gear train [7]. The spur gear is cylindrical in form and has teeth, which are of involute form in most cases. The spur gear is the simplicity in design, economies of manufacture and maintenance, and absence of end thrust. They impose only radial loads on the bearings. If noise is not a serious design problem, spur gears can be used of almost any spaces. A common arrangement of spur gear is an external gear and pinion combination [2].

## III. BASIC TERMS OF SPUR GEAR

i. MODULE: Module of a gear is defined as ratio of diameter to number of teeth. m = D/N

ii. FACE WIDTH: The width along the contact surface between the gears is called the face width.
iii. TOOTH THICKNESS: The thickness of the tooth along the pitch circle is called the tooth thickness.
iv. ADDENDUM: The radial distance between the pitch circle and the top land of the gear is called the addendum.
v. DEDENDUM: The radial distance between the pitch circle and the bottomland of the gear is called the dedendum.
vi. PRESSURE ANGLE: The angle between the common normal to two gear teeth at the point of contact and the common tangent at the pitch point [2].

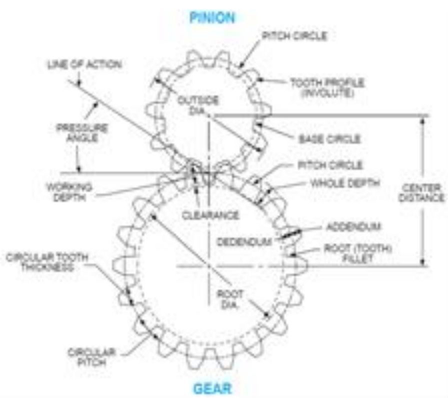



Fig 2. Nomenclatures of Spur Gear [7]

## IV. FORCE ON GEAR TEETH FOR SPUR GEAR

The power is received from engine by the clutch shaft rotating at engine speed. The power is transmitted from the clutch shaft to the output shaft. There is a torque in the shaft that can be computed from the following equation [3].

$$M_t = \frac{9550 \times kW}{rpm} \qquad \ldots\ldots (1)$$

The transmitted force acts tangential to the pitch surface of the gear, actually transmits torque and power from the driver to the driven gear and acts in a direction perpendicular to the axis of the shaft carrying the gear [3].

$$F_t = \frac{M_t}{D/2} \qquad \ldots\ldots (2)$$

In the design of gear for strength, the pitch diameter is unknown; the following form of the Lewis equation may be used [3].

$$S_{ind} = \frac{2 M_t}{m^3 k \pi^2 y N} \qquad \ldots\ldots (3)$$

where, $M_t$= torque on weaker gear, Nm
N = number of teeth on weaker gear
m = module, mm

The allowable stress for gear tooth design depends on the selected material and pitch line velocity (V).

$$S_{all} = S_o\left(\frac{3}{3+V}\right), \text{ for } V \langle 10 \text{ m/sec} \qquad \ldots\ldots (4)$$

$$S_{all} = S_o\left(\frac{6}{6+V}\right), \text{for } 10 \text{ m/sec} \langle V \langle 20 \text{ m/sec} \qquad \ldots\ldots (5)$$

$$S_{all} = S_o\left(\frac{5.6}{5.6+\sqrt{V}}\right), \text{for } V \rangle 20 \text{ m/sec} \qquad \ldots\ldots (6)$$

Apply the Lewis equation as the following factor:
i. The Lewis equation is applied only to the weaker of two gears.
ii. When the gears are made of same material, the pinion is weaker.
iii. When the gears are made of different materials, the Lewis equation is used to that gear for which deciding factor ($S_o y$) is less.
iv. The product ($S_o y$) is called strength factor of the gear.
v. Face width (b) may be taken as $b_{min}$ to $b_{max}$ [1].

The dynamic analysis, as proposed by Buckingham equation.

$$F_d = \frac{21 V (bC + F_t)}{21 V + \sqrt{bC + F_t}} + F_t \qquad \ldots\ldots (7)$$

where, $F_d$ = dynamic load, N
b = face width, m
C = dynamics Factor (N/m)

The endurance load is calculated by the following equation.

$$F_o = S_o b y \pi m \qquad \ldots\ldots (8)$$

In this equation, $S_o$ is based on average stress concentration values. To insure the durability of a gear pair, the tooth profile must not have excessive contact stress as determines by the wear load $F_w$ [3].

$F_w = D_p bKQ$ ….. (9)

where, $D_p$= pitch diameter of smaller gear , m

K = stress factor for fatigue, Pa

Q = ratio factor

$$Q = \frac{2\,N_g}{N_g + N_p} = \frac{2\,D_g}{D_g + D_p} = \frac{2\,V.R}{V.R + 1} \quad \text{….. (10)}$$

$$K = \frac{S_{es}^2 \sin\phi}{1.4}\left[\frac{1}{E_p} + \frac{1}{E_g}\right] \quad \text{….. (11)}$$

where, $S_{es}$ = surface endurance limit of a gear pair, MPa

$E_p$ = modulus of elasticity of pinion material, MPa

$E_g$ = modulus of elasticity of gear material, MPa

Ø= pressure angle

The surface endurance limit may be estimated from

$S_{es} = 2.75(BHN)_{avg} – 70$ ….. (12)

Where BHN may be approximated by the average brinell hardness number of the gear and pinion up to a BHN of about 350 for steel. The wear load $F_w$ is an allowable load and must be greater than the dynamic load $F_d$. Dynamic forces in the teeth is greater than the transmitted force. The dynamic analysis is proposed by Buckingham [1].

## V. DESIGNING OF SPUR GEAR TOOTH PROFILE

Number of teeth = N
Pressure angle = Ø
Take module = m
Pitch circle diameter =D
Addendum =0.8 m
Dedendum =1 m
Tooth thickness =1.5708
Working depth = 1.6 m
Minimum total depth = 1.8 m
Clearance =0.2 m
Fillet radius =0.4 m
Outer (Add: circle) radius =Addendum + Pitch circle radius
Root (Ded: circle) radius =Pitch circle radius – Dedendum
Base circle radius =Pitch circle radius – Addendum

TABLE 1.Results data on reverse speed spur gears

| Sr.No. | Gear Terminology | Pinion | Idler | Gear |
|---|---|---|---|---|
| 1 | No. of Teeth | 14 | 31 | 39 |
| 2 | PCD ( mm) | 56 | 124 | 156 |
| 3 | Pressure Angle | 20° | 20° | 20° |
| 4 | Module (mm) | 4 | 4 | 4 |
| 5 | Face Width (mm) | 30 | 30 | 30 |
| 6 | Tooth Thickness | 6.283 | 6.283 | 6.283 |
| 7 | Working Depth (mm) | 6.4 | 6.4 | 6.4 |
| 8 | Whole Depth (mm) | 7.2 | 7.2 | 7.2 |
| 9 | Addendum (mm) | 3.2 | 3.2 | 3.2 |
| 10 | Dedendum (mm) | 4 | 4 | 4 |
| 11 | Clearance (mm) | 0.8 | 0.8 | 0.8 |
| 12 | Fillet radius (mm) | 1.6 | 1.6 | 1.6 |

Fig 3. Modelled Pinion

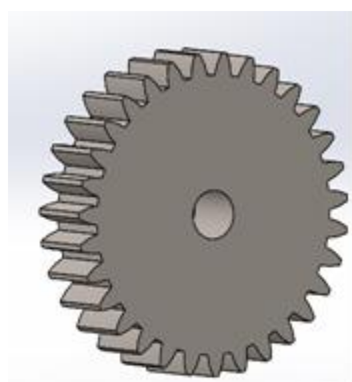


Fig 4. Modelled Idler

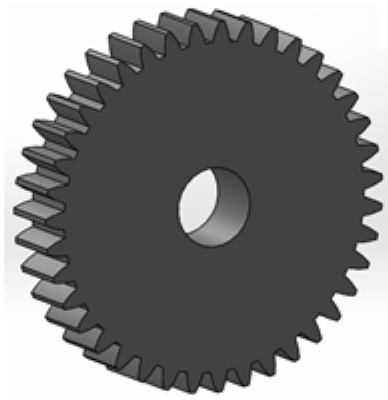


Fig 5. Modelled Gear

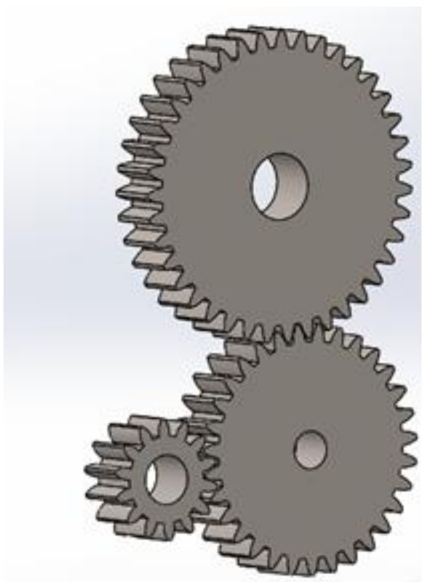


Fig 6. Meshing of Gears

## VI. CALCULATION OF TOOTH LOAD BY CHANGING MODULE

$M_t$= 218.882 Nm
$S_o$= 740 MPa
Np= 14
y = 0.108

*A. If module, m = 3.5 mm*

Then, $D_p$= 49 mm
$F_t$= 8934 N

$$V = \frac{\pi\, D \times rpm}{60} = 7.34 m/s$$

b = 38 mm

Endurance tooth load

$F_o$= 740×38×0.108×π×3.5
= 33393 N

Dynamic tooth load
Take C = 114 kN

$$F_d = \frac{21 \times 7.34((38 \times 114) + 8934)}{(21 \times 7.34) + \sqrt{(38 \times 114) + 8934}} + 8934$$

= 16526 N

Wear tooth load
Take K = 6.459 MPa
Q = 1.378
$E_p$=$E_g$= 207 GPa
$F_w$ = 49×38×6.459×1.378
= 16571 N

*B. If module, m = 4 mm*

Then, $D_p$= 56 mm
$F_t$= 7817 N
V =8.39 m/s
b = 30 mm

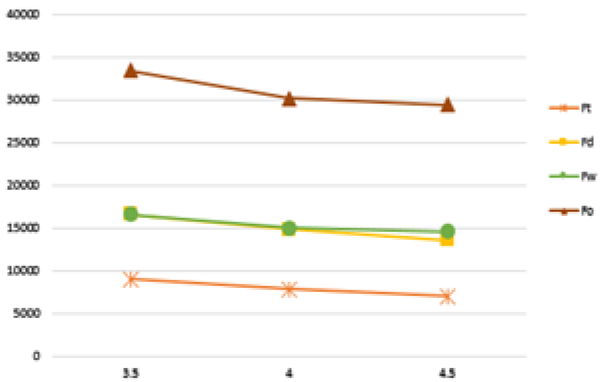
Endurance tooth load
$F_o$= 30129 N
Dynamic tooth load
$F_d$= 14832 N
Wear tooth load
$F_w$= 14951 N

*C. If module, m = 4.5 mm*

Then, $D_p$= 63 mm
$F_t$= 6949 N
V =9.43 m/s
b = 26 mm

Endurance tooth load
$F_o$= 29376 N
Dynamic tooth load
$F_d$= 13546 N
Wear tooth load
$F_w$= 14577 N

The above result data of tooth loads with respect to various modules are compared in the Fig. 7. It can be seen that the module is inversely proportional to the tooth load. In this Fig.7, if the value of module is greater, the value of tooth load is lesser.




Module, mm

Fig 7. Tooth Load Vs module

## VII. CALCULATION OF TOOTH LOAD BY CHANGING FACE WIDTH

Module, m = 4 mm

$F_t = 7817$ N

*A. If face width, b = 32 mm*

Endurance tooth load

Fo= 740×32×0.108×4

= 32138 N

Dynamic tooth load

$$F_d = \frac{21 \times 8.39((32 \times 114) + 7817)}{(21 \times 8.39) + \sqrt{(32 \times 114) + 7817}} + 7817$$

$$= 149457 \text{ N}$$

Wear tooth load

$F_w$ = 56×32×6.459×1.378

= 15949 N

*B. If face width, b = 34 mm*

Endurance tooth load

$F_o$= 34146 N

Dynamic tooth load

$F_d$= 15062 N

Wear tooth load

$F_w$= 16945 N

*C. If face width, b = 36 mm*

Endurance tooth load

$F_o$= 36155 N

Dynamic tooth load

$F_d$= 15176 N

Wear tooth load

$F_w$= 17941 N

*D. If face width, b = 38 mm*

Endurance tooth load

$F_o$= 38164 N

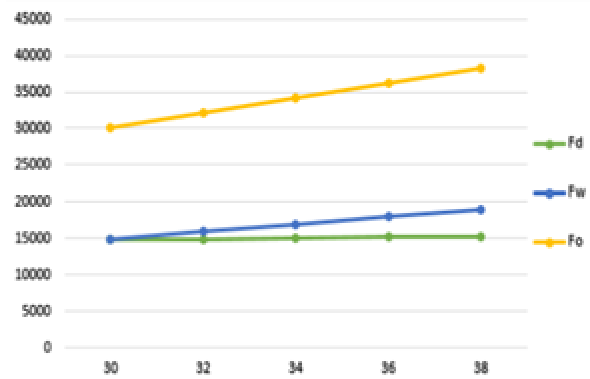
Dynamic tooth load

$F_d$= 15290 N

Wear tooth load

$F_w$= 18938 N

The above result data of tooth loads with respect to various face width are compared in the Fig. 8. It can be seen that the value of face width is directly proportional to the tooth loads.




Face width, mm

Fig 8. Various tooth loads, F w.r.t face width, b

## VIII. CONCLUSION

In this paper, the design of reverse speed gears of Light Truck gearbox has been described. For the design of reverse speed gears, spur gear design is calculated by changing the module and face width. In this design, structural rigidity combined with lightness must always be the first consideration for the durability of the wearing parts and smoothness of running.

Gears are designed with AISI 5160 OQT-400 Alloy Steel, which has brinell hardness 627, yield stress 1790 MPa and ultimate stress 2220 MPa [8]. The required design calculation of spur gears can also be calculated through both strength and dynamic check.

The reverse gear ratio is 2.8.The strength of the gear teeth is determined from Lewis equation. The requirement design calculation of spur gear can also be calculated through both strength and dynamic check [5]. Therefore, this material is proper. Spur gears teeth of transmission gear box must have sufficient strength and they will not fail under static loading.

## REFERENCES

[1] Hall, Holowenko, Laughlin, "Theory and Problems of Machine Design", SI (Metric) Edition, SOS series Schoum Publishing Co; New York, (1961).

[2] Khurmi, R.S. and Gupta, J.K, "A Textbook of Machine Design", First Multicolour Edition, Eurasia Publishing House (Pvt) Ltd, New Delhi, (2015).

[3] Hlae Yi Myo, Ma, :Design and Stress analysis of Gear Box (Light Truck)", M.E. Thesis, Mandalay Technological University, (2014).

[4] Ei Ei San, Ma, "Design Calculation of Five Speed Gear Box (Light Truck)", M.E. Thesis, Mandalay Technological University, (2008).

[5] Ye Htut, Mg, "Design of Gear Box for Mazda Jeep", M.E. Thesis, Technological University (Maubin), (2009).

[6] Aung Kyaw Kyaw, Mg, , "Design of Gear Box for Mazda Jeep", M.E. Thesis, Technological University (Thanlyin), (2009).

[7] Parveen Kumar And Harsh Raghuvanshi, "Design & Analysis of a Spur Gear in Different Geometric Conditions", International Journal of Engineering And Advanced Technology (IJEAT), ISSN: 2249-8958, Volume-3, Issue-2, December 2013.

[8] Khin Khin Thant, Ma, "Stress Analysis on Spur Gear Teeth for Various Contact Ratio", National Conference on Science and Engineering, YTU, July 2017.