ISSN 2395-1621

Design and Analysis of Gearbox for Tractor Transmission System

#1Rahul Mokal, #2R.V. Mulik, #3S.B. Sanap

1rbmokal13.scoe@gmail.com
2mulik.edl@araiindia.com
3sbsanap.scoe@sinhgad.edu

#13Sinhgad college of Engineering, Vadgaon Bk, Pune.
#2PTE Dept., Automotive Research Association of India, Pune, Maharashtra, India

ABSTRACT

The tractor gear box plays an important role for various torque transmission to assure wider range of speed. Gearbox transmits power from engine to differential of tractor through propeller shaft. Generally tractor consists of gearbox with various configurations. The main objective of Paper is to design tractor gear box of 16 forward speed and 16 reverse speed. Paper work contains analytical method and Duty cycle analysis technique of the tractor gear box. In this Paper work, the significant parameters of gear box like speed, torque etc., are considered while modelling the gearbox. The modelling as well as the analysis of the gearbox done by using ROMAX software. Validation of the same has been done with the help experimental published data.

Keywords- **Duty cycle, speed, torque, ROMAX etc.**

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

A transmission or gearbox provides speed and torque conversions from a rotating power source to another device using gear ratios. The most common use is in motor vehicles, where the transmission adapts the output of the internal combustion engine to the drive wheels. In British English term transmission refers to whole drive train, including gearbox, clutch, propeller shaft (for rear wheel drive), differential and final drive shafts. 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 transmissions are also on pedal bicycles, fixed machines, and anywhere else rotational speed and torque to be adapted. Often, a transmission will have multiple gear ratios (or simply gears), with the ability to switch between them as speed varies. The switching may be done manually (by the operator), or automatically. Directional (forward and reverse) control may also be provided. Contemporary automobile manual transmission typically use four to six forward gears and one reverse gear. Transmission for heavy trucks and other heavy equipment usually have at least 9 gears so the transmission can offer both a wide range of gears and close gear ratios to keep the engine running in the power band. Some heavy vehicle transmissions have dozens of gears, but many are duplicates, introduced as an accident of combining gear sets, or introduced to simplify shifting. Generally tractor consists of gearbox with various configurations like 6 forward and 6 reverse, 8 forward and 8 reverse, 12 forward and 4 reverse etc. The main objective of Paper is Design and Duty Cycle Analysis Tractor Gear box of 16 forward speed and 16 reverse speed.

I. PREPARATION OF LAYOUT

Given Layout consists of input shaft, lay shaft and output shaft with number of gears mounted. First to fourth gears are mounted on lay shaft while pinions of the same are mounted on output shaft. Output shaft is followed by gear train which is used to obtain high and low speed for every gear step (i.e. 1st to 4th) as shown in figure below.

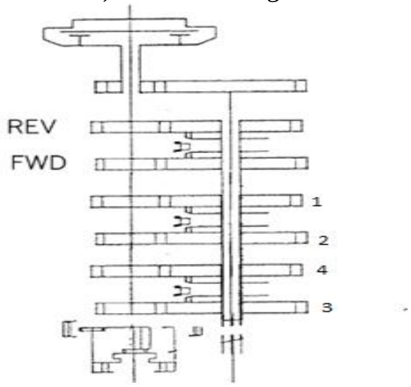

Fig.1. Schematic layout of 8 by 8 tractor gearbox

Proposed layout contains two more pairs of forward and reverse gear kept in adjacent to original reverse and forward gear pair.

II. GEOMETRY DESIGN

A. Design of Helical gear sets:[6]

Centre distance (C.D.) = $0.5\sqrt[3]{\max.torque}$ (1)

No. of pinion teeth = $\frac{15}{gear\,ratio+1}$ (2)

No. of gear teeth = $pinion\,teeth \times gear\,ratio$ (3)

Pinion diameter = $\frac{2 \times CD}{gear\,ratio+1}$ (4)

Gear diameter = $D = d \times gear\,ratio$ (5)

Calculated bending stress

$$= \frac{F_t}{b \times m_n} \times K_a \times K_v \times K_{H\beta} \times K_{F\alpha} \times Y_F \times Y_s \times Y_{\beta X} \times Y_{DT} \quad (6)$$

Contact Stress = $\sigma_{H1} = Z_B \sigma_{H0} \sqrt{\frac{F_t (u+1)}{d_1 b u}}$ (7)

Where,

K_A = Application factor,
K_v = Dynamic factor,
$K_{H\beta}$ = Face load factor,
YF = Tooth form factor,
Ys = Stress correction factor,
Y_B = Rim thickness factor,
Y_β = Helix angle factor,
Y_{DT} = Deep tooth factor,
Y_{NT} = Life factor,
Y_X = Size factor,
Z_H = Zone factor,
Z_B and Z_D for helix angle= Single tooth factor = 1,
Z_E = Elasticity factor,
$Z\epsilon$ = Contact ratio factor,
Z_β = Helix angle factor.

With help of above equations, following details of the all gear set are calculated.[1],[2]

TABLE.1.
GEAR GEOMETRY

Gear Name	Zg	Zp	Normal module(mn)	Gear Ratio
Reverse 1 (new)	24	44	1.81	1.83:1
Forward 1 (new)	31	37	2.39	1.19:1
Reverse	22	37	1.81	1.68:1
Forward	28	41	2.36	1.46:1
1st	42	14	2.91	3:1
2nd	40	20	2.71	2:1
3rd	40	29	2.45	1.37:1
4th	34	35	2.45	0.97:1

B. Design of Shafts:

Torque transmitted by shaft is calculated by,

$$T = \frac{P \times 60}{2\pi N} \quad (8)$$

Maximum bending moment, M is calculated by considering all the forces in vertical and horizontal plane.

Equivalent twisting moment is calculated by,

$$T_e = \sqrt{M^2 + T^2} \quad (9)$$

Finally diameter of the shaft is calculated by,[6]

$$d = \sqrt[3]{\frac{16\,T_e}{\pi \times \tau}} \quad (10)$$

C. Ratios from epicyclic:

Epicyclic gear train is connected in ahead of output shaft which gives low ratio of 4.09:1 and high ratio of 1:1. Following table shows required conditions for getting low and high ratios:

TABLE 2.
EPICYCLIC GEARTRAIN RATIO

Drive	Sun	Sun
Output	Planet carrier	Planet carrier
Held securely	Annulus gear	Planet gear
Ratio	Low ratio = i= $1+\frac{Z_R}{Z_S}$	High ratio = 1:1

III. GEOMETRY MODELLING

ROMAX DESIGNER is a virtual product development and simulation environment for the design, analysis and NVH exploration of the complete transmission – gear boxes, bearings, drive trains and all components.[1]

Model generation and analysis has done in steps which start with:

- Detailing of all the shafts which includes drive shaft, main shaft, lay shaft and reverse idler shaft details.
- Detailed gear geometry parameters including position of each other.
- Detailed bearing specification and position of each
- Material selection for shaft and gears including their heat treatment procedures.
- Lubricant details to be specified

- Standard factors, gear quality grade and method of calculation.
- Duty cycle for gearbox which includes the driving cycles of each gear for specified no of hours.

Figure below shows an isometric view of Romax gearbox model.

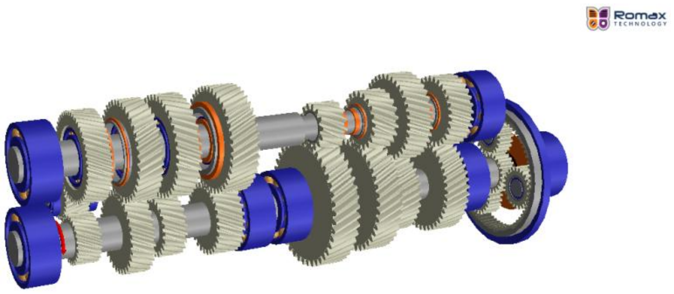

Fig.2. ROMAX model – Isometric view

Figure below shows an power flow diagram for 3rd gear in forward.

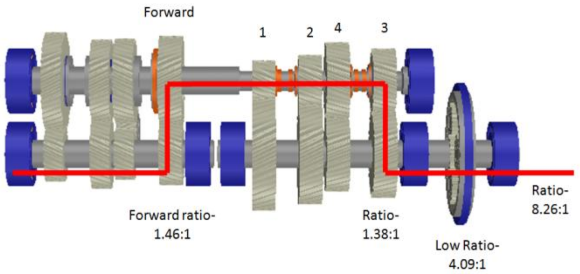

Fig.3. Power flow Diagram for forward 3rd speed

IV. Duty cycle analysis and Results

A. *Contact and bending life:*

Table below shows gear summary or life summary for required life when gearbox is running in forward 3rd gear (low ratio) condition.

TABLE 3.
CONTACT LIFE, BENDING LIFE, COMBINED LIFE

Gear	Contact Life		Bending Life		Combined Life	Pass/fail?
	Left	Right	Left	Right		
	(hrs)					
Wheel 9	N/A	N/A	N/A	N/A	N/A	Pass
Wheel 8	N/A	N/A	N/A	N/A	N/A	Pass
Wheel 4 th	1338.0616	N/A	N/A	N/A	1338.0616	Pass
Wheel 3 rd	838.0679	N/A	N/A	N/A	838.0679	Pass
Wheel 2 nd	N/A	254.1756	N/A	5.9231e7	254.1756	Pass
wheel 1 st	102.0786	N/A	4.3037e7	N/A	102.0786	Pass
Wheel 1	N/A	514.7906	N/A	N/A	514.7906	Pass
Pinion 9	N/A	N/A	N/A	N/A	N/A	Pass
Pinion 8	N/A	N/A	N/A	N/A	N/A	Pass
Pinion 4 th	1377.4164	N/A	N/A	N/A	1377.4164	Pass
Pinion 3rd	607.5992	N/A	N/A	N/A	607.5992	Pass
Pinion 2 nd	N/A	127.0878	N/A	5.7401e7	127.0878	Pass
pinion 1 st	34.0262	N/A	1.2811e6	N/A	34.0262	Pass
Pinion 1	N/A	351.5643	N/A	N/A	351.5643	Pass
idler	N/A	N/A	N/A	N/A	N/A	Pass

B. *Deflection Results:*

Figure below shows an deflection results of gearbox when it is running in forward 3rd gear (low ratio) condition.

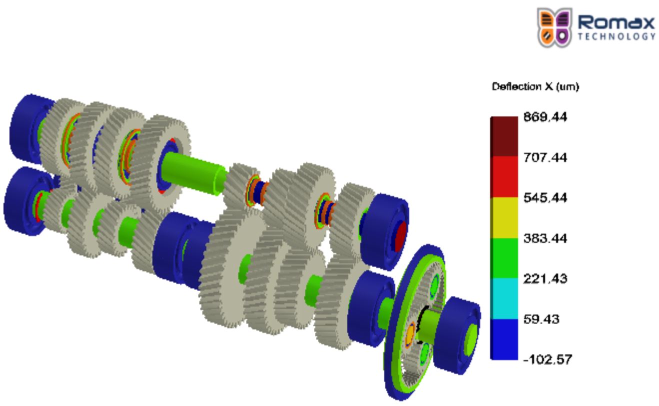

Fig.4. Deflection results in X (mm)

C. *Duty cycle Safety factor Summary :*

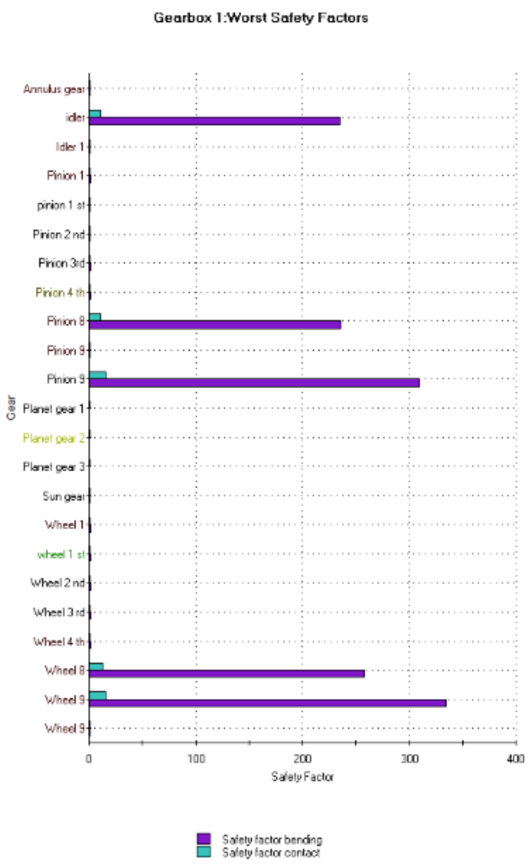

Fig.5. Duty cycle safety factor summary

TABLE 4.
DUTY CYLE SAFETY FACTOR SUMMARY

Gear	Safety Factor Contact		Safety Factor Bending	
	Left	Right	Left	Right
Wheel 9	>1000.0	12.397	>1000.0	238.689
Wheel 8	>1000.0	11.845	>1000.0	225.274
Wheel 4 th	1.245	22.470	1.692	596.759
Wheel 3 rd	1.181	21.066	1.493	544.382
Wheel 2 nd	>1000.0	1.029	>1000.0	1.538
wheel 1 st	1.249	15.267	1.529	512.025
Wheel 1	>1000.0	1.114	>1000.0	1.880
Pinion 9	>1000.0	12.161	>1000.0	221.195
Pinion 8	>1000.0	9.925	>1000.0	225.760
Pinion 4 th	1.249	22.546	1.695	599.224
Pinion 3rd	1.137	20.297	1.433	530.651
Pinion 2 nd	>1000.0	0.94801	>1000.0	1.296
pinion 1 st	1.249	13.470	1.192	437.402
Pinion 1	>1000.0	1.067	>1000.0	1.823
idler	>1000.0	9.643	>1000.0	205.031

V. CONCLUSIONS

This paper discusses in detail the design and duty cycle analysis of gears and shafts also selection of bearing for 16 forward speeds and 16 reverse speeds gearbox of tractor application. Given gear ratios are tailored specifically for tractor application.

1. Gears are analysed for safety parameters like root safety, flank safety and scuffing safety by using Romax software.
2. Also gear pair meshing contact pressure and stress distribution was analysed.
3. Factor of safety for all gear pairs are in acceptable range.
4. Shaft are analysed for its deflection, bending and stress distribution.

ACKNOWLEDGMENT

I would like to thank all engineers who helped us in Power train engineering department of ARAI, Pune. I would like express my special thank to Mr. S.S. Ramdasi and N.V. Marathe (HOD, PTE, ARAI) who gave me the golden opportunity to do this wonderful project, who also helped me in doing a lot of research and I came to know about so many new things we are really thankful to them.

REFERENCES

[1] Design Analysis and Development of a New 5F/1R Synchromesh Rear Mounted Gearbox for Mini Truck for an Emerging Markets, Gunjegaonkar, D. S., Bhattacharya, S and Sunil R. Sequeira Greaves Cotton Ltd., India, SAE Paper No. 2009-26-021

[2] Selection of Gear Ratio for Smooth Gear Shifting Jaideep Singh, k.v.v. rao Srinivasa and Jagmindar Singh Mahindra & Mahindra Ltd., SAE International, 2012-01-2005.

[3] .Gearbox Design by means of Genetic Algorithm and CAD/CAE Methodologies, Elvio Bonisoli, Mauro Velardocchia, Sandro Moos, Stefano Tornincasa and Enrico Galvagno Politecnico di Torino, SAE International, 2010-01-0895. R. E.

Sorace, V. S. Reinhardt, and S. A. Vaughn, “High-speed digital-to-RF converter,” U.S. Patent 5 668 842, Sept. 16, 1997

[4] Analytical Approach to Gear Reverse Engineering (Spur and Helical) Sagar Jadhav Mahindra & Mahindra Ltd., Amit Sandooja Mahindra Engineering Services, SAE International, 2012-01-0812.

[5] Gearbox Design by means of Genetic Algorithm and CAD/CAE Methodologies, Elvio Bonisoli, Mauro Velardocchia, Sandro Moos, Stefano Tornincasa and Enrico Galvagno Politecnico di Torino, SAE International, 2010-01-0895.

[6] Achieving Balanced Design of Transmission Gears for Bending Life, Contact Life and NVH for Agricultural Tractor Applications, Y. Subbaiah, P. N. Rao and R. B. Verma Tractors and Farm Equipment Limited, SAE international, 2008-01-2632.

[7] Maitra, G.M. “Handbook of Gear Design”.

[8] Dudley, Darle W. “Handbook of Practical Gear Design”.

