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Design of Transmission System of Tractor for same Load Carrying Capacity

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Abstract: The aim of this work is to design a transmission system for same load carrying capacity which will increases efficiency of the system by fully synchronized gear train. In order to avoid interfacing the module of reverse gear is changed from 3.8 to 4.2. This helps to improve the overall product life and eases the operational aspects by reducing the cost also.
Keywords: Transmission system, Gear Design, Shaft, Bearing

I. INTRODUCTION

Transmission system is used for torque transmission. Basic gearbox has 4 or 6 forward gear and reverse gear [1]. While moving tractor in the with loaded trolley torque which require to move in reverse direction is more than first forward direction movement. To change this torque, it requires to change gear ratio of gears. To change gear ratio, we need to change number of teeth of gears. Another problem is reverse direction sliding mechanism the idle gear edge wears due to sudden engage during the gear change from first gear to reverse gear. By changing gear ratio, design of gears for reverse direction motion [2-4]. Also, the problem of idle gear edge wear by changing from sliding gear mechanism to constant mesh gear mechanism [5]. Design of transmission system according to the changes and making it easy operation without any major changes in the current system.

This work is basically based on a tractor transmission system which we have done as industrial defined project at Trishul Tractors Pvt. Ltd.. As company using old generation transmission system, we try to modify it by changing small parts of this system. When tractor is connected to the loaded trolley, it requires higher torque transmission to run in reverse direction than the first forward direction. Also try to eliminate problem of wear out of gear tooth. By doing this project we try give best solution to the problem of organization for implement of their product and better product life.

II. ANALYSIS OF CURRENT DESIGN

A. Forces on the Trailer

As seen in above Figure 1 For pulling a component F*sinθ acts downwards along with the weight m*g and therefore increases the normal reaction N=Normal reaction is equal to sum of all the vertical forces. And friction is directly dependent on Normal reaction; More is the frictional force. This force F*sinθ acts upwards along with the weight m*g and therefore decreases the normal reaction N. Therefore, the frictional force is reduced. As per fig. 1 For pushing a component, there is one component of force that adds to the weight of the body and hence there is more friction. From this we can say that it is easier to pull than push. That phenomena is same for tractor trolley (trailer) also. Tractor with trolley moves in reverse direction means pushing the trolley requires more power. For more power transmission more torque is required. This torque transmission is done by the transmission system means gearbox. For reducing to that torque requirement first we need to calculate torque requirement for first forward gear and reverse gear that is done in following topic.

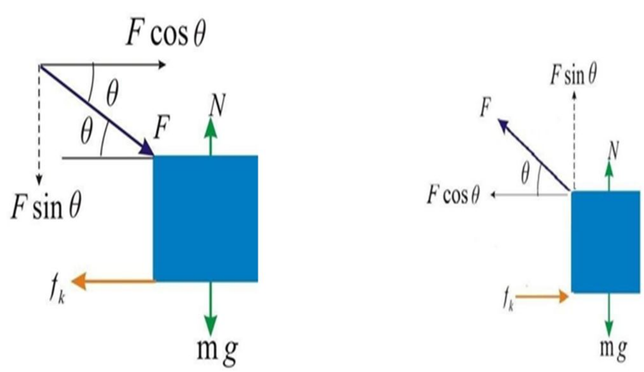

Fig. 1 Free-Body diagram for push and pull

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B. Drawbar Pulling and Pushing Forces

DP = T*R/(r-RR)

RR = GVW*R/1000

RR = rolling resistance, R = gear reduction ratio, T = engine torque, r = radius of the drive tire)R = Rolling resistance on the surface

GVW = Gross vehicle weight

Torque (N.m) = 95.488 * power (kw) / speed (rpm) Torque = 232.2 N.m

Power = 8.48 kw

RR=3100*0.4/1000 RR=4.05

DP=232*3.8/(14-4.05) DP=110.2 N.m

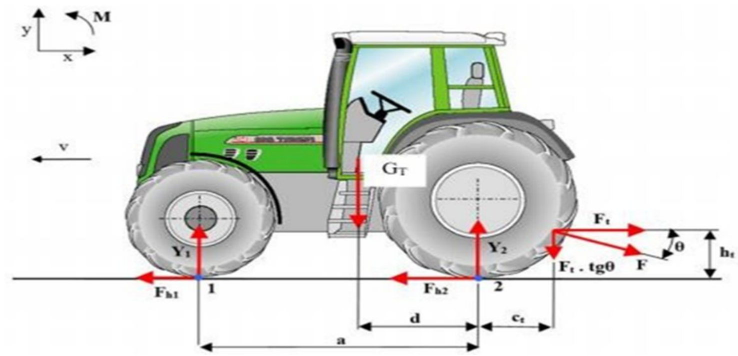

Fig. 2 Drawbar Push and Pull Force

As per the block data if we change force direct at angle 5 the change in torque the forward and reverse is about 8%. So the require pushing force is 108.82 N.m. The new, R (gear reduction ratio) need change 3.8 to 4.2 As per our new design we change the module of the gear and make it 3.5 and gear for the gear teeth is 40 and pinion 13. The available torque after changing the gear ratio is 110.89 N.m. Which is in the feasible range.

TABLE 1

Trailer's dimensions and load capacity

Particular Type	Single Axle 2- wheeler semitrailer	
Dimension	Overall length	3100 mm / 4025 mm(chassis)
	Overall Width	1900 mm
	Overall height	1700 mm
Load Capacity	Max. load	60 KN
	Trailer weight load	13 KN
	Gross load	73 KN
Axle	M.S. Axle beam of square cross section having side width of 75mm & length 1700mm.	
Tires	2- Wheels	10" (width) X 20" (radius)

C. Current Gear Data Used in Tractors

FIG. 3 Main shaft & reverse shaft with gears

Figure 3 consist of main shaft, reverse shaft and idle gear of the current deign gearbox, which is used by a company. In this 4 forward gear pairs and 1 reverse gear pair. This design is a semi- synchronized gearbox. In this 1-4 forward gears are connected in synchronize mechanism and reverse gear have sliding gear mechanism so it has separate idle gear. 4 Different gear ratios for various speeds. For reverse gear it is same as 1st forward gear ratio. Details of number of teeth on pinion and gear, diameter of that pinion and gear, module and different gear ratios as per the reduction in speed are given in the following table 2. Here module will be same for all gear pairs which is 3.7.

TABLE 2
DATA OF GEARS

Gear No.	Zg	Zp	Dg	Dp	Gear ratio
1 (Reverse)	37	13	137	50	3.80
1 (Forward)	37	13	137	50	3.80
2	33	19	126	73	2.37
3	27	25	103	95	1.26
4	22	30	82	114	1

III. IMPLEMENTATION OF DESIGN

A. Calculation for Gears for Reverse transmission

1) *Pinion C45 steel*
a) σ_u = 630MPa HB = 215
b) GEAR C45 steel
c) σ_b = 210 MPa, E=2.15 X 105 N/ mm^2 GEAR RATIO i = 3.5
2) *Number Of Teeth*
a) Teeth on Pinion = 13 Teeth on Gear = 40
3) *Tangential Load*
b) Power to be transmitted = 8.94 KW, Input speed =2660 rpm
c) Ko =1 (steady load) F_t=(P/V) K_o
d) F_t= [(8940)/ [(π m (13) (2600))/ (60 x103)] x 1
e) V = [π d1N1/ (60 x 103)] F_t = 1350.45/m
f) F_t = 385.84 N
4) *Initial Dynamic Load*
g) $F_d = F_t \times C_v$
h) Assume, V_m = 3 m/s $V_m <$ 10 m/s F_d = (1350.4/3.5) x 2
i) C_v = (3+Vm)/3, F_d = (2011.3/m) C_v= 2
5) *Beam Strength*
a) S = [σ_b] by π m [$\sigma_b = \sigma_o/3$] [σb] = [(σ_u)/3] = 210mPa
b) Face value, b =30 mm Pitch diameter, d= 140mm Velocity, v= 2.237 m/s
6) *Recalculated Beam Strength*
a) F_s= [210 x 15 x0.13 x π x 3.5] F_s=4500.405 N
7) *Accurate Dynamic Load*
a) $F_d = F_t$+ [21V (b1F_t)] F_t = {p/V} = 670.54
b) B = 30mm
c) V = 2.237 m/s Assume Carefully
d) Cut Gear: e=0.025 and c=296.5

F_d = 385.84 + [(46.977(51118.04))/46.977+71.54)] F_d = 3699.19 N

$F_d < F_s$

Design is safe. Module of reverse gear pair is changed from 3.7 to 3.5 as shown by CAD geometry in Fig. 4 (Table 3)

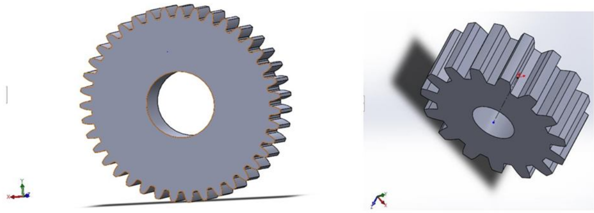

FIG. 4 Reverse gear of main shaft & idle gear

TABLE 3
Data Of Gears Changed After Deisgn Calculations

Gear No.	Zg	Zp	Dg	Dp	Gear ratio
1 (Reverse)	40	13	140	46	4.20
1 (Forward)	37	13	137	50	3.80
2	33	19	126	73	2.37
3	27	25	103	95	1.26
4	22	30	82	114	1

B. Engine Shaft Design Improvement

1) N (min) = 2660 rpm $M_t = P \times 60/ 2\pi$

   $N = 1.3 \times 10^3 \times 60/(2 \times \pi \times 2600) = 67.74 \times 10^3$ N.mm

   $P_t = 2M_t /D = 2 \times 67.10 \times 10^3/38 = 3531.7$ N $P_n = P_t /\cos 20 = 37583.56$ N

   $Mb = P_nL/4 = 37583.56x100/4$ $M_b = 26.742x103$ N.mm

   $M_{teq} = \sqrt{}M_b^2 + M_t^2$

   $M_{teq} = 62.86x10$^3 N.mm

2) $T = 16M_t/\pi d^3 = 55 = 16 \times 32.86 \times 10^3 /\pi$ d3

   d = 34.89mm

   d = 35mm NHW=529 rpm

3) $M_t=85.09x10^3$ N.mm

4) $P_t = [(2 \times 27.09 \times 103)]$

   $Pn=(P_t)/\cos 20=1067.7$ N

   $M_b=(P_n \times L)/4 = 66.09 \times 10^3$ N.mm

   $M_{tor}= 78.031$ N.mm

To improve gearbox design, we need to change reverse gear mechanism by changing reverse gear in synchronizing manner with the main shaft. Due to this reverse gear which is attached to main shaft is rotating in reverse direction. It creates relative speed is doubled compare to first gear rotation. Tractor is working on low speed, main problem is torque sustainability. So to solve this we need to redesign main shaft as per the change. For that we need to increase diameter of main shaft rod at reverse gear placement and provide with the higher speed bearing which help to reduce the friction and heating problem. Main Shaft need to design as per the maximum torque transfer capacity. Also because company buy most of the standard product separately like buying gears, main shaft, reverse shaft and then assemble it. We redesign the most of the part which help to improve the efficiency and also more durable. So, based on that part we redesign the shaft which suitable for the fully-synchronize gearbox (Fig. 5 and 6)

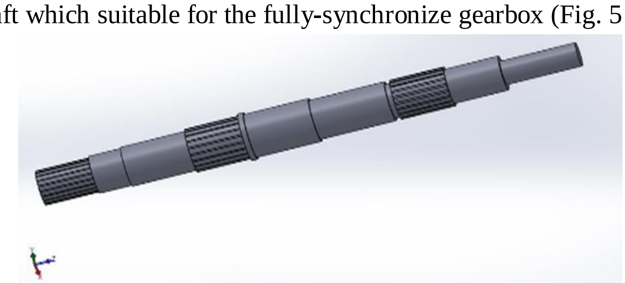

FIG. 5 CAD DESIGN OF MAIN SHAFT

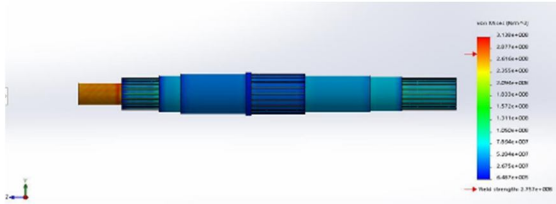
Fig. 6 Main shaft for maximum yield strength

Here reverse shaft also changed and placement of gears changed due to make this mechanism fully synchronize. This above figure of reverse shaft position of gears changed and that is in the following manner from right hand:

1st – 1st (reverse) – 2nd – 3rd – 4th

Based on the reverse gear high relative motion which cause the vibration to prevent the vibration for certain level it place between the 2nd gear and 1st gear. Also gear changing slide also need to redesign based on the new design the reverse and 2nd gear have share same slider for the first gear slider used on the other side. (fig. 7 and 8)


Fig. 7 CAD Design of REVERSE SHAFT

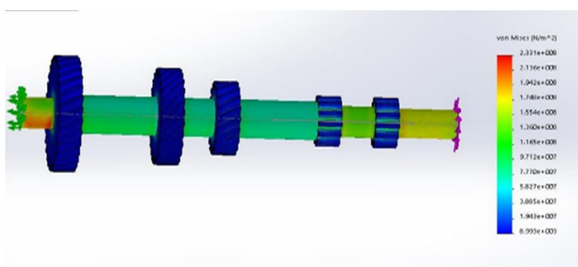
Fig. 8 Reverse shaft for maximum yield strength

C. Selection of Bearing

Needle bearing more reliable because it has a higher supports load and also have longer life in oil with foreign material. Also support higher load and high reliability.

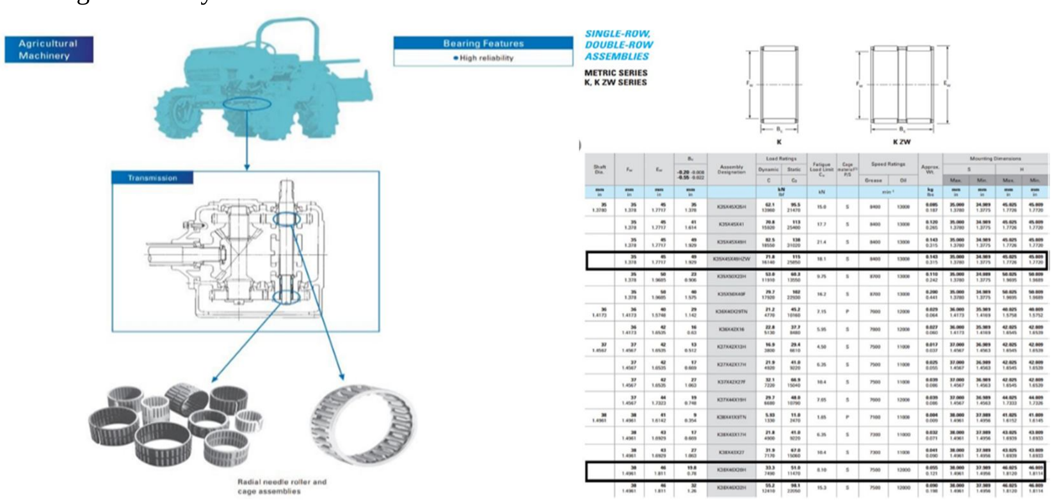

Fig. 9 Needle Bearing and Catalogue for Selection [6-7]

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D. Synchronisers

Synchronizers can be structured by the number of cones used. The single-core, dual-core and triple-cone synchronizers and the descriptions of the single components. The synchronization process always follows the same sequences. The sleeve is moved by the shift fork towards the gear to be engaged. As long as there is a speed difference between the sleeve/hub-system and the gear wheel the sleeve is blocked by the blocker ring and the synchronizer rings create a friction torque. Best synchronizer for this transmission system is single core synchronizer. Which is given in following figure. When the speeds are synchronized the sleeve can be moved further and engages into the spline of the engagement ring at the gear wheel. Following figure 10 shows that how to place gears on shaft and which are the needed part for attachment like, hub, insert spring, sleeve, synchronize ring, needle bearing, etc as shown in Fig. 9. Fig. 10 shows the final assembly of transmission system.

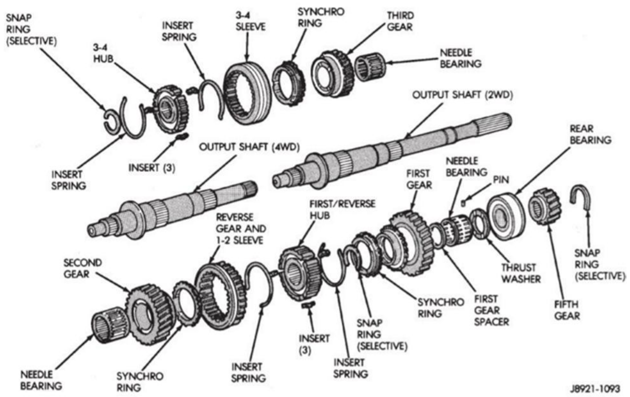

Fig. 10 Gear attachment on shaft

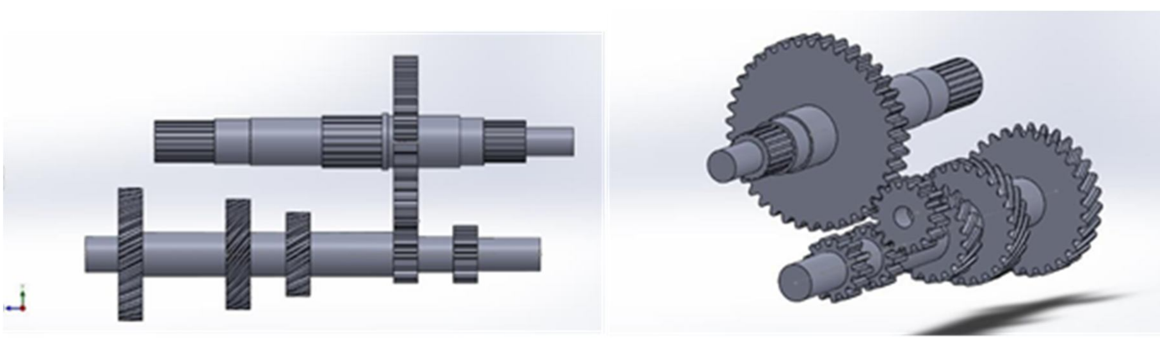

Fig. 11 Synchronized gear mechanism

IV. CONCLUSIONS

For producing required torque we changed the gear ratio of the reverse gear from 3.8 to 4.2 Also to avoid interfacing we also changed the module of the gear which change from the module of the gear form 3.7 to 3.5 By changing the module the gear is also weaken so we also need to change the material form higher standard material C45 steel. Changing the module gear engaging and interference problem solved. By changing the gear ratio from 3.8 to 4.2 we acquired the pushing force of 110.89 N.m. Which is more feasible from the calculations. The different torque generate by reverse and forward gear. Also changing design of shafts make gear train fully synchronized, but design of chasing, gears for forward transmission will remain same. Other parts like synchronize ring, hub, insert ring etc. will also remain same. Which is efficient to do the above changes in the design of transmission system.

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