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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 increase 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 $F \sin \theta$ acts downwards along with the weight $m \cdot g$ 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 $F \sin \theta$ acts upwards along with the weight $m \cdot g$ 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.

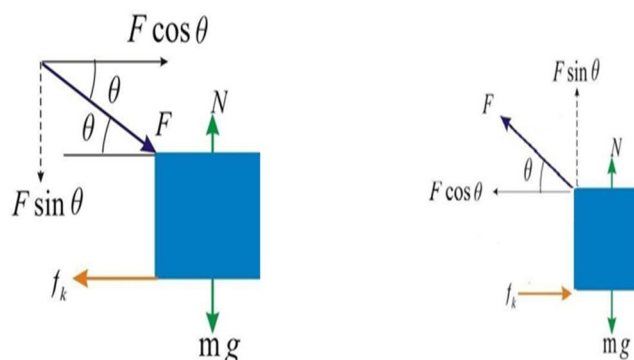


Fig. 1 Free-Body diagram for push and pull

B. Drawbar Pulling and Pushing Forces

$$DP = T \cdot R / (r \cdot RR)$$

$$RR = GVW \cdot R / 1000$$

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

$$\text{Torque (N.m)} = 95.488 \cdot \text{power (kw)} / \text{speed (rpm)} \quad \text{Torque} = 232.2 \text{ N.m}$$

$$\text{Power} = 8.48 \text{ kw}$$

$$RR = 3100 \cdot 0.4 / 1000 \quad RR = 4.05$$

$$DP = 232 \cdot 3.8 / (14 \cdot 4.05) \quad DP = 110.2 \text{ N.m}$$

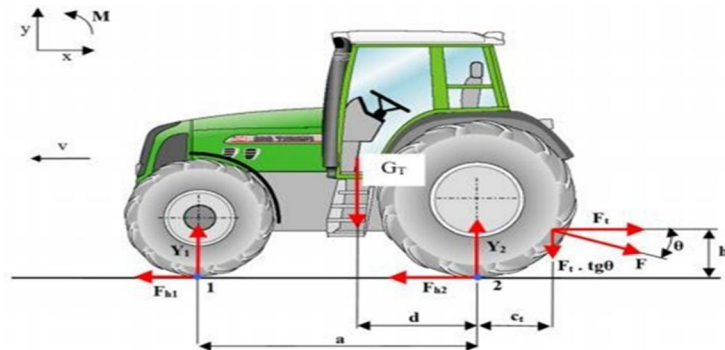


Fig. 2 Drawbar Push and Pull Force

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) $\sigma_u = 630\text{MPa}$ HB = 215

b) GEAR C45 steel

c) $\sigma_b = 210\text{ MPa}$, $E=2.15 \times 10^5\text{ N/mm}^2$ 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) $K_o = 1$ (steady load) $F_t = (P/V) K_o$

d) $F_t = [(8940) / [(\pi m (13) (2600)) / (60 \times 103)]] \times 1$

e) $V = [\pi d1N1 / (60 \times 103)] F_t = 1350.45/m$

f) $F_t = 385.84\text{ N}$

4) Initial Dynamic Load

g) $F_d = F_t \times C_v$

h) Assume, $V_m = 3\text{ m/s}$ $V_m < 10\text{ m/s}$ $F_d = (1350.4/3.5) \times 2$

i) $C_v = (3+V_m)/3$, $F_d = (2011.3/m)$ $C_v = 2$

5) Beam Strength

a) $S = [\sigma_b]$ by πm $[\sigma_b = \sigma_o/3]$ $[\sigma_b] = [(\sigma_u)/3] = 210\text{mPa}$

b) Face value, $b = 30\text{ mm}$ Pitch diameter, $d = 140\text{mm}$ Velocity, $v = 2.237\text{ m/s}$

6) Recalculated Beam Strength

a) $F_s = [210 \times 15 \times 0.13 \times \pi \times 3.5] F_s = 4500.405\text{ N}$

7) Accurate Dynamic Load

a) $F_d = F_t + [21V (b1F_t)] F_t = \{p/V\} = 670.54$

b) $B = 30\text{mm}$

c) $V = 2.237\text{ m/s}$ Assume Carefully

d) Cut Gear: $e = 0.025$ and $c = 296.5$

$F_d = 385.84 + [(46.977(51118.04))/46.977+71.54] F_d = 3699.19\text{ N}$

$F_d < F_s$

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)

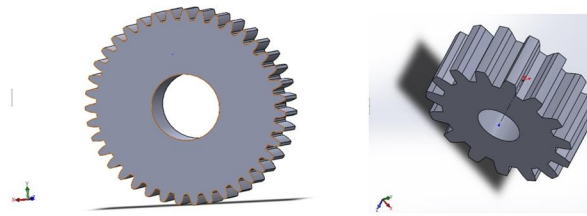


FIG. 4 Reverse gear of main shaft & idle gear

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

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

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

$M_b = P_n L / 4 = 37583.56 \times 100 / 4$ $M_b = 26.742 \times 10^3 \text{ N.mm}$

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

$M_{teq} = 62.86 \times 10^3 \text{ N.mm}$

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

$d = 34.89 \text{ mm}$

$d = 35 \text{ mm}$ $NHW = 529 \text{ rpm}$

3) $M_t = 85.09 \times 10^3 \text{ N.mm}$

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

$P_n = (P_t) / \cos 20 = 1067.7 \text{ N}$

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

$M_{tor} = 78.031 \text{ N.mm}$

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)

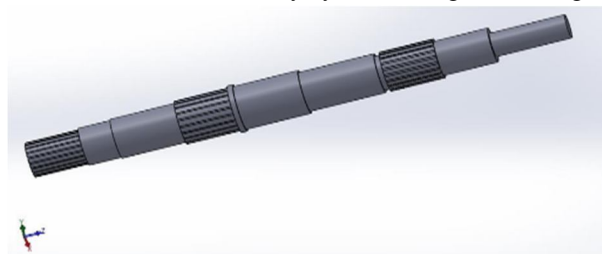


FIG. 5 CAD DESIGN OF MAIN SHAFT

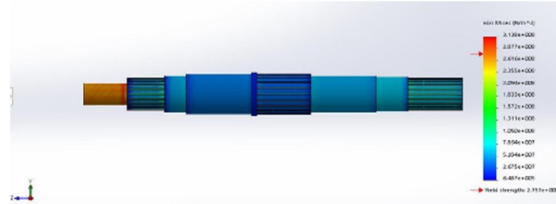


Fig. 6 Main shaft for maximum yield strength

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

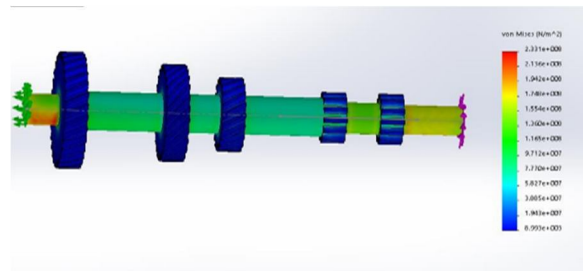



Fig. 8 Reverse shaft for maximum yield strength

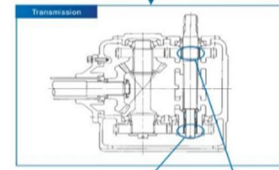
C. Selection of Bearing

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.


Agricultural Machinery



Transmission



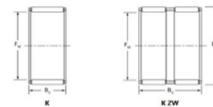
Radial needle roller and cage assemblies



Bearing Features

- High reliability

SINGLE-ROW, DOUBLE-ROW METRIC SERIES K, K ZW SERIES



Bore Dia. (mm)	Bore Dia. (in)	Outer Dia. (mm)	Outer Dia. (in)	Bore Tolerance (mm)	Bore Tolerance (in)	Outer Tolerance (mm)	Outer Tolerance (in)	Pitch Circle Diameter (mm)	Pitch Circle Diameter (in)	Number of Balls	Pitch Diameter (mm)	Pitch Diameter (in)	Speed Ratings		Approx. Weights		Mounting Dimensions				
													min. (mm)	max. (mm)	min. (kg)	max. (kg)	min. (mm)	max. (mm)	min. (mm)	max. (mm)	
10	0.3937	16	0.6299	0.008	0.0003	0.012	0.0005	12.5	0.4921	10	12.5	0.4921	10	1000	1500	0.015	0.025	10	10	10	10
12	0.4724	18	0.7087	0.008	0.0003	0.012	0.0005	15	0.5906	12	15	0.5906	12	1000	1500	0.02	0.035	12	12	12	12
15	0.5906	22	0.8661	0.008	0.0003	0.012	0.0005	18	0.7087	15	18	0.7087	15	1000	1500	0.03	0.05	15	15	15	15
18	0.7087	28	1.1024	0.008	0.0003	0.012	0.0005	22	0.8661	18	22	0.8661	18	1000	1500	0.045	0.075	18	18	18	18
20	0.7874	30	1.1811	0.008	0.0003	0.012	0.0005	24	0.9448	20	24	0.9448	20	1000	1500	0.055	0.09	20	20	20	20
22	0.8661	32	1.2598	0.008	0.0003	0.012	0.0005	26	1.0235	22	26	1.0235	22	1000	1500	0.065	0.105	22	22	22	22
25	0.9843	36	1.4173	0.008	0.0003	0.012	0.0005	30	1.1811	25	30	1.1811	25	1000	1500	0.09	0.14	25	25	25	25
28	1.1024	40	1.5748	0.008	0.0003	0.012	0.0005	34	1.3387	28	34	1.3387	28	1000	1500	0.12	0.18	28	28	28	28
30	1.1811	42	1.6535	0.008	0.0003	0.012	0.0005	36	1.4173	30	36	1.4173	30	1000	1500	0.135	0.2	30	30	30	30
32	1.2598	44	1.7322	0.008	0.0003	0.012	0.0005	38	1.496	32	38	1.496	32	1000	1500	0.15	0.22	32	32	32	32
35	1.3679	48	1.9297	0.008	0.0003	0.012	0.0005	42	1.6535	35	42	1.6535	35	1000	1500	0.18	0.26	35	35	35	35
38	1.496	50	2.0084	0.008	0.0003	0.012	0.0005	44	1.7322	38	44	1.7322	38	1000	1500	0.2	0.28	38	38	38	38
40	1.5748	52	2.0871	0.008	0.0003	0.012	0.0005	46	1.8109	40	46	1.8109	40	1000	1500	0.22	0.3	40	40	40	40
42	1.6535	54	2.1658	0.008	0.0003	0.012	0.0005	48	1.8896	42	48	1.8896	42	1000	1500	0.24	0.32	42	42	42	42
45	1.7717	56	2.2445	0.008	0.0003	0.012	0.0005	50	1.9683	45	50	1.9683	45	1000	1500	0.27	0.35	45	45	45	45
48	1.89	60	2.402	0.008	0.0003	0.012	0.0005	54	2.1258	48	54	2.1258	48	1000	1500	0.31	0.4	48	48	48	48
50	1.9683	62	2.4807	0.008	0.0003	0.012	0.0005	56	2.2045	50	56	2.2045	50	1000	1500	0.33	0.42	50	50	50	50
52	2.047	64	2.5594	0.008	0.0003	0.012	0.0005	58	2.2832	52	58	2.2832	52	1000	1500	0.35	0.44	52	52	52	52
55	2.1658	68	2.7169	0.008	0.0003	0.012	0.0005	62	2.4407	55	62	2.4407	55	1000	1500	0.39	0.48	55	55	55	55
58	2.2445	70	2.7956	0.008	0.0003	0.012	0.0005	64	2.5194	58	64	2.5194	58	1000	1500	0.41	0.5	58	58	58	58
60	2.3232	72	2.8743	0.008	0.0003	0.012	0.0005	66	2.5981	60	66	2.5981	60	1000	1500	0.43	0.52	60	60	60	60
62	2.402	74	2.953	0.008	0.0003	0.012	0.0005	68	2.6768	62	68	2.6768	62	1000	1500	0.45	0.54	62	62	62	62
65	2.5209	76	3.0317	0.008	0.0003	0.012	0.0005	70	2.7555	65	70	2.7555	65	1000	1500	0.47	0.56	65	65	65	65
68	2.6396	80	3.1892	0.008	0.0003	0.012	0.0005	74	2.913	68	74	2.913	68	1000	1500	0.51	0.6	68	68	68	68
70	2.7179	82	3.2679	0.008	0.0003	0.012	0.0005	76	2.9917	70	76	2.9917	70	1000	1500	0.53	0.62	70	70	70	70
72	2.7966	84	3.3466	0.008	0.0003	0.012	0.0005	78	3.0704	72	78	3.0704	72	1000	1500	0.55	0.64	72	72	72	72
75	2.9153	88	3.5041	0.008	0.0003	0.012	0.0005	82	3.2279	75	82	3.2279	75	1000	1500	0.59	0.68	75	75	75	75
78	3.034	90	3.5828	0.008	0.0003	0.012	0.0005	84	3.3066	78	84	3.3066	78	1000	1500	0.61	0.7	78	78	78	78
80	3.1127	92	3.6615	0.008	0.0003	0.012	0.0005	86	3.3853	80	86	3.3853	80	1000	1500	0.63	0.72	80	80	80	80
82	3.1914	94	3.7402	0.008	0.0003	0.012	0.0005	88	3.464	82	88	3.464	82	1000	1500	0.65	0.74	82	82	82	82
85	3.3097	96	3.8189	0.008	0.0003	0.012	0.0005	90	3.5427	85	90	3.5427	85	1000	1500	0.67	0.76	85	85	85	85
88	3.4284	100	3.9764	0.008	0.0003	0.012	0.0005	94	3.7002	88	94	3.7002	88	1000	1500	0.71	0.8	88	88	88	88
90	3.5067	102	4.0551	0.008	0.0003	0.012	0.0005	96	3.7789	90	96	3.7789	90	1000	1500	0.73	0.82	90	90	90	90
92	3.5854	104	4.1338	0.008	0.0003	0.012	0.0005	98	3.8576	92	98	3.8576	92	1000	1500	0.75	0.84	92	92	92	92
95	3.7037	108	4.2913	0.008	0.0003	0.012	0.0005	102	4.0151	95	102	4.0151	95	1000	1500	0.79	0.9	95	95	95	95
98	3.8224	110	4.3700	0.008	0.0003	0.012	0.0005	104	4.0938	98	104	4.0938	98	1000	1500	0.81	0.92	98	98	98	98
100	3.9011	112	4.4487	0.008	0.0003	0.012	0.0005	106	4.1725	100	106	4.1725	100	1000	1500	0.83	0.94	100	100	100	100

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.

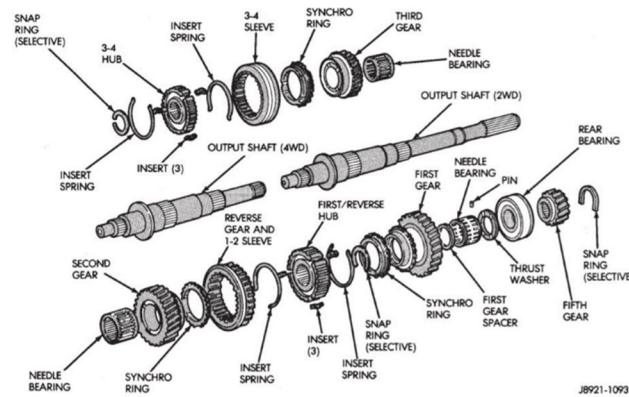


Fig. 10 Gear attachment on shaft

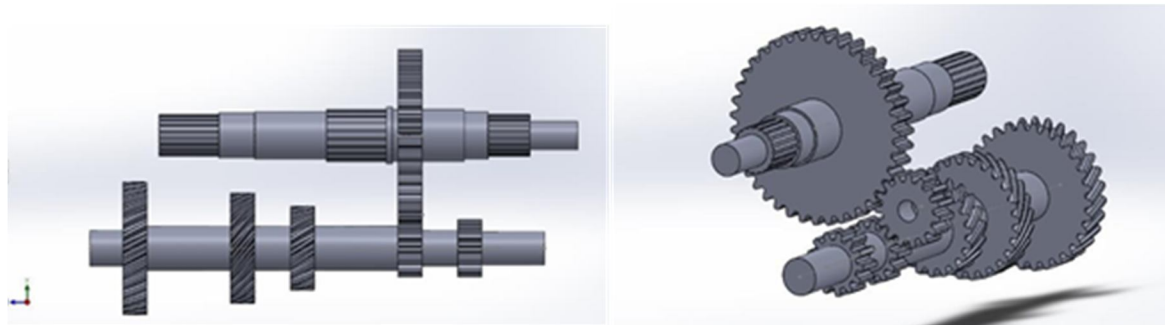


Fig. 11 Synchronized gear mechanism

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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