Scientific Journal of Impact Factor (SJIF): 4.72

e-ISSN (O): 2348-4470
p-ISSN (P): 2348-6406

International Journal of Advance Engineering and Research Development

Volume 5, Issue 01, January -2018

Design of Two Stage Planetary Gear Train for High Reduction Ratio

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Abstract:- *Presently for the automation of car parking system in the multistory building like malls, Cinema Theater at helical gear motor of 1.5 kW, input power 1410 rpm as input 25% rpm with safety factor 1.5 used. The gearbox for this application shall be compact, should have high torque density shall be lubricated for life and operation shall be noiseless. But this is not possible in helical gear system as planetary gearboxes can use frequently to match the inertias, lower the motor speed, boost the torque, and at the same time provide a sturdy mechanical interface for pulleys, cams, drums and other mechanical components. This paper presents the design of planetary gear systems, consideration to be given while determining the reduction ratios of the gear box, minimum and maximum reductions per stage of planetary gear pairs. Also, the case study to determine the reduction ratio of the two-stage gearbox is discussed. The planetary gear system is designed, stimulated through SolidWorks and actual testing is done.*

Keywords –*Planetary Gears, Gear train, IS 4460, Torque density.*

I. INTRODUCTION

Gears are defined as toothed wheel or multi-lobed coins which transmit power and motion from one shaft to another by means of successive engagement of teeth gears are classified according to the relative position of shaft -

1. Spur gears- In spur gears, the teeth are cut parallel to the axis of the shaft. As the teeths are parallel to the axes at the shaft spur gears are used only when shafts are parallels.
2. Helical gears- The teeths of gears are cut at the angle with the axis of the shaft the angle is known as helix angle. The shafts are parallel to each other in this the profile is involute in a plane perpendicular to the tooth element.

Epicyclic or planetary gear train -

Planetary gear system normally consists of a centrally pivoted sun gear a ring gear and several planets found blow sun and ring gear now industrial applications demands high torque in contact a high torque /volume and light a high torque to weight ratio in planetary gears the torque density can be increasing adding more planets a planetary gear with say three planets can transfer three times the torque of a similar sized fixed axis standard spur gear system. The applied load to planetary gears are distributed onto multiple gear mesh points means the load is supported by n contacts (n = no planet gears) increasing the torsional stiffness at gear train by factor n. hence it lowers the lost motion compared to similar size standard gear trains.

High rotational stiffness is important for application of positioning accuracy and repeatability requirements, especially under fluctuating loading condition. Hence planetary gears are used for such application.The smaller gears in the planetary system result in lower inertia.Compared to same torque rating gearbox, planetary is smaller by a square of number of planets.The load is branched into multiple gears meshes locations.

II. SELECTION OF GEAR BOX FOR THE OPERATION

Here in this condition, the torque density is high, so an Epicyclic gear train is used which will easily tolerate the fluctuating load, have High torque density and compactness in design can be obtained. It is having compaction which gives Light weight. It have High torsional stiffness, High repeatability and positioning accuracy, Low volume, and Low inertia.

To design a gear train for input speed 1410 rpm and output speed 25 rpm.

Reduction ratio

$$R = \frac{\text{Input rpm}}{\text{Output rpm}}$$
$$= \frac{1410}{25}$$
$$= 56.4$$

To determine the number of teeth.

Let initial gear ratio be 10:1

There are three types of Epicyclic arrangements

1. Planetary – In this type of arrangement, the input shaft is connected to the sun gear, the output is obtained from the planets. The internal gear in the annular ring is held fixed. Due to space

consideration and size of gears, the maximum ratio is 12:1. Gear ratio lies between 3:1 to 12:1. If gear ratio exceeds 12:1 the size of planet gear increases, so planet size increases and the arrangement becomes more complicated so gear ratio is usually below 12:1

2. Star arrangement - In a star arrangement, the carrier is fixed. The sun and planet relative speeds are determined by the numbers of teeth in each gear. Here the ratio lies between 2:1 and11:1
3. Solar arrangement- In this arrangement the internal gear is fixed input is given to the planets and output is taken from the sun. The ratio range is between 1.2:1 and 1.7:1

So for the first stage, we select planetary arrangement
Determining number of teeth
For ratio 10:1 maximum no of planets can be 3 (because these are having greater interference).

$$D_s + 2\,D_p = d_a$$
$$Z_1 + 2\,Z_2 = Z_3$$

Let number of teeth for 20 pressure angle for sun be 18

$$Z_1 = 18$$
$$18 + 2\,Z_2 = Z_3$$
$$\text{Gear ratio for epicyclic } = \frac{PCD\ of\ ring + PCD\ of\ sun}{PCD\ of\ sun}$$

$$10 = 1 + \frac{Na}{18} \text{Where } N_a = Z_3$$
$$9{\times}18 = N_a = Z_3$$
$$N_a = 162$$
$$Z_3 = 162 \text{ teeths}$$
$$Z_2 = 72 \text{ teeth}$$

Therefore, $Z_1 = 18$ $Z_2 = 72$ $Z_3 = 162$ teeths

III. MATERIAL SELECTION

The materials which are used for the gears depends upon the service factor and strength like wear or noise conditions etc. and they came in metallic and nonmetallic form.Our application is industrial use so metallic gear are selected. Commercially available metals are steel, ci and bronze. In these case hardening steel is widely used for industrial gears because of its excellent wearing properties and high bending strength. So we also case hardening steel 20mncr5 as per pin 17210 with the tensile strength of 1200mpa.

Reasons –

1. Low production cost
2. High stability
3. Surface finish.

Case carburizing and hardening to high wear resistance for long service of life. The hardness of 56 to 63 HRC is selected as per AGMA-6019
Lewis form factor for 18 teeths and20 full depth involute,

$$V = 0.3204$$
$$V = \frac{\pi d_p\ N}{60000}$$
$$V = 1.328 \text{ m/s}$$

Velocity factor, which gear is manufactured by hobbing, shaping and milling. BHN of material is 600

$$Cv = \frac{6}{6 + v}$$
$$Cv = \frac{6}{6 + 1.328m}$$

Now, $$Fw = dp * b * Q * k$$
$$Q = 2 * \frac{zg}{zg + zp}$$
$$Q = 1.6$$
$$K = 0.16 * \left(\frac{BHN}{100}\right)^2$$
$$K = 5.76$$
$$\text{Let } b = 10 \text{ m}$$
$$Fw = dp * b * Q * k$$
$$Fw = 1658.88\ m2\ N$$

$\sigma_p\ Y_p < \sigma_g\ Y_g$ for same material, pinion is weaker.

For, $$Fb = \sigma * b * m * y$$
$$= \frac{Sut}{3} * 10 * m * m * \left[0.484 - \frac{2.87}{zp}\right]$$

$$=1298\ m^2$$

Where, $Fb<Fw$

The gear pair should be designed for bending,

$$Feff = \left(Cs * \frac{Cm}{Cv}\right) * Ft$$

For heavy shock and uniform operation Here $C_s = 1.75$

$$Feff = \left(1.75 * \frac{1.3}{\frac{6}{6+1.328*m}}\right) * \left(\frac{1128.75}{m}\right)$$

$$2.02 * m2 = 6 + 1.328 * m$$

$$m = 2$$

Now, $c = K_e\left[\frac{E_p.E_g}{E_p+E_g}\right]$

$$c = 11655e\ \ N/mm$$

$$\phi_p= m+0.25\sqrt{d_p}$$

$$e_p= 20.375\ microns$$

$$e_g= 16.1.25(m+0.25\sqrt{d_g})=22.25\mu m$$

$$E=e_p+e_g$$

$$= 42.65*10^{-3}\ mm$$

$$c = 11655(e_p*e_g)$$

$$c = 496.79\ N/mm$$

Force is, $F_d=\frac{21V(bc+F_{tmax})}{21V+\sqrt{bc+F_{tmax}}}$

$$F_{tmax} = C_s*C_m*F_t$$

$$F_{tmax}= 1283.95\ N$$

$$V = 2.656\ m/s$$

Effective force,

$$F_d=\frac{21V(bc+F_{tmax})}{21V+\sqrt{bc+F_{tmax}}}$$

$$B=10m=20mm$$

$$C=496.79\ N/mm$$

$$F_d=3870.10\ N$$

$$F_{eff}= F_{tmax}+F_d$$

$$=5154.05\ N$$

Force at bearing $F_b = N_f*F_{eff}$

$$N_f = 1.007$$

$$m = 2.5$$

$$F_{tmax}= K_a*K_m*F_t$$

$$F_{tmax}=1027.16\ N$$

$$V = 3.32\ m/s$$

$$C = 11665\ e\ N/mm$$

$$d_p = 45\ mm$$

$$d_g = 180\ mm$$

$$e_p= 21.22\ m$$

$$e_g= 21.22\ m$$

$$F_{eff}= 6218.37N$$

$$F_b = N_f*F_{eff}$$

$$N_f = 1.3$$

Spur gear train

Beam strength, $Fb = \sigma * b * m * y$

$$Y = 0.484- \frac{2.87}{Z_p}$$

$$Fb = 1298\ m2N$$

Wear force,

$$F_w= d_pbQK$$

$$Q = \frac{2Z_g}{Z_g+Z_p}$$

$$F_w = 2068.56\ m^2\ N$$

Effective load

$$F_{eff} = \frac{K_a . K_m . F_t}{K_v}$$
$$F_{eff} = 0.1328\ m/s$$
$$F_b = N_f * F_{eff}$$
$$m = 3.1\ mm$$

Hence design is safe.

IV. DESIGN OF PINION SHAFT (2 STAGE)

This shaft connects the output of the planetary gear arrangement to the input cef the 2nd stage (i.e. pinion) the output of the planetary gear arrangement is taken from the planets. The material used is carbon steel (30c8) since the material used is ductile, max shear stress theory is used to design this shaft. The length of the shaft is taken as 150mm and the pinion is mounted at the end of the shaft.
Material - (30c8) carbon steel.
Yield strength (set) = 400mpa
From previous calculations
Torsional moment (Mt) =101575.1 Nmm

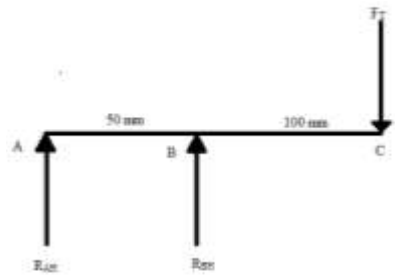
Tangential force, $F_t = \frac{M_T}{d_p/2}$

$= 2788.345$

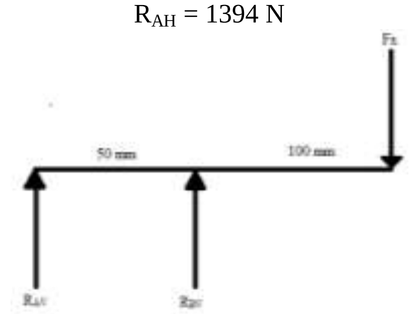
Radial force, $F_R = F_T \tan\phi$

$= 1014.873$ N

Since pinion is small, the weight of the pinion is negligible compared to the radial force. Hence, weight is neglected.
The tangential force will act in the horizontal plane while radial force will act in the vertical plane.
Horizontal plane



$$R_{AH} + R_{BH} = 2788.34$$
$$R_{BH} = 4182\ N$$
$$R_{AH} = 1394\ N$$



$$R_{AV} + R_{BV} = 1014.873\ N$$
$$R_{BV} = 1522.31\ N$$
$$R_{AV} = 504.435\ N$$

Bending movement will be maximum at B

$$M_{bH} = 50F_T = 139417\ mm$$

Maximum Bending movement

$$M_b = \sqrt{M_{bH}^2 + M_{bv}^2}$$
$$M_b = 148364.48\ Nmm$$

Maximum principle shear stress

$$\tau = 0.5 \frac{S_{yt}}{N_f}$$

$$\tau = 40 \text{ Mpa}$$

Equivalent torsional movement

$$\tau = \frac{16M_e}{\pi({d_o}^3 - {d_i}^3)}$$

$$d_e = 28.90 \text{ mm}$$

A standard shaft of on 30 is selected
Outer diameter =30mm
Inner diameter =18mm

V. SELECTION OF KEYS

Key is selected based on the values from the design data book. The material of the key is same as that of the shaft i.e. carbon steel.
Material = carbon steel 38c8
Width (w) =8mm height (h) =7mm length (l) =40mm
Checking for shear stress

$$\tau_{sk} = 0.3syt$$

$$\tau_{sk} = 120mpa$$

$$8.71 << \tau_{sk}$$

The key is safe in crushing stresses

VI. SECTION OF BEARING

6.1 Bearing a

From the shaft design calculation

$$R_A = \sqrt{{R_{AH}}^2 + {R_{AV}}^2}$$

$$R_A = 1482.46 \text{ N}$$

Based on our application, a load multiplication factor of 1.75 was selected.

$$P_N = 2594.31$$

Since our application will be used for approx.8 hrs. A day, the life of the bearing was selected as 18000hrs.

$$L_{10} = \frac{60n*L_{H10}}{10^6}$$

$$L_{10} = 152.28 \text{ million revolutions.}$$

$$L_{10} = \left(\frac{C}{P_A}\right)^3$$

$$C = 13853.82 \text{ KN}$$

The dynamic load capacity is 13.85 KN
Bore diameter =40mm
Based on our calculation, skf bearing c206 was selected.

6.2 Bearing B

$$R_B = \sqrt{{R_{BH}}^2 + {R_{BV}}^2}$$

$$R_B = 1.75\ R_B$$

$$L_{10} = \left(\frac{C}{P_B}\right)^3$$

L_{10} = 152.28 million revolutions.
Based on our calculations, SKF Bearing C4016 is selected.

6.3 Design of output shaft (2 stage)

Material is same for all the shaft of the entire gearbox. The length is taken as 200mm.
Material-carbon steel 30c8
Yield strength (set) =400mpa
Factor of safety = 5
From the gear design calculation, we found out that

$$M_t = 56*\text{Input Torque}$$

$$M_t = 568820.56 \text{ Nmm}$$

Tangential force,

$$F_t = \frac{M_t}{d_{g/2}} = 2788.34 \text{ N}$$

$$\text{Weight } w = \frac{\pi}{2} * d_g^2 * b * \rho * g$$

$$= 414.83 \text{ N}$$

We have selected a standard shaft of od = 50mm and id =30mm

VII. Selection of key

Key is made up of same material as that of the shaft i.e. carbon steel (30c8)
The dimensions of the key are selected from the design data book.
W = 14mm
H= 9 mm
Length= 40mm
Checking for shear stress

$$\tau_{sk} = 0.3*S_{yt} = 120 \text{ MPA}$$
$$\tau = \frac{F_t}{w*l} = 4.98\text{Mpa}$$
$$4.98 << 120 \text{ Mpa}$$

The key is safe
Checking for crushing stress

$$\sigma_c = 2\tau_{sk} = 240 \text{ MPa}$$
$$\sigma_c = \frac{2F_t}{h*l} = 15.49 \text{ MPa}$$
$$15.49 << 240 \text{ Mpa}$$

The key is safe in both the situation.
Bearing selection
Bearing at a
From the shaft design calculation

$$R_A = \sqrt{R_{AH}{}^2 + R_{AV}{}^2}$$
$$= 6267.02 \text{ N}$$
$$L_{10} = \frac{60n*L_{H10}}{10^6}$$
$$= 27 \text{ million revolutions.}$$
$$L_{10} = \left(\frac{C}{P_A}\right)^3$$
$$C = 32.901 \text{ KN}$$

Dynamic load capacity is 32.901KN. The bore diameter is 50mm. therefore the bearing 6210 satisfies our needs from catalogue.
Bearing at B

$$R_B = \sqrt{R_{BH}{}^2 + R_{BV}{}^2}$$
$$R_B = 5483.64 \text{ N}$$
$$L_{10} = \left(\frac{C}{P_A}\right)^3$$
$$C = 16.45 \text{ KN}$$

Based on the application, the load multiplication factor is 1.75
Dynamic load capacity is 16.45KN
Bore diameter is 50mm
The bearing 6010 from the catalogue is suitable for our application.

VIII. DESIGN OF CASING

The casing was designed with an aim of achieving compactness and providing fixed support to the bearings. Grey cast iron was chosen as the material for the casing due to the following reasons.

1. Easy castability
2. Vibration absorbing capability
3. Tough
4. Easily available and low cost.

In order to reduce the complications involved in the casing designing the 2nd bearing on the input shaft was fixed in the carrier connecting the 3 planets. Thus, the inner race rotates with the input speed (1410rpm). The 2nd stage of spur gears was kept out of the casing since its speed is very low (i.e. 250 rpm)
The casing is drum-shaped and made up of 2 parts which are bolted together during assembly. On one of the parts, holes are drilled along the outer circumference, which is used to bolt the internal gear and casing together. The input shaft bearing is press-fitted in the hub of this part. For the 2nd part, hubs are provided on both the sides. The 2 bearing of the intermediate shaft is fitted in these hubs.
The total size of the planetary gear arrangement is 550mm. therefore the casing was designed to drum shaped with diameter 650mm to provide sufficient clearance. The length of the drum is 140mm. bolting arrangements are made to fix it on the ground.

IX. 3D MODELLING

The 3D modelling is carried out with dimensions obtained analytically and modelling is done in SOLIDWORKS software with front, side view and exploded view of planetary gear box

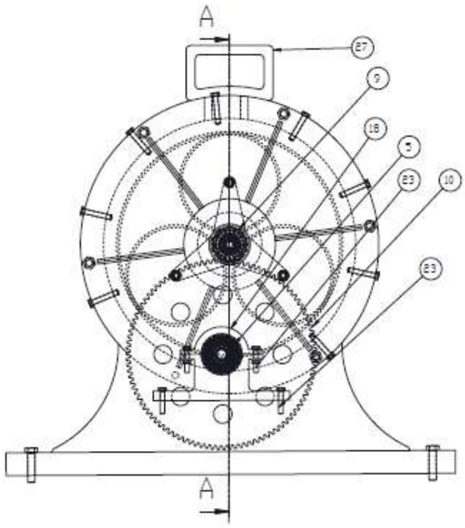


Fig 9.1 Front view of planetary gear box

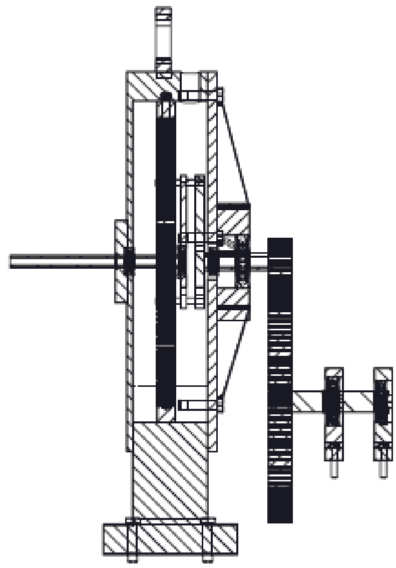


Fig 9.2 Side view of planetary gear box

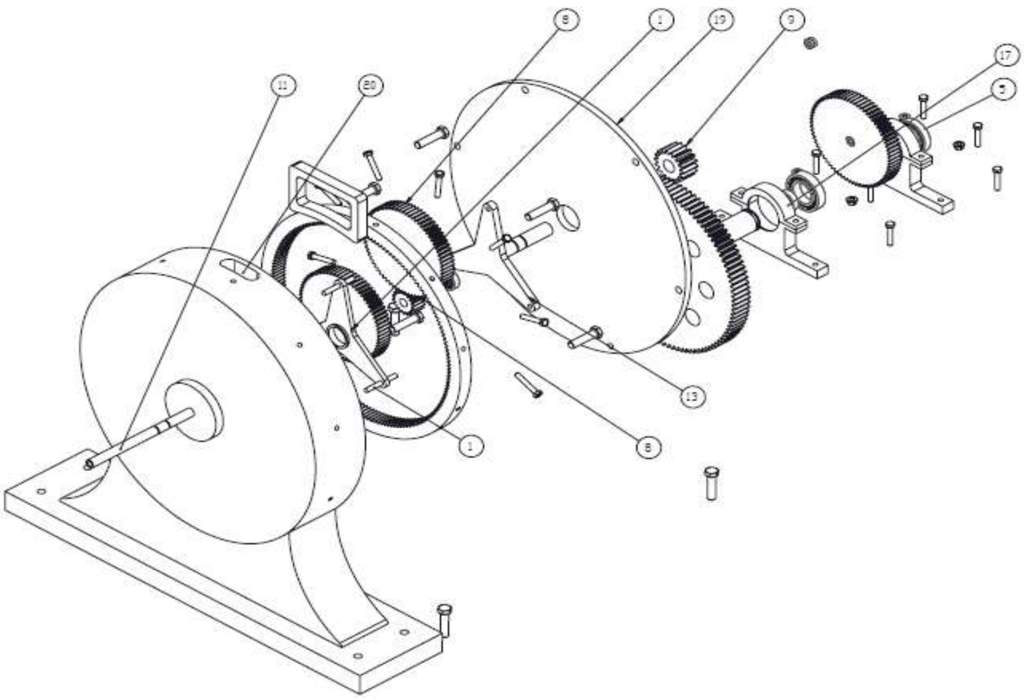


Fig 9.3 Exploded view of planetary gear box

X. LUBRICATION

Fixed axis spur gears will exhibit lubrication starvation and quickly fail it running at high speeds because lubrication is slung away. Hence pressurized forced lubrication system is required.
Grease lubrication is impartial is spur gears because of tunneling effect in which the grease lubrication overtime is pushed away. In and cannot flow back into the mesh. In planetary systems, the lubricant cannot escape it is continuously redistributed pushed or pulley or mired into gear contacts ensuring safe lubrication practically in any mounting position. Hence planetary gearbox can be grease lubricated for life. This feature is inherent in planetary gearing because of relative motion between different gears making up of the arrangement.

XI. MANUFACTURING PROCESS

Component	Material	Process
Sun, planets (spur gears)	20MnCr5	1. Blank forging, shaft hole 2. drilling, boring, reaming 3. keyways, slotting, finishing of flank 4. teeth rough and fine hobbing 5. Heat treatment, case hardening.
Internal gear	20MnCr5	1. blank forging 2. Slotting, finishing, milling 3. gear shaping 4. heat treatment case carburizing
Shaft	20MnCr5	1. hot rolling 2. cold rolling and precision 3. Turning, facing and finishing.
Carrier	20MnCr5	1. casting, milling 2. drilling 3. welding of pins
Casing cast iron (grey)	20MnCr5	1. casting, 2. milling 3. drilling and threading for holes

XII. CONCLUSION

Above study & results enables us to draw the following conclusion

1. Reduction ratio 78:1 is achieved in 2 stages for planetary gears which was in 3 stage with helical gears, hence planetary gear system becomes compact as one stages is eliminated.
2. Planetary gears have uniform strength (factor safety) in bending against equivalent helical gears.
3. The volume of planetary gear system for the given torque was 49 % less than equivalent helical gear system.
4. The weight of planetary gear system for the given torque was 49% less than equivalent helical gear system.
5. The torque density (Torque/volume or Torque/ weight) of planetary gear system for the given torque was 96% higher than equivalent helical gear system.

XIII. ACKNOWLEDGEMENTS

Thanks are expressed to the Maharashtra Institute of Technology, Pune stafffor allowing to study on this subject and their full hearted co-operation in providing all the necessary data and detailed guidance during this work.

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