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Design and Analysis of Planetary Gearbox for Industrial Concrete Mixer

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ABSTRACT- An epicyclic gear train or a planetary gear train is a gear train in which one gear is fixed and the meshing gear has a motion composed of two parts, namely, a rotation about its own axis and a rotation about the axis of the fixed gear. The fixed gear is called the sun gear, whereas the revolving gear is called the planet gear. Epicyclic gear trains achieve high reduction ratio in a small and power dense package, [1]. Gears in the Epicyclic gear trains are one of the most critical components in the mechanical power transmission system in which failure of one gear will affect the whole transmission system, thus it is very necessary to determine the causes of failure in an attempt to reduce them. The different modes of failure of gears and their possible remedies to avoid the failure are mentioned in J.R. Davis (2005), Khurmi & Gupta (2006), P. Kannaiah (2006) as bending failure (load failure), Pitting (contact stresses), scoring and abrasive wear. Epicyclic Gear Trains have been used in Industry for their many advantages which includes high torque capacity, comparatively smaller size, lower weight, improved efficiency and highly compact package [2]. This research paper helps in designing the planetary gear pair required for an industrial concrete mixer, since instead of giving an individual power source to cause the planetary motion, a gear drive can be used which will reduce the input power to be supplied to the machine, thus improving its efficiency of mixing or mechanical efficiency. This paper also helps in determining the probable cause by which the gear pair will fail, as well as countermeasures to prevent its failure.

KEYWORDS- Concrete, Gearbox, Industrial, Mixer, Planetary

I. INTRODUCTION:

It is examined that load sharing capability is not equal in the planetary gear train. These Gear Trains are extensively used for the transmission and are the most critical component in a mechanical power transmission system. They play a very vital role in all the industrial areas, any failure in the gear train leads to a total system failure, thus identifying the causes and optimizing to get the best performance is very necessary. The advantages of epicyclic gear trains are higher torque capacity, lower weight, small size and improved efficiency of the planetary design. As the weigh is 60%, and half the size of a conventional gear box, it is very likely to have a misconception that it is not as strong. Thus the loads have to be minimum to reduce the stresses in the gear train[2]

The analysis consists of finding out the module and dimensions of the gears required to achieve the necessary purpose i.e. concrete mixing. Since the gears are made of 50C4, and rotate at a very low rpm, their failure due to pitting or wear strength will not be the main criterion of consideration. Instead, the Lewis equation for bending strength will be the main criterion of design, and finding out the value of module from this equation, we will substitute it in the wear strength equation to find out the value of Brinell Hardness Number, and then compare it to the value of existing BHN of the material so as to determine the need of case hardening.

II. METHODOLOGY:

As we know that the gear is one of the most critical components of the power transmission system, failure in the gear will affect the whole transmission system and thus it is necessary to optimize the gear for low load operation and its effective delivery of power transmission. The gear design has been done in SolidEdge and the figure below shows the general layout of the gear pair that we have to design. [2]

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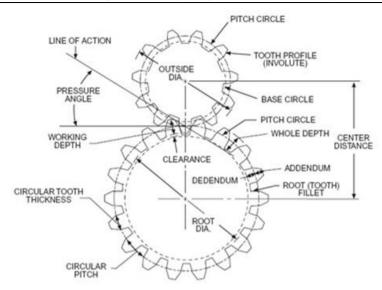


Figure 1: General Layout of Intended Gear Pair

The main acting loads on a gear pair are as the Tangential Load, the Effective Load, the Bending load or Beam Strength, and the Pitting or Wear Strength load.

Module: It is the ratio of the pitch circle diameter (in millimeters) to the number of teeth. It is usually denoted by m, where m = D / T D=Pitch Circle Diameter, T = Number of Teeth The recommended series of modules in Indian Standard are 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40 and 50. The modules 1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18, 22, 28, 36 and 45 are of second choice. [3], [4]

Systems of Gear Teeth:-The following four systems of gear teeth are commonly used in practice. $14 \frac{1}{2}^{\circ}$ Composite systems, $14 \frac{1}{2}^{\circ}$ Full depth involute systems, 20° Full depth involute system and 20° Stub involute system. The tooth profile of the 20° full depth involute system may be cut by hobs. The increase of the pressure angle from $14 \frac{1}{2}^{\circ}$ to 20° results in a stronger tooth, because the tooth acting as a beam is wider at the base. The 20° full depth involute system has a strong tooth to take heavy loads, thus was selected. [3], [4], [5]

Gear Material:-The materials which are used for the gears depend upon the service factor and strength like wear or noise conditions etc. and they come in metallic and non-metallic form. For industrial purposes metallic gears are used, commercially can be obtained in steel, cast iron and bronze. Out of these, 50C4 steel was chosen, which has Sut= 660MPa and Hardness= 250BHN. This material is chosen because it has high strength, toughness, can be accurately machined and be easily surface hardened. These properties are essential because in elements like gears, the surface is heavily stressed whereas the core stresses are comparatively less. [1]

III. DESIGN OF GEAR PAIR:

Tangential Load (Pt):- Tangential tooth load is also called the beam strength of the tooth. It is the load acting perpendicular to the radial tooth load (Pr) and normal tooth load (Pn) as shown in figure.

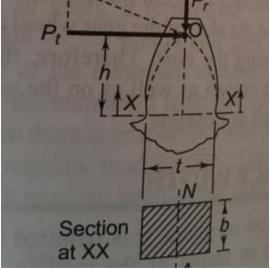


Figure 2: Tangential Tooth Load Direction

Teeth on pinion is chosen as 18, because for 20° full depth involute system, the minimum number of teeth required to prevent interference is 18. Since the required gear ratio is 3, the number of teeth on the gear are given by 18*3=54. The input power is 2 HP, which is approximately equal to 1.5 KW. The pinion rotates with 45 rpm around the gear and with 15 rpm about its own axis. Therefore, the higher of these 2 values i.e. 45 was taken as criterion for design of gear as per the Beam Strength equation.

Twisting Moment $M_t = (60*10^6*KW)/2*\pi*np$

Where np= speed of pinion rotation, rpm

Therefore, on substituting the above values, we get

 $M_t = (60*10^6*1.5)/2*\pi*45 = 316.2*10^3 \text{ N-mm}$

Then, tangential load is given by $P_t = 2*M_t / \mathrm{Dp}$

Where Dp= diameter of pinion = m*Zp, m = module of the gear pair, Zp= no. of teeth on the pinion

On substituting above values, $P_t = 2*316.23*10^3 / m*18 = (35*10^3) / m$

Now pitch line velocity $v = (\pi^* Dp *np)/(60*10^3) = (\pi^* m^* Zp *np)/(60*10^3)$

 $v = \pi * m * 18 * 45 / (60 * 10^3) = 42.4 * 10^3 m$

Since gears are manufactured by hobbing and requirement of module for this particular application is within the range 3-6, the velocity factor Cv is chosen as 3/(3+v). Therefore, on substituting the above value, we get the value of Cv as $3/(3+(42.4*10^{-3*m}))$.

Applying Lewis Equation, Sb= σ_b *b*m*Y, where b = face width = 10*m, Y= Lewis form factor, selected on the basis of the number of teeth, and σ_b = permissible bending stress in MPa

Earle Buckingham suggested that the permissible bending stress is approx. $1/3^{\rm rd}$ of the ultimate tensile strength of the material and therefore, $\sigma_b = 660/3 = 220$ MPa. The value of Lewis form factor is 0.308 for 18 number of teeth, and this is chosen from the standard table as shown below.

Z Y Z Z 15 0.289 27 0.348 55 0.415 16 0.295 28 0.352 60 0.421 17 0.302 29 0.355 65 0.425 18 0.308 30 0.358 70 0.429 19 0.314 32 0.364 75 0.433 0.320 20 33 0.367 80 0.436 90 21 0.326 35 0.373 0.442 0.446 100 0.330 37 0.380 22 0.458 39 0.386 150 23 0.333 200 0.463 0.389 40 24 0.337 0.471 300 0.399 45 0.340 25 0.484 Rack 0.408 50 0.344 26

TABLE 1: Value of Lewis Form Factors

Substituting all these values in above equation,

Sb= 220*10*m*m*0.308= 712.8*m^2

Assuming factor of safety (FOS) as 1.5, we get the equation Sb= FOS* Peff

Where Peff= effective force = (Cs*Pt)/ Cv, where Cs is the service factor, which is taken as 1, since the machine is run at 0-3 hours per day at a light shock load as per the following table.

TABLE 2: Value of Service Factors

Type of load	Type of service			
	Intermittent or 3 hours / day	8 – 10 hours per day	Continuous 24 hours per day	
Steady	0.8	1.00	1.25	
Light Shock	1.00	1.25	1.54	
Medium Shock	1.25	1.54	1.80	
Heavy Shock	1.54	1.80	2.00	

Substituting values in above equation

 $712.8*(m^2) = 1.5*3/(3+(42.4*10^-3*m))*(35*10^3)/m$

Solving for the value of m, we get m= 3.944 mm. The next standard module is 4mm, therefore 4mm is chosen.

Substituting the value m=4mm in above equation of effective force, we get the value

Peff = $3/(3+(42.4*10^{-}3*4))*(35*10^{3})/4=8281.8 \text{ N}$

Wear Strength: The failure of gear tooth due to pitting occurs when the contact stresses between two meshing teeth exceed the surface endurance strength of the materials. It is based on Hertz theory of Contact stresses and the value of wear force Sw is given by

Sw= Dp*b*Q*K, where Q= ratio factor = (2* Zg)/ Zg+Zp, where Zg is number of teeth on the gear, and K= load stress factor = 0.16 (BHN/100)^2, for steel gears with 20° pressure angle.

Q = (2*54)/54+18 = 1.742

Therefore, substituting Peff* FOS = Sw, we get the value of BHN as

 $8281.8*1.5 = 4*18*10*4*0.16*(BHN/100)^2*1.742$

Solving for BHN, we get the value as 423. 95. Since the value of BHN for 50C4 is restricted to 250, these gears will need to be case hardened to prevent their failure due to pitting. [1]

IV. SELECTION OF LUBRICANT:

Proper lubrication of the gear teeth is essential for the satisfactory performance and durability of the gears. Gears are lubricated by grease or straight mineral oils. Grease is used as the lubricant for the gears having lower pitch line velocity. [1]

Since the gears are rotating at low rpm but with a higher torque, the lubricant selected should be able to withstand the higher load as well as maintain the rpm of the gear pair. For this particular purpose, grease is generally chosen as the lubrication. For the purpose of this paper, the grease called EP-4 was chosen, as this doesn't chemically react with 50C4 gears and has the following properties:

TABLE 3: Properties of EP-4 Lubricant

Typical Properties

	Mobilux EP 004
NLGI Grade	00
Thickener Type	Lithium
Color, Visual	Brown
Penetration, Worked, 25°C, ASTM D 217	415
Viscosity of Oil, ASTM D 445	
cSt @ 40°C	160
Timken OK Load, ASTM D 2509, lb	40
4-Ball Wear, ASTM D 2266, Scar, mm	0.5
4-Ball Weld Load, ASTM D 2596, Kg	250
Dropping Point, D 2265, C	
Rust Protection, ASTM D 6138, Distilled Water	0-0

V. RESULTS AND DISCUSSION:

As can be seen from the above calculations, the gear pair with module 4 is chosen for this particular application. However, the material will have to be case-hardened to prevent the failure of the gear pair due to pitting. Hence, necessary heat treatment is provided to improve the hardness of the gear surface. Following are the dimensions of the gear pair designed for this purpose:

TABLE 4: Final Dimensions of Gear Pair

Dimension	Pinion	Gear
No. of teeth	18	54
Pitch Circle Diameter (mm)	72	216
Addendum (mm)	4	4
Dedendum (mm)	5	5
Clearance (mm)	1	1
Working Depth (mm)	8	8
Whole Depth (mm)	9	9
Tooth Thickness (mm)	6.3	6.3

VI. CONCLUSION:

Thus, we have designed a planetary gear pair for the purpose of mixing of concrete in an industrial mixer. We have also determined the values of dimensions of the gear pair, as well as analyzed the various modes of failure by which the gear pair will fail. Case hardening is a requirement, since there is a large disparity between the values of material BHN and required BHN. We have also studied the properties of the lubricant which is to be used in the gearbox, and the reason for its selection.

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