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Design, Load Analysis and Optimization of Compound Epicyclic Gear Trains

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Abstract: - 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) [17], Khurmi & Gupta (2006) [19], P. Kannaiah (2006) [18] [20] as bending failure (load failure), Pitting (contact stresses), scoring and abrasive wear, in any case it is related to the loads acting on the gear and this research deals with the Optimization of the gear design leading to the reduction in the load failure of the gears. Further, table.1 explains the different areas of research carried out by different authors on Epicyclic gear trains. This study carried out in this research shows the optimization analysis of the epicyclic gear train in INDIA to reduce load failure. The analysis is restricted to the optimization of gear train through load analysis of the gears, pinions and annulus including the sun and plant gears, and finding out the optimal load conditions for the gear train to perform effectively without leading to load failure. 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, however there has not been a comprehensive study of its load bearing performance with respect to different parameters such as module, material, and power of the epicyclic gear trains [16] [17]. This research paper provides an attempt in filling that gap in aiming to get the epicyclic gear trains load performance on different parameters. This process helps in finding the optimized design for the epicyclic gear trains in which it has the best performance without any failure and with minimum Loads acting on the gears. The main aim of this research investigation is to optimize the epicyclic gear train through load analysis, to prevent load failure from happening in the future.

Keywords: - *Optimization, Planetary Gear Trains, Tangential Tooth load, Wear tooth load, Dynamic tooth load, Static tooth load*

I. INTRODUCTION

A Planetary or Epicyclic Gear Trains comprises of one or more planet gears revolving around a sun gear. Usually, an epicyclical gearing systems are employed to achieve high reduction ratio in a small and power dense package. 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. The epicyclic gear train model is taken from BHEL, and some of its parameters have been modified to optimize its performance. The gear train consists of five external gears and 4 internal annulus gears, including sun and planet gears forming an epicyclic gear train. The present work on epicyclic gear trains carries out the design of all the gears, Shafts, keys and the loads are calculated for individual gears in the epicyclic gear train system. The analysis is divided into three parts, in which the first part, the power of 10HP is taken for the whole epicyclic gear train with four different modules (3, 4, 5, 6) for four different loads Tangential Tooth Load (Wt), Dynamic Tooth Load (Wd), Static Tooth Load (Ws) and Wear Tooth Load (Ww) with Cast Iron as the base material. The same process was conducted for the Power 15 HP and 20 HP while keeping the other parameters (material/module-3, 4, 5, 6) as constant. As the condition was stated that for preventing Gear failure the Static Tooth Load (Ws) and the dynamic Tooth Load (Wd) should be greater than the Wear Tooth Load (Ww) [15] [19], This condition is analyzed for the entire gear train and optimized for to get the least loads on the gears. As these Gear trains are subjected to high loads during their operation they are subjected to high stresses in the process which may cause failure, thus calculating the loads for different modules and for different power levels will show us the best optimized design of the gear train. This paper shows the optimization of gear trains with varying the modules and power of the entire gear train.

Table.1 shows the prominent authors who contributed to the analysis of Gears

Author	**Description of the work carried out**
S. Avinash [1]	Load Sharing behavior in epicyclic gear trains
P. Sunyoung [17]	Failure analysis of a planetary gear train
A. kiril [12]	Alternative method for analysis of complex compound planetary gear train
C. Yuksel [7]	Dynamis tooth load of planetary gear sets
M. Rameshkumar [16]	Load Sharing analysis of High-Contract-Ratio in Spur Gear
B. Gupta [15]	Contact stress analysis of spur gear
A.R. Hassan [14]	Contact stress analysis of spur gear teeth pair

II. COMPUTATIONAL METHODOLOGY

The Compound Epicyclic Gear train in Fig.1 is taken from BHEL and the parameters are altered for the optimization purposes. The Gears, arms, keys and annulus are designed in Solidworks which is shown in Fig.1 (a). Fig.1 (b) shows the general diagram showing all the positions of gears, annulus, shafts and arms. This model of the epicyclic gear train failed due to the high loads acting on the gears. 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.

Loads in an epicyclic gear train are divided into four parts: Tangential Tooth Load (Wt), Static Tooth Load (Ws), Dynamic Tooth Load (Wd) and Wear Tooth Load (Ww).

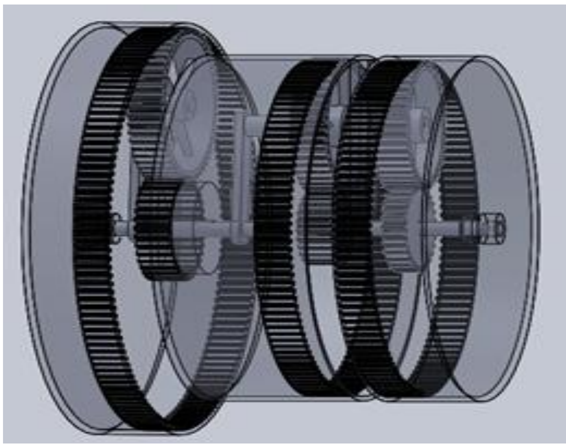


Fig.1(a)

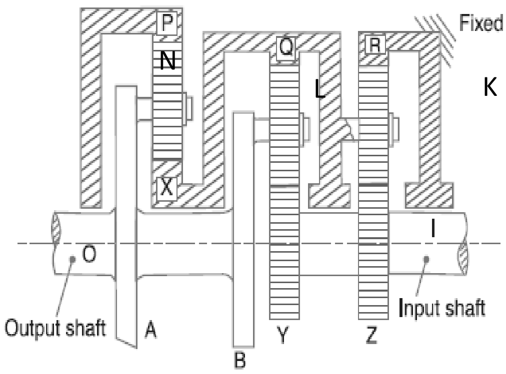


Fig.1(b)

Fig.1 Figure shows the positions of all the internal and external gears including Sun and Plant Gears, (a) Figure shows the Design made in Solidworks software after theoretical calculations, (b) Shows the 2D representation of the Epicyclic gear trains showing all the Gears, Pinion, Annulus including sun and planet gears

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, from which modules 3, 4, 5 and 6 were selected for the design optimization of gears [18] [20].
Systems of Gear Teeth:-The following four systems of gear teeth are commonly used in practice. 14 ½° Composite systems, 14 ½° 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 ½° to 20° results in a stronger tooth, because the tooth acting as a beam is wider at the base. The 20° stub involute system has a strong tooth to take heavy loads, thus was selected [16] [15] [18].

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. Among these Cast iron is widely used because of its excellent wearing properties, in which Cast Iron with UTS 480 Mpa, Elongation 6-16% was selected because of its long service life, high wear resistance, low production cost, high stability and surface finish [17][13]

Table.2 Shows all the parameters of Epicyclic Gear Train from Gears Z to Gear P (9 Gears) for module 4

Parameters (mm)	Gear Z	Gear k	Gear R	Gear Y	Gear L	Gear Q	Gear X	Gear N	Gear P
No. of Teeth (T)	30	45	120	24	48	120	36	54	114
Pitch Diameter (D)	120	180	480	96	192	480	144	216	576
Circular Pitch (Pc)	12.56	12.56	12.56	12.56	12.56	12.56	12.56	12.56	12.56
Face Width (b)	26.11	40	40	40	40	40	60	40	40
Module (m)	4	4	4	4	4	4	4	4	4
Steady Load (Wt)	795.78	795.78	795.78	994.76	994.76	994.76	3315.7	452.14	452.14
Pressure angle (∅)	20	20	20	20	20	20	20	20	20
Increment Load (Wi)	2897.9	3936.1	3936.1	3763	3796.4	3796.4	3229.2	4206.1	4206.1

Table.3 Shows the parameters of Gear modeling for modules 3, 4, 5, 6

Parameters (mm)	M=3	M=4	M=5	M=6
Addendum (1× m)	3	4	5	6
Dedendum (1.25× m)	3.75	5	6.25	7.5
Working depth (2× m)	6	8	10	12
Total depth (2.25× m)	6.75	9	11.2	13.5
Tooth thickness (1.507× m)	4.52	6.28	7.535	9.04
Clearance (0.25× m)	0.75	1	1.25	1.5
Fillet radius (0.4× m)	1.2	1.6	2	2.4

Table.4 Shows all the Gears, Annulus, Sun and Planet Gears in the Gear Train

S.no	Gear	Annulus	Sun	Planet
1	Z	R	Z	K
2	K	Q	X	L
3	Y	X	Y	N
4	L	P		
5	N			

III. DESIGN AND LOAD OPTIMIZATION OF GEARS

The following are the sample design and load calculations for analysis of Gear-Z having module 4 and power 10 HP, from the Epicyclic Gear train, in which the calculations of the four loads acting on the gears (Tangential Tooth Load (Wt), Static Tooth Load (Ws), Dynamic Tooth Load (Wd) and Wear Tooth Load (Ww)) are calculated, for the module-(4). The parameters for the gears are mentioned in Tables 2, 3 and 4. As the below calculations are performed, similarly the calculations are done for the rest of the eight gears for modules 3, 4, 5 and 6. Now this process is repeated for Power 10HP, 15 HP, 20 HP. All the results are tabulated for Table.6 to Table.14, and the Graphs are plotted from Graph.1 to Graph.9, showing the Load variance on the different modules at different power levels.

DESIGN OF Z -GEAR (pinion):

Teeth of Gear Z(T_Z)=30; Diameter of Gear Z (D_Z)=120 mm = 0.12 m; Speed of Gear Z (N_Z)=1500 r.p.m

Pitch Line Velocity $V = \frac{\pi D_Z N_Z}{60}$ = (π x 0.12 x 1500) / 60 = 9.4247 m/s

Tangential Load(W_T):- Tangential tooth load is also called the beam strength of the tooth. It is the load acting perpendicular to the radial tooth load (Wr) and normal tooth load (Wn) [16] [18] as shown in the fig.3

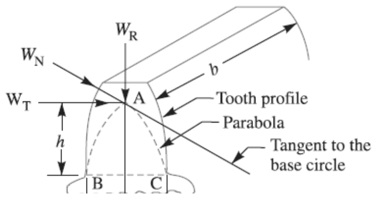


Fig.2 Shows the tangential tooth load direction on the gear tooth profile

$W_T = \frac{P}{V} \times CS = (\frac{7500}{9.4247})$ x 1.5 = 1193.671 N

Where W_T = Permissible tangential tooth load in N, P = Power transmitted in watts, v = Pitch line velocity in m/s; $\frac{\pi DN}{60}$, D = Pitch circle diameter in m, N = Speed in r.p.m., and C_S = Service factor (Cs is taken from the Table.5)

Table.5 Shows Service Factor (Cs) for different loads

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

Applying Lewis Equation [15] [16] [18]
$W_T = \sigma_W \times b \times P_c \times y$ Where σ_W=Permissible working stress, b=Gear tooth face width,
Pc=Circular Pitch = π× m (module), y=Lewis form factor
$\sigma_W = \sigma_O \times C_V$ Where σ_O=Allowable Static Stress, Cv= Velocity Factor, σ_O = 90 N/mm^2 as material was nodular Cast iron)
C_V = (4.58) / (4.58+V) = 0.327 Where V= Pitch Line Velocity (9.4247 m/s)
y = 0.154 - (0.912 / T_Z) = 0.1236 Where Tz = Teeth of Gear Z
$P_C = \pi \times m = \pi \times 4$
Substituting the values in the equation
⇒ 1193.671 = 90 x 0.327 x b x π x 4 x 0.1236
⇒ b = 26.11 mm

Dynamic Tooth Load (W_D):- The dynamic tooth loads act due to inaccuracies in tooth spacing, tooth profiles and deflection of tooth under loads [18]. The formulae for (W_D) is given below as
$W_D = W_T + W_I$ Where W_D=Total Dynamic Tooth Load, W_T =Steady Load due to transmitted torque, W_I=Increment Load due to dynamic action.
W_T = (P / V) = (7500 / 9.4247) = 795.781 N Where P=Power, V= Pitch line velocity
$W_I = \frac{K_3 \times V(b \times C + W_T)}{\sqrt{K_3 \times V + (b \times C + W_T)}}$
Where (K_3 = 20.67), V=Pitch line velocity, b=face width (mm), C=Deformation factor (n/mm) [18]
$C = \frac{e}{K_1[(\frac{1}{Ep}) + (\frac{1}{Eg})]}$ Where e=Tooth error (mm), K_1= Factor of Gear Teeth for 20 ^{0} full involute system, E_p=Module of elasticity of Pinion, E_g= Module of elasticity of Gear, (K_1=9; E_P= E_G=164000 N/mm^2) [18]
C = 0.127 / {9 [(1/ 164000) + (1/ 164000)]} C = 115.711 N/ mm Subtituting all the results in W_I
$W_I = \frac{\{20.67 \times 9.4247\ (26.11 \times 115.711 + 795.781)\}}{20.67 \times 9.4247 + \sqrt{(26.11 \times 115.711 + 795.781)}}$ = 2897.939 N Substituting W_T, W_I in W_D
⇒ W_D = 795.781 + 2897.939 = 3693.72 N
Static Tooth Load (W_S):- The static tooth load (beam strength or endurance strength of the tooth) is derived from lewis formula with the substitution of elastic limit stress (σ_e) instead of Permissible working stress (σ_w). It is said that for preventing tooth breakage (Ws) should be greater than Dynamic tooth load (Wd).
$W_S = \sigma_e \times b \times P_C \times y$ Where σ_e = Elastic limit stress (σ_e = 175 N/mm^2), b= Face width, P_C= Circular pitch (π x 4) y=Lewis form factor [18]

W_S = 175 x 26.11 x (πx 4) x 0.1236 = 7096.974 N

Wear Tooth Load (W_W):- It is maximum load that a gear tooth can carry without premature wear. It depends upon the curvature of tooth profile, elasticity and surface fatigue limit of the gear material. It uses Buckingham equation [18] [16].

$W_W = D_Z . b .Q .K$ Where D_Z = Pitch circle diameter of Gear Z, b= Face width, Q = Ratio factor for external or internal gears, K = Load stress factor or material combination factor T_G= Teeth of Gear, T_P = Teeth of pinion

$Q = (2T_G) / (T_G + T_P) = (2 \times 45) / (45 + 30) = 1.2$

$K = \sigma^2_{es} \times \sin\varnothing / 1.4 [(1/E_P) + (1/E_G)]$ Where σ = Surface endurance limit, ∅=Pressure angle, E_P= Young modulus of elasticity of Pinion, E_G= Young modulus of elasticity of Gear

$K = 630^2 \times \sin 20 / 1.4[(1/164000) + (1/164000)] = 1.1824\ N/mm^2$ Substituting the values in W_W

$\Rightarrow W_W = 120 \times 26.11 \times 1.2 \times 1.1824 = 4445.634$ N

The results of the four loads derived for the Gear–Z can be seen in Table.6 and Graph.1, in power 10HP for module4. Similarly the loads for rest of the eight gears for module 3, 4, 5 and 6 for the power 10 HP, 15 HP and 20 HP respectively can be inferred from Tables 6 to 14 and Graphs 1 to 9.

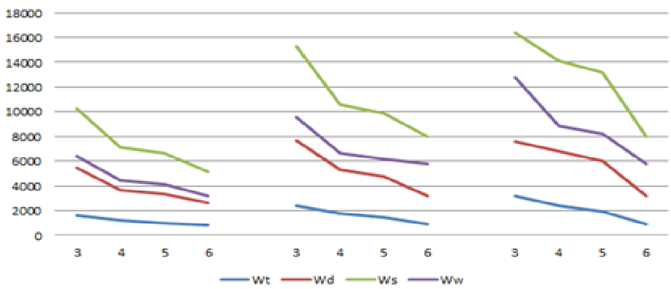
IV. RESULTS AND DISCUSSION

As the resulting loads are calculated for the Gear-Z of Epicyclic Gear Train having module 4, similarly the four loads W_T, W_D, W_S, W_W are calculated for all the rest of the eight Gears and Annulus including sun and planet gears. This process is repeated for different modules (3, 4, 5, 6) with all the 9 Gears in the Epicyclic Gear Train. Now this whole procedure is conducted for three different power levels (10 HP, 15 HP, 20 HP). All the results are tabulated and graphs are plotted accordingly from Table.6 to Table.14 and from Graph.1 to Graph.9. Where the condition is that'' For safety against tooth breakage, the Static Tooth Load (Ws) should always be greater than Dynamic tooth load (Wd), also the Dynamic tooth load (Wd) should not be more than the wear tooth load (Ww) otherwise the gear will fail [18], also the least loads are observed for all the Gears to get the optimized design in the Epicyclic Gear Train. Below are the graphs plotted for the loads where

P=Power (HP); M=Module; Loads (N) Wt = Tangential Tooth Load; Wd = Dynamic Tooth Load; Ws = Static Tooth Load; Ww = Wear tooth Load

Table.6 Shows the different parameters of Gear-Z

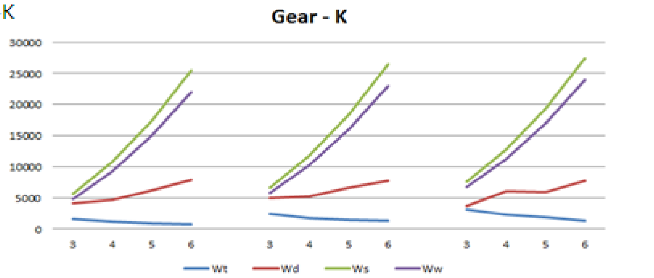
P		Wt	Wd	Ws	Ww
10	3	1591.68	5454.16	10267.1	6431.44
10	4	1193.67	3693.72	7096.97	4445.63
10	5	954.93	3310.22	6633.53	4155.33
10	6	797.784	2668.02	5133.143	3215.46
15	3	2372.6	7645.2	15303.6	9586.3
15	4	1779.3	5288.3	10578.8	6626.7
15	5	1423.4	4772.6	9887.11	6193.4
15	6	935.56	3218.55	7952.45	5812.36
20	3	3162.13	7601.9	16365.26	12776.3
20	4	2371.4	6823.6	14098.83	8831.6
20	5	1897.1	6049.8	13148.83	8236.22
20	6	935.56	3218.55	7952.45	5812.36



Graph.1 Shows the different parameters of Gear-Z with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.7 Shows the different parameters of Gear-K

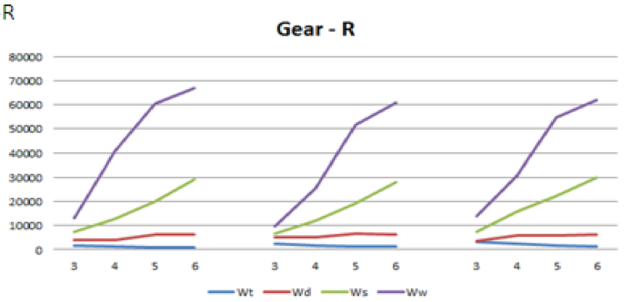
P	M	Wt	Wd	Ws	Ww
10	3	1591.68	4163.799	5615.487	4746.464
10	4	1193.67	4731.909	10760.87	9215.94
10	5	954.93	6149.585	17376.35	14962.4
10	6	795.784	7937.417	25461.95	21985.5
15	3	2372.6	4982	6615.48	5746.46
15	4	1779.3	5228.1	11760.8	10215.9
15	5	1423.49	6661.7	18376.87	15962.8
15	6	1285.3	7725.69	26461.95	22985.6
20	3	3162.21	3693.2	7615.5	6746.2
20	4	2371.2	6002.2	12760.8	11215.2
20	5	1897.1	5912.2	19376.3	16962.4
20	6	1285.3	7725.7	27461.9	23985.6



Graph.2 Shows the different parameters of Gear-K with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.8 Shows the different parameters of Gear-R

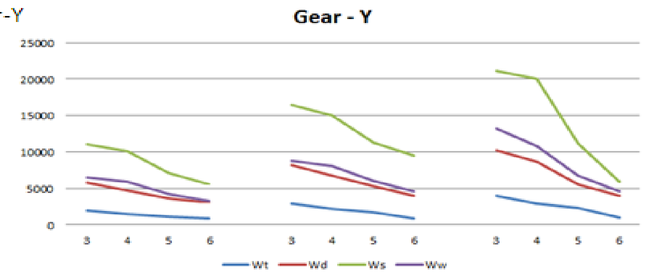
P	M	Wt	Wd	Ws	Ww
10	3	1591.6	4163.7	7243.8	13044.61
10	4	1193.6	4131.9	12878	40523.75
10	5	954.9	6149.5	20121.9	60568.35
10	6	796.1	6256.6	28975.5	66785.6
15	3	2372.6	4980	6615.48	9746.4
15	4	1779.3	5228.8	11878	25486.8
15	5	1423	6661.8	19121	51652.2
15	6	1318.65	6256.6	27975.5	60785.6
20	3	3162.2	3693.3	7443.8	14044.6
20	4	2371.4	6002.2	15878.2	30523.7
20	5	1897.2	5912.6	22121.2	54568.2
20	6	1318.6	6256.6	29975.5	61785.6



Graph.3 Shows the different parameters of Gear-R with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.9 Shows the different parameters of Gear-Y

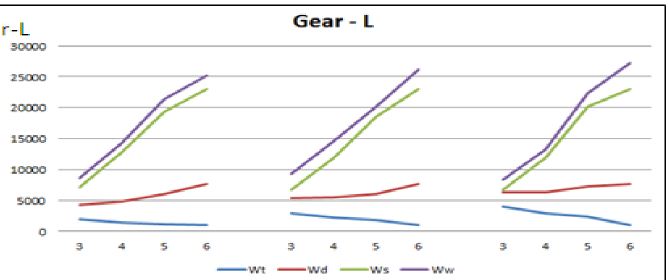
P	M	Wt	Wd	Ws	Ww
10	3	1989.7	5860.1	11041.4	6534.4
10	4	1492	4757.7	10073.7	5961.7
10	5	1193.6	3707.5	7097.7	4200.5
10	6	930.9	3038.2	5521.5	3276.7
15	3	2966	8144.5	16457.6	8787.1
15	4	2224.1	6718.03	15015	8086.2
15	5	1779.3	5307.5	11273.3	6049.4
15	6	943.4	4038.2	9521.5	4567.7
20	3	3952.9	10283.2	21092.6	13195.4
20	4	2964.2	8630.1	20012.8	10843.8
20	5	2371.4	5618.4	11237.4	6775
20	6	994.7	4038.2	5921.5	4567.7



Graph.4 Shows the different parameters of Gear-Y with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.10 Shows the different parameters of Gear-L

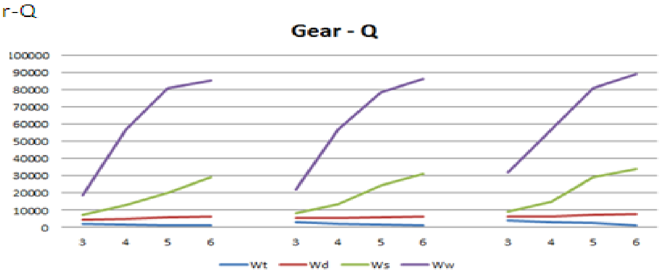
P	M	Wt	Wd	Ws	Ww
10	3	1989.7	4338.9	7126.3	8652.1
10	4	1492	4791.1	12857.3	14265.3
10	5	1193.6	6020.9	19365.6	21365.3
10	6	994.7	7706.17	22958.7	25174.4
15	3	2966	5318.2	6679.8	9356.2
15	4	2224.1	5549.3	11875.2	14562.3
15	5	1793.3	6063.9	18555	20145.3
15	6	994.78	7706.1	22958.7	26174.4
20	3	3952.9	6294.8	6679.8	8365.3
20	4	2946.2	6317.3	11875.2	13256.3
20	5	2371.4	7301.8	20158	22365.3
20	6	994.7	7706.1	22958.7	27174.4



Graph.5 Shows the different parameters of Gear-L with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.11 Shows the different parameters of Gear-Q

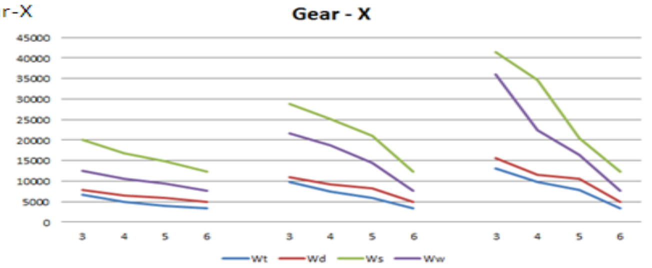
P	M	Wt	Wd	Ws	Ww
10	3	1992.3	4341.2	7243.8	18924.8
10	4	1502.2	4802.1	12878	56577.2
10	5	1195.6	6022.6	20121	80680.3
10	6	998.3	6203.6	28975.5	85123.8
15	3	2966	5318.2	8443.8	21924.3
15	4	2224.1	5559.1	13478	56755.2
15	5	1779.6	6070.2	24521.9	78680
15	6	994.78	6203.6	30954.5	86132.8
20	3	3953.2	6296.1	9258.1	31927.4
20	4	2949.5	6319.4	14768	56759.7
20	5	2369.1	7303.8	29121.6	80681.2
20	6	993.1	7709.9	33902.5	89133.3



Graph.6 Shows the different parameters of Gear-Q with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.12 Shows the different parameters of Gear-X

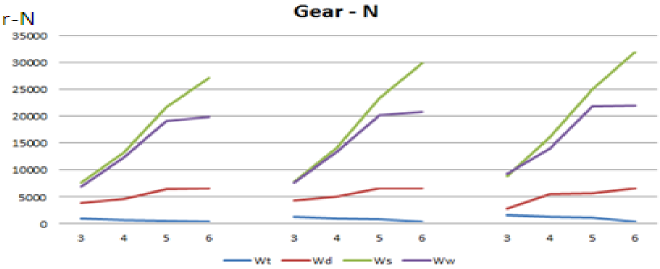
P	M	Wt	Wd	Ws	Ww
10	3	6631.6	7809.8	20069.3	12566.1
10	4	4973.6	6545	16843.9	10546.6
10	5	3979.4	5928.2	14912	9337
10	6	3316.6	4906.4	12272.3	7684.2
15	3	9885.6	10901.4	28764.3	21622.02
15	4	7414.1	9125.1	25110.23	18635.21
15	5	5932	8256.2	21136.21	14365.21
15	6	3316.6	4906.4	12272.3	7684.2
20	3	13174.9	15689.6	41452.6	36007.3
20	4	9881	11600.3	34559	22325.32
20	5	7905.9	10482.2	20472.1	16325.32
20	6	3316.62	4906.42	12272.3	7684.2



Graph.7 Shows the different parameters of Gear-X with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.13 Shows the different parameters of Gear-N

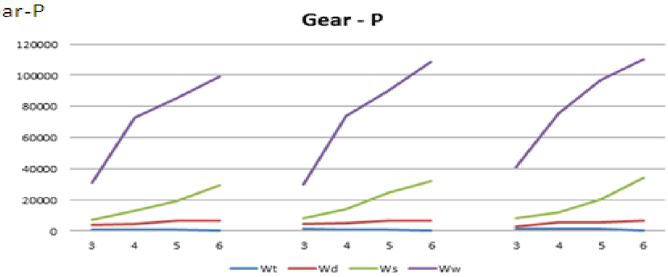
P	M	Wt	Wd	Ws	Ww
10	3	904.34	3866.88	7658.32	6895.76
10	4	678.21	4658.31	13256.3	12259.12
10	5	542.587	6367.75	21654.3	19154.88
10	6	442.11	6524.52	27164.8	19879
15	3	1348.22	4375.21	7783.21	7593.21
15	4	1011.45	5045.21	14059.9	13297.87
15	5	808.15	6632.221	23280.1	20111.32
15	6	442.11	6524.5	29834.9	20776.6
20	3	1523.12	2842.19	8783.7	9223.2
20	4	1347.4	5440.25	16000.2	13997.1
20	5	1077.13	5669.9	24945.2	21777.6
20	6	442.11	6524.52	31974.9	21988.6



Graph.8 Shows the different parameters of Gear-N with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

Table.14 Shows the different parameters of Gear-P

P	M	Wt	Wd	Ws	Ww
10	3	904.34	3866.8	7303.2	30864.8
10	4	678.2	4658.3	12983	72628.4
10	5	542.5	6367.7	19286	85511.8
10	6	452.1	6517.7	29213	98942.7
15	3	1348.07	4375.21	7956.43	30053.4
15	4	1011.45	5048.16	14073.4	73578.4
15	5	808.81	6632.08	24836.7	90361.8
15	6	452.17	6517.7	31947.6	108942.7
20	3	1523.1	2842.2	8194.3	40864.3
20	4	1347.4	5440.3	12059.9	75478.3
20	5	1077.9	5669.9	20286.8	97294.7
20	6	452.2	6517.7	34213	110065.3



Graph.9 Shows the different parameters of Gear-P with three different power levels (10 HP, 15 HP, 20 HP) for modules 3, 4, 5 and 6.

As all the loads (Wt, Wd, Ws and Ww) were calculated for the gears it was seen that static tooth load (Ws) and wear tooth load (Ww) were greater than dynamic tooth load (Wd) for all the gears and thus the design of the Gear train is safe. It is observed that in the Sun Gears (Z, X, Y), the least loads can be seen at the module 6, but of the plant gears and annuluses the least loads were observed at module 3.

V. CONCLUSION

The purpose of this research paper is to determine the optimal design of the gear train with the load analysis carried out in the gear trains by varying the module (3, 4, 5, 6) for all the gears for three different power levels 10 HP, 15 HP and 20HP. On further examination of the loads for the gears which were plotted from Table 6 to 14 and Graphs 1 to 9, we can notice that the Wear tooth load (Ww) for all the gears in the gear train is higher than the Dynamic tooth load (Wd), and the Dynamic Tooth load (Wd) is less than Static tooth load (Ws) for all the gears in the system. As this condition has to be true for safety against tooth failure, thus we can state that the design is safe. We can observe in Graphs Z, Y and X that the loads are decreasing as the module is increasing and the least load is observed on module 6, as those are the sun gears in the gear train. Also it is observed that the rest of the gears and annulus in graphs K, R, L, Q, N and P that the loads are increasing as the module increases and the lease load is observed on module 3. This was seen consistent with the power level 10HP, 15 HP and 20 HP. Furthermore it is also observed that in Graphs Q, P and R, the wear tooth load is greater than the static tooth load which is why the teeth of the annuls should be of a higher wear resistant material like cast iron as was suggested in section II of this research paper. Thus as the design satisfy the condition that Static Tooth Load (Ws) should always be greater than the Dynamic Tooth Load (Wd) also the Wear tooth load (Ww) should not be less than the Dynamic tooth load (Wd), the proposed design is safe and the least load conditions being at the least module (3, in this condition) is preferred for the annulus design and the planet gears where as a higher module (6, in this condition) is preferred for the sun gears design for all power levels.

VI. ACKNOWLEDGEMENT

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VII. FUTURE SCOPE

The Research could be further extended through further optimization by varying different materials, also taking into consideration the working conditions and different gears (helical, worm etc.). The loads with different materials could be further analyzed and compared with the practical loads and stresses derived on the site.

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