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Design, Load Analysis and Optimization of Epicyclic Gear Trains

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ABSTRACT

Mechanical power transmission systems rely heavily on Epicyclic gear trains, as their failure can compromise the entire system. Consequently, identifying and mitigating the causes of gear failure is crucial. Various gear failure modes, including bending pitting (contact stresses), failure (load failure), abrasive wear, and scoring, are discussed in literature by J.R. Davis (2005) [24], Khurmi & Gupta (2006) [23], and P. Kannaiah (2006) [21][22]. These failures are often linked to the loads acting on the gears. This research focuses on optimizing gear design to reduce gear load failure. Table 1 summarizes various research efforts on Epicyclic gear trains conducted by different authors. This study delves into the optimization of epicyclic gear trains in India to minimize load failure. The analysis focuses on optimizing the gear train through load analysis of the pinions, gears, and annulus, including the sun and planet gears. The goal is to determine the optimal load conditions for the gear train to function effectively without succumbing to load failure. Epicyclic Gear Trains are widely used in industry due to their numerous advantages, including high torque capacity, improved efficiency, relatively smaller size, lower weight, and a highly compact package. However, there has been no comprehensive study of their load-bearing performance with respect to various parameters such as rotational speed, material, and power. This research paper aims to fill this gap by investigating the load performance of epicyclic gear trains under different parameters such as loading conditions and rotational speeds. This process helps in identifying the optimized design of epicyclic gear trains, ensuring optimal performance and minimizing gear loads and choosing correct rotational speed. The primary objective of this research investigation is to optimize epicyclic gear trains through load analysis to prevent future load failures.

1. INTRODUCTION

Planetary gear trains, also known as epicyclic gear trains, are composed of sun gear and one or more planet gears rotating around it. These gear systems are commonly employed to achieve high devaluation ratios within a compact and power-dense package. Studies have shown that load sharing is not uniform within planetary gear trains. These gear trains are widely used for power transmission and are considered crucial components in mechanical power transmission systems. They play a crucial role in various industrial sectors, and any failure in the gear train can lead to a complete system shutdown. Therefore, finding the root causes of gear train failures and optimizing their performance is essential. Epicyclic gear trains offer several advantages, including lower weight, higher torque capacity, smaller size, and improved efficiency compared to conventional gearboxes. Their compact design, which is approximately 60% lighter and half the size of a conventional gearbox, may lead to misconceptions about their strength. However, to minimize gear train stresses, it is crucial to maintain Minimum loads. The epicyclic gear train model used in this study was obtained from BHEL (Dilawer et al., 2013) [25], and some parameters were modified to enhance its performance. The gear train comprises five external gears and four internal annulus gears, including the sun and planet gears, forming an epicyclic gear train configuration. The current work on epicyclic gear trains involves the design of all gears along with the calculation of loads for individual gears within the epicyclic gear train system. The analysis is divided into three parts. The first part involves analysing the performance of the entire epicyclic gear train under a power of 10 HP at three rotational speeds (2000 RPM, 1500 RPM, and 1000 RPM) for four different loads:wear tooth load (Ww), static tooth load (W_s),dynamic tooth load (W_t), and tangential tooth load (W_t). The material used for this analysis is cast iron. The same process was repeated for power levels of 15 HP and 20 HP while keeping other parameters constant. To prevent gear failure, the dynamic tooth load (W_s) and static tooth load (W_d) should be greater than the wear tooth load (Ww) [15] [19]. This condition was analysed for the entire gear train and optimized to achieve minimal loads on the gears. Because these gear trains operate under heavy loads, they experience substantial stresses that can lead to failure. Therefore, calculating loads for different rotational speeds and power levels provides insights into the optimal design of the gear train. This paper demonstrates the development of gear trains by varying the rotational speeds and power of the entire gear train.

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

Author	**Description of the work carried out**
C. Yuksel [3]	Dynam is tooth load of planetary gear sets
S. Avinash [8]	Load Sharing behavior in epicyclic gear trains
M. Ramesh kumar [9]	Load Sharing analysis of High-Contract-Ratio in Spur Gear
A. R. Hassan [10]	Contacts tress analysis of spur gear teeth pair
A.kiril [13]	Alternative method for analysis of complex compound planetary gear train
P. Sunyoung [24]	Failure analysis of a planetary gear train
B.Gupta[11]	Contacts tress analysis of spur gear
Liu, Hu Ran [26]	Fundamental Equations of Load Deviation of the Gears in Compound Gearing
S. Prayoonrat, D. Walton [27]	Practical approach to optimum gear train design

2. COMPUTATIONALMETHODOLOGY

The compound epicyclic gear train depicted in Figure 1 was sourced from BHEL (Dilawer et al., 2013) [25], with modifications made to its parameters for optimization purposes. The gears were designed using Solid Works software, as shown in Figure 1a. Figure 1b presents a general diagram illustrating the arrangement of gears, annulus, shafts, and arms were also sourced from (Dilawer et al., 2013). The original epicyclic gear train model experienced failure due to excessive loads acting on the gears. Given the critical role of gears in power transmission systems, gear failure can compromise the entire system, necessitating gear optimization to achieve low-load operation and efficient power transmission. Loads in an epicyclic gear train are categorized into four types: wear tooth load (W_w), dynamic tooth load (W_d), tangential tooth load (W_t), and static tooth load (W_s).

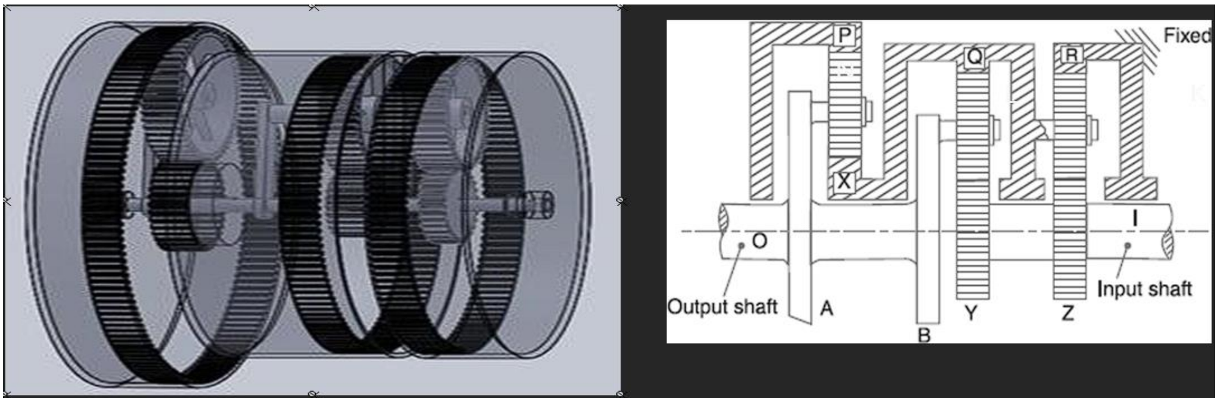


Fig.1(a) Fig.1(b)

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 represents the ratio of the pitch circle diameter (in millimetres) to the number of teeth. It is typically stand by m, where m = D / T, with D representing the pitch circle diameter and T representing the number of teeth. The Indian Standard recommends the following series of modules given in table 2. From which module 4 was selected for gear design optimization [22] [21].

Gear Tooth Systems: Four primary systems of gear teeth are commonly employed in practical applications: 14 ½° full depth in volute systems,14 ½° composite systems, 20° full depth in volute systems, and 20° stub in volute systems. The tooth profile of the 20° full depth in volute system can be cut using hobs. Increasing the pressure angle from 14 ½° to 20° results in a stronger tooth due to its wider base, enhancing its beam-like properties. The 20° stub in volute system offers a robust tooth capable of handling heavy loads, leading to its selection [9] [11] [22].

Table. 2 shows recommended modules

Recommended Modules	2nd Choice for Modules
1	1.125
1.25	1.375
1.5	1.75
2	2.25
2.5	2.75
3	3.5
4	4.5
5	5.5
6	7
8	9
10	11
12	14
16	18
20	22
25	28
32	36
40	45
50	

Table. 3 shows then parameters of gear modelling for module 4.

Parameters (mm)	Module=4
Addendum (1 × m)	4
Dedendum (1.25 × m)	5
Working Depth (2 × m)	8
Total Depth (2.25 × m)	9
Tooth Thickness (1.507 × m)	6.28
Clearance (0.25 × m)	1
Fillet Radius (0.4 × m)	1.6

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

Gear	Annulus	Sun	Planet
Z	R	Z	K
K	Q	X	L
Y	X	Y	N
L	P		
N			

Gear Material: Gear materials are chosen based on factors such as service factor, strength requirements, wear resistance, and noise considerations. Metallic and non-metallic materials are available. For industrial applications, metallic gears are commonly used, and commercially available options include steel, cast iron, and bronze. Among these materials, cast iron is widely utilized due to its excellent wear properties. In this study, cast iron with a Ultimate Tensile Strength of 480 MPa and an elongation of 6-16% was selected due to its high stability, high wear resistance, low production cost, long service life, and superior surface finish [24] [12].

1. DESIGN AND LOAD OPTIMIZATION OF GEARS

The following design and load calculations illustrate the analysis of Gear-Z, a component of the epicyclic gear train, with a module of 4 and a power rating of 10 HP and Rotational Speed of 1500 RPM. The calculations encompass the four loads acting on the gears: static tooth load (W_s), dynamic tooth load (W_d), tangential tooth load (W_t),and wear tooth load (W_w). Gear parameters are provided in Tables 2, 3, and 4. Similar calculations are performed for the remaining eight gears, spanning Rotational Speeds of1000 RPM, 1500 RPM and2000 RPM. For power levels, repeat this procedure of 10 HP, 15 HP, and 20 HP. The results are presented in Tables 6 to 14, and Graphs 1 to 9 illustrate the load variations across different Rotational Speeds and power levels.

DESIGNOFZ-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 ($V = \pi \times D_Z \times N_Z$) / 60= ($\pi$ × 0.12 × 1500) / 60 = 9.42477796 m/s

<u>Tangential Load(Wt): -</u> Tangential tooth load is also known as the beam strength of the tooth. It is the load acting perpendicular to the normal tooth load (Wn) and radial tooth load (Wr) [9] [22] as shown in fig:5.

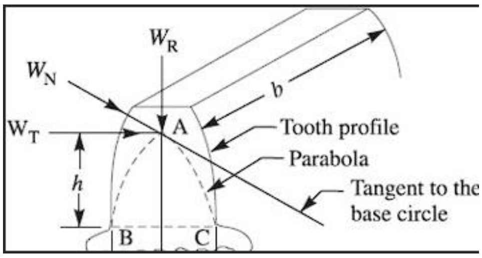


Fig.5 Shows the tangential tooth load direction on the gear tooth profile

Table.5Shows Service Factor(C_s)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

By applying Lew is Equation [11] [9][22]

$W_t = \sigma_W \times b \times P_c \times y$

Where σ_W= Permissible working stress, b=Gear tooth face width,

$\sigma_W = \sigma_O x C_V$

Where σ_O= Allow able Static Stress, C_v=Velocity Factor, σ_O=90N/mm^{2}as material was nodular Cast iron)

P_c = Circular Pitch=π× m(module), y = Lew is form factor

C_V= (4.58)/ (4.58+V) = 0.327031247, Where V = Pitch Line Velocity (9.42477m/s)

Y=0.154-(0.912/T_Z)=0.1236

WhereT_z = Teeth of Gear Z

$P_C = \pi \times m = \pi \times 4$

Substituting the values in the equation,

$$1187.29588 = 90 \times 0.327 \times b \times \pi \times 4 \times 0.1236$$

$$b= 25.97160951mm$$

Dynamic Tooth Load (W_d):- Inaccuracies in tooth spacing, tooth profiles, and tooth deflection under loads cause the dynamic tooth loads[22]. 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) = (7460 / 9.424777) = 791.5305836N, Where P=Power, V= Pitch line velocity

$$W_I = \frac{K3 \times V \times (b \times C + WT)}{K3 \times V + \sqrt{(b \times C + WT)}}$$

Where (K3 = 20.67), V=Pitch line velocity, b=face width(mm), C=Deformation factor (n/mm) [22].

$C = \frac{e}{K1(\frac{1}{EP}+\frac{1}{EG})}$, Where e = Tooth error(mm) = 0.127, K_1=Factor of Gear Teeth for 20^0 full in volute 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) [22].

$$C = \left(\frac{0.127}{9 \times (\frac{1}{164000}+\frac{1}{164000})}\right)/10 = 115.7111111 \text{ N/mm.}$$

$$W_I = \frac{20.67 \times 9.424777 \times (25.9716 \times 115.71111 + 791.530)}{20.67 \times 9.424777 + \sqrt{(25.9716 \times 115.71111 + 791.530)}} =$$

2884.408059 N. Substituting W_T and W_I,

$W_d = W_T + W_I$ = 791.5305836 + 2884.408059 = 3675.939 N.

Static Tooth Load (W_s):- The Lew is formula is used to compute the static tooth load, also known as the beam strength or endurance strength of the tooth, and elastic limit stress (σe) is used in place of permissible working stress (σw). There is a claim that in order to prevent tooth breaking, the work load (Ws) should exceed the 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/mm2), b= Face width, P_C= Circular pitch (π ×M) y=Lew is form factor [22]

W_S=175 ×25.9716 × (π × M) × 0.1236 =7059.358671N.

Wear Tooth Load (W_w):- It is the highest load that a gear tooth can support before wearing down too soon. It depends upon the parameters like curvature of tooth profile, elasticity and surface fatigue limit of the gear material. It uses Buckingham equation [22] [9].

$W_w = D_Z \times b \times Q \times K$. 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 × 45) / (45 + 30) = 1.2

K = σ^2_{es} × sin∅ / (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 × sin 20 / (1.4 / (1/164000) + (1/164000)) = 1.1824 N/mm^2

Substituting the values in W_w

W_w = 120 × 25.9716 × 1.2 × 1.1824 = 4422.341 N

The results of the four loads acquire for the Gear–Z can be seen in Table:6 and Graph:1, in power 10HP formodule4. Similarly, the loads for the rest of the eight gears for rotational speeds1000rpm, 1500rpm, 2000rpm for the power 10 HP, 15 HP and20HP respectively can be in ferred from Tables 6to14 and Graphs1 to 9.

3. RESULTS AND DISCUSSION

The load calculations performed for Gear-Z of the epicyclic gear train, module 4, were extended to determine the four loads (W_d, W_s, W_w, W_t) for the remaining eight gears and annulus, including the sun and planet gears. This process was repeated for different rotational speeds (1000rpm, 1500rpm and 2000rpm) for all nine gears in the epicyclic gear train. The entire procedure was then conducted for three different power levels (10 HP, 15 HP, and 20 HP). The outcomes are tabulated and plotted from Table 6 to Table 14 and Graph 1 to Graph 9. To ensure safety against tooth breakage, the dynamic tooth load (W_d) should always exceed the static tooth load (W_s), and the dynamic tooth load (W_d) should not surpass the wear tooth load (W_w) to prevent gear failure [22]. Additionally, the optimized design of the epicyclic gear train aims to minimize loads across all gears. The graphs below illustrate the load variations for different power levels (P) and rotational speeds (N), with loads represented in W_d = Dynamic Tooth Load, Newtons (N): W_t = Tangential Tooth Load, W_w= Wear Tooth Load, and W_s = Static Tooth Load.

Table: 6 and Graph:1 shows the different results for Gear Z

P	N	Wt	Wd	Ws	Ww
10	1000	1780.9	4254.8	8213.67	5146.46
10	1500	1187.3	3675.9	7059.36	4422.34
10	2000	890.47	3335.5	6482.2	4060.75
15	1000	2671.4	6050.9	12320.511	7718.1947
15	1500	1780.9	5292.2	10589.038	6633.5119
15	2000	1335.7	4841.4	9723.3014	6091.1705
20	1000	3561.9	7742.2	16427.348	10290.926
20	1500	2374.6	6829.7	14118.717	8844.6825
20	2000	1780.9	6286.1	12964.402	8121.5606

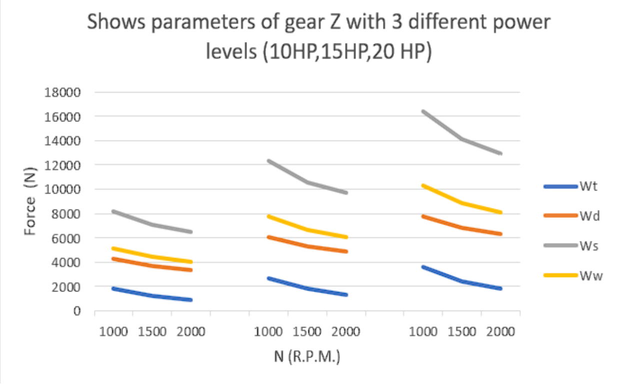


Table: 7 and Graph: 2 shows the different results for Gear K

P	N	Wt	Wd	Ws	Ww
10	1000	1187.3	3522.9	7059.4	7431.36
10	1500	791.53	3056	6289.8	6621.27
10	2000	593.65	2793.9	5905	6216.22
15	1000	1780.9	5079.1	10589	11147
15	1500	1187.3	4454	9434.7	9931.9
15	2000	890.47	4097.9	8857.6	9324.33
20	1000	2374.6	6561.9	14119	14862.7
20	1500	1583.1	5801.6	12580	13242.5
20	2000	1187.3	5364.2	11810	12432.4

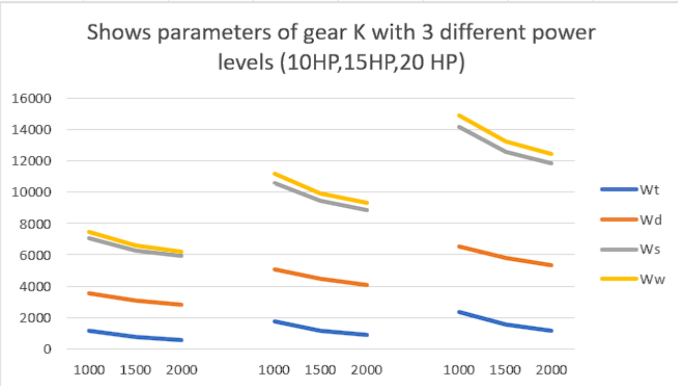


Table: 8 and Graph:3 shows the different results for Gear R

P	N	Wt	Wd	Ws	Ww
10	1000	445.24	2416	5616.5	5400.9
10	1500	296.82	2193	5327.9	5123.4
10	2000	222.62	2076.6	5183.6	4984.6
15	1000	667.85	3564.5	8424.7	8101.3
15	1500	445.24	3252.4	7991.8	7685.1
15	2000	333.93	3088.3	7775.4	7476.9
20	1000	890.47	4688.1	11233	10802
20	1500	593.65	4295.9	10656	10247
20	2000	445.24	4088.2	10367	9969.3

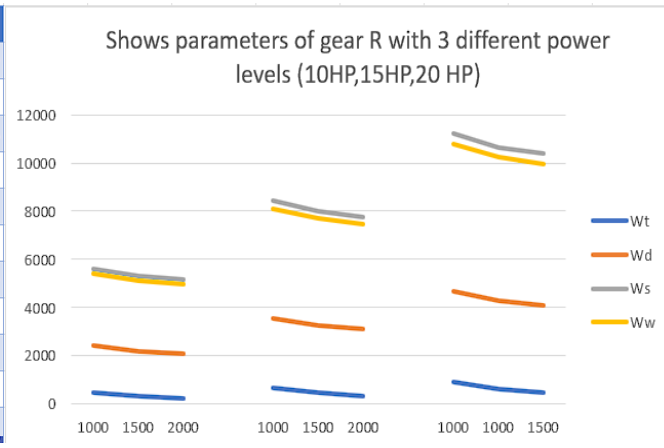


Table: 9 and Graph: 4 shows the different results for Gear Y

P	N	Wt	Wd	Ws	Ww
10	1000	2226.2	4740.6	9079.4	5387.1
10	1500	1484.1	4107.6	7636.5	4531
10	2000	1113.1	3726.3	6915.1	4102.9
15	1000	3339.3	6690.8	13619	8080.6
15	1500	2226.2	5866.1	11455	6796.4
15	2000	1669.6	5369	10373	6154.4
20	1000	4452.4	8518.5	18159	10774
20	1500	2968.2	7526.2	15273	9061.9
20	2000	2226.2	6932.3	13830	8205.8

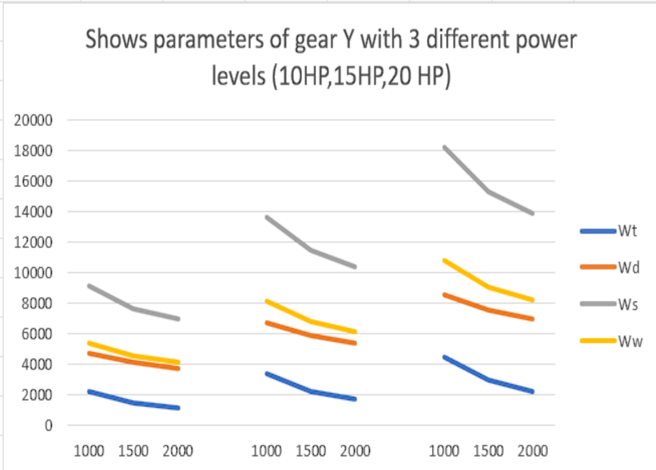


Table: 10 and Graph: 5 shows the different results for Gear L

P	N	Wt	Wd	Ws	Ww
10	1000	1113.1	3422.9	6915.1	7554.6
10	1500	742.06	2974.4	6193.6	6766.4
10	2000	556.54	2724.3	5832.9	6372.3
15	1000	1669.6	4944.9	10373	11332
15	1500	1113.1	4342.2	9290.4	10150
15	2000	834.82	4001.2	8749.3	9558.5
20	1000	2226.2	6398	13830	15109
20	1500	1484.1	5663.2	12387	13533
20	2000	1113.1	5243.3	11666	12745

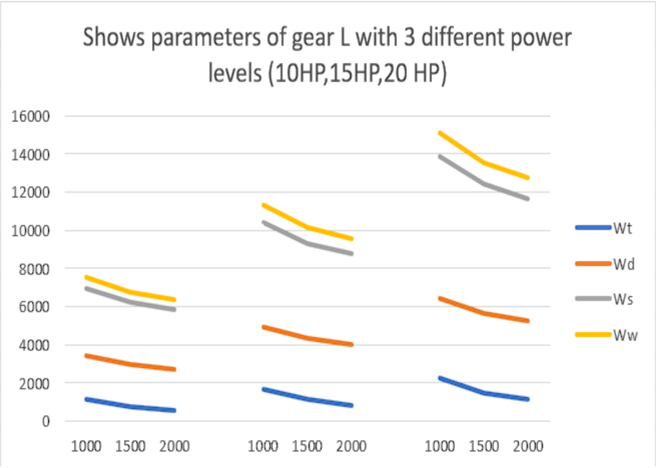


Table: 11 and Graph: 6 shows the different results for Gear Q

P	N	Wt	Wd	Ws	Ww
10	1000	445.24	2416	5616.5	5658.1
10	1500	296.82	2193	5327.9	5367.4
10	2000	222.62	2076.6	5183.6	5222
15	1000	667.85	3564.5	8424.7	8487.1
15	1500	445.24	3252.4	7991.8	8051
15	2000	333.93	3088.3	7775.4	7833
20	1000	890.47	4688.1	11233	11316
20	1500	593.65	4295.9	10656	10735
20	2000	445.24	4088.2	10367	10444

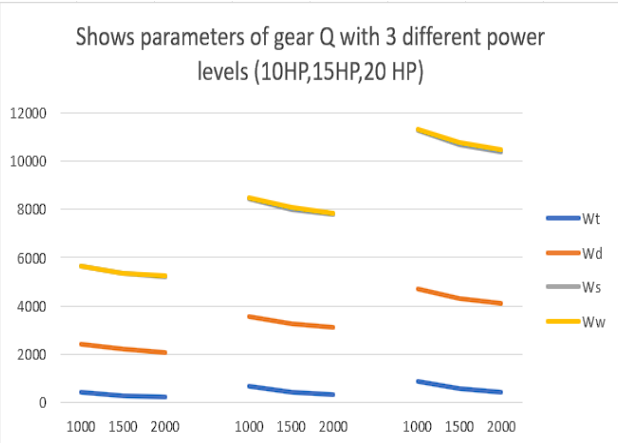


Table 12 and graph 7shows the different results for Gear X

P	N	Wt	Wd	Ws	Ww
10	1000	1484.1	3902.9	7636.5	5514.6
10	1500	989.41	3373	6674.6	4820
10	2000	742.06	3068	6193.6	4472.7
15	1000	2226.2	5585.6	11455	8271.9
15	1500	1484.1	4885	10012	7230
15	2000	1113.1	4476.1	9290.4	6709
20	1000	2968.2	7178	15273	11029
20	1500	1978.8	6332	13349	9640
20	2000	1484.1	5834.8	12387	8945.3

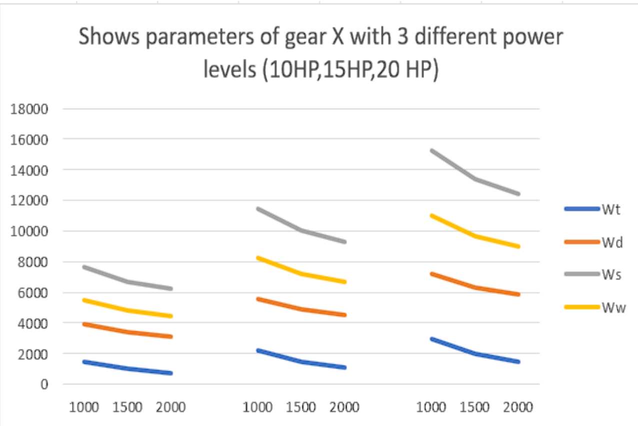


Table 13 and graph 8shows the different results for Gear N

P	N	Wt	Wd	Ws	Ww
10	1000	989.41	3251.4	6674.6	7673.2
10	1500	659.61	2836.1	6033.3	6935.9
10	2000	494.71	2607.2	5712.7	6567.3
15	1000	1484.1	4713.6	10012	11510
15	1500	989.41	4151.9	9050	10404
15	2000	742.06	3838	8569	9851
20	1000	1978.8	6114.8	13349	15346
20	1500	1319.2	5426.9	12067	13872
20	2000	989.41	5038.6	11425	13135

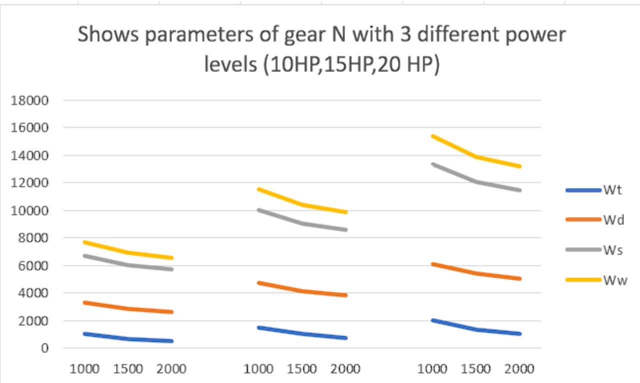
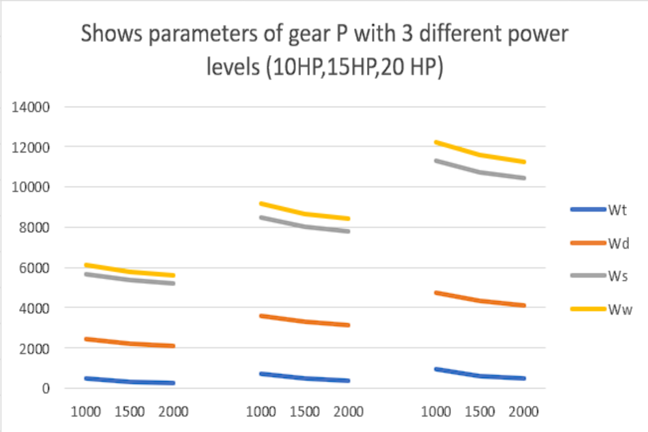


Table 14 and graph 9 shows the different results for Gear P

P	N	Wt	Wd	Ws	Ww
10	1000	468.67	2454.9	5662	6112.8
10	1500	312.45	2221.9	5358.3	5784.9
10	2000	234.33	2100	5206.4	5620.9
15	1000	703	3618.9	8493	9169.2
15	1500	468.67	3293.4	8037.4	8677.3
15	2000	351.5	3121.6	7809.6	8431.3
20	1000	937.34	4756.6	11324	12226
20	1500	624.89	4348	10717	11570
20	2000	468.67	4130.9	10413	11242



As all the loads (W_w, W_s, W_t and W_d) were calculated for the gears it was seen that wear tooth load (W_w) and static tooth load (W_s) were greater than dynamic tooth load (W_d) 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 canbeseenatthe2000 RPM and at 10 HP, also for the plant gears and annul uses the least loads were observed at 10 HP at 1000 RPM. It is also seen that for Sun Gears Static tooth load (W_s) was the highest load while for the plant gears and annul uses wear tooth load (W_w) was the highest load.

4. CONCLUSION

The objective of this research paper is to identify the optimal design for the gear train by analyzing the loads across different rotational speeds (2000 RPM, 1500 RPM, 1000 RPM) for all gears under three power levels: 20 HP, 15 HP, and 10 HP. Upon examining the loads for the gears plotted from Table: 6 to Table: 14 and Graphs 1 to 9, it is evident that the wear tooth load (W_w) for all gears in the gear train exceeds the dynamic tooth load (W_d), and the dynamic tooth load (W_d) remains lower than the static tooth load (W_s) for all gears in the system. This condition, which is crucial for preventing tooth failure, is satisfied, indicating that the design is safe. Graphs X, Y, and Z reveal that the loads increase as the power level increases, with the minimum load observed at 2000 RPM, corresponding to the sun gears in the gear train. Additionally, Graphs K, L, Q, N, and P demonstrate that the wear tooth loads (W_w) for the remaining gears and annulus is higher than other loads, with the maximum load observed at gear N at 1000 RPM and 20 HP. Conversely, it is seen that static tooth load (W_s) is maximum for plant gears. This trend is consistent across power levels of 10 HP, 15 HP, and 20 HP. Also, Graphs K, L, Q, N, P illustrate that the wear tooth load surpasses the static tooth load for these gears. This observation suggests that the teeth of these gears should be made from a material with superior wear resistance, such as cast iron, as recommended in Section 2 (Computational Methodology) of this research paper. Since the proposed design satisfies the condition that static tooth load (W_s) should always exceed wear tooth load (W_w) and dynamic tooth load (W_d)should not be less than dynamic tooth load (W_d), it is considered safe. The minimum load conditions, observed at the lowest power level (10HP in this case), are preferred for the optimal working of gears with higher rotational speeds (2000 RPM) in this case.

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