Design of a Planetary Reduction Gearbox for a Hydraulic Roller Press of the Cement Industry

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Abstract**—** Gearbox has been standardised by well-developed and established Standards like AGMA and DIN. A detailed guidelines and recommendations are available for varieties of application for design as well as manufacturing and quality. But, design of gearbox for a new application is always a challenging task. Understanding of application loading condition is needed in designing of accurate, economical and reliable gearbox. In addition, load is fluctuating with time. Such application requires fatigue conditions to take into consideration. Studies are required to estimate loading conditions. Such studies will help in designing application specific conditions, and may lead to better performance, less cost and compact design. Design of gearbox by taking into consideration of equivalent torque found from the fatigue loading of the Hydraulic Roller Press is presented in this Paper. To develop loading characteristics of an application, detailed study of application function is required. This application study of present application of hydraulic roller press is carried in this Paper. Loading pattern for particular application is developed by observation of power v/s time curve. DIN 3990 Standard is used to make the design more accurate and standardized.

Keywords: Gearbox, Hydraulic Roller Press, Cement

I. INTRODUCTION

In the present work, an attempt has been made to create an indigenous design of a planetary gearbox for a hydraulic roller press of the cement industry. Hydraulic roller press is used in cement manufacturing process to crush hard balls of clinker. In present application, the loading characteristics are estimated which is varying within a large range with respect to time. For such application, fatigue considerations are also vital to heed. If design of the gearbox is carried out with the consideration of the maximum power and torque, then the design will be over safe. Therefore, uniqueness about this work is that, concept of equivalent torque obtained from the loading characteristics is used for the design of the gearbox, which makes design optimized.

Cement manufacturing process is a continuous process. Thus, hydraulic roller press works continuously 24×7. If it stops it working even for a small amount of time, it will cause large loss to industry. Due to this continuous fatigue loading high reliability is required for gearbox used for hydraulic roller press. DIN standard gives accurate and economical design by increasing strength to weight ratio of gearbox.

The illustration of a problem of design of gearbox which requires a large reduction ratio of 49.77:1 with a large output torque of 760488 N.m is given in this paper.

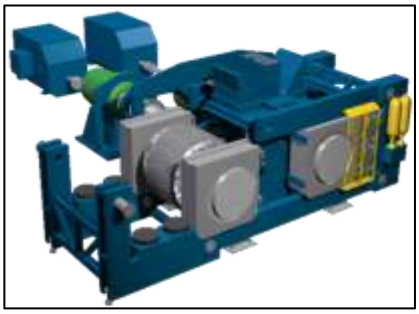


Fig. 5.1: Hydraulic Roller Press

In roller press, very high torque is required to crush hard balls of clinker while motor runs at high speed and low torque. Therefore, to increase the torque and reduce the speed, gear box is required. The Hydraulic Roller Press as shown in Figure 5.1 involves exposing feed material to a very high pressure over a short time. The pressure exerted forms micro cracks in the feed particles, leading to the generation of a large amount of fine material. Whenever the applied pressure exceeds the crushing strength of clinker, the crushing of clinker takes place. The final product is called Mineral Cake Slab.

Cement Industry applications demand high torque in compact (a high torque/volume) and light (a high torque/weight ratio) package. In planetary gears, torque density can be increased by adding more planets through multiple gear mesh points. This means a planetary gear with say three planets can transfer three times the torque of a similar sized fixed axis standard spur gear system. [1]

The applied load to planetary gears are distributed onto multiple gear mesh points means the load is supported by N contacts (where N = number of planet gears) increasing the torsional stiffness of the gear train by factor N. Hence, it lowers the lost motion compared to similar size standard gear trains. High rotational stiffness is important for applications with positioning accuracy and repeatability requirements; especially under fluctuating loading conditions. [1]

The smaller gears in planetary system result in lower inertia. Compared to a same torque rating standard gearbox, it is a fair approximation to say that the planetary gearbox inertia is smaller by the square of the number of planets. [1]

In planetary systems, the lubricant cannot escape. It is continuously redistributed, pushed and pulled‖ or mixed into the gear contacts, ensuring safe lubrication practically in any mounting position and at any speed. Hence, planetary gearboxes can be grease lubricated for the life. This feature is inherent in planetary gearing because of the relative motion between the different gears making up the arrangement. In short, the planetary gears systems have high torque density, compact, low inertia and can be grease lubricated for life, which are demands of industrial applications. [1]

If the annulus is held stationary and the sun gear is used as the output, the planet carrier will be the input. The gear ratio in this case will be $\frac{1}{1+\frac{A}{S}}$. This is the lowest gear ratio attainable with an epicyclic gear train. [1]

Pedrero et al. [2] developed strength model for bending and pitting calculations because the calculation methods for internal gear calculations are based on the same equations as that of external gears and small adjustments are done to account the internal gear geometry.

D. R. Salgado et al. [3] presented a paper focuses on design of planetary gear train transmissions to extend the constant power range and for high speed machining. Three types of constraints are taken into consideration to make a design: 1) constraints involving gear size and geometry- it takes into consideration of limitation of range of face width of gear and limitation of minimum of number of teeth of gear to avoid undercutting.2) Constraints to avoid interference during assembly of planetary gear train.3) Contact and bending stress limits.

Kawalec et al. [4]developed comparative analysis between ISO and AGMA tooth root strength evaluation methods.

Miryam B. Sanchez et al. [5] presented a paper in which critical value of stress and the critical load conditions have been obtained and complete analysis of tooth bending strength have been carried out. Recommendation for calculation of load capacity for spur and helical gear is proposed in this paper.

Francesca cura et al. [6] proposed a method to calculate in-operation application factor KA in gears subjected to variable loading conditions. Gears are subjected to fatigue loading thus it is important to take into consider this fatigue loading conditions. In this paper loading pattern has been developed of application in its service stage. Using the standard ISO 6336-6 methodology, application factor is found out for that loading condition. This gives more realistic and accurate design. Methodology for design of gear according to ISO 6336-1 to ISO 6336-3 for pitting and bending criteria is described in this paper. Comparison of design application factor and in-operation applications factor is made.

X. Li and G.R. Symmons et al. [7] presented a method for minimizing the centre distance of a helical gear set of prescribed power capacity using a fixed ratio of face width to PCD and formulae contained within AGMA procedures. It is a method that has been developed based on transforming the design constraints into direct limiting boundaries on design variable and general monotonic relation between centre distance and maximum power capacity of gear. Pitting resistance and bending strength constraints are transformed into lower and upper limits of centre distance. Also, maximum helix angle, minimum face contact ratio, minimum number of teeth of pinion and discrete set of modules are used as constraints. Instead of using minimum volume of gear as function, using centre distance is more appropriate as it can directly relate with pitting and bending strength values.

Seok-chul Hwang et al. [8] presented a paper in which contact stress analysis for a pair of mating gears during rotation is done. Variation of contact stress during rotation from 2.7° to 27° is observed and comparison is made between contact stress at lowest point single tooth contact(LPSTC) found using FEA method and contact stress found using AGMA equation. He found that contact stress found using FEA model is more severe than that of AGMA standard. When gears are meshed and revolved the stress becomes maximum around LPSTC point of contact teeth and then reduces. Maximum stress is generated at contact point and then propagated along line of action.

AGMA method determines pitting and bending strength of the gear based on empirical formulas. AGMA calculations are proven by field experiments. In AGMA in most of cases fatigue calculation are based on proven fatigue data.

DIN Standard is complex in a sense that it will consider all loading conditions acting on gear, thus it is universal in nature. But at the same time, it is flexible as it gives freedom to designer to select suitable method according to application and its loading conditions [9] [10] [11] [12] [13] [14].

DIN 3990 Standard introduces different influence factors that takes into account manufacturing deviations dimensions, material quality and heat treatment, alignment, lubrication and all deviations from ideal condition when it is put into service [9]- [14].

For the first step of design process equivalent torque is found from loading pattern of driven application and design of gears is done based on equivalent torque according to DIN 3990-6 [14]. If loading characteristics cannot be determined, then design process based on application factor can be selected from DIN 3990-1 [9]. According to DIN-3990, minimum safety factors should be selected accurately and for that deep knowledge of the application is required. The more accurately all minimum safety factors and parameters are determined, the more reliable the result of calculation and smaller the reserve capacity of material and more economical is the design.

From the literature study, the following research gap has been found...

1) Very less research has been done on the design of the gearbox based on equivalent torque and there is no research done on the design of the gearbox for a particular hydraulic roller press application from the practical data of the loading pattern observed at its service stage.
2) Lot of research is done on the design of gears as per AGMA and DIN Standard. But no research has been done on the design of gears as per DIN 3990 using the concept of equivalent torque.

Hence, it is decided to do research on the design of gears of a planetary reduction gearbox as per DIN 3990 standard using the concept of equivalent torque concept. Following two objectives are decided from above research gap.

1) To develop engineering specifications for a planetary gearbox for a Hydraulic Roller Press of the cement industry.
2) To develop a design of gearbox with an approach of design for manufacturing and assembly.

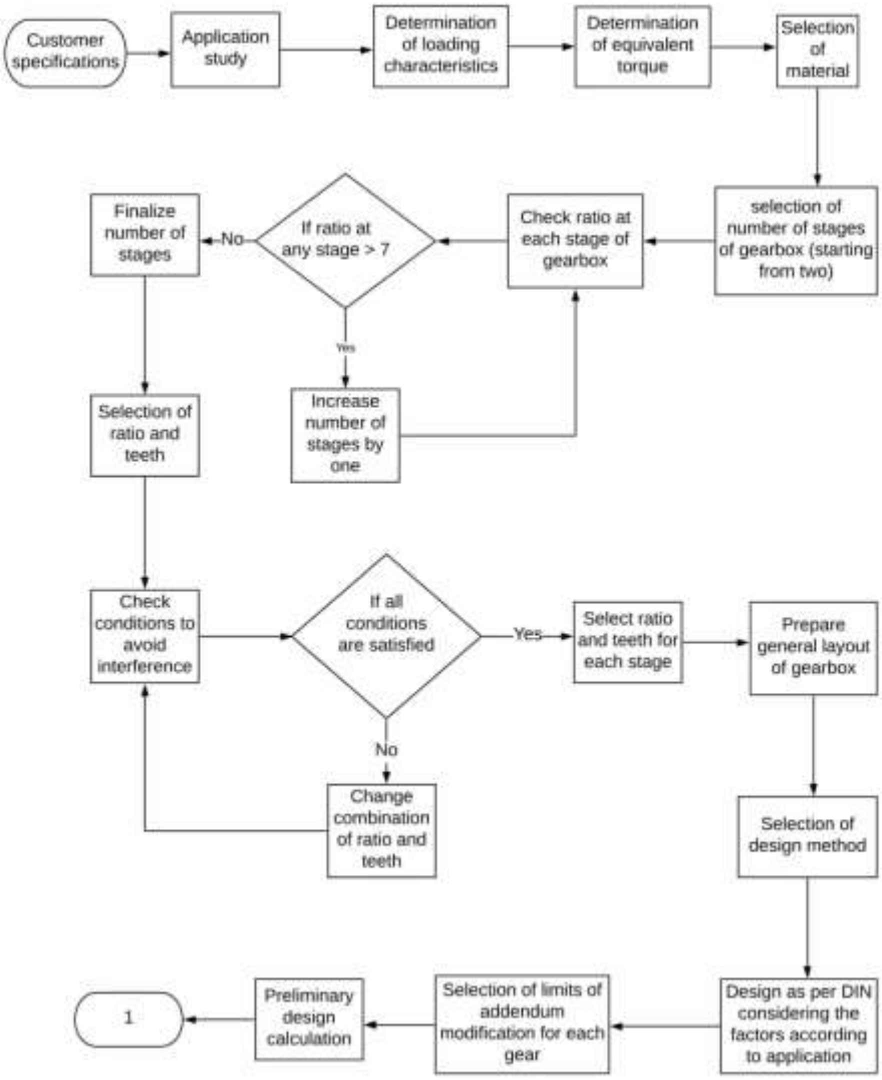
II. DESIGN METHODOLOGY FOR GEARS

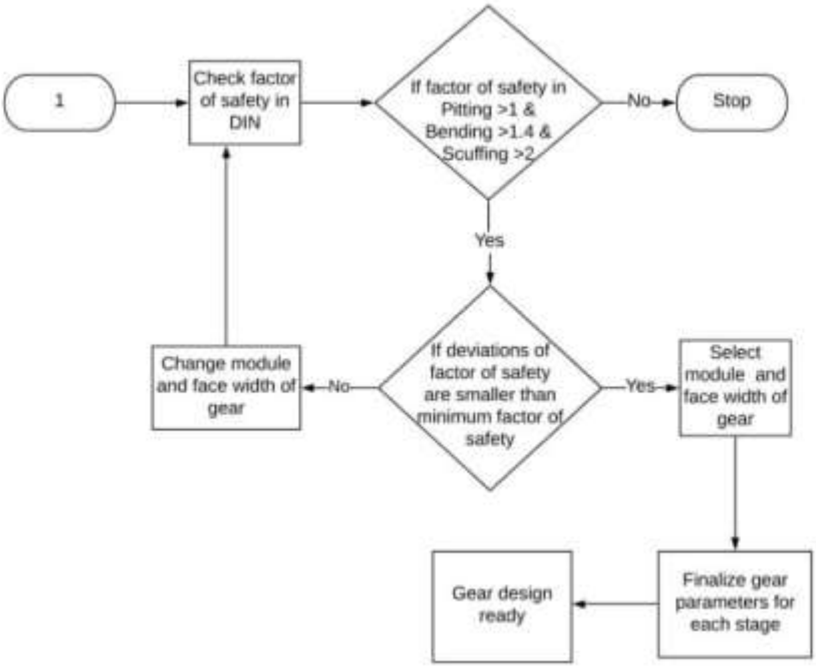
Before starting design of gearbox the factors which are causing fluctuation in loading on hydraulic roller press need to know to understand fatigue loading on gearbox.

A. Factors causing fluctuation in load on Hydraulic Roller Press

- Throughput capacity (1300-1550 tonnes per hour)
- Clinker size (3 to 25 mm)
- Feed temperature (<130 °C)
- Crushing strength and hardness of clinker
- Moisture content of clinker
- Harder particles cause 80% increase in power consumption compared to softer particles
- Metal percentage in the clinker

B. Flow chart





C. Design specifications of the gearbox

Specification	Magnitude and unit
Roller diameter	1920 mm
Roller Speed	19.89 rpm
Roller velocity	2 m/s
Maximum motor speed	990 rpm
Service factor	2.5 as per DIN/AGMA Standard
Design temperature	-20 to 40 °C
Gear box efficiency	96%
Motor power	1760 KW
Absorbed Power	1584 KW
Torque For Gearbox	760487 Nm
Max. torque for gear box	1901218 Nm
Duty	24 X 7

Table 2.1: Design Specification of Gearbox

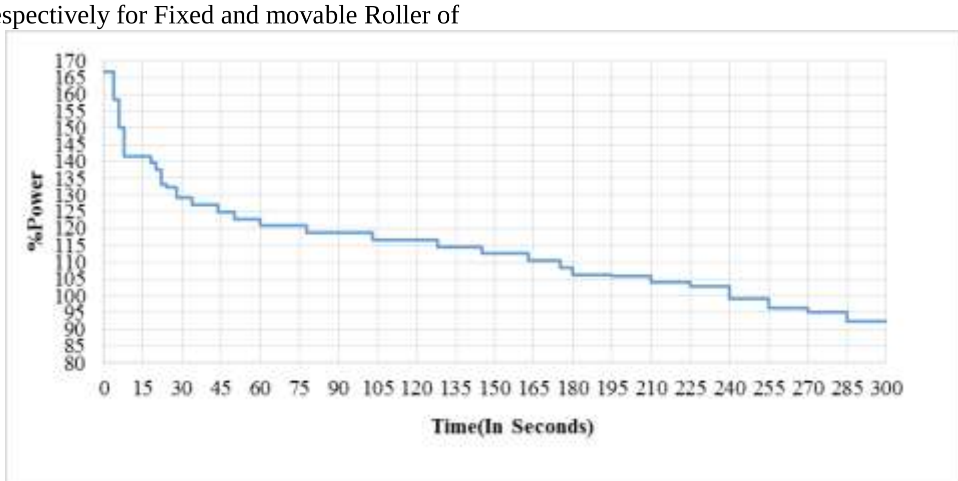
D. Development of loading pattern of Hydraulic Roller Press

- The observation of the Power v/s Time Curve for 1 hour & 1 Month respectively for Fixed and movable Roller of a Hydraulic Roller Press and the reasons for load fluctuations.
- The maximum power occurring during 1 month is also incorporated into loading cycle of 5 minute for fixed and movable roller respectively. The data is arranged in decreasing order of power as per consideration in DIN 3990-6 [14].
- From the observations of the curves loading pattern is developed for a 5-minute cycle as below:

From above graph we can see that maximum load occurring during one month is 166.67% for 5 seconds and minimum load is 91% for 15 seconds.

As load acting on movable roller is higher than that acting on fixed roller final design of gearbox is made for movable roller. This loading pattern can be simplified by dividing it into bins using palmgrem miner's rule [14]. According to this rule equivalent torque can be found out using equation below:

$$\sum_i \frac{n}{N} = 1.0 \text{ [14]}$$



Graph 2.1: %Power V/s Time (Movable Roller)

$$T_{eq} = \left(\frac{n_1 T_1^p + n_2 T_2^p + n_3 T_3^p + \dots}{n_1 + n_2 + n_3 + \dots}\right)^{\frac{1}{p}} \text{ [14]}$$

Where,
n_i = the number of load cycles
T_i = Torque of individual bin
p = slope of Wohler-damage line
For Pitting p = 6.610
For Tooth root p = 8.738
Therefore, equivalent torque calculated is,
Teq = 122.34%
Selection of materials

SR. NO.	Component	Material
1	PINION	18CrNiMo7-6 +QT
2	GEAR WHEEL, PLANET WHEEL	18CrNiMo7-6 +QT
3	ANNULUS	42CrMo4+H&T

Table 2.2: Selection of Materials

E. Selection of the number of stages of the gearbox

- The calculated reduction ratio of this gearbox i $=\frac{990}{19.89}$ = 49.77.
- The requirement of torque for this particular gearbox is high as per specifications. When there is need to transmit such a large torque with high reduction ratio ranging from 40:1 to 125:1, a compound planetary gearbox is required.
- The advantages of using planetary gear train are the compactness in a single stage, lightweight, and higher torque transmitting capacity.
- If only single stage planetary gearbox is used, then due to larger reduction ratio, the torque transmitting capacity would be smaller.
- If reduction ratio will increase above 7:1 in a single stage, then size of sun gear will decrease and it limits the torque transmitting capacity of gearbox and transmits only smaller torque.
- Using ratio of 4:1, size of sun and planet gears will be equal. This transmits larger amount of torque. In addition, 5:1 and 6:1 ratio will give higher torque transmitting capacity.
- At last stage, smaller ratio of a planetary stage gives smaller shaft diameter. Adding 3rd planetary stage will acquire more space therefore. If we add one more planetary stage, then it consumes more space also as per shaft orientation if we provide one helical pair then require ratio can be achieved in smaller space. In addition, an input shaft is offset from the output shaft, there for adding the helical gear pair will complete the 3 stage compound gearbox.
- At first stage, a spur or helical gear pair can be selected. But considering here the limitations of the spur gear pair such as a sudden contact over entire width of the tooth and larger noise. Where as in helical gear pair a contact is gradually taking place over entire face width of the gear.

F. Selection of Ratio

To avoid the interference while assembly, following conditions are needed to be satisfied

1) $\frac{\text{No of teeth of internal gear + No of teeth of sun gear}}{\text{number of planet gears}}$ = should be integer number [15]
2) (No of teeth of annulus gear - No of teeth of planet gear) - (No of teeth of sun gear + No of teeth of planet gear) = 0 or 1 or 2 [15]
3) $m(Z_S + 2) < m(Z_A + Z_S)*\sin\frac{\pi}{n}$ where n is number of planet gears [15]

We have checked conditions for 64 combinations of ratio and teeth and from that, only 6 combinations can fulfill conditions. For all the combinations of the gear teeth and ratio, gear parameters are obtained and analysis has been done.

If the teeth of sun pinion are 19 or more than that, it results into larger annulus diameter and larger overall weight of the gear box. But using 19 teeth of sun pinion gives an advantage that it can accommodate 4 number of planets in 2nd stage thus if in future torque requirement of gearbox will increase by even 10% then it can fulfil that need with the same design by adding just one planet. Therefore, by considering this future requirement, final teeth of sun pinion selected is 19.

For 3rd stage, if we use number of teeth less than 25 it will result into module higher than 20. But, it will make manufacturing tough. The third stage can also accommodate one more planet if the number of teeth of sun pinion is 25. Thus, it can fulfill future requirement of higher torque.

3rd stage(Planetary)	4	Z_s	25
2nd stage(Planetary)	5.67	Z_s	19
1st stage(Helical)	2.1944444	Z_1	21
Overall reduction ratio	49.80		

Table II.1: Final Selected Ratio and Teeth

G. Criteria for gear design

Gear is such an application where high hardness is required at surface to withstand large contact load and decrease wear. At the same time, it requires high toughness at its core to absorb shocks due to fatigue loading. Therefore, gear requires high hardness at surface and high toughness at core. Gears are transmitting power by its rolling action. Due to that, gear is subjected to load at contact point. This load induces stresses in gear. In order to design gears, it is very important to understand stresses, which are acting on gears and identify predominant stresses acting on it. It is vital to know the basic modes of failure of gears. Based on these modes of failure there are basic three criteria of gear design:

1) Bending strength
2) Pitting
3) Scuffing

H. Preliminary Gear Design Equations

1) For Spur Gears

Bending Criteria [16]

$$m = \sqrt[3]{\frac{4000T_1}{z_1^2\left(\frac{b}{d_1}\right)\sigma_{bp}}} \text{ (in mm)}$$

Pitting Criteria [16]

$$m = \left(\frac{1}{z_1}\right)\sqrt[3]{\frac{2000(u+1)T_1 y_m^2 y_p^2}{u\left(\frac{b}{d_1}\right)p_{cp}^2}} \text{ (in mm)}$$

Optimum face width [16]

$$b = 10m$$

Tangential force F_t [16]

$$F_t = \left(\frac{2000T}{d}\right) \text{ (in mm)}$$

Nomenclature [16]:
m = module (mm)
T_1= Torque in kN
z_1 = Number of teeth of Pinion
d^1= Pitch Circle Diameter of Pinion(mm)
σbp = Permissible bending stress u = Reduction Ratio
y_m = Material coefficient
y_p = Pitch point coefficient
P_{cp} = Allowable surface stress

2) For Helical gears

Bending Criteria [16]

$$m_n = \sqrt[3]{\frac{4000T_1(\cos\beta)^2}{z_1^2\left(\frac{b}{d_1}\right)\sigma_{bp}}} \text{ (in mm)}$$

Pitting Criteria [16]

$$m_n = \left(\frac{\cos\beta}{z_1}\right)\sqrt[3]{\frac{2000(u+1)T_1 y_m^2 y_p^2}{u\left(\frac{b}{d_1}\right)p_{cp}^2}} \text{(in mm)}$$

Pitch circle diameter [16] (d) (in mm)

$$d = \frac{zm_n}{\cos\beta}$$

Minimum face width [16]

$$b = \frac{1.15\,p}{\tan\beta}$$

Pitch point factor y_p [16]

$$y_p = \sqrt{\frac{\cos\beta_b}{(\cos\alpha_t)^2\tan\alpha_{tw}}}$$

$$\tan\beta_b = \tan\beta\cos\alpha_t$$

Tangential force F_t [16]

$$F_t = \left(\frac{2000T}{d}\right) \text{ (in mm)}$$

Nomenclature [16]:
m_n = module(mm)
β_b = Base Helix Angle in degree
α_t = Transverse Pressure angle in degree
β = Helix Angle in degree
α_{tw} = Working Pressure angle in degree
p = circular Pitch (mm)

I. Calculation of Nominal Stress Numbers [13]

- From DIN 3990-5
- $\sigma_{H\ lim}$ = Endurance limit for contact stress (N/mm²) = 1500 N/mm²
- $\sigma_{F\ lim}$ = Endurance limit for root bending stress (N/mm²) = 500 N/mm²

J. Addendum Modification (Profile Shift) [17]

Addendum Modification has advantages such as:
1) It increases load carrying capacity of gear as it increases tooth thickness at root.
2) It decreases chance of undercutting of gear.
3) It is helpful to get desired center distance.
4) It results in decrease in gear size.
5) It decreases slipping of tooth flank thus decreases wear of gear.

III. General Layout of Gearbox

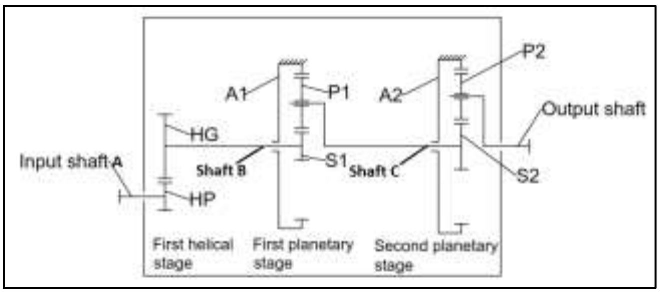


Fig. 3.1: General Layout of Gearbox

IV. Preliminary Gear Design

A. Stage 1 preliminary Calculation

Stage 1:Helical	
Input parameters	
Z1	21
i_1	2.1944444
β	0.2269
σ_{ut}	1050
σ_{bp}	280
Z2	46
b/d	0.7263535
T1	18693.591

Calculated parameters	
m(std)	12
pitch circle dia. (d)	259
pitch circle dia. (d)	567
b	188
Ft	144352.0566
FOS in bending	2.183178462
Torque transmitting capacity	40811.44596
Base Circle Dia. Of pinion(mm)	243
Base Circle Dia. Of gear(mm)	531
Tip Diameter of pinion(mm)	295
Tip Diameter of gear(mm)	598
Root diameter of pinion(mm)	241
Root diameter of gear (mm)	544.2
Velocity of pinion	13.431
Contact ratio	2.610050902
Cent. Dist.(pinion and gear)	421.5329753
tooth height(pinion)	27
tooth height(gear)	26.9

B. Stage 2 &3 preliminary Calculation

Input Parameters		
Parameter	Stage 2 Planetary	Stage 3 Planetary
Zs	19	25
i_2	5.67	4
σ_{ut}	1050	1050
σ_{bp}	280	280
T2	13674.016	58148.753
b/ds	0.5263158	0.4
Za	89	75
Zp	35	25
N	3	4

Parameters	Stage 2 Planetary	Stage 3 Planetary
m(std)	14	18
Pitch dia.(ds)	266	450
Pitch dia.(dp)	490	450
Pitch dia.(da)	1246	1350
working pitch dia.(ds)	267	461
working pitch dia.(dp)	492	461
working pitch dia.(da)	1251	1383
b	140	180
Ft	102427.0852	252272.2452
FOS in bending	2.66894527	1.755153732
Torque transmitting capacity	36495.2	102060
Base Circle Dia. Of sun(mm)	250	423
Base Circle Dia. Of planet(mm)	460	423
Base Circle Dia. Of annulus(mm)	1171	1269
Tip dia. of sun(mm)	301.56	497.88
Tip dia. Of Planet(mm)	513.8	497.88
Tip dia. Of annulus(mm)	1217.110486	1347.802724
root dia. Of sun pinion(mm)	239.56	416.88
root dia. Of planet wheel(mm)	452.8	416.88
root dia. Of Annulus(mm)	1286.98	1431
Contact ratio (Sun planet)	1.571005936	1.523151207
Contact ratio (Planet annulus)	1.236165277	1.8321088
Center distance of Sun planet	379.6552432	460.9613621
Center distance of Planet annulus	379.6552432	460.9613621
Velocity of sun planet	6.309504521	1.921329544
tooth height(sun)	31	40.5
tooth height(planet)	30.5	40.5
tooth height(Annulus)	34.93475682	41.5986379

C. Design of Gear Using DIN 3990 Standard

- The gear designed using the preliminary calculation are checked in DIN 3990 standard.
- The Factor of Safety are calculated according to DIN 3990 Standard [9]- [12].

Criterion	Minimum factor of safety as per DIN 3990
Bending	1.4
Pitting	1
Scuffing (Integral temp. Method)	2

1) Stage:1 Factor of safety: m = 12 mm and face width = 188 mm

Criteria	DIN safety factors
Pitting	1.770318307
Bending	3.744507027
scuffing(integral temp method)	2.987478793

2) Stage:2 Factor of safety: m = 14 mm and face width = 140 mm

	DIN safety factors		
Criteria	Sun	Planet	Annulus
Pitting	1.657733962	2.25	
Bending	4.439692553	2.813099	3.917143
Scuffing(integral temp. method)	2.608877133	2.608877	

3) Stage:3 Factor of safety: m = 18 mm and face width = 180 mm

	DIN Safety Factors		
Criteria	Sun	Planet	Annulus
Pitting	1.480059472	1.48	
Bending	2.127782201	1.4894475	2.58169
scuffing(integral temp method)	2.650092728	2.6500927	

D. Iteration using DIN 3990 Standard

1) Stage: 1 Helical gear pair

	Iter. 1	Iter. 2	Iter. 3	Iter. 4	Iter. 5
Module	11	10	9	8	8
Facewidth	172	157	141	125	130
Criteria	DIN safety factors				
Pitting	1.5945	1.4138	1.2047	1.0270	1.0399
Bending	3.5110	2.4126	1.8169	1.3708	1.4063
Scuffing(Int. Temp.)	2.8733	2.7436	2.6439	2.4631	2.4548

2) Stage: 2 Planetary gear pair

Iteration-1 module=12 mm face width = 120 mm			
	DIN safety factors		
Criteria	Sun	Planet	Annulus
Pitting	1.28658212	1.7465	
Bending	2.71728251	1.8559321	2.634921
Scuffing(integral temp. method)	2.41079216	2.4107922	
Iteration-2 module = 11 mm face width = 110 mm			
Pitting	1.11931208	1.5194	
Bending	2.00747342	1.4757145	2.126984
Scuffing(integral temp. method)	2.28457279	2.2845728	
Iteration-3 module = 10 mm face width = 100 mm			
Pitting	0.95852814	1.3012	

Bending	1.42903025	1.1223464	1.660952
Scuffing(integral temp. method)	2.17911493	2.1791149	
Iteration-4 module = 10 mm face width = 140 mm			
Pitting	1.05982327	1.43875	
Bending	1.76166957	1.42376	1.462635
Scuffing(integral temp. method)	2.29928722	2.2992872	

Combination	Sun pinion weight	Planet weight	Annulus weight	Total weight
m=11 mm b=110 mm	3479.15	12015	3939.31	43463.5
m=10 mm b=140 mm	3832.56	13237.5	4333.64	47878.6

- As Total weight of gear in 2nd choice is more than 1st one final module and face width we have selected is:
- m = 11 mm b = 110 mm

3) Stage: 3 Planetary gear pair

Iteration-1 module = 16 mm face width =160 mm			
	DIN safety factors		
Criteria	Sun	Planet	Annulus
Pitting	1.217448397	1.217	
Bending	1.573025655	1.101117959	1.9140845
Scuffing(integral temp. method)	2.482968132	2.482968132	
Iteration-2 module =16 mm face width =190 mm			
Pitting	1.295961032	1.296	
Bending	1.762614087	1.40121	1.83338028
Scuffing(integral temp. method)	2.557916354	2.557916354	
Iteration -3 module =14 mm face width = 197 mm			
Pitting	1.102535804	1.1025	
Bending	1.372421327	0.960694929	1.193967
Scuffing(integral temp. method)	2.433758395	2.433758395	

Combination	Sun pinion weight (in grams)	Planet weight (in grams)	Annulus weight(in grams)	Total weight(in grams)
m=18 mm b=180 mm	27850.42	27850.42	15621.71	154873.81
m=16 mm b=190 mm	23244.92	23244.92	12604.649	128829.249

As Total weight of gear in 1st choice is more than 2nd one final module and face width we have selected is:

m = 16 mm b = 190 mm

V. FINAL GEAR TOOTH GEOMETRY

A. Stage 1:Helical Gear Pair

Parameters	Helical pinion	Helical Gear
Module(mm)	8	8
Pitch circle diameter(mm)	172	378
working pitch diameter(mm)	176.159	385.872
Helix angle(°)	13	13
Number of teeth	21	46
Face width(mm)	130	130
Base circle diameter(mm)	161	354
Tip diameter(mm)	196	399
Tooth height(mm)	18	18.1
Normal pressure angle(°)	20	20
Bore diameter(mm)	NA	230
Root diameter(mm)	160	362.8

Table V.1: Stage: 1 Helical Gear tooth Geometry Parameters

B. Stage 2: Planetary Gear Pair

Parameters	Sun pinion	Planet Gear	Annulus
Module(mm)	11	11	11
Pitch circle diameter(mm)	209	385	979
working pitch diameter(mm)	210	387	983
Number of teeth	19	35	89
Face width(mm)	110	110	110
Base circle diameter(mm)	196	362	920
Tip diameter(mm)	236.94	403.7	956.301
Tooth height(mm)	24.25	23.75	27.4845
Normal pressure angle(°)	20	20	20
Root diameter(mm)	188.44	356.2	1011.27

Table V.2: Stage: 2 Planetary Gear tooth Geometry Parameters

C. Stage 3: Planetary Gear Pair

Parameters	Sun pinion	Planet Gear	Annulus
Module(mm)	16	16	16
Pitch circle diameter(mm)	400	400	1200
working pitch diameter(mm)	410	410	1229
Number of teeth	25	25	75
Face width(mm)	190	190	190
Base circle diameter(mm)	376	376	1128
Tip diameter(mm)	442.56	442.56	1198.05

Tooth height(mm)	36	36	36.976 6
Normal pressure angle(°)	20	20	20
Root diameter(mm)	370.56	370.56	1272

Table V.3: Stage: 3 Planetary Gear tooth Geometry Parameters

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