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Design of Compound Planetary Gear Train

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Abstract: Planetary gear is widely used in many sectors of industries. Gears in the planetary gear trains are one of the most critical components in which failure of one gear will affect the whole transmission system, thus it is very necessary to determine the causes of failure to reduce it. Planetary Gear Trains have been used in Industry for their many advantages which includes high torque/weight ratio, comparatively smaller size, improved efficiency and highly compact package. This research will help in design calculation of compound planetary gear pair according DIN 3960 and DIN 3990 standard.
Keywords: Solid Works, DIN Standard, Kisssoft

I. INTRODUCTION

Planetary gearing is a gear system consisting of one or more outer gears, or planet gears, revolving about a central, or sun gear. Typically, the planet gears are mounted on a movable arm or carrier, which itself may rotate relative to the sun gear. Planetary gearing systems also incorporate the use of an outer ring gear or annulus, which meshes with the planet gears. Planetary gears are typically classified as simple or compound planetary gears. Simple planetary gears have one sun, one ring, one carrier, and one planet set. Compound planetary gears involve one or more of the following three types of structures: meshed-planet (there are at least two more planets in mesh with each other in each planet train), stepped-planet (there exists a shaft connection between two planets in each planet train), and multi-stage structures (the system contains two or more planet sets). Compared to simple planetary gears, compound planetary gears have the advantages of larger reduction ratio, higher torque-to-weight ratio, and more flexible configurations. The different types of failure of gears and their possible remedies are mentioned in [1] S. Jyothirmai, [2] K. Aslantas, [3] S. H. Chang, [4] Osman Asi 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. [11] Dr. Alexander Kapelevich had focused on analysis and design of planetary gear arrangements that provide high gear ratios. A special, two-stage planetary arrangement may use a gear ratio of over one hundred thousand to one. [12] Bernd-Robert Höhn presents planetary gear transmissions; designed according to ISO 6336, optimized in terms of efficiency, weight and volume, and calculated using low-loss involute gears as well as the maximum feasible number of planets.

This paper presents analysis and design of planetary gear arrangements that provide extremely high gear ratios. The real scope of this paper is on design parameter, gear ratio, gear tooth combination, stress, and factor of safety.

II. COMPUTATIONAL METHODOLOGY

The compound planetary gear train taken from the Elecon Engineering shown in figure 2. The sun gear, planet gear, annulus are design in SolidWokrs software. The arrangements of planetary gear train are shown in figure 1. Figures 1 present differential-planetary arrangements with compound planet gear train. In the arrangement, the sun gear is engaged with planet 2a and planet 2a is engaged with stationary ring gear 3a. Planet 2a and planet 2b are making compound planet gear. Planet 2b is engaged with rotating ring gear 3b. 3b is the output gear box. There are one-stage differential-planetary arrangements that provide much higher gear ratios. In these arrangements the output shaft is connected to the second rotating ring gear rather than the carrier, as with the planetary gear.

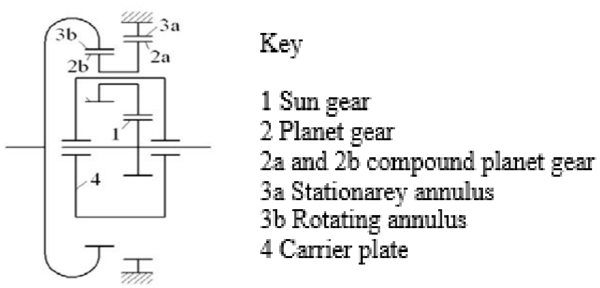

Figure 1 2d arrangement of planetary gear train

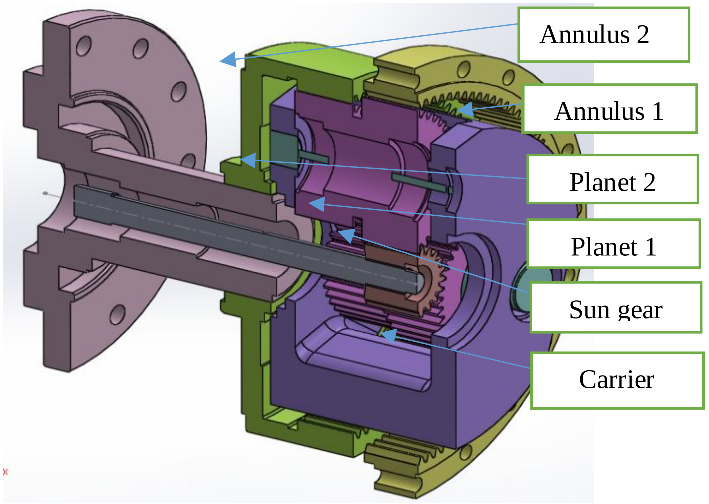

Figure 2 Compound planetary gear train

Table 1 Specification considered for gears

Parameters	Sun	Planet 1	Annul-us 1	Planet 2	Annul-us 2
No. of teeth	20	46	112	40	106
Pitch dia.	65	149.5	364	130	344.5
Module	3.25	3.25	3.25	3.25	3.25
Center distance	107.2	107.25	107.25	107.25	107.25
Face width	53	53	53	55	55
Pressure angle (^{o})	20	20	20	20	20
Base dia.	61.08	140.49	342.04	122.16	323.74
Outside dia.	72.9	155.10	357.5	135.73	338.00
Root dia.	58.40	140.61	372.12	121.11	352.62

Generally many company used kisssoft software for gear design. Kisssoft have module to calculate gear design for only simple planetary gear train. But in kisssoft software there is no module for calculation of compound planetary gear. Here, all calculation are done according to DIN 3960 standard. This paper focus on geometry calculation of all gears. In this arrangement, we can considered two stage i.e. 1-2a-3a that's make simple planetary gear and 2b-3b that's make compound planetary gear train. So we can calculate data in kisssoft software for simple planetary gear train and we can compare geometry data between numerical calculation and kisssoft software.

III. DESIGN CALCULATION

For gear mechanism, reduction ratio is ratio of no. of teeth on pinion to the no. of teeth on gear. Gear reduction mechanism reduce the speed of driven shaft to that of driving shaft. Here, we used table method to find the reduction ration in compound planetary gear train. In table, all value show the rpm of gear. In the table first assumption is done by fixing carrier plate and giving one rotation to sun gear. Remaining gear speed calculated according to sun gear. Then we give unknown factor X as a speed of sun gear. Then we added speed of carrier plate that is Y. The reduction ratio for this gear train is ratio of speed of sun gear to the speed of annulus gear.

Table 2 reduction ratio of gear train

Rpm					
Action	Carrier	Sun(1)	Planet(2a/2b)	Annulus(3a)	Annulus(3b)
Fixed carrier	0	1	$-\frac{Z_1}{Z_{2a}}$	$-\frac{Z_1}{Z_{3a}}$	$-\frac{Z_1}{Z_{2a}}\times\frac{Z_{2b}}{Z_{3b}}$
Fixed carrier	0	X	$-\frac{Z_1}{Z_{2a}}X$	$-\frac{Z_1}{Z_{3a}}X$	$-\frac{Z_1}{Z_{2a}}\times\frac{Z_{2b}}{Z_{3b}}X$
Add y	Y	Y+X	$Y-\frac{Z_1}{Z_{2a}}X$	$Y-\frac{Z_1}{Z_{3a}}X$	$Y-\frac{Z_1}{Z_{2a}}\times\frac{Z_{2b}}{Z_{3b}}X$

Ratio $u = \frac{\text{Speed of Sun Gear}}{\text{Speed of Annulus}} = \frac{Y+X}{Y-\frac{Z_1}{Z_{2a}}\times\frac{Z_{2b}}{Z_{3b}}X}$ or

$$\text{Reduction ratio, } u = \frac{1+\frac{Z_{3a}}{Z_1}}{1-\frac{Z_{2a}\times Z_{3a}}{Z_{2a}\times Z_{3a}}}$$

$$u = \frac{1+\frac{112}{20}}{1-\frac{40\times 112}{46\times 106}} = 81.26$$

Speed of carrier:

Revolution of annulus 1 is 0. So, $Y-\frac{Z_1}{Z_{2a}}X = 0$

$$Y = \frac{Z_1}{Z_{2a}}X$$

$$X = \frac{112}{20}Y = 5.6Y$$

Speed of sun gear: $X + Y = 1800$

$6.6Y = 1800$

$Y = 272.72$ rpm

Speed of Planet:

$$\frac{N_p}{N_s} = -\frac{T_s}{T_a}$$

$$N_p = -\frac{20}{46}\times 1800$$

$$N_p = 782.60 \text{ rpm}$$

Speed of Annulus 2(3b):

$$\frac{N_{3b}}{N_s} = -\frac{1}{u}$$

$$\frac{N_{3b}}{1800} = -\frac{1}{81.26}$$

$N_{3b} = 22.15$ rpm

Table 4 shows calculation equation for external gear and table 5 shows calculation equation for internal gear and external gear.

Table 3 Calculation of gear geometry for external gear

No.	Item	Symbol	Formula
1	Module	m	-
2	Pressure angle	α	-
3	Number of teeth	N	-
4	Total profile shift	Σx	
5	Coefficient of profile shift	$x_{1,} x_2$	-
6	Inv α		tan α – α
7	Involute function	Inv α_w	$2\tan\left(\frac{x_1+x_2}{z_1+z_2}\right)+\text{inv}\,\alpha$
8	Working pressure angle	α_w	Find from involute function table
9	Center distance increment factor	y	$\frac{z_1+z_2}{2}\left(\frac{\cos\alpha}{\cos\alpha_w}-1\right)$
10	Center distance	a_x	$\left(\frac{z_1+z_2}{2}+y\right)\times m$
11	Pitch diameter	d	Zm
12	Base diameter	d_b	d cos α
13	Working pitch diameter	d_w	$\frac{d_b}{\cos\alpha_w}$
14	Addendum	$h_{a1,} h_{a2}$	$(1 + y – x_2)$ m, $(1 + y – x_2)$ m
15	Whole depth	h	$[2.25 + y – (x_1 + x_2)]m$
16	Outside diameter	d_a	$d + 2h_a$
17	Root diameter	d_f	$d_a – 2h$

Table 4 Calculation of gear geometry for external and internal gear

No.	Item	Symbol	Formula
1	Module	*m*	-
2	Pressure angle	*α*	-
3	Number of teeth	*N*	-
4	Total profile shift	*Σx*	-
5	Coefficient of profile shift	x_1, x_2	-
6	Inv α		*Tan α – α*
7	Involute function	*Inv* α_w	$2\tan\left(\frac{x_1+x_2}{z_1+z_2}\right)+inv\,\alpha$
8	Working pressure angle	α_w	Find from involute function table
9	Center distance increment factor	*y*	$\frac{z_1+z_2}{2}\left(\frac{\cos\alpha}{\cos\alpha_w}-1\right)$
10	Center distance	a_x	$\left(\frac{z_1+z_2}{2}+y\right)\times m$
11	Pitch diameter	*d*	*Zm*
12	Base diameter	d_b	*d cos α*
13	Working pitch diameter	d_w	$\frac{d_b}{\cos\alpha_w}$
14	Addendum	$h_{a1,} h_{a2}$	$(1 + x_1)$ *m*, $(1 – x_2)$ *m*
15	Whole depth	*h*	2.25 *m*
16	Outside diameter	d_{a1}(Ext. gear) d_{a2}(int. gear)	$D_1 + 2ha_1$ D_2 - $2h_{a2}$
17	Root diameter	d_{f1}(Ext. gear) d_{f2}(int. gear)	D_{a1} - $2h$ D_{a2} - $2h$

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IV. COMPARISON BETWEEN CALCULATED DATA AND KISSSOFT DATA

Here, we cannot calculate whole data for this type of compound planetary gear train in kisssoft software. But we can calculate data for the first stage simple planetary stage in kisssoft software. Here, we have compared geometry data and various factor required for safety between calculated data and kisssoft data. Here, we have assumed two pairs of gear combination i.e. 18-40-106 and 20-46-112.

Table 5 Geometry data for 20-46-112

	sun		Planet 1		Annulus 1	
parameters	Kisssoft data	analytical data	Kisssoft data	analytical data	Kisssoft data	analytical data
No of teeth	20		46		112	
Module (mm)	3.25		3.25		3.25	
Pcd (mm)	65	65	149.5	149.5	364	364
Reference center distance	107.25	107.25	107.25	107.25	107.25	107.25
Speed (rpm)	1800	1800	782.6	782.60	0	0
Center distance based on profile shift (mm)	107.62	107.56	107.62	107.56	107.62	107.56
Working pressure angle	20.45	20.45	20.45	20.45	20.45	20.45
Base dia (mm)	61.08	61.08	140.48	140.49	342.04	342.04
Whole depth (mm)	7.23	7.24	7.23	7.24	7.30	7.31
Outside dia (mm)	73.19	72.9	155.22	155.1	357.5	357.5
Root dia (mm)	58.40	58.4	140.61	140.61	372.15	372.15
Contact ratio	1.58	1.6	1.58		1.95	1.9
Torque(N-mm)	360751.2	360751.2	-	-		2020206.7
Tangential force(N)	3700.01	3700.01	3700.01	3700.01	3700.01	3700.012

Table 6 Geometry data for 18-40-106

	Sun		Planet 1		Annulus 1	
parameter	Kisssoft data	analytical data	Kisssoft data	analytical data	Kisssoft data	analytical data
No of teeth	18		40		106	
module	3.25		3.25		3.25	
pcd	58.5	58.5	143	143	344.5	344.5
Reference center distance	100.75	100.75	100.75	100.75	100.75	100.75
speed	1800	1800		736.36	0	0
Center distance based on profile shift	101.04	101.04	101.04	101.04	101.04	101.04
Working pressure angle	20.45	20.46	20.45	20.46	20.45	20.46
Base dia	54.97	54.976	134.37	134.38	323.72	323.72
Whole depth	7.21	7.22	7.21	7.22	7.12	7.13
Outside dia	66.51	66.36	148.72	148.56	338	338
Root dia	51.90	51.90	134.11	134.11	352.62	352.62
Contact ratio	1.56	1.63	1.56/1.9	1.36/1.93	1.95	1.93
Torque(N-mm)	360751.2	360751.2	-	-	2124423.7	2124423.7
Tangential force(N)	4111.12	4111.12	4111.12	4111.12	4111.12	4111.12

In this study, comparison of analytical value and Ansys value were considered. Calculations were done based on DIN 3960 for this gear train. Now in order to justify the calculated data, we need to compare the analytical calculation with kisssoft data and it was found very close.

V. CONCLUSION

In this paper, attempt has been made to find safe design and to compare calculated data through analytical approach based on DIN 3960 as well as kisssoft software.

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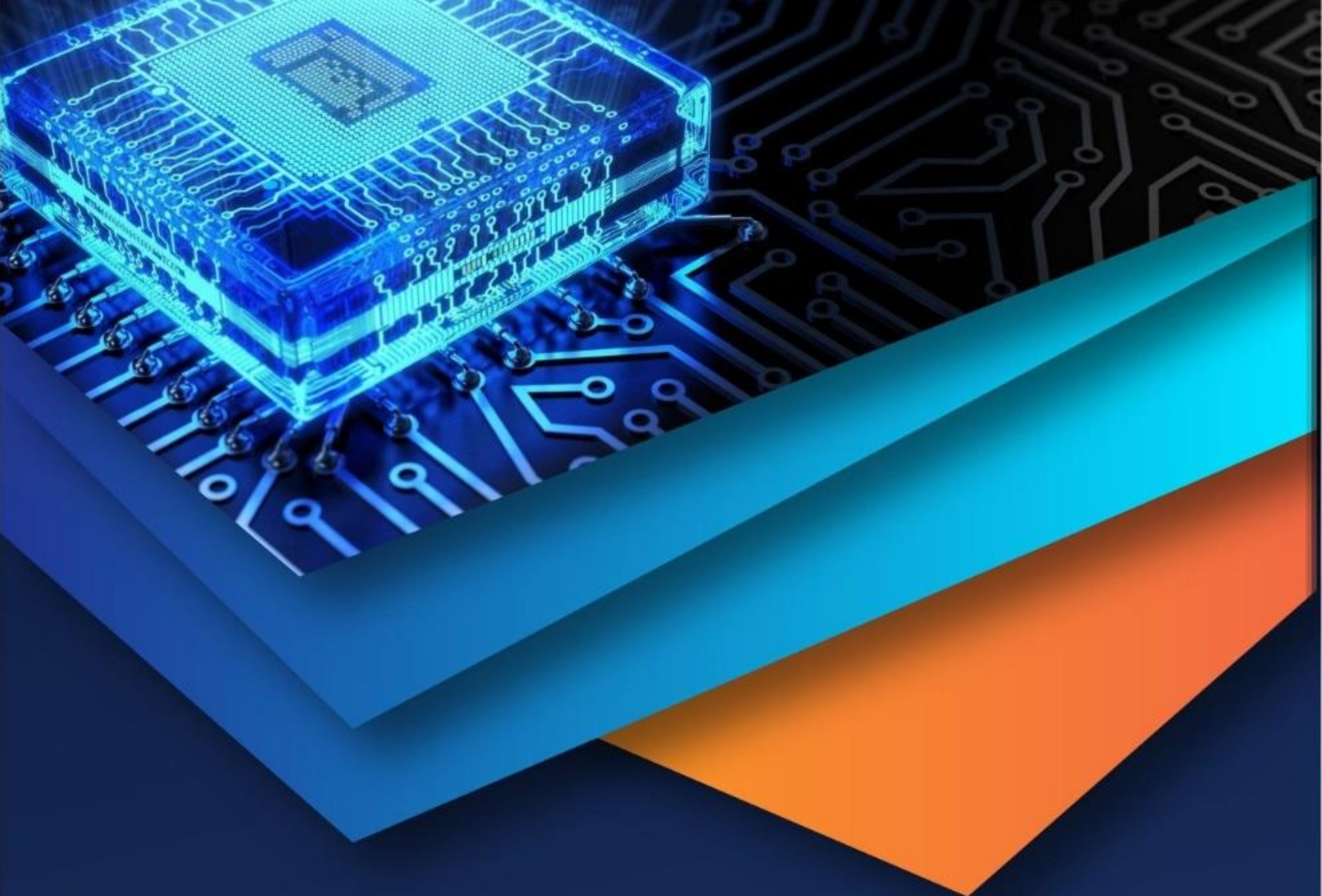

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