Design, Analysis and Optimization of Planetary Gearbox: A Review

Shailesh Chaudhari[1] Jayveer shah[2] Tarun Garg[3]
[1]P.G.Student [2]Assistant Professor [3]G.M. Mechanical Maintenance
[2]Mechanical Engineering Department
[1]Gujarat Technological University, India [2]Veerayatan Group of Institutions, India [3]Welspun Steel Ltd, India

Abstract- Planetary gear systems normally consist of a centrally pivoted sun gear, a ring gear, and several planet gears found between the sun gear and ring gear. Compared to traditional gear boxes, the planetary gear systems have some advantages. Planetary gear systems possess larger efficiency in small volumes because of the compact combination of gears in the planetary gear system. 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. Optimum modeling for analysis is needed to carry out a more precise analysis result. Therefore, the modeling for strength and durability analysis is focused on gear train part.

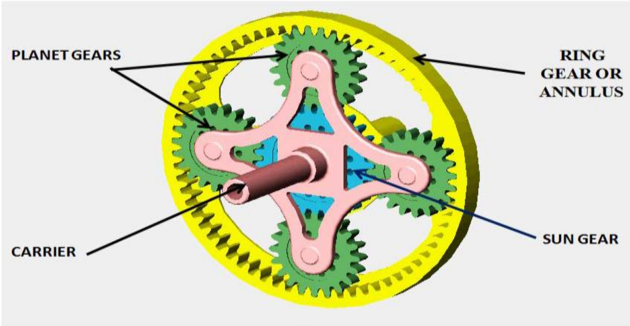
Key words: Complex planetary gear system, optimization of planetary gearbox, Weight reduction

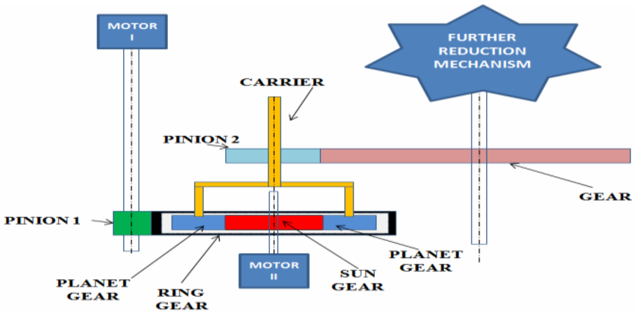
NOMENCLATURE

- No. of teeth on Gear (Z)
- No. of teeth on Sun Gear (S)
- No. of teeth on Planet Gear (P)
- No. of teeth on Pinion Gear 1 (P1)
- No. of teeth on Pinion Gear 2 (P2)
- No. of teeth on Ring Gear External (RE)
- No. of teeth on Ring Gear Internal (RI)
- Total Gear Reduction Ratio (TR)
- T_S = number of teeth on sun
- T_P = number of teeth on planet
- T_A = internal number of teeth on annulus
- NA = speed of annulus NS = speed of sun
- N_P = speed of planet N_C = speed of carrier

I. INTRODUCTION

An epicyclic gear set has some gear or gears whose center revolves about some point.Here is a gearset with a stationary ring gear and three planet gears on a rotating carrier.The input is at the Sun, and the output is at the planet carrier. The action is epicyclic, because the centers of the planet gears revolve about the sun gear while the planet gears turn.Finding the gear ratio is somewhat complicated because the planet gears revolve while they rotate.





A. *Refer Figures :*

- Sun: The central gear.
- Planet Gear: Peripheral gears, of the same size, meshed with the sun gear and Annulus.
- Planet carrier: Holds one or more peripheral planet gears, of the same size, meshed with the sun gear.
- Ring Gear/Annulus: An outer ring with inward-facing teeth that mesh with the planet gear or gears.

Especially in applications requiring:

- High reducation ratios
- High torque transmission
- High redial loads on output s

II. LITERATURE SURVEY

The assembly industries of today use electrical and pneumatic nutrunners for many of their fastening applications. The nutrunner market is full of competing manufacturers doing their best to make their product stand out and be ahead of the competition. Atlas Copco have been making nutrunners since early 1900's and are always trying to find ways to improve the performance of their tools. One step in this improvement process is to map and analyze the losses of the in-tool planetary gearbox. This Bachelor of Science thesis will investigate the losses occurring and with an experimental test rig try to explain why they take place and how to minimize them.Power losses with unwanted heating of the tool as a result are unavoidable in all rotating machines. But knowledge about the losses gives a fundament to improve the efficiency and thus reducing the heat generated. In Atlas Copcos current range of tools, a major part the generated losses are located in the planetary gearbox.

The main objective with this thesis work is to analyze and understand the losses existing in the gearbox and from this knowledge develop a model to help designing a more efficient gearbox. The model made should be valid to planetary gearboxes of different sizes and gearings. The work will result in an understanding of the occurring losses and a Matlab program, usable for simulating the power losses in the gearbox. [1]

Light-weight construction and consideration of available resources result in gearbox designs with high load capacity and power density. At the same time, expectations for gear reliability are high. Additionally, there is a diversity of planetary gears for different applications. Gears with one or more stages and with one or more gearbox inputs and outputs are not uncommon.Furthermore, different kinds of teeth exist: e.g., spur and helical gears, and also double-helical gears are doable. For the mounting of shafts and gearings, roller bearings and sliding bearing are used .

All of these conditions require exceptional and robust design criteria, including maximum load and dynamic loads under different load situations. Experience with drivetrains with stiff foundations and constant, external loads is not directly applicable, due to unique boundary conditions, dynamic excitation of the structure, and changing influences by external conditions.[2]

For the dimensioning of highly stressed toothings the analysis of load distribution and the definition of tooth flank modifications belongs to the principal tasks. Similar problems appear at the evaluation of toothing damages and failure modes of whole gears. Although there are a large number of standards for the calculation of spur gears, It is necessary to have special and powerful calculation software which is reflecting the force-- deformation--relation for every point of the contact area more precisely. The cause is the divergence from the conjugated toothings at gear wheels of spur and planetary gears. Often the flanks aremodified in height and width direction. With thesemodifications the load--dependent deformations of the toothing and the surroundings as well as toothing errors, position errors of the housing boreholes andbearing clearance can be compensated. Also thegear noise as well as the load capacity is influencedin a positive way.

For the determination of the load distribution in planetary gear stages the deformation analysis is a more complex task as for spur gear stages. The deformation of the wheel body as well as the adjacent structures and the planet carrier can’t be calculated efficiency in an analytical way. They need to be investigated with FE calculations or extended model approaches. [3]

A flexible pin mechanism considered in the paper consists of a planet gear, a sleeve (integrated with a hydraulic bearing) and a pin. the section view of a stage of planetary gear set with the flexible pin mechanism, designed by ITRI. There are totally eight planets in the planetary gear set. The focus on designing the flexible pin mechanism is not only its stiffness but also its deformation under a given load. The stiffness affects the load sharing among the planet gears of the planetary gear drive, while the deformation is important for the load distribution on the tooth flanks. In order to analyze the influences of the geometrical parameters of the flexible pin on the deformation and the stress, a simplified model with corresponding design parameters is introduced .[4]

In spite of the number of investigations devoted to gear research and analysis there still remains to be developed, a general numerical approach capable of predicting the effects of variations in gear geometry, shear, wear and bending stresses. The objective of this work is to use ANSYS to de- velop theoretical models of the behavior of planetary gears in mesh, to help to predict the effect of gear tooth stresses and deflection. The main focus of the current research as developed here is to develop and to determine appropriate models of contact elements, to calculate various stresses and using ANSYS and compare the results with theoretical.

The project work mainly deals with:

- Checking of wear stresses & bending stresses using IS 4460 equations for sun gear.
- Force calculations for planetary gear
- Calculate the values for sun gear tooth for bending, shear, wear & deflection using theoretical method.
- Generation of gear tooth profile in Pro-E3.
- Create 3D model of circular root fillet & trochoidal root fillet of gear tooth for simulation using Pro-E3.
- Importing Pro-E model in ANSYS in IGES format.
- Comparison of the results of the 3D analyses from ANSYS with the theoretical values.
- Comparison of ANSYS results in circular root fillet & trochoidal root fillet.[5]

A. *Calculations for Planetary System Speeds:*

Let,
T_S = number of teeth on sun
T_P = number of teeth on planet
T_A = internal number of teeth on annulus
N_A = speed of annulus
N_S = speed of sun
N_P = speed of planet
N_C = speed of carrier
Speed of annulus N_A is given as

To account for the combined rotation and revolution of the planet gear, we use a two step process.
First, we lock the whole assembly and rotate it one turn Counter-Clockwise. (Even the Ring, which we know is fixed)

We enter this motion into a table using the convention:

CCW = Positive

CW = Negative

	Sun	Planet	Ring	Arm
Rotate Whole Assembly CCW	+1	+1	+1	+1

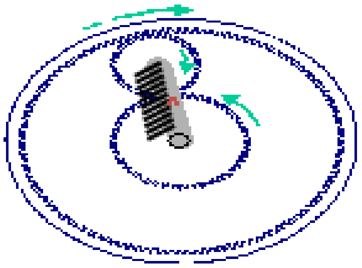
The Tabular (Superposition) Method

Next, we hold the arm fixed, and rotate ***whichever gear is fixed during operation*** one turn clockwise.

Here, we will turn the Ring clockwise one turn (-1), holding the arm fixed.

- The Planet will turn N_{Ring} / N_{Planet} turns clockwise.
- Since the Planet drives the Sun, the Sun will turn (N_{Ring} / N_{Planet}) x (-N_{Planet} / N_{Sun}) = - N_{Ring} / N_{Sun} turns (counter-clockwise).
- The arm doesn't move.

We enter these motions into the second row of the table



	Sun	Planet	Ring	Arm
Rotate Whole Assembly CCW	+1	+1	+1	+1
Hold Arm, Rotate Ring CW	N_{Ring} / N_{Sun}	- N_{Ring} / N_{Planet}	-1	0

	Sun	Planet	Ring	Arm
Rotate Whole Assembly CCW	+1	+1	+1	+1
Hold Arm, Rotate Ring CW	N_{Ring} / N_{Sun}	- N_{Ring} / N_{Planet}	-1	0
Total Motion	1 + N_{Ring} / N_{Sun}	1 - N_{Ring} / N_{Planet}	0	+1

Finally, we sum the motions in the first and second rows of the table.

Now, we can write the relationship:

$$n_{Sun} = 1 + \frac{N_{Ring}}{N_{Sun}} \times n_{Arm}\ , \quad or$$

$$n_{Arm} = \frac{1}{1 + \frac{N_{Ring}}{N_{Sun}}} \times n_{Sun}$$

If the Sun has 53 teeth and the Ring 122 teeth, the output to input speed ratio is +1 / 3.3 , with the arm moving the same direction as the Sun.

The configuration shown here, with the input at the Sun and the output at the Ring, is <u>not</u> epicyclic.

It is simply a Sun driving an internal Ring gear through a set of three idlers.

1) The gear ratio is:

$$\frac{n_{ring}}{n_{sun}} = \frac{-N_{sun}}{N_{planet}} \times \frac{N_{planet}}{N_{ring}} = \frac{-N_{sun}}{N_{ring}}$$

n = speed; N = # Teeth

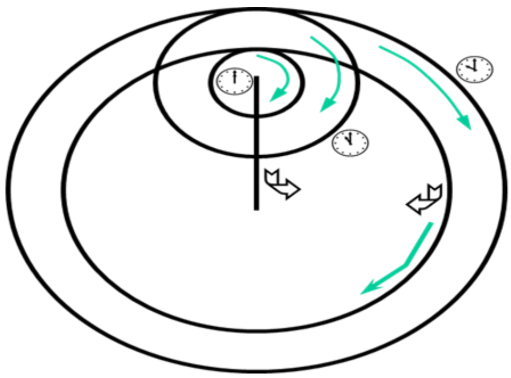
Where the minus sign comes from the change in direction between the two external gears.

If the Sun has 53 teeth and the Ring 122 teeth, the ratio is -1 / 2.3 .

Example:

We turn Ring 4 one turn CW (-1).

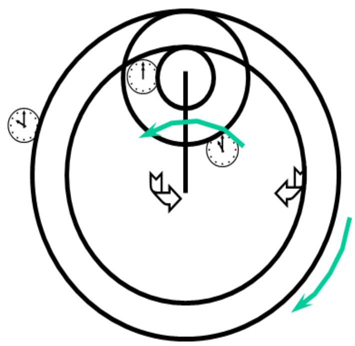
- Ring 4 drives Gear 3, which turns + N_4/N_3 x (-1) = - N_4/N_3 rotations.
- Gear 2 is on the same shaft as Gear 3, so it also turns - N_4/N_3 rotations.
- Gear 2 drives Ring 1, which turns + N_2/N_1 x n_2 = + N_2/N_1 x (- N_4/N_3) = - N_2N_4/N_1N_3 rotations.
- The arm was fixed, so it does not turn



We enter these motions into row two of the table:

Then we add the two rows to get the total motion

	Ring 1	Gear 2	Gear 3	Ring 4	Arm 5
Rotate Whole Assembly CCW	+1	+1	+1	+1	+1
Hold Arm, Rotate Ring CW	- N_2N_4/N_1N_3	- N_4/N_3	- N_4/N_3	-1	0



Then we add the two rows to get the total motion

	Ring 1	Gear 2	Gear 3	Ring 4	Arm 5
Rotate Whole Assembly CCW	+1	+1	+1	+1	+1
Hold Arm, Rotate Ring CW	$-N_2N_4/N_1N_3$	$-N_4/N_3$	$-N_4/N_3$	-1	0
Total Motion	$1 - N_2N_4/N_1N_3$	$1 - N_4/N_3$	$1 - N_4/N_3$	0	+ 1

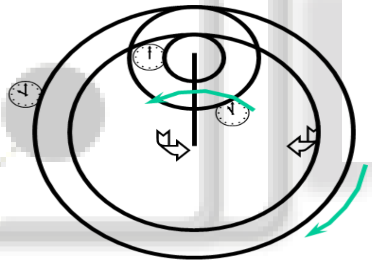
And we can write the relationship:

$$n_{Ring1} = 1 - \frac{N_2N_4}{N_1N_3} \times n_{Arm}, \quad or$$

$$n_{Arm} = \frac{1}{1 - \frac{N_2N_4}{N_1N_3}} \times n_{Ring1}$$

For our example

Ring 1: N_1 = 100 Teeth
Gear 2: N_2 = 40 Teeth
Gear 3: N_3= 20 Teeth
Ring 4: N_4 = 78 Teeth



We compute:

$$n_{Arm} = \frac{1}{1 - \frac{N_2N_4}{N_1N_3}} \times n_{Ring1}$$

$$n_{Arm} = \frac{1}{1 - \frac{40 \times 78}{100 \times 20}} \times n_{Ring1}$$

$$n_{Arm} = \frac{1}{1 - 1.56} \times n_{Ring1} = \frac{-1}{0.56} \times n_{Ring1}$$

$$n_{Arm} = -1.786 \times n_{Ring1}$$

The output arm rotates almost twice as fast as the input ring, and in the opposite direction.

– Output direction is dependent on the numbers of teeth on the gears!

III. CONCLUSION

For the evaluation of strength and durability of gear element stress concentration factor are generally used. In order to optimize the weight of planetary gearbox sun and pinion gear, the stress induced in sun and pinion gear must be studied. The review of previous research permits to conclude that the analysis of of sun pinion, planet pin and planet carrior and the optimization of planet carrior and planet.

By static analysis of mentioned parts, all parts are in safe limit.in this, the failure criteria used is Von Mises Stress theory which demonstrates that the parts are safe within the safe limit of yield strength of the material of parts.

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