ISSN 2395-1621

Investigation of effect of rim thickness on strength of annular gear of planetary gearbox under dynamic conditions

#1M. B. Raut, #2Prof. S. L. Shinde

1mbr191288@yahoo.co.in
2samirlshinde@yahoo.co.in

#12Department of Mechanical Engineering, Savitribai Phule Pune University, Pune, India

ABSTRACT

Planetary gears nowadays, finding more and more applications as it is useful for transmitting power with large speed reductions or multiplications. Hence, Research in planetary gears has become necessity. The rim thickness of the annular gear is one of the important parameter of planetary gearbox to meet design objectives. An optimum rim thickness is required to meet the strength criterion at the same time to reduce mass of the gearbox and to save on material cost. Repetitive run-ups, time varying loading and speed conditions are very common in many applications of planetary gearboxes. Hence, in this paper major attention is given to the dynamic conditions. Experiments are performed to find out the strain using strain gauges on annular gear of planetary gearbox. On the other hand, a finite element model of an annular gear is developed on a commercial FEA software. The results obtained in the FEA software are validated against experimental results. The validated FE model then can be used to make the conclusions on how much the rim thickness must be kept.

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

***Keywords*— annular gear, dynamics, FEM, planetary gearbox, rim thickness.**

I. INTRODUCTION

Planetary gears have a great ability of power transmission, because of its compact design, high power density, and coaxial arrangement of the input and the output shafts, planetary gear drives play a very important role for power transmission.Research developments in planetary gears become a necessity in order to improve efficiency and compactness and to decrease noise and price. Most studies were devoted to stationary condition where loads and speeds are constants. However, repetitive run up, time varying loading and speed conditions are very common in many industrial applications of planetary gears which imply non stationary operations. If we add excessive applied torque, manufacturing or installation errors, the transmission will be subjected to instability and severe vibrations.Dynamic analysis of planetary gears is essential for eliminating noise and vibration problems of the products they are used in. The dynamic forces at the sun-planet and annular-planet meshes are the main sources of wear problems. Although planetary gear sets have generally more favorable noise and vibration characteristics compared to parallel-axis gear systems, planetary gear set noise still remains to be a major problem. The dynamic gear mesh loads that are much larger than the static loads are transmitted to the supporting structures, in most cases, increasing gear noise. Larger dynamic loads also shorten the fatigue life of the components of the planetary gear set including gears and bearings.As planetary gear sets possess unique kinematic and geometric properties, they require specialized design knowledge. One type of the key parameters, the rim thickness of the gears, must be defined carefully in order to meet certain design objectives, the rim of the each gear forming the planetary gear set must be as thin as possible in order to minimize mass. Besides reducing mass it adds gear flexibility through reduced rim thickness. It is well known that flexible internal gear helps improve the load sharing amongst the planets when a number of manufacturing and carrier errors are present. Thus reducing rim thickness of gears, improve functionality of the gear set by minimizing adverse effects of gear.

However these benefits come at the expense of increased gear stresses. Thus the practical design question is how thin gear rim thickness should be without any durability problems is not possible to answer based on static analysis alone. It is expected that behavior of planetary gear sets changes under dynamic conditions, as the system flexibility is increased, potentially increasing gear stresses to a certain level.Model is created in Creo Parametric software and imported in ANSYS for simulation. After simulation the validated FE model can be used to make predictions regarding stresses and strains in the annular gear, which will facilitate designing annular gear of optimum thickness.

II. EXPERIMENTAL SETUP

Experimental setup is shown in the following fig.1. Planetary gear box contains sun gear, three planet gears and annular gear. In this gear box, a sun gear is input and the carrier is output. The planetary gear box is driven by the 3 HP motor having maximum 3000 rpm. The flexible coupling is used in between output shaft of motor and input shaft of planetary gear box. Brake drum dynamometer is attached on the output shaft of planetary gear box and two S type load cells are used to measure torque with a two decimal accuracy. Wherein one load cell is fixed to the frame and other is attached to the screw for applying tensile force on it. The complete frame is attached to the concrete foundation with the help of foundation bolts to give rigidity and damp the vibration produced by the gear box.

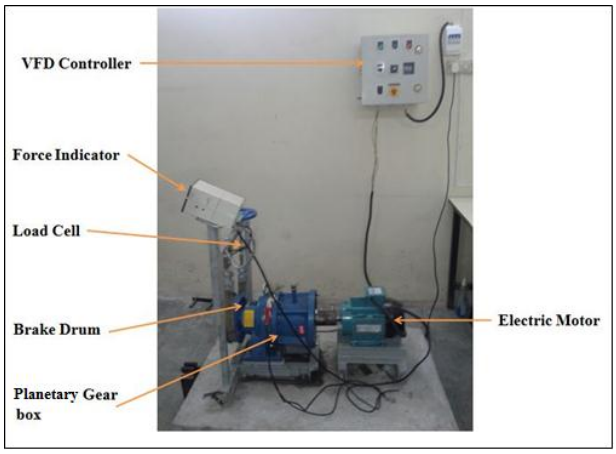

Fig. 1Actual experimental setup

DW-43 FFT analyser is used to measure strains developed over the annular gear, and read over the computer with DEWESoft software. Strains are measured using strain gauges, strain gauge conductors are thin strips of metallic film deposited on a non- conducting substrate material called the carrier. The strain gauge is connected into a Wheatstone bridge circuit with a combination of four active gauges. Strain gauges are stuck on to the outer surface of the annular gear. The annular gear is fixed with the help of eight bolts to the gear box casing. Specifications of Planetary gearbox are

TABLE I
SPECIFICATIONS OF PLANETARY GEARBOX

Parameter	Sun gear	Planet Gears	Annular gear
Number of teeth	10	19	50
Module	3.25	3.25	3.25
Pressure angle	20^0	20^0	20^0
Diameter of pitch circle (mm)	35.75	63.38	162.50
Root Diameter (mm)	27.62	55.25	170.62
Outer Diameter (mm)	42.25	69.88	156.00
Tooth height (mm)	7.31	7.31	7.31
Face width (mm)	31	30	31
Young's modulus (MPa)	2.05 x 10^5	2.05 x 10^5	2 x10^5
Poisson's ratio	0.29	0.29	0.29
Material	SAE 8620	SAE 8620	EN8 / AISI1040
Density (kg/m^3)	7850	7850	7845
Yield Strength (N/mm^2)	833	833	415

Variable-frequency drive is used to control AC motor speed and torque by varying motor input frequency and voltage.

III.F.E MODELLING & ANALYSIS

Simulation is done on ANSYS software, to find out the stresses in the annular gear under dynamic conditions. The results obtained with simulation are validated with experimental results. The validated FE model is then used to find minimum thickness of the rim for operating conditions. Further, the FE model can also be used to predict optimum number of bolts.

The thickness of the rim is very small as compared to the diameter of the rim, it allows us to make use of plane stress assumption. For this work, stresses are calculated in ANSYS software by using plane stress condition. PLANE 42 type of element is used to simulate plane stress condition.

Further, we can also observe that the rim is having symmetry about X and Y axes. Under such conditions it is better to go for half or quarter symmetric boundary conditions. This not only gives us chance to use fine mesh but also reduces computational efforts required by the CPU by approximately half and quarter respectively.

To develop the plane stress model an annular rim gear surface is first extracted from the solid model of rim gear. This surface is then imported to ANSYS software. Meshing is done by using quad dominant technique. (Node count = 3352, element count = 3041)

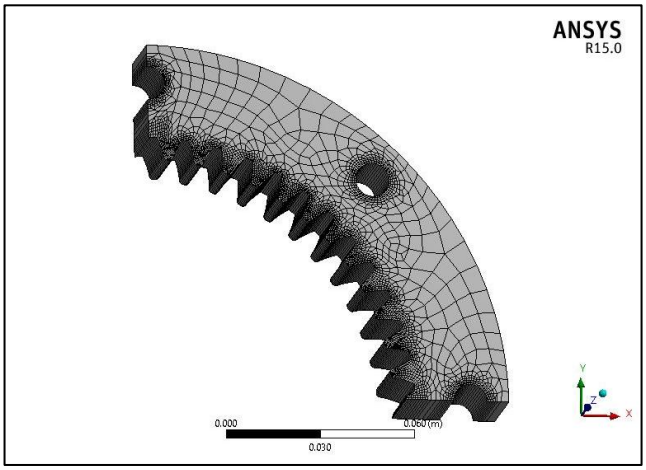

Fig. 2 Meshed FE model of quarter annular gear

To simulate quarter symmetry the nodes lying on x-axis are constrained not to move in y direction and nodes lying on y-axis are constrained not to move in x direction. In addition the boltholes are fixed. The boundary conditions are summarised in the Fig.3.

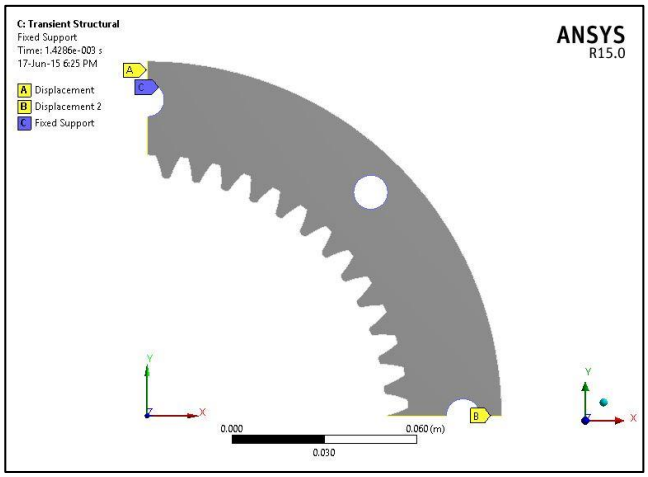

Fig. 3 FE model of quarter annular gearwith boundary conditions

To simulate dynamic nature of the problem transient structural type of analysis is done. As the planet gear rotates the force changes its location and jumps from one teeth to the next. The time required for one planet gear to move from one teeth to next teeth is calculated based on speed of the planet gear and number of teeth of annular gear. A time varying force is then applied on all teeth to simulate the dynamic nature of force. Refer Fig. 4 Force value is calculated using torque applied (40 Nm) at the belt dynamometer.

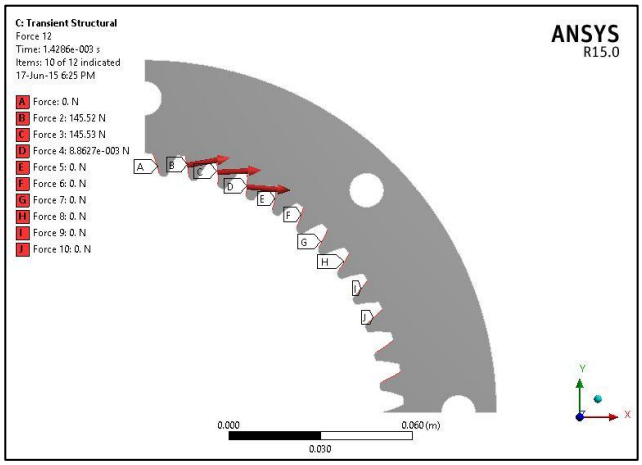

Fig. 4Forces appliedon the teeth of the annular gear

The analysis is done for 2500 rpm, considering only one planet. Such transient structural FE model is then solved to obtain strain produced at the probing location.

IV.RESULT & DISCUSSION

The experiments results were recorded using DW43 measuring instrument. Sample results for 2500 rpm are shown in Fig.5

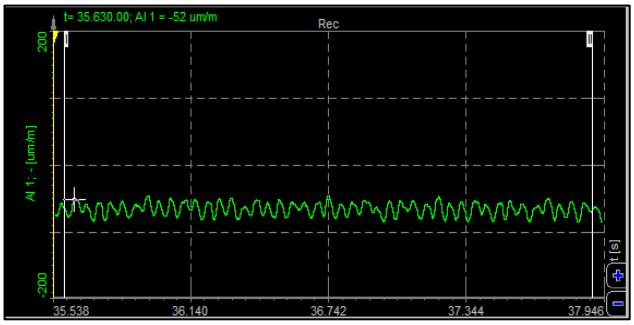

Fig. 5 Result obtained in DEWESoft (for 2500 rpm)

Similar results were obtained by varying motor speed viz. 1500, 2000, 2500 rpm. The cyclic nature seen in the strain value shows how strain value varies as the planet gear passes from the probing region. The strain value increases as the planet passes from the nearest teeth of the annular gear where the strain gauge is attached. It again achieves its normal value till the next planet comes near to it. The average increase in strain value is recorded and is summarised in the Table II.

TABLE II
EXPERIMENTAL RESULTS

Speed of motor (rpm)	Strain recorded (µm/m)
1500	2.49
2000	2.87
2500	2.6

After doing transient structural analysis in ANSYS strain results are obtained as shown in the Fig. 6. The strain value obtained is 2.564 x 10^{-6} m/m which is close to 2.6 µm/m obtained in the experimental results.

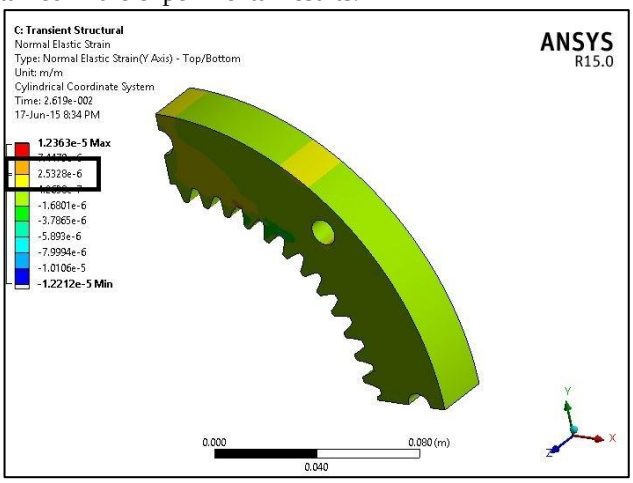

Fig. 6 Result obtained in DEWESoft (for 2500 rpm)

V.CONCLUSION

When comparing the experimental results with the results obtained using ANSYS software shows that they are reasonably similar. Thus the ANSYS model can be used to predict stresses after design modifications also.

Stresses obtained in the current design are very less. Further reduction of the annulargear rim thickness can be done without hampering the strength of annular gear.

ACKNOWLEDGMENT

I am thankful to Prof. S. L. Shinde sir for his valuable guidance during this research work. I am also thankful to

Prof. S. B. Patilsir of Department of Mechanical Engineering, SCOE, Pune for his cooperation throughout this research work.

REFERENCES

[1] Takeshi Ishida, Teruaki Hidaka, TakahiroMikami, Hideharu Takizawa, 1986, "Theoretical analysis of nominal stress in rim of planet gear." Technology reports of the Yamaguchi University, 3(5), 377-384.

[2] Kahraman, A., Kharazi, A. A., Umrani, M., 2003, "A deformable body dynamic analysis of planetary gears with thin rims", Journal of Sound and Vibration, 262, pp. 732-768.

[3] Bodas A., Kahraman A. 2001, "Influence of manufacturing errors and design parameters on the static planet load sharing behaviour of planetary gear sets", International Motion and Power Transmission Conference, Fukuoka, Japan.

[4] Chang S.H., Huston R.L., Coy J.J., 1982, "A finite element stress analysis of spur gears including fillet radii and rim thickness effects", Transactions of the American Society of Mechanical Engineers, Journal of Mechanical Design, 82-WA/DE-35, pp. 1–4.

[5] Yuksel C., Kahraman A., 2004, "Dynamic tooth loads of planetary gear sets having tooth profile wear", The University of Toledo, Toledo.

[6] Hammami A., Del Rincon A. F., Rueda F. V., Chaari F., Haddar M., "Dynamic behavior of two stages planetary gearbox in non stationary operations", Dynamics of Mechanical systems research unit-National School of EngineerinsSfax, Tunisia.

[7] Tsai S. J., Hwang G. L., Yeh S. Y., 2011, "An Analytical Approach for Load Sharing Analysis of Planetary Gear Drives", 13th World Congress in Mechanism and Machine Science, Guanajuato, Mexico. pp 1-8.