Wear Investigation of Polymer Gears on Various Operating Conditions

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Abstract*- Gear is a machine element used to transmit motion and power among rotating shafts by means of progressive engagement of projections called teeth. Gears are classified according to the relative position of the axes of the shaft, category of gearing, peripheral velocity of the gears and position of teeth on gear surface. Presently gears are suffered by backlash the amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the pitch circles, undercut a condition in generated gear teeth when any part of the fillet curve lies inside of a line drawn tangent to the working profile at its lowest point and interference is an important aspect of kinematics of gearing. When the gear tooth attempts to dig below the base circle of mating gear then the gear tooth action shall be non-conjugate and violate the fundamental law of gearing this non-conjugate action is called the interference.In this paper, wear analysis of a standard polymer gears made-up of Nylon 66 material is studied against mild steel gear for various loading conditions. Experimental setup is prepared for testing the wear rate of gears at various speeds & loads. The results obtained from experimental analysis are then analyzed in Minitab 19 software using Taguchi Orthogonal L9 array for determining the best material among the three polymers in terms of wear sustenance.*

Keywords- Wear, Nylon 66,Minitab 19, Taguchi Orthogonal Array, Wear analysis setup.

I. INTRODUCTION

Gears are the most common means of transmitting power in the modern mechanical engineering world. They vary from a tiny size used in watches to large size used in lifting mechanisms and speed reducers. Toothed gears are used for modifying the speed and power ratio as well as direction between input and output.

Fig 1 shows two mating gear teeth, in which

- Tooth profile 1 drives tooth profile 2 by acting at the immediate contact point K.
- N1N2 are the joint normal of the two profiles.
- N1 is the foot of the perpendicular from O1 to N1N2
- N2 is the foot of the perpendicular from O2 to N1N2.

Although the two profiles have different velocities V1 and V2 at point K, their velocities along N1N2 are equal in both magnitude and direction. Otherwise the two tooth profiles would

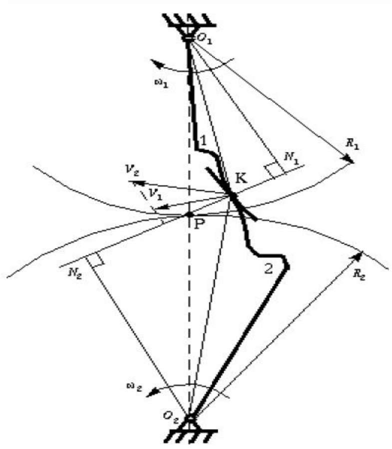

Fig.1 Law of Gearing

separate from each other. Therefore, the intersection of the tangency N1N2 and the line of center O1O2 is at point P. Thus, the relationship between the angular velocities of the driving gear to the driven gear, or velocity ratio, of a pair of mating teeth is Point P is very important to the velocity ratio, and it is called the pitch point. Pitch point divides the line between the line of centers and its position decides the velocity ratio of the two teeth. The above expression is the fundamental law of gear-tooth action.

Mating gear teeth acting against each other to produce rotary motion are similar to cams. When the tooth profiles, or cams, are designed so as to produce a constant angular-velocity ratio during meshing, these are said to have conjugate action. In theory, at least, it is possible arbitrarily to select any profile for one tooth and then to find a profile for the meshing tooth which will give conjugate action. One of these solutions is the involute profile, which, with few exceptions, is in universal use for gear teeth.

There are two forms of tooth profile commonly used: -

a) Cycloidal teeth
b) Involute teeth

An advantage of the Cycloidal teeth over the involute one is that wear of Cycloidal tooth is not as fast as with involute tooth. For this reason, gears transmitting very large amount of power are sometimes cut with Cycloidal teeth. On the other hand, involute teeth are very easy to manufacture and the actual distance between the centers may deviate slightly from the theoretical distance without affecting the velocity ratio or general performance. Because of this distinct advantage, gears with involute cut teeth are used much more than those with Cycloidal teeth.

II. LITERATURE SURVEY

It's the foremost preliminary step for proceeding with any research work writing. While doing this go through a complete thought process of your Journal subject and research for its viability by following means:

Wright et al. [1] in this paper, the meshing stiffness of spur gear pairs, considering both global tooth deflections and local contact deflections, is evaluated at any point of contact and approximated by an analytical, simple function. With this function, the load distribution ratio was calculated and compared with preceding results obtained from the hypothesis of minimum elastic potential energy, considering the tooth deflections, but neglecting the Hertzian deflections. Critical bending and contact stresses from both models are too compared for standard and high contact ratio spur gears.

Kim et al. [2]. In this paper contact stresses were found out by FEM method and experimental method by using the polariscope and the FEM results are compared with experimental results for validation.

Singh et al. [3] The objective of this thesis is to study the various stress state of spur gear. They mentioned that gears are machine elements used to transmit power between rotating shaft by means of engagement of projection called teeth. The author's calculated the tangential and radial forces acting on various points of the spur gear. By using Ansys software bending stress and contact stress on the tooth of spur gear drive were found

Clerico et al. [4] The objective of this paper is to summarize and update the spur- gear life analysis and to relate the analysis to some of the experimental results from NASA tests. The authors concluded that the gears used in aircraft and other applications may fail from scoring, tooth fracture due to bending fatigue, or surface pitting fatigue.

Hoskins et al. [5] in this paper the gear fitted in the gearbox armoured tracked vehicle is vulnerable to considerable fatigue damage over its life period due to the dynamic excitations caused by the terrain undulations, the rotating wheel and track assemblies. For this purpose, firstly static analysis of the model was carried out to validate the model and the boundary conditions correctness. Further Modal analysis is carried out to determine the dynamic characteristics of the gear model. The random load time history is transformed in to frequency domain using Fast Fourier transform to obtain load Power Spectral Density (PSD). Then the stress PSD response is obtained at critical node from the random vibration analysis. Once the spectrum of stress variation is found given input to the fatigue analysis and fatigue life is determined by FE package ANSYS 11.0.

III. PROBLEM STATEMENT

There is no specific investigation done on wear analysis of polymers gears like Nylon 66, against metallic gear. For this purpose, the behaviour of three different standard polymer gears are needed to be observed against a standard mild steel gear for various loading conditions.

IV. OBJECTIVES

1. To develop geometric design of experimental set-up in any solid modeling software like CATIA, CREO, Unigraphics etc.
2. To manufacture experimental setup.
3. To test gears at various speeds and loads.
4. To interpret the results obtained from experimental analysis in Minitab 19 using Taguchi Orthogonal L9 array for identifying the wear sustenance ability of the materials.

V. SELECTIONOF MATERIAL AND METHODOLOGY

The Driven Gear Selected for Testing is Nylon 66 and the Driver Gear selected for testing is Mild Steel. Some of the material specifications are as follows:

Table 1: Material Properties

Sr.No.	Properties	Nylon 66	MS
1	Hardness (BHN)	118-120	106.8
2	TensileStrength(N/mm²)	85	67.70
3	Flexural Yield Strength (MPa)	145-310	40
4	Elongation at brake (%)	5-640	13
5	Melting Point(Celsius)	260	1470
6	Thermal conductivity(W.m⁻¹.K⁻¹)	0.53	46
7	Tensile Modulus (MPa)	5500	240
8	Density (Kg/m³)	1400	7860
9	Yield Stress(N/mm²)	82.8	285
10	Poisons Ratio	0.39	0.3

The Gear Specifications of driver and driven Gears are as follows:

Sr. No	Specification	Driven gears	Driver Gears
1	Tooth profile	Involute	Involute
2	Material	Nylon66	MS
3	Normal Module	1.5	1.5
4	Normal Pressure angle	200	200
5	Number of teeth	66	66

After selection of material the next step is to design the shaft for both the Gears using the above specified values.

VI. DESIGN AND DEVELOPMENT OF EXPERIMENTAL SETUP

The analytical calculations performed for calculating the speed ratio, center to center distance between the two shafts having involute gears attached to it, & torque are as follows:

A. Design Calculations for Shaft:

Step 1: Calculation for finding out the speed ratio (i):

$i = Z_g/Z_p = 66/66 = 1$

(where, Z_g& Z_p are number of teeth on driver and driven gear respectively)

Step 2: Calculation for finding out the Center Distance (a)

$a=m(z_p+z_g)/2\cos\Psi$

$a=1.5(15+66)/2\cos20=64.65mm$

Step 3: Calculation for finding out the pitch circle diameter& torque:

$D=\frac{p}{\sin(180/z)} = \frac{9.525}{\sin(180/15)} = 45.81$ mm

Top diameter (D_a)

$(D_a)_{max}=D+1.25 =45.81+1.25*9.525=57.72$ mm

Tooth flank radius

$(r_e)_{max}= 0.008(Z^2+180)= 0.008*(152+180) = 3.24$ mm

$(r_e)_{min}= 0.12(Z+2) = 0.12* (15+2) =2.04$ mm

Tooth height above the pitch polygon

$(h_a)_{max}=0.625\ p-0.5+0.8p/Z$

$=0.625*9.525-0.5+0.8*9.525/15=5.95mm$

$(h_a)_{min}=0.5(p)=0.5(9.525)=4.76$ mm

$P= 2\pi NT/60$

$150=2\pi*300*T/60$

T=4.77 N-m is the required torque.

Step 4: Calculation for finding out the diameters of both the shafts.

Diameter of mild steel shaft (MS)

$T= 16T/\pi d^3$

$142.5= 16*4.77/ \pi d^3$

D=8.12mm

D=10 mm

Diameter of Shaft (Nylon 66)

$T= 16T/\pi d^3$

$41.25=16*4.77/ \pi d^3$

D=12.27mm

D=14 mm

From the data obtained by performing the analytical calculation, the geometry model of the experimental setup is prepared in a solid modeling software which is CATIA. Fig. 1 depicts the .jpg file of the actual model created in CATIA software.

B. Design of experimental setup in CATIA V5

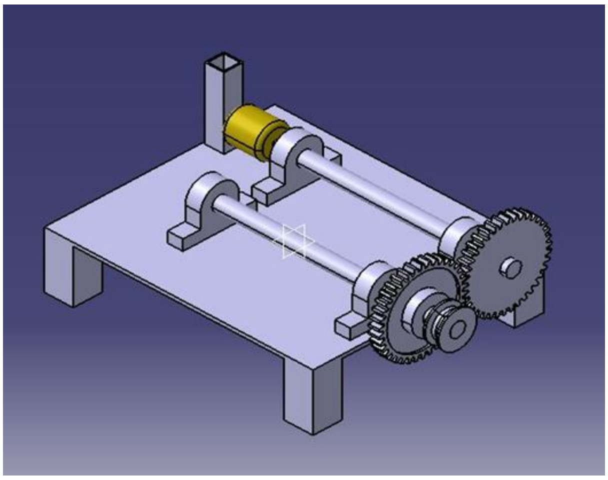

Fig 1: 3-Dimensional model created in CATIA V5 software

VII. EXPERIMENTAL TESTING

By creating the 3D geometry in CATIA V5 software, 2D drawing sheets for manufacturing where easily generated, that were used to convey the technical information to fabricators. Fig 2. shows the actual experimental setup manufactured in a fabrication shop.

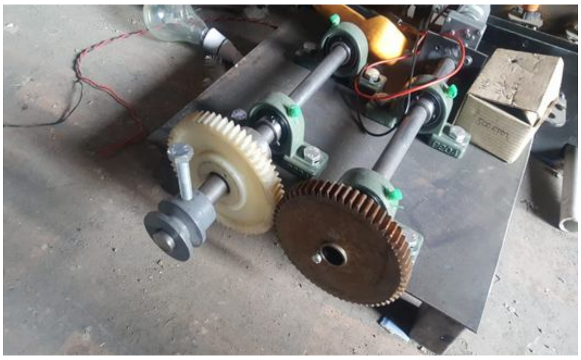

Fig1:Experimental Setup

The testing of the setup is carried out in time intervals by changing the load attached to the pulley and keeping the speed constant for calculating the wear rate. Fig 3. below shows the tachometer reading taken while performing tests on Experimental Setup

Fig2:Tachometer Reading in RPM

VIII. RESULTS

The testing conditions monitored while carrying out the experimental analysis is shown in Table 3. These readings a taken with the help of well calibrated measuring instruments.

Table 2: Testing Conditions

Sr. No.	Test conditions	Readings
1	Applied Load (N/mm)	9.81,19.62
2	Rotation Speed (rpm)	300
3	Environment	Air
4	Revolution	10
5	Temperature	Room Temperature
6	Humidity	35–55%

The test results obtained after engaging the nylon66 gear against mild steel gear at an applied load of 9.81 N/mm is shown in Table 4. below.

Table 4: Test results for Nylon66 against mild steel

Sr.No.	No. of Cycles	Wear(mm)
1	5000	0.10100
2	10000	0.18900

The test results obtained after engaging the nylon66 gear against mild steel gear at an applied load of 19.62 N/mm is shown in Table 5. below

Sr. No.	No. of Cycles	Wear(mm)
1	5000	0.1112
2	10000	0.2110

The results obtained from experimental analysis are then analyzed in Minitab 19 software using Taguchi Orthogonal L9 array. The equation of the Taguchi Orthogonal L9 array is given below which is programmed in Minitab 19 software to generated graphs for specific wear rate determination as shown in Fig 4 & 5 at applied load of 9.81 N/m & 19.62 N/m respectively.

$$W_s = \frac{W_v}{2zmbN_T}$$

where Wv is the wear volume (mm3), z is the number of pinion teeth, m is the module (mm), b is the tooth width (mm), and $N_T$is the total number of revolution (rev).

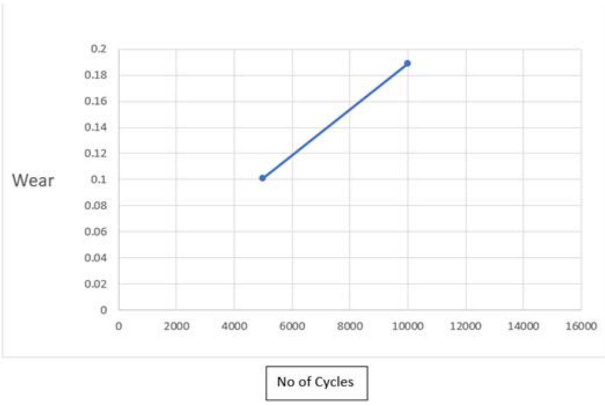

Fig 3: Wear rate of nylon66 against mild steel at 9.81N/mm

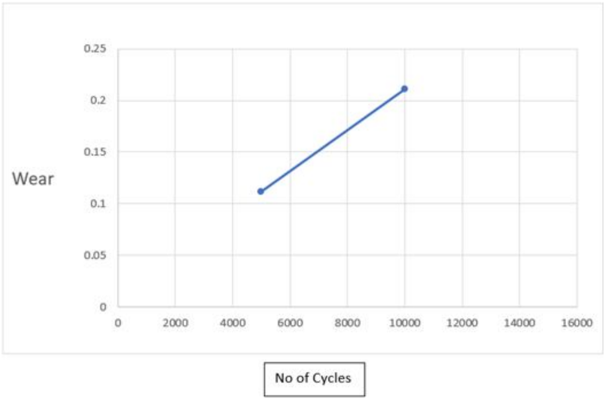

Fig 4: Wear rate of nylon 66 against mild steel at 19.62N/mm Load

IX. CONCLUSIONS

From the tests it has been found that, for the same testing time, wear is higher when the test is interrupted at regular intervals. When plastic-to-metal transfer occurs, the ultimate wear of the plastic hardly varies with the nature of the metallic material.If the load, and consequently the temperature, is very high, the nylon layer on the steel will melt almost completely and the contact takes place with liquid plastic lubricant.

X. ACKNOWLEDGMENT

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