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REVIEW OF PERFORMANCE ANALYSIS OF CRANKSHAFT THRUST WASHER BEARING MATERIAL

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ABSTRACT

This research experimentally quantifies and maps the behaviour of thrust washer under various conditions. The bearing is to be test at controlled loads and speeds for a governed period of time or until failure. In present work thrust washer bearing materials testing is to be carried out under dry as well as lubricated condition at two loads and four velocities. Aim of the present work is to find out the wear and coefficient of friction of bearing materials under dry as well as lubrication conditions and suggest the suitability of the thrust washer bearing material on crankshaft. For finding the wear and friction, we used pin on disc machine. The testing is to be carried out by maintaining the similar condition of testing as in case of actual engine. Therefore we have to use disc material as material of shaft i.e. steel no.1045. The disc is hardened and tempered . Pin on disk machine records frictional force, temperature and height loss from which the wear rate and coefficient of friction can be calculated. The results obtained from the experiment are analyzed and effect of load and speed on both the bearing materials along with wear trends obtained with respect to time is studied and suitable Thrust washer bearing material is chosen.

I. INTRODUCTION

Thrust washers are washers designed to prevent movement along the axis of a shaft. These rugged washer shaped flat bearings are used to prevent wheels from moving sideways on axles whenever the bearing that handles the radial load such as a bushing or roller bearing has no specific provision for axial or thrust loads. Typical sideways or axial loads are encountered whenever turning a corner and the vehicle is thrust sideways towards the outside of the curveThe thrust washer are two semicircular metal pieces that fit over the crankshaft on either side of the upper half of the rear main crankshaft bearing . they are held up into a groove in the bearing cap and cannot be seen with the rear bearing cap in place unless of course they are in bottom of oil pan. Friction bearings a thrust surface on the shaft rubs against a similar surface on crankcase . a precision insert main bearing may have a thrust flange for the crank to rub against it thrust washer use. Half thrust washer are often used which is inserted both side of radial bearing in recess in housing in bearing cap in present trend in smaller engine size is mainly toward the lower cost solution loose thrust washer which has always been standard in large engine foe reason of manufacturing.

1.1 Crankshaft thrust washer Material

Aluminium Base material

Aluminium silicon are the most important ones mainly due to their excellent combination of properties such as good castability, good surface finish, light weight, fewer tendencies to oxidation, lending to modification, low coefficient of thermal expansion, high strength-to-weight ratio and good corrosion resistance. It is more tolerance derbies and oil. Aluminium based material (SAE 783) is softer and therefore it gets deformed during punching of thrust washers.

1. Copper Base material

Copper alloy are give combination of properties such as strong , high thermal conductivity , machinability ,corrosion resistance and ductility.copper base alloy give the longer life. While copper based material is much harder but corrosive hence coated by tin (Sn) layer.

1.2 Failure of crankshaft thrust washer

1.2 Wearing thrust washer

crankshaft preventing it from moving forward and backward more than the specific amount. the every time press the clutch paddle push the whole crankshaft against rear thrust washer clutch depress or sitting at stop light with car in gear and pushed in cause great wear .eventually it wear thin. When it thin enough it can slip past rear bearing cap and drop out. Once is out crankshaft move further forward and the front thrust washer can follow the first onto the oil pan. without the rear thrust washer in place the flange on the crankshaft starts to wear against bearing cap because it is thicker than the part of the block above it.

1.3 Problem statement

Friction and wear between mechanical components has long been of great interest to engineers and scientists. It is commonly known that excessive wear of components can lead to altered performance and premature failure of machinery. Friction is likely to also affect the efficiency of systems by converting mechanical energy into non-recoverable thermal energy. Hence, it is of great importance that bearings, which are designed to decrease the friction and wear between contacting mechanical components, perform to a level acceptable for their individual application.

Crankshaft thrust washers form an important part of the main journal bearing that transmit and resolve axial forces in rotating mechanisms to keep components aligned along the shaft. They are designed to prevent movement along the axis of a shaft. These crankshaft thrust washers are very thin nearly in size of 2 to 4 mm in thickness. Hence there are high chances of them getting distorted during punching operation. Presently two types of thrust washers are used. One is Aluminium based and other is copper based material. Both are manufactured by different manufacturing processes. Aluminium based material is manufactured by casting and rolling method while copper based material is manufactured by sintering and rolling. Aluminium based material (SAE 783) is softer and therefore it gets deformed during punching of thrust washers and hence its rejection rate is high thus uneconomical. While copper based material is much harder but corrosive hence coated by tin (Sn) layer. Thus two different families of bearing materials are used here.

In case of engine it is difficult to test both the bearing materials due to variation in operating conditions of the engine. Hence these two materials are tested on pin on disc apparatus and a comparative analysis of these two families is performed.

1.4 Objectives

The purpose of this work is to develop an understanding of the physical phenomena that governs the behavior and life of a thrust washer bearing. The project work has following objectives:

1. Testing of the candidate bearing materials for friction and wear under various loads and various speeds in dry and lubricated conditions.
2. Experimentation on Pin on Disc apparatus to investigate effect of wear and coefficient of friction.
3. Formulation of approximate wear model using FEM software**.**
4. Comparative analysis of results for two thrust washer materials and studying the effect of parameters like load and sliding velocity on wear rate of these materials
5. Employing regression analysis for correlating the experimental data with the assumptions.

II. LITERATURE REVIEW

R L Jacksonand I Green [1]This research experimentally quantifies and maps the behaviour of various thrustwasher configurations under various conditions. The bearings are tested at controlled loads andspeeds for a governed period of time (up to 14 h) or until failure. The experimental results showthat at some loads and speeds the bearing operates with a near full hydrodynamic film andunder more harsh conditions it operates in the boundary lubrication regime. The experimentalresults indicate that coatings enhance performance by decreasing friction and thus decreasingthe heat generated. By decreasing the generated heat, the physical mechanisms of thermoelasticinstability and thermoviscous distress are less likely to occur. Using bronze will decrease thefriction between the bearings, although itmayalso decrease the life of the bearing.However, usingmore than one round washer appears to not significantly benefit washer-bearing performance. From the paper following points were studied:

i. Average effective coefficient of friction increases with load and speed
ii. At lowest load, bearing operates with low coefficient of friction.
iii. With increasing load & speed maximum bearing temperature increases.
iv. Coatings like niborn (Nickel & Boron) can enhance overall performance of bearing
v. Bearing life can be extended by use of washers having low coefficient of friction and adding a low friction coating

Robert L. Jackson and Itzhak [2] Green this study addresses the mechanisms that distress a flat-facedThrust washer bearing system .This washer bearing system separates Helical gear and its carrier within a gear set. It was foundthat the bearing can experience distress by the combination ofrotation speed, axial load, and the sequence and rate of theirapplication .distress define s a sudden rise in the real-timefrictional torque and temperature.The various tests suggest the presence of hydrodynamic effectsat certain rotational speeds and axial load combinations markedby decrease and the calculated effective coefficient of friction withdecrease velocity.

In the tested cases, a distinct increase in thecoefficient of friction occurs at the instant of distress. Three types of tests are conducted here

i. Rotational speed is kept constant and axial load is increased till distress is achieved.
ii. Speed is increased while load is kept constant
iii. Both load and speed are increased simultaneously

At distress point coefficient of friction increases suddenly indicating fluid film breakdown.There is presence of hydrodynamic effect produced by bearings at certain combination of load and speed

M. Sudheer, Ravikantha Prabhu, K. Raju, and Thirumaleshwara Bhat [3] in this paper

- The dry sliding friction and wear behaviour of epoxy hybrid composites.
- The influence of normal load, sliding velocity, and whisker content on both friction coefficient and specific wear rate was investigated on a pin-on-disc machine.
- The tests were conducted at ambient conditions based on the 3 × 3 (3 factors at 3 levels) full factorial design.

From this paper the concept of factorial design of experiments was studied where different parameters are taken into consideration and effect of individual as well as group of parameters is studied.

H .SO [4] The obvious contradictions include the following. The results between the arrangements of the rotating pin and the stationary pin under the same load and speed are different. The bulk temperatures of the rubbing specimens increase with the duration of testing, which may eventually arrive at a steady state. However, before the wear condition reaches a steadystate, it will have continuously varied. Moreover, the friction coefficient increases with sliding speed when the applied load on the rubbing specimens is over certain levels. All thesecontradictions can be reasonably explained with the accurate prediction of bulk and flash temperatures at the contact area. To this end, this paper provides a more reasonable method for the calculation of temperatures and the real and apparent contact areas.

has considered

i. The effect of sliding speed and normal load on wear.
ii. The paper also provides reasonable method to calculate real and apparent contact areas.
iii. Compared the appearance of the rubbed surface of the pin specimen.

E.M. Bortoleto [5] in his paper presents a computational study based on the linear Archard's wear law and finite element modelling (FEM), in order to analyze unlubricated sliding wear observed in pin on disc tests.

- The Global Incremental Wear Model (GIWM) considers Archard's formulation in a global scale, performing an incremental removal of material, based on contact pressure results, mechanical properties and geometrical characteristics of the sliding system, providing results as function of elastic deformation and wear.
- After defining the body and counter body geometry and material properties the following boundary conditions and loads were applied to the finite element model
- The model calculated tangential loads, friction loads resulting from the difficulty in the tangential movement of the pin with respect to the disc.

Xuejin Shen, Yunfei Liu, Lei Cao, Xiaoyang Chen [6]in this paper the thermo-mechanical finite element analysis method, a thermal wear simulation program was designed for the wear properties analysis of spherical plain bearing with self-lubricating fabric liners. In the program, the classical Archard wear model was applied to analyze the dynamical wear process of the bearing, and Abaqus scripting interface was used to simulate the progressive accumulation of wear between contact surfaces. The position of maximum wear depth occurs at the central contact region and this is in close agreement with test results. The relative error of the maximum wear depth between the FEA prediction and experimental results is a little more than 10%. It is shown that the complex nonlinear wear process can be simulated with a series of discrete quasi static models and the wear simulation program could be used to analyze the practical mechanical and tribological properties of the spherical plain bearings.

that the complex nonlinear wear process can be simulated with a series of discrete quasi static models. The 3D thermo-mechanical finite element model and 2D wear simulation program designed could provide practical mechanical and tribological analysis tools to predict wear problems for spherical plain bearings.

Priit Po˜dra , So¨ren Andersson [7] in this paper Wear of components is often a critical factor influencing the product service life. Wear prediction is therefore an important part of engineering. The wear simulation approach with commercial finite element (FE) software ANSYS is presented in this paper. A modelling and simulation procedure is proposed and used with the linear wear law and the Euler integration scheme. Good care, however, must be taken to assure model validity and numerical solution convergence. A spherical pin-on-disc unlubricated steel contact was analysed both experimentally and with FEM, and the Lim and Ashby wear map was used to identify the wear mechanism. It was shown that the FEA wear simulation results of a given geometry and loading can be treated on the basis of wear Coefficient sliding distance change equivalence. The finite element software ANSYS is well suited for the solving of contact problems as well as the wear simulation. The actual scatter of the wear coefficient being within the limits of ±40–60% led to considerable deviation of wear simulation results. These results must therefore be evaluated on a relative scale to compare different design options. The integration time step is a critical parameter regarding the reliability of simulation results. Too long steps cause erratic results and possibly the un convergence of FEA procedure. Too short intervals take too much computing time. A simple simulation time step optimisation routine was developed, evaluating the integration step duration for every solution step individually on the basis of the fixed maximum wear increment. The wear mechanism must be considered and its changes must be foreseen during the simulation process. The Lim and Ashby wear map can be used for steels. Assuming the linear wear law to be valid, the FEA wear simulation results for a given contact geometry anda given load can be treated on the basis of wear coefficient sliding distance change equivalence. Due to the model simplifications and the real deviation of input data, the FEA wear simulation results should be evaluated on a relative scale to compare different design options, rather than to be used to predict the absolute wear life.

III. THEROTICAL ANALYSIS

3.1 Crankshaft Thrust Washer specification

Swift Dzire specification

Gear Speed of vehicle: - TGR (economical Speed)

1st Gear = 15 km/hr 3.545

2^{nd} Gear = 25 km/hr	1.904
3^{rd} Gear = 40 km/hr	1.233
4^{th}Gear = 60 km/hr	0.911
5^{th} Gear = 80 km/hr	0.725

Power = 74 bhp @ 4000 rpm

Torque = 200 N.m @ 2000 rpm

Tyre Dia. = 61.96 cm = 0.6196m

Circumference = 194.65 cm = 1.9465 m

Calculate (rpm) :-

1^{st} Gear = 15 km/hr

Road wheel dia. = D

Circumference = PiD

Rpm = $\frac{V*1000}{60*Pid}$

1) 1^{st} Gear (15 km/hr)

Rpm = $\frac{V*1000}{60*Pid}$

Rpm = $\frac{15*1000}{60*1.946}$ = 128.435

Engine rpm = rpm x FDR x TGR

=128.435 X 1 X 3.545

(rpm_1) = 4555.30rpm

2) 2^{nd} Gear (25 km/hr)

Rpm = $\frac{V*1000}{60*Pid}$

Rpm = $\frac{25 X 1000}{60 X 1.976}$ = 215.10

Engine rpm = rpm X FDR X TGR

=215.10 X 1.67 X 1.904

(rpm_2) = 683.94 rpm

3) 3^{rd} Gear (40 km/hr)

Rpm = $\frac{40 X 1000}{60 X 1.946}$ =344.16

= 344.16 X 2.67 X 1.233

(rpm_3) = 1133.01 rpm

4) 4^{th} Gear (60 rpm)

Rpm = $\frac{V*1000}{60*Pid}$

Rpm = $\frac{60 X 1000}{60 X 1.946}$ = 516.244

Engine rpm = 516.244 X 4 X 0.911

(rpm_4) = 1881.19 rpm

5) 5^{th} Gear (80 rpm)

Rpm =$\frac{80 X 1000}{60 X 1.946}$ = 688.32

Engine rpm = 688.32 x 5.33 X 0.725

(rpm_5) = 2659.84 rpm

3.3 Load Calculation

1) Load 1

GAS FORCE (F_s) [25]

F_s= P * 3.14 * d_p^2 = 0.1 * 3.14 * 69.61^2

= 2988.95 N

Weight of crankshaft (F_1)

F_1 = 13.28 kg =13.28 * 9.81 =130.27 N

Weight of connecting rod (F_2)

F_2 = 240 gm = 0.240 * 9.81 * 4 = 9.41 N

Weight of piston (F_3)

F_3 = 440 gm = 0.440 * 9.81 *4 = 17.26 N

Total Weight

W = $F_s + F_1 + F_2 + F_3$

= 3145.89 N

Load only one side

W_1 = W / 2

= 1572.94 N

2.Load 2 (W_2)

W_2 = p * ld

= 7* 2.78 * 42

= 817.32 N

Sliding Velocity (v)

V = rω

Where

V = linear velocity

r = radius

ω = angular velocity rad/s

V = r $\frac{2\pi N}{60}$

V= r x N x 0.10472

1) 1^{st} Gear (v_1) :-

V_1 = 0.031 x 455.30 x 0.10472= 1.46 m/s

2) 2^{nd} Gear (v_2) :-

V_2 = 0.031 x 0.10472 x 683.94= 2.22 m/s

3) 3rd Gear (v_3) :-

V_3 = 0.031 x 0.10472 x 1133.01= 3.675 m/s

4) 4th Gear (v_4) :-

V_4 = 0.031 x 0.10472 x 1881.19= 6.105 m/s

5) 5th Gear (v_5) :-

V_5 = 0.031 x 0.10472 x 2659.84= 8.63 m/s

3.4 Dimensionless Analysis [7]

3.4.1 Coefficient of friction (f_{fc})

(f_{fc})= 0.78 – 0.13 log (ύ)

ύ = dimensionless normalised velocity

$$ύ = \frac{v\, r_o}{a_0}$$

v = sliding velocity (m/s)

r_o = apparent contact area radius = 0.0015(m)

a_o = thermal diffusivity (m^2/s)

Aluminium Thrust washer Bearing material

1. Coefficient of friction $(f_{fa})_1$

a_o = thermal diffusivity (m^2/s)= $74.2*10^{-6}$ m^2/s

$$ύ_1 = \frac{v\, r_o}{a_0}$$

$ύ_1 = \frac{1.47*1.5*10E-3}{74.6E-6}$

= 29.71

$(f_{fa})_1$= 0.78 – 0.13 log (29.71)

=0.33

2. Coefficient of friction $(f_{fa})_2$

$ύ_2 = \frac{2.22*1.5*10E-3}{74.6E-6}$

= 45

$(f_{fa})_2$= 0.78 – 0.13 log (45)

=0.28

3. Coefficient of friction $(f_{fa})_3$

$ύ_3 = \frac{3.675*1.5*10E-3}{74.6E-6}$

= 74.5

$(f_{fc})_3$= 0.78 – 0.13 log (74.5)

=0.21

4. Coefficient of friction $(f_{fa})_4$

$ύ_4 = \frac{6.105*1.5*10E-3}{74.6E-6}$

= 123.75

$(f_{fc})_4$= 0.78 – 0.13 log (123.75)

=0.15

5. Coefficient of friction $(f_{fc})_5$

$ύ_5 = \frac{8.63*1.5*10E-3}{74.6E-6}$

= 174.93

$(f_{fc})_5$= 0.78 – 0.13 log (174.93)

=0.108

3.4.2 Dimensionless Wear Rate

$$\frac{V}{s} = K\frac{P}{H}$$

V= volume wear (m^3)

K= wear coefficient = $1*10^{-5}$

H= hardness of softer material =$539.4*10^6$ N/m

$$\frac{V}{s} = 1*10E-5\,\frac{1572.94}{539.4E6}$$

$$\frac{V}{s} = 2.91 * 10^{-11}\ m^3$$

A = $1.38*10^{-3}$ m^2

Dimensionless wear rate (Q)

1. At S= 1.47 m/s

$Q_1 = \frac{V}{sA} = \frac{2.91E-11*1.47}{1.38E-3}$= 3.09 * 10^{-8}

2 At S= 2.22 m/s

$Q_2 = \frac{V}{sA} = \frac{2.91E-11*2.22}{1.38E-3}$=4.67* 10^{-8}

3. At S= 3.675 m/s

$Q_3 = \frac{V}{sA} = \frac{2.91E-11*3.675}{1.38E-3}$= 7.74* 10^{-8}

4. At S= 6.105 m/s

$Q_4 = \frac{V}{sA} = \frac{2.91E-11*6.105}{1.38E-3}$= 1.28* 10^{-7}

5. At S= 8.63 m/s

$Q_5 = \frac{V}{sA} = \frac{2.91E-11*8.63}{1.38E-3}$= 1.81* 10^{-7}

Copper Thrust washer Bearing material

a_o = thermal diffusivity (m^2/s)= $1.11*10^{-4}$ m^2/s

3.4.3 Coefficient of friction $(f_{fc})_1$

1. Coefficient of friction $(f_{fc})_1$

$ύ_1 = \frac{1.47*1.5*10E-3}{1.11E-4}$

= 19.86

$(f_{fc})_1$= 0.78 – 0.13 log (19.86)

=0.39

2. Coefficient of friction $(f_{fc})_2$

$$\acute{\upsilon}_2 = \frac{2.22*1.5*10E-3}{1.11E-4}$$

= 30

$(f_{fc})_2$= 0.78 – 0.13 log (30)

=0.33

3. Coefficient of friction $(f_{fc})_3$

$$\acute{\upsilon}_3 = \frac{3.675*1.5*10E-3}{1.11E-4}$$

= 49.66

$(f_{fc})_3$= 0.78 – 0.13 log (49.66)

=0.27

4. Coefficient of friction $(f_{fc})_4$

$$\acute{\upsilon}_4 = \frac{6.105*1.5*10E-3}{1.11E-4}$$

= 82.5

$(f_{fc})_4$= 0.78 – 0.13 log (82.5)

=0.20

5. Coefficient of friction $(f_{fc})_5$

$$\acute{\upsilon}_5 = \frac{8.63*1.5*10E-3}{1.11E-4}$$

= 116.62

$(f_{fc})_5$= 0.78 – 0.13 log (116.62)

=0.16

3.4.4 Dimensionless Wear Rate

$$\frac{V}{s} = K\frac{W}{H}$$

V= volume wear (m^3)

K= wear coefficient = $1*10^{-5}$

H= hardness of softer material =$620*10^6$ N/m^2

$$\frac{V}{s} = 1*10E-5\frac{1572.94}{620E6}$$

$= 2.537 * 10^{-11}$

A = $1.38*10^{-3}$ m^2

Dimensionless wear rate (Q)

1. At S= 1.47 m/s

$$Q_1 = \frac{V}{sA} = \frac{2.537E-11*1.47}{1.38E-3}$$

$= 2.69*10^{-8}$

2. At S= 2.22 m/s

$$Q_2 = \frac{V}{sA} = \frac{2.537E-11*2.22}{1.38E-3}$$

$= 4.06*10^{-8}$

3. At S= 3.675 m/s

$$Q_3 = \frac{V}{sA} = \frac{2.537E-11*3.675}{1.38E-3}$$

$= 6.72*10^{-8}$

4. At S= 6.105 m/s

$$Q_4 = \frac{V}{sA} = \frac{2.537E-11*6.105}{1.38E-3}$$

$= 1.11*10^{-7}$

5. At S= 8.63 m/s

$$Q_5 = \frac{V}{sA} = \frac{2.537E-11*8.63}{1.38E-3}$$

$= 1.57*10^{-7}$

IV. CONCLUSION

The table shows friction coefficient (COF)and Dimensionless wear rate (Q)obtained from various sliding velocities and 1572.94 N and 817.32 N load.

Sliding velocity	Aluminium Base Thrust Washer			Copper base Thrust washer		
V (m/s)	COF	$Q_{1572.94 N}$	$Q_{817.32}$	COF	$Q_{1572.94 N}$	$Q_{817.32}$
1.47	0.33	$3.09*10^{-8}$	$1.60*10^{-8}$	0.39	$2.69*10^{-8}$	$1.40*10^{-8}$
2.22	0.28	$4.37*10^{-8}$	$2.42*10^{-8}$	0.33	$4.08*10^{-8}$	$2.12*10^{-8}$
3.675	0.21	$7.74*10^{-8}$	$4.02*10^{-8}$	0.27	$6.72*10^{-8}$	$3.50*10^{-8}$
6.105	0.15	$1.28*10^{-7}$	$6.67*10^{-8}$	0.20	$1.11*10^{-7}$	$5.83*10^{-8}$
8.63	0.108	$1.81*10^{-7}$	$9.42*10^{-8}$	0.16	$1.57*10^{-7}$	$8.24*10^{-8}$

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