An experimental investigation of various materials on thrust washer bearing operation

R L Jackson[1*] and **I Green**[2]

[1]Department of Mechanical Engineering, Auburn University, Auburn, AL, USA

[2]George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, GA, USA

The manuscript was received on 25 April 2007 and was accepted after revision for publication on 12 June 2007.

DOI: 10.1243/13506501JET294

Abstract: This research experimentally quantifies and maps the behaviour of various thrust washer configurations under various conditions. The bearings are tested at controlled loads and speeds for a governed period of time (up to 14 h) or until failure. The experimental results show that at some loads and speeds the bearing operates with a near full hydrodynamic film and under more harsh conditions it operates in the boundary lubrication regime. The experimental results indicate that coatings enhance performance by decreasing friction and thus decreasing the heat generated. By decreasing the generated heat, the physical mechanisms of thermoelastic instability and thermoviscous distress are less likely to occur. Using bronze will decrease the friction between the bearings, although it may also decrease the life of the bearing. However, using more than one round washer appears to not significantly benefit washer-bearing performance.

Keywords: hydrodynamic bearings, coatings, failure analysis, automotive transmission fluid, bronze, steel, polytetrafluoroethylene, surface coatings

1 INTRODUCTION

The goal of this research is to investigate the physical phenomena that distress a thrust washer bearing system (Fig. 1). This research also intends to provide performance maps for different types of thrust washers. This is a continuation of a body of work that includes both experimental [**1–3**] and numerical [**4**, **5**] investigations. The current work will expand on these results by including additional tests which have considered the effect of various materials and washer configurations. Work concerning flat thrust washer bearings as described is scarce, although a review of previous work is given in references [**1**] to [**4**]. It should be noted that the current body of works differs from previous works [**6–9**] in that the washers are able to tilt and not loaded axisymmetrically.

In the application which the current work focuses on, the thrust washer bearing system bears the load produced by the planets of a helical planetary gearset within an automatic transmission. The thrust washer bearing system consists of one or two flat-faced washers that are placed between a helical gear and its carrier. Because of the non-axisymmetric loading caused by the helical gears, the gears and washers are tilted in relation to the carrier, forming a converging and diverging gap (Fig. 1). Due to such tilt, there are areas of concentrated contact between the components. From this point forward, 'bearing' will refer to the thrust washer bearing system within the test rig, unless specified otherwise.

The behaviour of the thrust washer bearing system and its lubrication in the automatic transmission between gears and their carriers is largely unknown, where the bearing is physically found to distress at an accelerated rate and from uncertain reasons. When distress occurs, the planetary gearset locks up and maintenance is required. Experience shows that the failed bearing in the transmission is sometimes completely worn away, thus leaving debris in the transmission fluid. Obviously, this result is undesirable and unacceptable.

To gain an understanding of the bearing behaviour under non-axisymmetric loading, a test rig was

Corresponding author: Department of Mechanical Engineering, Auburn University, 270 Ross Hall, Auburn, AL 36849-5341, USA. email: robert.jackson@eng.auburn.ed

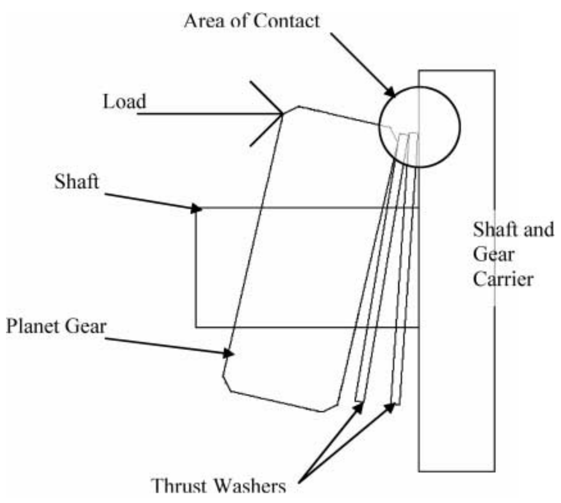

Fig. 1 Diagram of expected contact loading conditions of thrust washer bearing

designed and built to provide a physical model. The test rig is described in detail in references [**1**] to [**3**]. The test rig controls the operational parameters governing the tribological behaviour of the washer. For given washer materials and surface finishes, the parameters that effect the life of the bearing and its tribological behaviour the most are believed to be thrust or axial load, rotational speed, lubrication supply, lubrication properties, and the geometry of the bearing. The test rig also records pertinent real-time data from the bearing. The frictional torque is recorded from the power output of the motor, since with increasing frictional torque the power needed to run the motor at a constant speed increases as well. The temperatures are recorded using thermocouples embedded near the bearing surface.

The experimental results from references [**1**] to [**3**] show that the thrust washer bearing is prone to severe and sudden distress at high loads and speeds. These are of the same trends predicted in references [**10**] and [**11**], and are shown in the progression of washer distress in Fig. 2. By visual inspection of the washer wear progression shown in Fig. 2, it is seen that at high speeds the wear changes from an abrasive type of wear to scuffing. The scuffing depicted in Fig. 2 is so severe that the washer at the highest load and speed

Load (N):	New	260.9	1239	1239
Speed (rpm):	New	1,300	1,300	13,000
Duration(cycles):	0	10^6	6,000	6,000

Fig. 2 Visual comparison of thrust washer wears under different loading conditions

has adhered or welded to the contacting part behind it. These points of severe distress are characterized by large and sudden increases in the bearing temperature and friction. The bearing, if not unloaded and shutdown immediately can also melt and weld together at this state.

A previous work [**5**] linked thrust washer bearing's point of distress to thermoelastic instability (TEI) and is marked by a sudden increase in the coefficient of friction (COF) and the bearing temperature [**1–3**]. The work on TEI that is probably most pertinent to the work is by Davis *et al.* [**12**]. They model TEI in thin disks and more specifically in mechanical clutches (referred to as DKS model). Created to model TEI in clutches, the DKS model does not consider lubrication or the dampening effects of wear, although it does model a geometry that is very similar to thrust washer bearings. DKS suggest that the region of the TEI is coupled between a critical bulk temperature (average bearing temperature) and operating speed. As the operating speed increases, the critical temperature required to cause the system to become unstable decreases. DKS also perform a parametric investigation into the effect of elastic modulus, thermal conductivity, and disk thickness, on the location of the thermoelastic stability threshold. Summarizing their results, increases in elastic modulus, thermal conductivity, and disk thickness pushes the stability threshold to higher temperatures and speeds. In reference [**5**], the DKS model is correlated to the currently considered thrust washer bearings and the results suggest that the thrust washer bearings operate well into the region of TEI.

In 1973, Dow and Burton [**13**] expanded their analytically modelled of TEI in the sliding contact between a thin blade or scraper and a half space [**14**] to consider the effect of wear. They analytically proved that the wear rate would dampen the TEI by changing the geometry. That should also be the case with the current thrust washer bearings.

While the bearing is in distress, material is often transferred between bearing surfaces and/or worn away. Under severe conditions the contacting surfaces can even weld together and cause the test rig to seize. Since this also occurs mostly at high speeds, it fits the definition of scuffing failure and wear as described by Williams [**15**]. Carper *et al.* [**16**] found experimentally that the mechanism of scuffing appeared to be characterizable by the surface temperature, film thickness, and the friction coefficient. Enthoven and Spikes [**17**] performed a study of scuffing which was able to measure the temperature *in situ* and also film the scuffing contact. According to their work, scuffing cannot be characterized by temperature and friction alone, and wear debris and film thickness should also be considered. Horng [**18**] also derived a methodology to predict the onset of scuffing in a rough surface contact which included the effects of temperature and film thickness. More recently Alzoubi *et al.* [**19**] studied the effect of a near frictionless carbon coating on preventing scuffing. It appears that low friction coatings are very effective at reducing the occurrence of scuffing. Low friction PTFE coatings are also tested in the current work with positive results. There are also many other works that study scuffing experimentally and analytically, and they all cannot be listed here.

It should be emphasized that this work is not intended to characterize the mechanism of TEI and scuffing, but rather map out the overall performance of different thrust washer bearings at different loads and speeds. At low speeds wear does occur, but parts rarely weld together and wear appears to occur due to abrasion rather than scuffing or adhesion. Due to the nature of washer distress, the material composition of the washers will have a large effect on washer performance due to changes in surface and material properties (elasticity, wear rate, conductivity).

2 CONSTANT LOAD AND CONSTANT SPEED TESTS

To decrease the amount of scatter that occurred in the variable load and speed tests presented in other works [**2**, **3**], the test methodology was changed to a constant load and speed methodology. Thus, the washer behaviour would no longer be affected by its load and speed history.

A time of 2 h for tests at 9100 rpm was chosen. At 9100 rpm, a two-hour test produces 1 092 000 cycles, in which one cycle is one revolution. This test duration was chosen because it was able to reproduce failures seen in the industrial application in a reasonable amount of time. For other tests to be comparable, they must be run for the same number of cycles. To produce the same number of cycles for each speed the tests must be run for the amounts of time indicated in Table 1.

2.1 Procedure

Based on tests previously conducted, it was found that at a certain loads and speeds the effective friction coefficient of the bearing rises suddenly from a value less than 0.04 to a value at or above 0.06. The coefficients may sometimes elevate for only a brief moment as if the bearing is able to dynamically correct itself and return to a more desirable state. The temperature at these elevated coefficients rises quickly and may cause damage to the test rig. It was found that the local temperature might rise above the boiling point of the lubricant in about 30 s. This boiling point causes lubricant in a gaseous or vapour form to exit the bearing and also the test rig. This occurrence is extremely undesirable since it drastically decreases the effectiveness of the lubricant. Thus, an average friction coefficient of 0.06 over a time period of 30 s is considered the distress point or failure criteria for these tests. The tests, however, was not always interrupted if this criterion is met.

Under certain conditions, the measured friction coefficient was high, but the bearing temperature did not rise significantly. In these cases the test was not interrupted until the temperature rose above a critical temperature, which could cause damage to the rig and its components. This critical temperature was found by trial and error during the course of the testing and is set at 91°C. It should be noted, that this is the temperature behind the bearing in the carrier, and that the temperatures in the actual bearing may be significantly higher. It is uncertain which of the two, temperature or frictional torque is a better indicator of wear or distress of the bearing. These tests will help determine that indicator.

The minimum time for a test is set at 1 min at speeds of 9100 rpm. Just as the maximum duration of tests were set to time periods that produce the same number of cycles, the minimum test period will also be set to produce the same number of cycles, according to the speed of the test. The minimum test periods at different speeds is given in Table 1.

A few different loading conditions will be used to obtain a statistical representation of the bearing behaviour and life. The initial goal is to obtain a good understanding of the failure mechanism of the bearing, than other bearing designs will be tested using this standard. These tests are within a range of speeds and loads, which are comparable to what the bearing carries in the transmission. The 16 different load and speed combinations vary widely over the capable loads and speeds of the test rig (Table 2). Initially, two tests will be performed at each load to bring the number of data points to 32. Then, if there is a large variation at certain loads and speeds, additional tests will be performed. Also, load combinations not included in the mesh may be tested to refine the mesh in those desired regions.

A new set of washers are used for each test and the carrier and gears are resurfaced or exchanged for new parts at regular intervals to prevent wear from affecting the tests. This interval has been determined by measuring the surface properties (roughness) after each test using a profilometer.

Table 1 Maximum and minimum test durations at each gear speed

Speed (rpm)	1300	5200	9100	13 000
Maximum test duration time (hr)	14	3.5	2	1.4
Minimum test duration (min)	7	1.75	1.0	0.7

Table 2 Axial loads and rotational speeds of life tests performed. Intersecting locations give PV (average pressure × average linear velocity in MPa·m/s) values for each combination

	Load			
Speed (rpm)	260.9 N (58.67 lbf)	750.4 N (168.7 lbf)	1240 N (278.7 lbf)	1729 N (388.7 lbf)
1300	0.972	2.794	4.617	6.439
5200	3.889	11.18	18.47	25.76
9100	6.806	19.56	32.32	45.08
13 000	9.722	27.94	46.17	64.40

2.2 Washer bearing configurations

A number of washer bearing configurations are tested using the above testing methods. These configurations are compared with each other. The following is a list of the configurations that will be tested. During a test, the washer configuration is placed between the gear and the carrier attachment (Figs 1 and 3). A list of the tested washer configurations is given in Table 3.

The washers are nominally 0.023 m at the outer diameter and 0.011 m at the inner diameter and 0.635 mm thick. The average roughness, Ra, of the bronze washers is 0.43 μm and Rq is 0.52 μm. For the steel washers Ra was 0.43 and Rq was 0.51. Roughness information for the other washer types is not available. Information about the hardness and material properties of the washers is also not available.

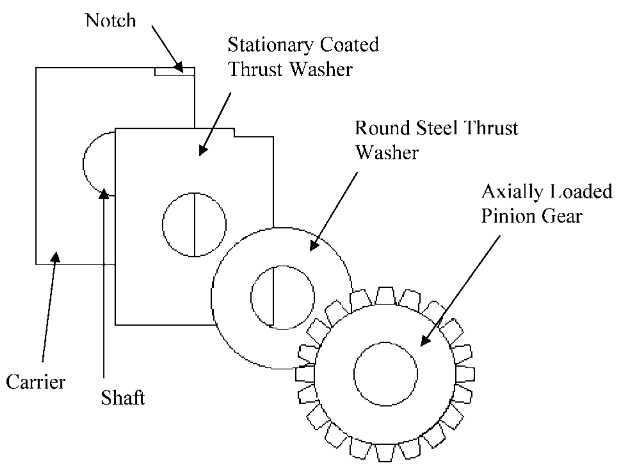

Fig. 3 Typical coated washer bearing configuration

Table 3 The various washer configurations tested in this investigation

1	One round steel washer, one round bronze washer (bronze washer is adjacent to carrier, as depicted in Fig. 1)
2	Two round steel washers
3	One round steel washer
4	One round bronze washer
5	One stationary steel washer (discontinued after poor performance)
6	One stationary bronze washer (discontinued after poor performance)
7	One stationary bronze washer and one round steel washer
8	One stationary PTFE coated washer, one round steel washer (stationary washer is adjacent to carrier) – see Fig. 3
9	One stationary nibron (nickel boron) coated, one round steel washer – see Fig. 3

Due to the possibility of long-term wear and run-in within the test rig, the tests performed are cycled through the different washer combinations. Four tests are performed using one combination and then the combination is switched. The load and speed combinations are also cycled randomly, such that every load and speed combination will have the same number of tests performed (the matrix will be complete) before a new matrix is begun. This cycle is continued until all the tests are performed. Certain washer combinations were discontinued if the testing suggests that the results will not provide useful information. Also, during initial testing, the regions of high loads and speeds are found to fail severely and to be very destructive to the test rig. Due to these concerns, once a load and speed combination results in a failure for a certain washer combination, the higher load and speed tests for that matrix is not performed.

The one stationary steel washer (case no. 5) and one stationary bronze washer (case no. 6) tests were discontinued due to severe failure at the lowest loads and speeds. This indicates that in comparison to the other tested configurations, these are not suitable for use. Since these combinations correspond to no round or free rotating washers, their sub par performance indicates that round washers do enhance washer-bearing performance.

3 EXPERIMENTAL RESULTS AND DISCUSSION

For each load and speed test, the average values for the COF and average temperature are calculated. The test duration and maximum temperature are also calculated for each load and speed test. The resulting values are then plotted versus load and speed in Figs 4 to 7.

The standard deviation between tests at the same conditions for temperature is 6.88 °C and for the COF 0.0472, although if the tests that distressed are not considered, these values reduce to 5.86 °C and 0.0176. The standard deviation reduces significantly probably because when the bearing does distress, scuffing

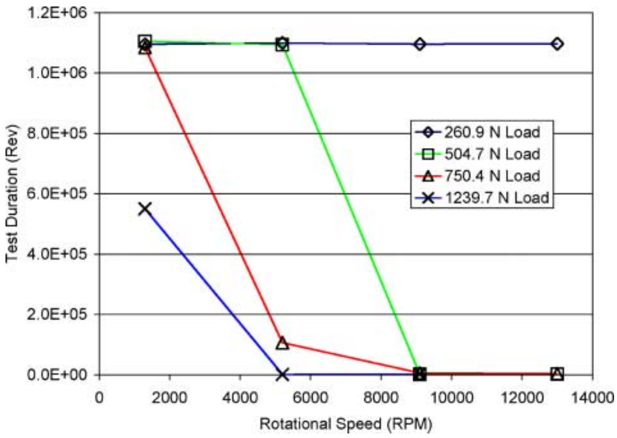

Fig. 4 Plot of test duration for single round steel washers as a function of load and speed

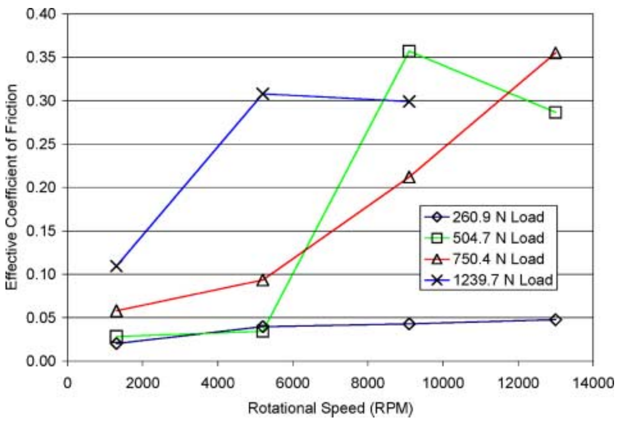

Fig. 5 Plot of average effective COF for single round steel washers as a function of load and speed

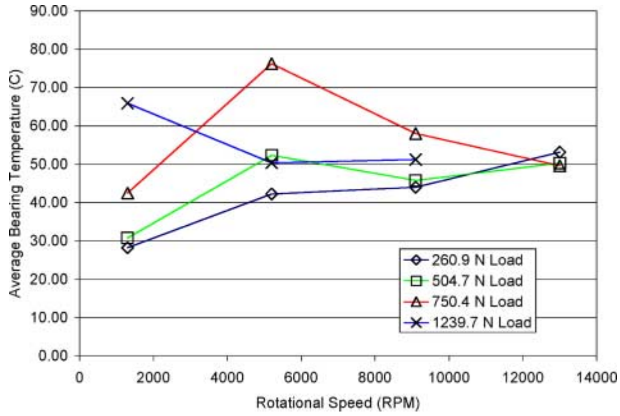

Fig. 6 Plot of average bearing temperature for single round steel washers as a function of load and speed

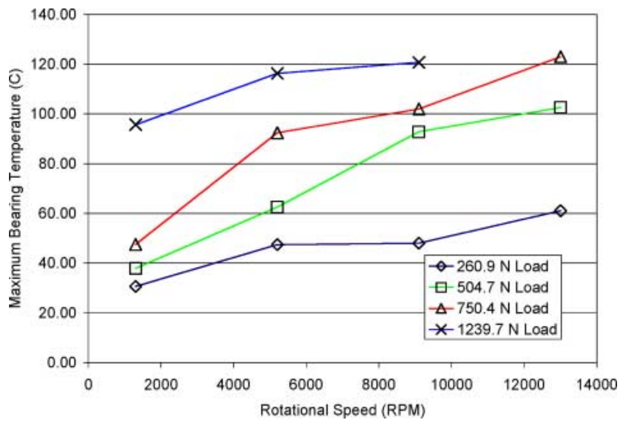

Fig. 7 Plot of maximum bearing temperature for single round steel washers as a function of load and speed

and excessive wear occurs, and the COF becomes less predictable. If the bearing operates at a point on the threshold of distress, due to slight differences in the tests the bearings may or may not distress. This will also increase the scatter for these loads and speeds, which are on the threshold of distress.

3.1 Single round steel washer results

As expected, for increasing loads and speeds the test duration decreases (Fig. 4). However, at the lowest load of 261 N, the washer operates for the full test duration for all speeds. This is probably due to sufficient hydrodynamic lubrication separating the parts and decreasing the friction and heat generation. For all except two tests the bearing either lasts the entire duration or distresses immediately. This indicates that the washer does not distress due to fatigue or other cyclic phenomena during these tests. Rather, the distress is attributed to a severe and immediate distress condition that is believed to be caused by thermo-elastic instability, thermo-viscous distress and scuffing [**5**]. These general trends are present in most of the tested bearings, except in the coated bearings, which reduce the friction and provide resistance to these thermally induced physical phenomena.

The average effective COF increases with load and speed as shown in Fig. 5. At the lowest load of 261 N, the COF appears to be low enough to suggest that the bearing is operating with a full hydrodynamic film between two of the bearing surfaces. As speed increases at 261 N and the film thickness increases (and the shear force of the lubricant), the COF also increases. For higher loads however, the film thickness cannot be sustained and the surface asperities contact. This contact causes the traction between the surfaces and the effective COF to increase. At the highest loads and speeds the high COFs are due to scuffing occurring between the surfaces.

There is no clear trend for the average bearing temperature shown in Fig. 6. When the bearing distresses, the temperature increases quickly until the test is stopped because the test interruption criteria is met. When this occurs, the maximum temperature (Fig. 7) is much higher than the average. For this reason the maximum temperature is much more reflective upon the state of the bearing at a certain load and speed. From this point forward the average temperature will not be plotted for each washer configuration.

The maximum bearing temperature increases with both load and speed (Fig. 7). The maximum temperature is for some loads and speeds greater than the test interruption temperature (see section 2.1) because the temperature often continues to climb after the test is cutoff. This is due to the delayed propagation of heat from the surface of the bearing to the thermocouples.

A Stribeck curve is also generated from the experimental results and shown in Fig. 8. The experimentally generated Stribeck curve follows the trends that are expected for a Stribeck curve operating in both the full film and boundary lubrication regimes. Figure 8 shows that at a Stribeck value of about 5×10^{-6} that the bearing begins operating in the mixed lubrication regime. At that point the effective COF increases sharply due to fluid film collapse and the resulting asperity contact. Also, looking back at the Stribeck curve generated from the variable loads and speed tests [**2**], it appears that those tests were run more in the region of the elbow at the bottom left of the curve. This is where the bearing transitions from full film lubrication to mixed and boundary lubrication.

3.2 Comparative results

From this point forward, the results of the single steel washer will be used as a reference with which to compare the other washer combinations. The single round steel washer combination is chosen because it is the simplest configuration that actually performs reasonably well. When coated washers were tested

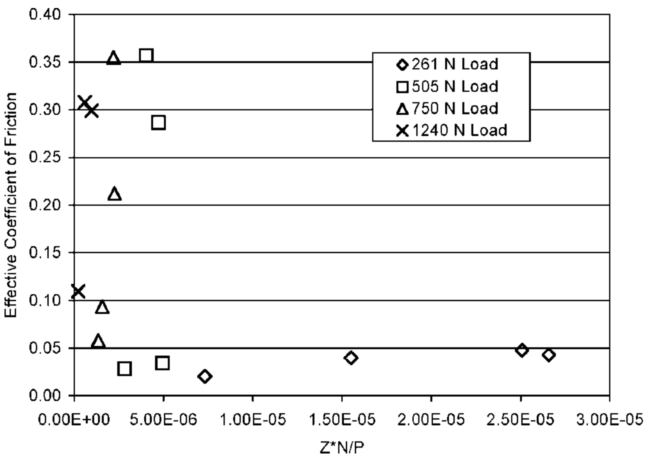

Fig. 8 Experimentally generated Stribeck plot for a single round steel washer

it was also with a single round steel washer and a stationary-coated washer.

The test duration for various loads and washer configurations at a rotational speed of 5200 rpm is plotted in Fig. 9. The two round steel performs better than the single steel, but the round steel and bronze perform worse than the single bronze. It is thus unclear if the number of washers used in the bearing increases or decreases performance. It appears that the use of more than one round washer provides no significant increase in performance in comparison to the single round washer configurations.

It appears that the two round steel and single round bronze test washer bearing configurations perform better than the single round steel and round bronze and steel configurations (Fig. 10). Again, even though certain washers seem to survive longer, the results show no clear advantage to using multiple round washers.

In Fig. 11, the effective COF follows similar trends for all the washer configurations, except at the highest load of 1240 N. At the highest load there are no

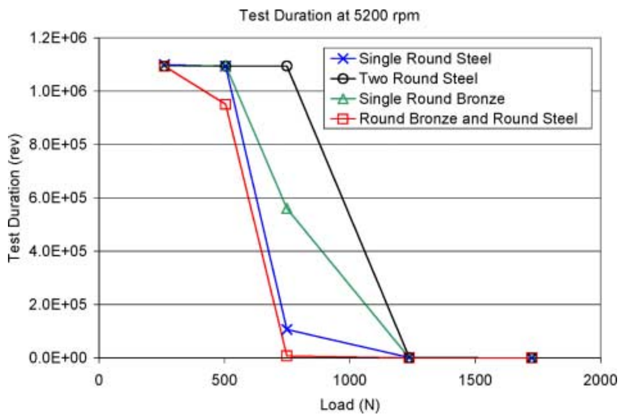

Fig. 9 Test duration plotted as a function of load and washer type

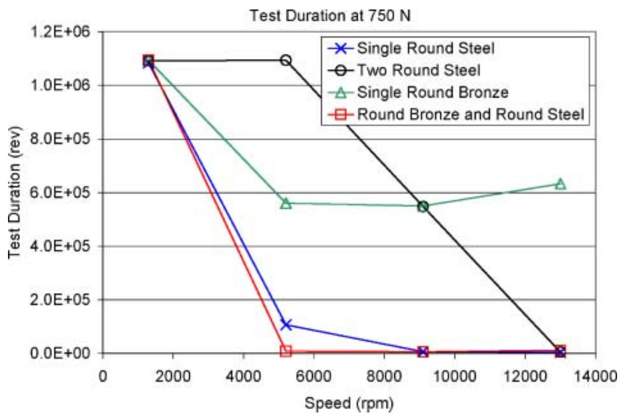

Fig. 10 Test duration plotted as a function of speed and washer type

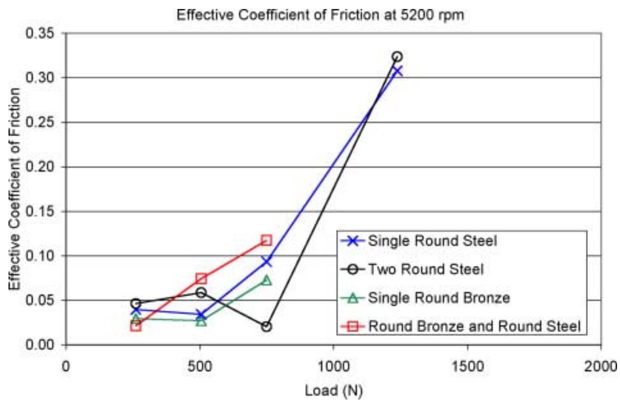

Fig. 11 Effective COF plotted as a function of load and washer type

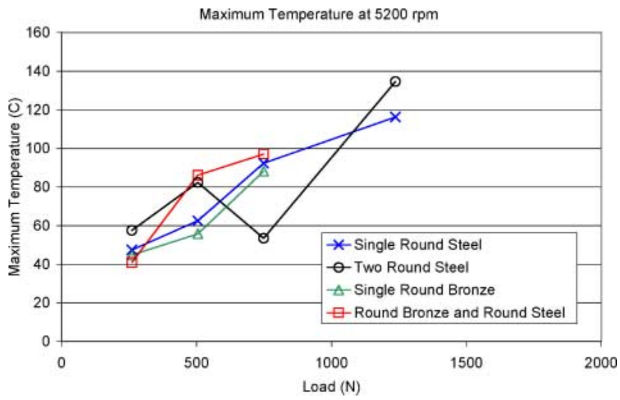

Fig. 12 Maximum bearing temperature plotted as a function of load and washer type

representative bronze tests because they distressed at a lower load. However, the steel tests show a significant increase in friction at the highest load. At the highest load the steel washers appear to begin to scuff and adhere due to high temperatures (Fig. 12). As will be more apparent in Fig. 13, the bronze washers provide some resistance to scuffing. This is because scuffing occurs more readily between two materials of similar structure which more easily bond.

It is apparent in Fig. 13, that at high loads the bronze washers significantly reduce the effective COF. This is thought to be due to an increased resistance to scuffing. The material selection does seem to have an effect on bearing performance, although using additional round washers appears to provide no benefit. At some conditions the two washer configurations appear to perform best, while for other conditions the single washer configuration seems to do better.

In Fig. 12, the maximum temperature of all the bearing configurations appears similar, except for the two round steel washers at 750 N. Also, there is no comparison at the highest load since tests were not run for the bronze bearings at those loads. Again in Fig. 14, when the speed is varied, there is no clear trend between number of washers, bearing material and maximum temperature. Although the bronze did improve bearing performance by reducing friction in Fig. 13, the bronze washers do not decrease significantly the overall temperature of the bearing. The bronze bearing just tends to scuff less once bearing distress has occurred. This does not mean that the bronze washer is more resistant to wear in general, because the bronze washer will still wear due to abrasive wear. The bronze will be less likely to cause the bearing to weld together and lock up.

Based on the results presented in Figs 8 to 14, it does appear that bronze washers provide a resistance to scuffing which could significantly reduce the possibility of the thrust washer bearings locking up when under distress. However, bronze washers also appear to distress at lower loads in some cases. There does not seem to be a significant benefit to adding additional round washers into the configuration.

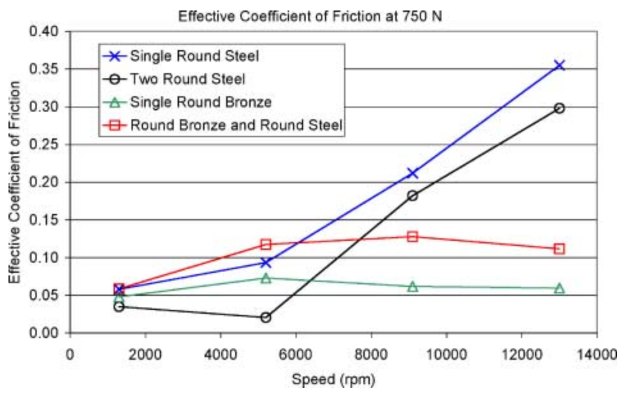

Fig. 13 Effective coefficient of friction plotted as a function of load and washer type

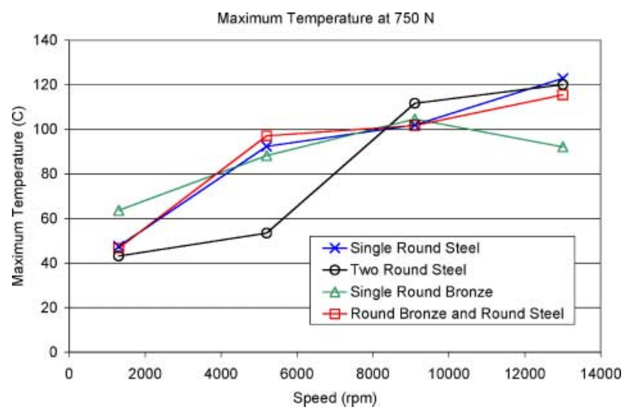

Fig. 14 Maximum bearing temperature plotted as a function of speed and washer type

3.3 The effect of coatings

The two coated (PTFE and nibron) stationary washers running with a single round steel washer are compared with the reference single round steel washers. The nibron coating consists of a nickel and boron plating which increases the hardness of the surface. A bronze stationary washer in place of the coated washers is also compared. The main effect the coatings have is to reduce friction. By simply reducing the friction other coupled effects will take place. Less frictional heat is generated and the coated bearing temperature will be less than the uncoated bearing temperature. By decreasing the bearing temperature, scuffing, and TEI are less likely to occur. The decrease in friction can also have the effect of decreasing wear and thus extending the life of the bearing.

As is shown in the following set of figures (Figs 15 to 20) coatings, especially PTFE, enhance bearing performance significantly. Nevertheless, coatings eventually wear away and leave the substrates below unprotected.

The nibron and PTFE coated washers appear to survive higher loads than the stationary bronze and

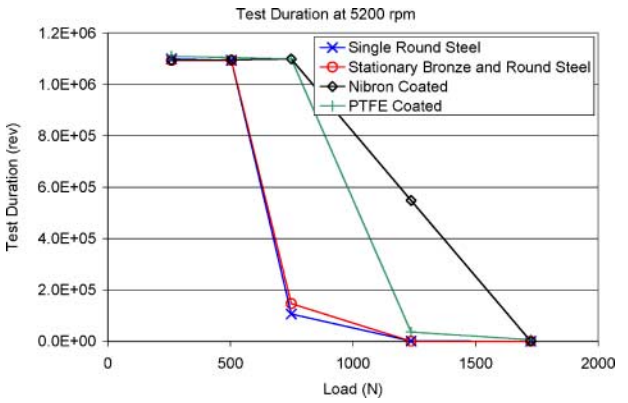

Fig. 15 Test duration plotted as a function of load and washer type

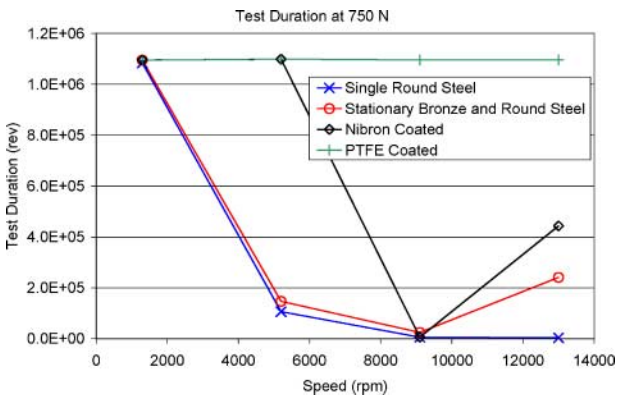

Fig. 16 Test duration plotted as a function of speed and washer type

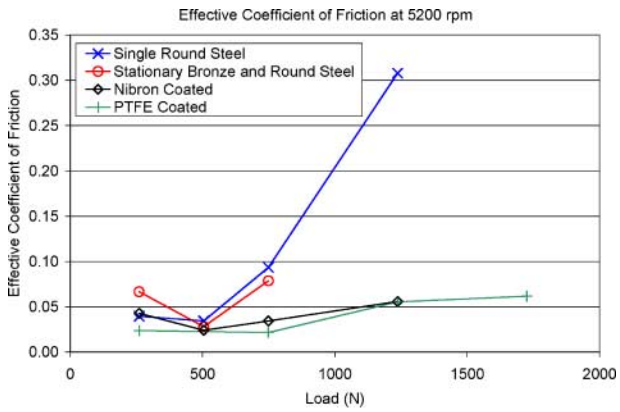

Fig. 17 Effective coefficient of friction plotted as a function of load and washer type

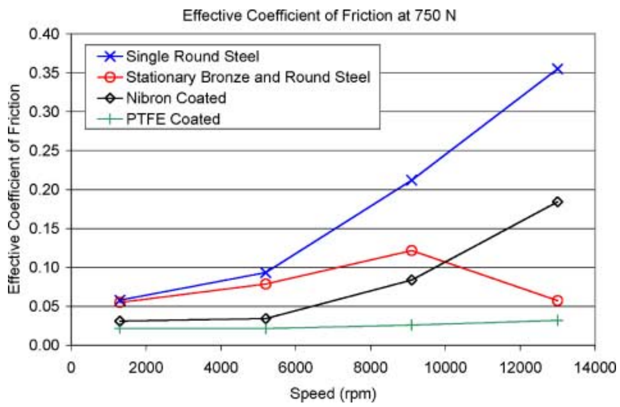

Fig. 18 Effective coefficient of friction plotted as a function of speed and washer type

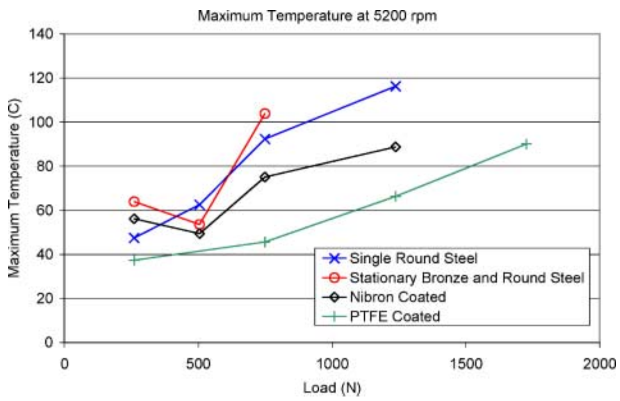

Fig. 19 Maximum bearing temperature plotted as a function of load and washer type

the reference single round steel washer configurations (Fig. 15). The stationary bronze washer does not seem to improve bearing performance significantly from the single round washer. Again in Fig. 16, the PTFE and Nibron coatings prove to enhance bearing performance. Actually, the PTFE coated bearings survived every load condition presented in the plot. It will be

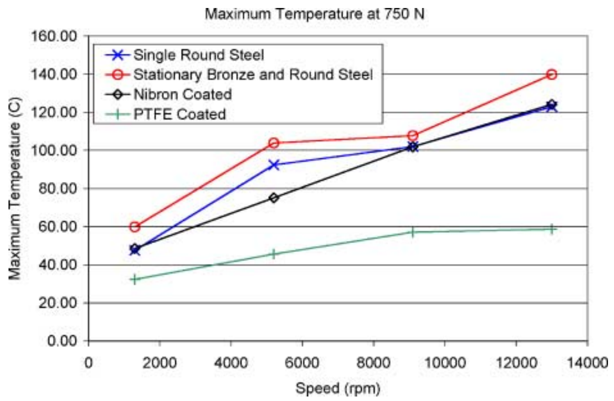

Fig. 20 Maximum bearing temperature plotted as a function of speed and washer type

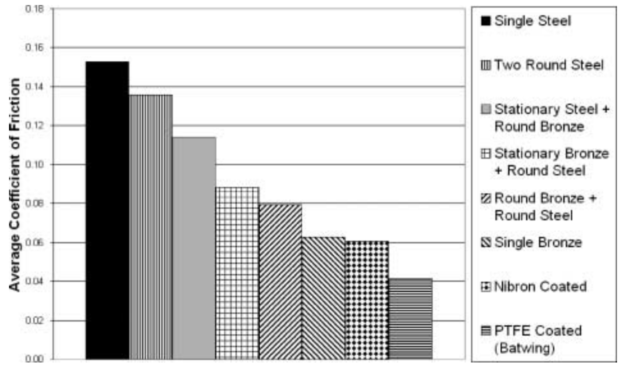

Fig. 21 The average coefficient of friction for all tests conducted for each washer combination

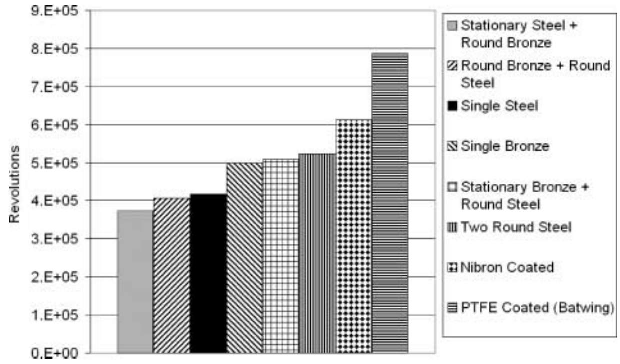

Fig. 22 The average bearing test duration for each type of tested bearing configuration

shown in subsequent plots (Figs 17 to 20) that the PTFE coated bearing performs superior to other bearing configurations for most criteria.

In Fig. 17 the effective COF for the reference single round steel washer configuration and various coated washers is plotted as a function load. The effective COF (about 0.30) reached by the single round steel washer is due to scuffing of the bearing faces. It appears from this plot that the nibron and PTFE coatings significantly reduce the friction coefficient and the inhibit scuffing at high loads. Since the bronze stationary washer failed before the coated washer and the single round steel washer, it is not apparent if it has improved washer performance significantly. Although for most loads, the effective COF of the bronze stationary bearing is larger than both coated bearings.

In Fig. 18, all the washer configurations show to have lower effective COF than the reference single round steel washer. Again the PTFE outperforms all other bearings at all speeds. At 13 000 rpm the stationary bronze performs better than the nibron coated bearing. The single round steel washer again appears to scuff at the highest speeds, whereas the bronze bearing and two coated bearings do not.

For the loads presented in Fig. 19, the PTFE coated bearing never reaches temperatures as high as the other bearings. Overall, the stationary bronze and single round steel produce the highest maximum temperatures. While the nibron coated bearing shows a slight improvement over the two uncoated bearings. For a varying speed test (see Fig. 20), the nibron coated bearing again only shows a slight improvement over the uncoated bearings. Again, the PTFE shows a much lower maximum temperature than the other bearings.

3.4 Combined results

For each washer configuration the average friction coefficient and average test duration are calculated (Figs 21 and 22). For loads and speeds that certain washer combinations were not tested at, test duration of 0 s was assigned. The washer combinations comprising of all steel components produced the largest average COF. This is because like materials tend to bond at the asperity contacts and requires larger shear forces to break apart. Adding bronze components decreased the average friction coefficient, and the single round bronze washer had the lowest. However, the coatings and especially the PTFE coating produced the lowest average COF. Lower coefficients of friction tend to improve bearing performance by generating less heat, although there are other factors involved as shown in Fig. 22 since the order of the washers changes.

Figure 22 plots the average test duration for each bearing configuration. In theory, bearings with longer average test durations should also have longer lives in the actual transmission. Clearly the PTFE coated washer performed much better than all washer combinations by having the longest average test duration. As with in Fig. 21, the Nibron coated bearing performs second best behind the PTFE coated. From there on it appears that multiple washers have slightly longer test durations than the other bearings. However, the differences between the two round steel, one round steel–one round bronze, and one round bronze appear to be insignificant.

4 CONCLUSIONS

The experimental results show that at some loads and speeds the bearing operates with a near full hydrodynamic film. The bearing behaviour can, thus, be characterized by the Stribeck curve. When the bearing is operating on the right side of the Stribeck curve the hydrodynamic lift is significant enough to separate the components with a fluid film. This decreases friction and extends the life of the bearing. If the bearing is operating on the left of the Stribeck curve, it has less hydrodynamic lift and is more likely to operate in the boundary lubrication regime. The bearing temperature increases with load and speed and will eventually cause the viscosity to decrease enough that the fluid film collapses (corresponding to the left side of the Stribeck curve). When the film collapses the temperature and friction increase very quickly and cause the bearing to distress (thermoviscous distress). If the test is not stopped immediately, the bearing components may weld together and cause bearing lock-up. This sequence of events can be explained by thermoviscous distress and TEI causing bearing distress and resulting in scuffing wear between the components. Besides the wear, these physical phenomena are captured in the numerical model [**5**] and also result in the bearing distressing quickly and severely at certain loads and speeds.

The experimental results indicate that coatings can enhance overall bearing performance. It is believed that coatings enhance performance by decreasing friction and thus decreasing the heat generated. By decreasing the generated heat, the physical mechanisms TEI and thermoviscous distress are less likely to occur. Using bronze will decrease the friction between the bearings, although it may also decrease the life of the bearing. However, using multiple round washer configurations appears to not significantly benefit washer-bearing performance. The best way to extend bearing life is by using a washer, which has a very low friction coefficient with other components or by adding a low friction coating.

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