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WEAR PERFORMANCE OF SELF-LUBRICATNG SLEEVE BEARINGS

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Summary. *The present work deals with the wear performance of six different self-lubricating bearing materials. These materials were selected as a typical representatives of the self-lubricating bearing materials currently used in different segments of the industry. The materials were tested in the form of 31.75 mm bushes on a specially designed test rig. Most of the materials were evaluated at a PV-value of 0.35 MPa . m/s for continuous operation for 200 hours. The frictional coefficient and temperature of each bearing were monitored and recorded through a digital data logger. The wear profiles were obtained after 200 hours of continuous operation with the help of a Talyrond. The wear profiles, radial wear rates and comparisons of bearing materials are reported in detail in this work. The comparison of wear measuring techniques revealed that the measurements obtained by a capacitance probe were non-reliable but the radial wear profiles obtained by a Talyrond were very reliable.*

Key Words: *Tribology, Wear, Self-lubricating material, Dry bearings, Friction.*

1. INTRODUCTION

The maximum *PV*-ratings for dry bearing materials are generally determined either on small sleeve bearing of a particular diameter or on other geometries as pin on disc, thrust washers or a flat specimen rubbing against a rotating shaft (Benzig et al., 1976). The results obtained on the above mentioned geometries are extrapolated to bearings of different lengths or diameters or both with the assumption that these bearings would give the same radial wear in the same running period (*H*) provided *PV*-rating is kept constant. This extrapolation has its dangers but if done intelligently quite useful predictions can be made for semi-quantitative purposes. Accelerated tests in which the simulated bearing conditions are multiplied by a factor of three or four or more are always suspect when applied to a dry sleeve bearing and quite misleading results can be made. The wear performance limitations of *PV*-criterion were reported in detail by Malik and Freeman (1972). Nevertheless, *PV* as the main criterion is still widely used in the industry and is reckoned quite convenient for practical use.

A review of the literature about various test rigs which were used to measure friction and wear (Benzig et al., 1976) refers to moving pin on a flat surface; rotating pin on disk; pin on rotating disk; cylinder on cylinder (face loaded); cylinder or pin on rotating cylinder; rectangular flat on rotating cylinder; disk on disk (edge loaded); or bushes of very small diameters (12.5 mm). Therefore, a special test rig was designed and developed for investigation of tribological properties of self-lubricating sleeve bearings by Malik et al. (1998a). This test machine can investigate dry bearings of different diameters ranging from 31.75 to 63.5 mm with different clearances and *L/D* ratios.

The wear rates of dry bearings reported extensively in the literature, more often than not, have been obtained by measuring radial wear along the load line using lineal gauging technique. This method is used due to its simplicity and ease of application. However, results obtained by such methods are likely to be an underestimate of the actual wear rates due to failure to establish (a) changing datum as a result of thermal expansion of the bearing and journal, (b) exact location and direction of maximum radial wear and (c) the presence of wear particles in the bearing loaded region.

2. WEAR ASPECT OF PV-CRITERION

According to adhesive wear, the total volume worn (V_w) in a sliding system will be proportional to the load supported (W) multiplied by the distance travelled (ℓ), i.e.,

$V_w \propto W.\,\ell \propto W.V.\,H$ where V is sliding speed and H is running time in hours.

The volume worn is considered of little direct value by the designers. A change in bearing wall thickness (R_w) is generally considered more practicable for the design of dry bearings. Therefore,

$R_w\,.\,A_2 \propto W\,.\,V.\,H$ where A_2 is bearing projected area and alternatively,

$R_w \propto (W/A_2)\,.\,V.\,H \propto P.\,V.\,H,$ or ,

$$R_w = k_r\,.\,P.\,V.\,H \qquad [1]$$

Where k_r is generally termed as bearing radial wear factor or radial wear coefficient or specific wear rate; $P = W/A_2 = W/LD$ is bearing nominal pressure; L is bearing length; D is bearing diameter; V is sliding speed; and H is the running time in hours.

The *PV*-criterion is a widely quoted measure of the performance of dry bearing materials (Pike and Conway-Jones, 1992). It is usually given as the maximum operating *PV*-value (MPa x m/s) allowable if the radial wear (R_w) is not to exceed a given allowable depth for a given period, say 0.125 mm in 1,000 hours. The specific wear rate or radial wear factor also increases with the surface temperature and the surface roughness and can also depend upon the material of the mating surface.

3. BEARING MATERIALS INVESIGATED

Six materials were selected for this investigation to give a reasonable variation of physical, mechanical, frictional and wear properties. These materials are a typical representative of all the groups of commercially available dry bearing materials. Table 1 lists the thermal and mechanical properties of these bearing materials.

Table 1. Properties of bearing materials investigated

No.	Identification	*E*, GPa	α mm/mm K	ρ kg/m^3	*k* W / mK	*c* J/kg K
A	Graphite filled metal (sintered material)	(1) 68.21	20.02×10^{-6}	6714.7	125.43	376.57
B	PTFE/lead-impregnated porous bronze	(1) 68.21	14.94×10^{-6}	6201.9	41.87	468.62
C	PTFE-filled carbon graphite	1.16	53.1×10^{-6}	2004.8	3.30	753.14
D	Ceramic typed filled PTFE	0.689	59.94×10^{-6}	2265.4	0.33	836.82
E	PTFE-cotton weave reinforced with a thermoset reinforced cotton backings	9.16	----------	(2) 2187.5	0.26	941.42
F	MoS_2 filled nylon	3.96	100.08×10^{-6}	1142.6	(3) 0.25	1255.2 (0°C) (3) 1464.4 (0 – 50 °C) 1882.8 (49 - 100 °C) 2301.3 (100-210 °C)

(1) The value quoted is for bronze. (2) The value quoted is for PTFE. (3) The values quoted are for Nylon 66.

4. EXPERIMENTAL SET-UP

The design requirements, details and general features of the bearing test rig shown in Fig. 1 are available in the reference (Malik et al., 1998a). This test rig can accommodate self-lubricating dry bearings of different diameters (*D*) ranging from 31.75 mm to 63.5 mm with different *L/D* and clearance ratios. The bearings can be tested dry or in a sparsely lubricated state.

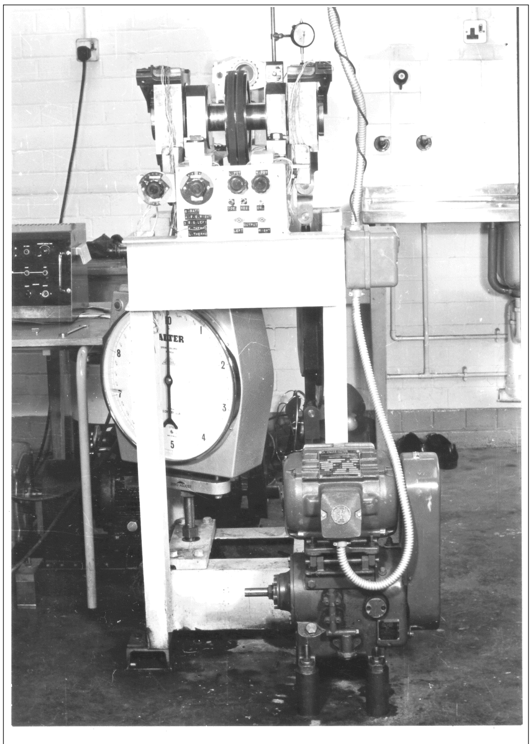


Fig. 1 Bearing Test Rig (Front view)

The test journal sleeves can be made from any suitable material for investigation. However, for this particular work, journal sleeves were manufactured from EN-56-AM stainless steel. In order to facilitate rapid dismantling and replacement of journal sleeves, these were made a light push fit on the mating shaft. These were held in position with a key passing through a 6 mm deep slot cut at the end of the journal sleeve and screwed to the end of main shaft. These journal sleeves can be given different surface finishes and the bearing clearance can be varied by grinding the journal in successive steps to decreasing diameters. The journal sleeves were degreased with trichloroethylene and air dried before any test. The test bearings were also cleaned and degreased with trichloroethylene prior to any test. The test rig was instrumented to measure bearing starting and kinetic friction; bearing, journal and housing temperature; bearing load; journal speed and bearing radial wear as reported in detail by Malik et al. (1998a).

Pre-test measurements

The diametral measurements were made with the help of a Universal Measuring machine Societe Genovoise – D which can read to an accuracy of 0.00025 mm. The measuring probe of this machine exerted a very light constant pressure during measurements. The journal sleeves were prepared by surface grinding within the range of 0.25-0.4 μ m CLA. The details for measurements of bearing diameter, journal sleeve diameter, bearing clearance and journal sleeve surface finish are available in Malik et al. (1998b).

5. EXPERIMENTAL PROCEDURE

The desired bearing load was applied separately to both the test bearings with the help of loading spring balances shown in Fig. 1. The variable speed drive was set to the desired speed setting. The driving motor was then started and the speed of shaft was adjusted to test speed. Bearing temperatures, journal sleeve temperatures, housing temperatures and bearing frictional torque were recorded by the data logger at various intervals up to 200 hours. The surrounding air temperature and humidity were recorded daily but the variations were small as most of the tests were conducted inside the laboratory environment. The average of the frictional torque obtained over the last 30 minutes of the test run was used for calculating the dynamic coefficient of friction.

Adequate instrumentation and the precise determination of various parameters were considered essential requirements in order to achieve realistic conclusions. Whereas the precise measurement of load, speed and bearing dimensions did not offer a great deal of difficulty, the precise measurements of friction and temperature represented the major effort of the problem. The instrumentation used here was capable of determining the frictional torque with an error of $\pm$ 0.05 N.m. The load and speed could be measured to an accuracy of $\pm$ 5 N and $\pm$ 0.1 revolution respectively.

Direct measurements of bearing interface temperature represent considerable difficulty especially for bearing materials of **low** thermal conductivity. In fact, the measurements of increase in ***real*** temperature in moving contact surfaces are available only on rare instances. The thermoelectric method of measuring contact temperature in which both the rubbing surfaces are used as the elements of the thermocouple had been applied widely. However, this method is not applicable to a dry bearing system which cannot constitute a thermocouple with the mating journal sleeve. Thus the following two approaches were adopted for the measurement of bearing temperature; (a) by embedding thermocouples in the test bearing loaded region and (b) by measurement of journal sleeve temperature during rotation. This arrangement was capable of measurement of temperature to an accuracy of $\pm$ 0.5^0 C. The details of test bearing, journal sleeve and embedded thermocouples were reported in detail by Malik et al. (1998a).

6. EXPERIMENTAL RESULTS

6.1 Method 1 - Preliminary procedure for the measurement of bearing radial wear

Preliminary measurements of bearing radial wear were made with the help of a capacitance probe mounted on the test housing adjacent to a specially made 105 mm long journal sleeve. The probe was clamped below the journal sleeve along the bearing load line. A general view of this arrangement is shown in Fig. 2. Any variation in distance between the test journal sleeve and capacitance probe was indicated by a Wayne-kerr electronic micrometer. The resolution of the micrometer depends upon the type of capacitance probe used. In this investigation, the capacitance probe used could indicate up to 0.0025 mm variation in distance.

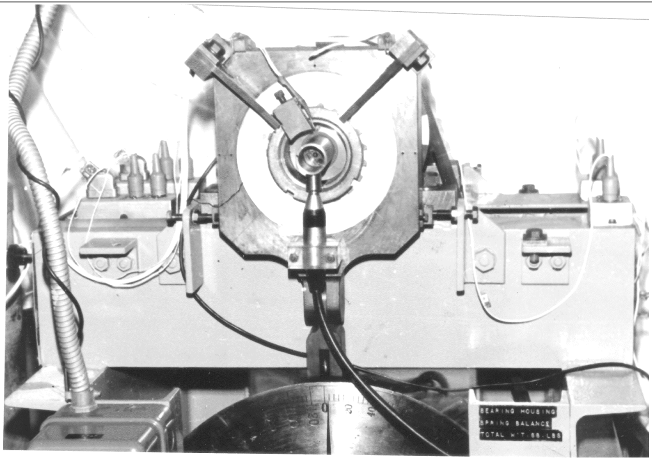


Fig. 2 Set-up used to measure bearing radial wear

Limitation of lineal gauging technique

The typical response of the wear measuring probe for three different *PV*-values is shown in Fig. 3. This indicates that during initial 30 to 40 minutes the approach between journal sleeve and capacitance probe decreased appreciably but increased afterward. The response of a capacitance probe or lineal gauging for measurement of radial wear can be divided into following two regions.

Primary approach $= X - Y \cong X$ (because $X >> Y$) and secondary approach $= Y - X \cong Y$ (because $Y >> X$), where

X = rate of expansion of journal sleeve + rate of thermal expansion of bearing wall thickness or bearing pad in the bearing loaded area.

Y = rate of bearing radial wear.

Therefore, it could be concluded that this technique will measure the primary approach X during initial few hours of a test. In between there will be a region where $X = Y$, but this will exist for a short period. The capacitance probe is likely to indicate the instantaneous wear rate (Y) of bearing material only when the system has gained thermal equilibrium. However, this technique will not indicate accurately the total wear during a test run.

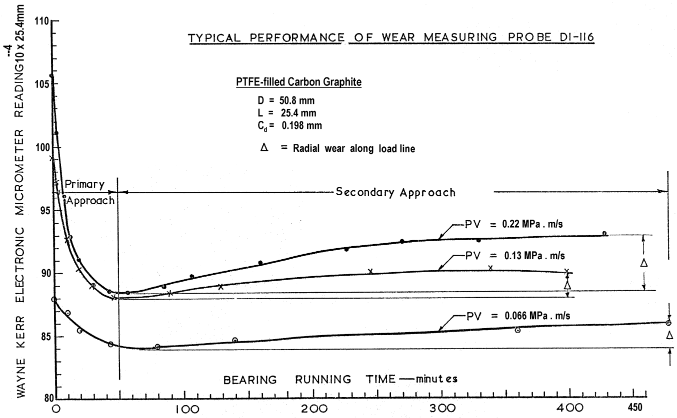


Fig. 3 Typical performance of wear measuring probe

Wear rates obtained by lineal gauging technique for Material "C"

The average values of wear rates obtained by this method for three different *PV*-values for 50.8 mm diameter bush of material "C" are shown in Fig. 3. The results are tabulated in Table 2. The results obtained by lineal gauging technique indicate that radial wear factor ($\boldsymbol{k_r}$) for this material varied between 4.24 x 10^{-3} and 9.99 x 10^{-3} mm/*PVH*. These results are based on the total radial wear obtained after a certain numbers of hours (≤ 10 hours) without taking into consideration the initial rapid wear during running-in period.

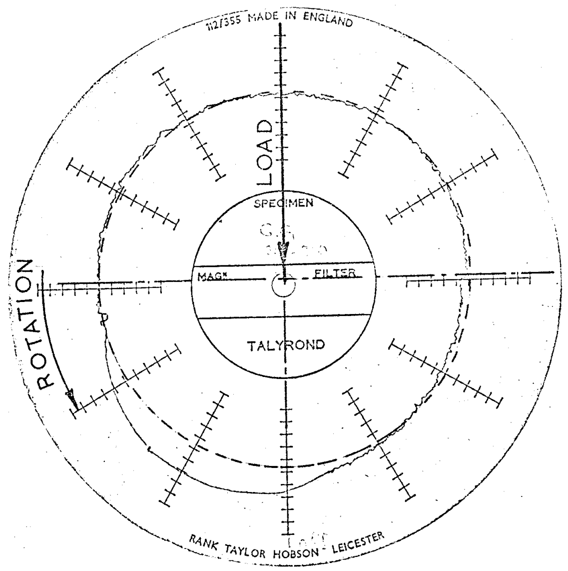
Subsequent tests made on a similar bearing of material "C" indicated that after initial bedding-in period, radial wear factor ($\boldsymbol{k}_r$) varied between 8.7 x 10^{-4} and 29 x 10^{-4} mm/*PVH*. The possible reason for this discrepancy appears to be the difficulty in obtaining a reliable estimate of bearing radial wear by a capacitance probe.

Table 2 Determination of radial wear rate for material "C"

No.	***P***, kPa	***V***, m/min	Test Duration, Hours	***PV***, MPa . m/s	r_w, mm	r_w / hr mm/hr	$k_r = r_w / PVH$
1	413.4	9.58	6.67	0.066	0.0044	6.60 x 10^{-4}	9.99 x 10^{-3}
2	826.8	9.58	10	0.132	0.0056	5.60 x 10^{-4}	4.24 x 10^{-3}
3	1378	9.58	6.2	0.220	0.0115	18.55 x 10^{-4}	8.43 x 10^{-3}

D = 50.8 mm; L = 25.4 mm; C_d = 0.199 mm

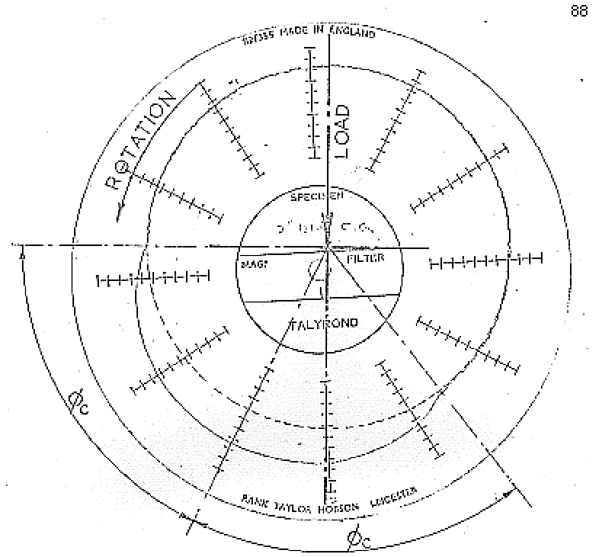
The wear tests made on another 50.8 mm diameter bearing of material "C" for a period of 200 hours revealed that radial wear factor (k_r) was indeed of the order of 8.7 x 10^{-4} mm/*PVH*. The bedding-in wear profile and wear profile obtained after a running period of 5 and 200 hours respectively for the foregoing bearing are shown in Figs. 4 and 5 respectively.



Magnification = 0.0127 mm/ division

PTFE filled Carbon grahite bearing
D = 50.8 mm;   L = 31.75 mm;
C_d = 0.0033 mm; PV = 0.35 MPa . m/s

Fig. 4 Bedding in wear after 5 hours



Magnification = 0.0254 mm / division

PTFE filled Carbon grahite bearing
D = 50.8 mm; L = 31.75 mm;
C_d = 0.0033 mm; PV = 0.35 MPa . m/s

Fig. 5 Wear profile obtained after 200 hours

This test further confirmed the limitations of lineal gauging technique for the correct measurement of radial wear and radial wear factor (k_r) for a sleeve bearing. Another inherent disadvantage of this system was found to be that accumulation of wear debris in the bearing loaded area affected the response of wear measuring probe and gave erratic readings for test runs of longer duration.

6.2 Method 2 - Wear profile measurements

Fig. 6 shows the schematic representation of the bearing wear which could occur in the present experimental apparatus. It indicates that maximum radial wear in a dry bearing mounted in a free housing will not occur along the load line. Equilibrium condition shows that the journal sleeve will be displaced in the case of a dry bearing in a sense opposite to the rotational motion (when a fluid lubricating film is present this displacement will be in the sense of rotation). Thus lineal gauging technique would indicate the vertical component (***r_w***) of the maximum radial wear (***R_w***) of the bearing. The determination of magnitude and location of maximum radial wear (***R_w***) would require measurements in the vertical as well as in the horizontal direction as shown in Fig. 6.

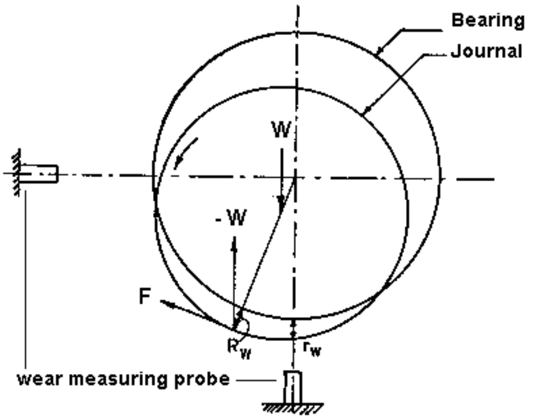


Fig. 6 Schematic representation of the bearing wear profile

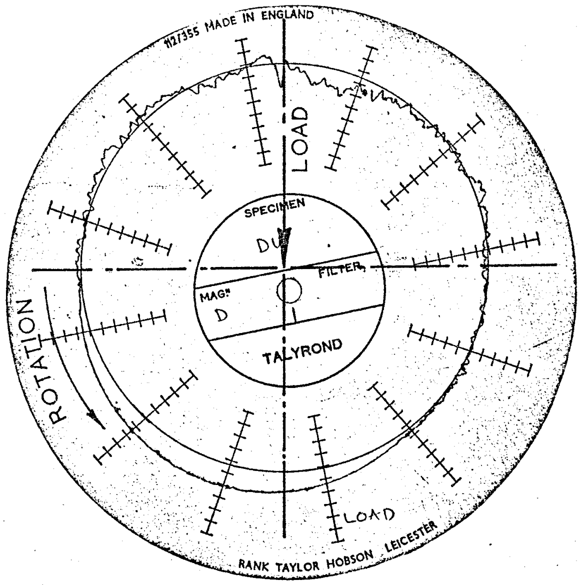
Determination of the geometrical form of the bushing wear was, therefore, considered desirable in order to differentiate between ***R_w*** and ***r_w***. A Talyrond proved useful for this purpose. During measurements on a Talyrond, fine variations of the surface texture of the bearing were cut out and only the general geometrical form of the test bearing was recorded. A general view of the set-up used is shown in Fig 7 and typical wear profiles obtained are shown in Figs. 4, 5 and 8 to 11. This method offered the advantage of location and measurements of ***R_w*** , ***r_w*** and wear contour angle with accuracy.

Fig. 7 Set-up used to measure the bearing circumferential wear profile

6.3 Wear rates comparison

Tests were conducted at *PV* = 0.35 MPa . m/s in order to compare the wear rates of the materials listed in Table 1. The clearance ratio was not constant for these bearing materials, but all the clearances were in the range of 0.048 to 0.254 mm. The clearance ratio of each bearing material was decided on the basis of previous tests (Malik et al., 1998b) so that each bearing could run effectively at *PV* = 0.35 MPa . m/s. Theoretical effect of bearing clearance ratio on the bearing wear rate was reported elsewhere by Malik and Martin (1987). The wear profiles obtained for sleeve bearings of materials “B”, “E”, “C” and “D” are shown in Figs. 8 to 11 respectively. These profiles were obtained after a continuous running of 200 hours.

The analysis of the wear results are summarized in Table 3. The frictional and thermal response of these materials obtained during 200 hours have been reported elsewhere (Malik and Dutra, 1999). The radial wear rates of materials “E”, “C” and “D” varied between 5.1×10^{-4} and 5.7×10^{-4} mm/hour. The wear rate of material “B” was about 1/5th of the wear rate of other materials investigated. These results showed clearly the superiority of bearing material “B” (PTFE/lead-impregnated porous bronze) over the other bearing materials tested. Material “E” was recommended by the manufacturer for low speed and high pressure conditions (i.e., 0.005 m/s and 137.8 MPa). The surface velocity in excess of 0.25 m/s is not

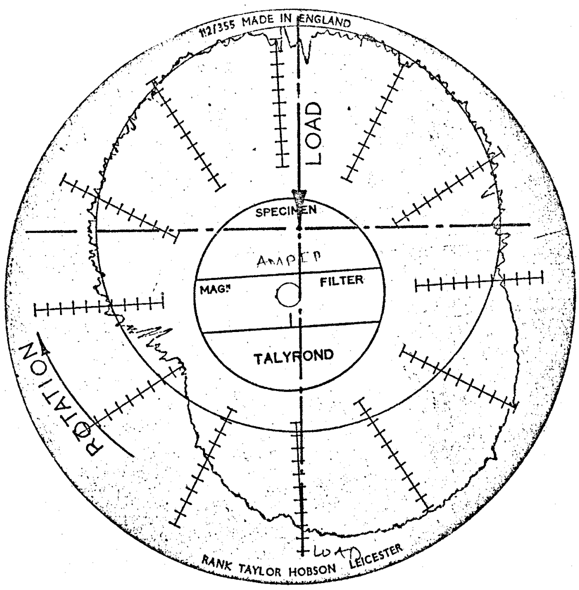


Magnification = 0.0127 mm/division

<u>"B" PTFE/lead-impregnated porous bronze</u>

D = 31.75 mm; L = 31.75 mm;
C_d = 0.048 mm; PV = 0.35 MPa . m/s

Fig. 8 Wear profile after 200 hours

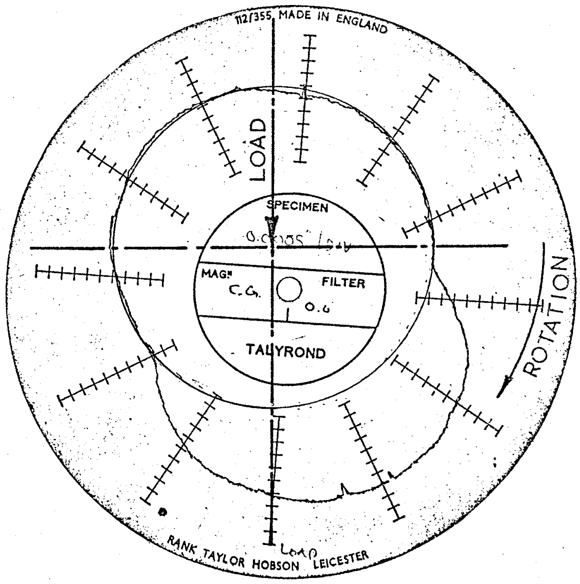


Magnification = 0.0127 mm/division

<u>"E" PTFE-cotton weave reinforced with a thermoset reinforced cotton backings</u>

D = 31.75 mm; L = 25.4 mm
C_d = 0.086 mm; PV = 0.35 MPa . m/s

Fig. 9 Wear profile after 200 hours

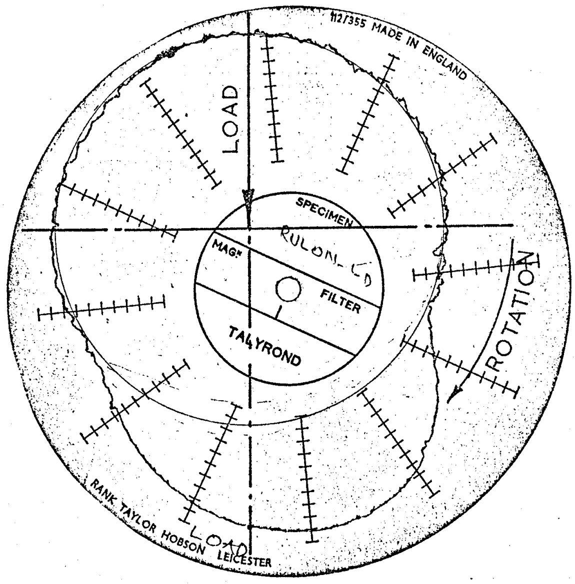


Magnification = 0.0127 mm/ division

<u>"C" PTFE-filled carbon graphite</u>

D = 31.75 mm; L = 31.75 mm;
C_d = 0.137 mm; PV = 0.35 MPa . m/s

Fig. 10 Wear profile after 200 hours



Magnification = 0.0127 mm/ division

<u>"D" Ceramic type filled PTFE</u>

D = 31.75 mm; L = 25.4 mm;
C_d = 0.257 mm; PV = 0.35 MPa. m/s

Fig. 11 Wear profile after 200 hours

Table 3 Wear Rates Comparison at *PV*-value of 0.35 MPa . m/s

Bearing Material	$\boldsymbol{R_w}$, mm	Wear rate/hr ($\boldsymbol{R_w}$/200)	Wear rate factor , $\boldsymbol{k_r} = \boldsymbol{R_w} / PVH$	Manufacturer rated *PV*-value
Material "B"	0.0216	1.08×10^{-4}	3.09×10^{-4}	0.56 MPa . m/s
Material "E"	0.108	5.4×10^{-4}	15.42×10^{-4}	0.70 MPa . m/s
Material "C"	0.102	5.1×10^{-4}	14.57×10^{-4}	0.70 MPa . m/s
Material "D"	0.114	5.7×10^{-4}	16.29×10^{-4}	0.53–0.70 MPa . m/s

considered suitable for this material. The high wear rate indicated by material "E" could be due to the sliding speed of 0.3 m/s used for this test. The wear rate factors ($\boldsymbol{k_r}$) in mm/*PVH* determined for the materials investigated along with manufacturer rated *PV*-values are also presented in Table 3.

6.4 Frictional performance

A difference in the frictional performance of these materials due to radial wear and running time has been reported in detail elsewhere (Malik and Dutra, 1999), however, only a summary is given here. The frictional performance of material "B" was found to be steady during the test period. The friction coefficient (μ) varied between 0.15 and 0.25 during initial running-in period but subsequently decreased to 0.15. The frictional performance of bearing material "E" was also steady but coefficient of friction (μ) increased from 0.2 to 0.28 due to bearing wear and running time. Similarly μ for material "C" increased from 0.2 to 0.32 during 200 hours of continuous operation. Material "D" indicated μ = 0.12 – 0.25 during initial running period (about 50 hours) but afterward frictional response was unsteady and μ increased to 0.35 as a combined result of wear and running time. The wear of rate of bearing material "D" was determined as 5.7 x 10^{-4} mm/hour (Fig. 11 and Table 3). Therefore, radial wear after 50 hours would be about 50 x 5.7 x 10^{-4} = 285 x 10^{-4} = 0.0285 mm. In other words, it could be concluded that bearing material "D" will not run steadily when 0.0285 mm of radial wear has taken place.

The hours taken by the bearing to wear to a radial depth of 0.125 mm is generally taken as the end of its useful life. The useful life of material "B"is generally considered by its manufacturer as the time taken to wear to a depth of 0.05 mm. The data summarised in Table 3 can be helpful in determining the radial wear rates of materials investigated and their useful life. However, it should not be extrapolated to bearings of different lengths or diameters or both with the assumptions that these bearings would give the same radial wear in the same running period.

6.5 Performance of material "A" and material "F"

The bearing material "A" showed a peculiarity not exhibited by any of the other dry bearing materials tested during this investigation. Its coefficient of friction decreased from 0.5 to 0.3 during initial few minutes of running period but afterward varied between 0.3 and 0.2. This was attributed to the high wear rates of the material observed and the accumulation of wear debris experienced in loaded area of the bearing. The friction peak ($\mu \approx 0.5$) was more prominent at high loads. The decrease in friction coefficient was attributed to the build up of a layer of fine wear particles of graphite and bronze as bearing indicated high wear rates. Bearing performance was noisy at *PV*-values higher than 0.25 MPa . m/s. The momentarily rise in the friction coefficient of the material "A" as reported elsewhere (Malik et al., 1998b) would probably make it suitable for braking conditions rather than continuous sliding. The results given by the material “A” were disappointing, not so much as regards the value of the bearing temperature but because of unsteady, erratic and troublesome running versus scoring of the journal sleeves and rapid wearing out of the bearing. This material did not prove a suitable mating material for EN-56-AM stainless steel. In general, the friction coefficient varied between 0.5 and 0.2 and decreased with increase in the operating *PV*-values. The material was not promising at lower *PV*-values, therefore it could not be investigated continuously for 200 hours.

The friction coefficient of the bearing material “E” varied between 0.2 and 0.35 and increased with the increase in the sliding distance as reported elsewhere (Malik and Dutra, 1999). The bearing material indicated unsteady frictional response at *PV*-value of 0.07 MPa . m/s and during initial stages of investigation. The clearance ratio was increased to 0.013 mm/mm but it did not improve its frictional performance. Therefore, it was not possible to test this material continuously for 200 hours at a *PV*-value of 0.35 MPa . m/s.

7. CONCLUSIONS

- The radial wear rates of materials “E”, “C” and “D” varied between 5.1×10^{-4} and 5.7×10^{-4} mm/hour. The wear rate of material “B” was about 1/5th of the wear rate of other materials investigated. These results showed clearly the superiority of bearing material “B” (PTFE/lead-impregnated porous bronze) over the other bearing materials tested. Material “B” showed very low wear rates and low coefficient of friction.
- The sintered metal bearing material “A” and MoS_2 filled Nylon material “F” proved to be disappointing against EN-56-AM stainless steel. It was not possible to test these materials continuously for 200 hours or at higher *PV*-values.
- The Talyrond proved extremely useful for the determination of circumferential wear profiles. This method offered the advantage of location and measurement of $\boldsymbol{R_w}$, $\boldsymbol{r_w}$ and wear contour angle with accuracy.
- The lineal gauging technique did not prove useful for the correct measurement of radial wear and radial wear factor ($\boldsymbol{k_r}$) for a sleeve bearing. The accumulation of wear debris in the bearing loaded area affected the response of wear measuring probe and gave erratic readings for test runs of longer duration.

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