Bearing Oil Delivery by Disk-Scraper Means

Certain bearing applications require the use of a lubrication system contained within the bearing housing and independent of any outside systems or power sources. Marine main propulsion lineshaft bearings often fall into this category and the use of a disk-scraper system for oil supply represents one solution. Tests on two sizes of such lubricating systems were conducted to determine flow rates and the influence of the primary variables involved. The test data is presented and correlated to assist in the design of this type of lubrication system.

Introduction

One means used to supply lubricant to bearings is a disk-scraper arrangement. This may be employed where a self-contained non-pressurized lubrication system is feasible and desired on a horizontal shaft unit. Such an arrangement consists of a disk, secured to and rotating with the shaft, the lower portion of which is submersed in the oil contained in the sump portion of the bearing housing. Oil adheres to the disk, is carried with the disk as it rotates, and is then scraped off in the upper area of the bearing housing where it is directed and flows by gravity to the bearing to be lubricated.

The test work reported here is from disk-scraper designs used in conjunction with marine main propulsion sleeve type lineshaft bearings. This is a relatively common application for such lubrication means, as a self-contained bearing and oil supply system is highly desirable and the operating conditions normally allow it to be used. In this application the disk-scraper system competes with oil rings, but is normally preferred for its more positive oil supply, particularly at low shaft speeds. Under conditions of low shaft speeds, as on turning gear, there have been instances of oil rings being inadvertently restrained from rotating, resulting in a loss of oil to the bearing. The restraint may come from a guide or a scraper, if present, and may be influenced by ship motions of roll and pitch, in such applications.

The disk-scraper arrangement discussed here avoids this problem as the disk is secured to and rotates with the shaft. This disk is commonly a separate split piece which clamps to the shaft but may also, of course, be an integral part of the shaft. A typical disk-scraper lubricated lineshaft bearing is outlined in Fig. 1.

Oil ring lubrication systems are widely used, particularly on horizontal shaft electric motors, and in this respect have been the subject of several technical papers. Test work in this area by Lemmon and Booser is reported in [1], and this also includes references on earlier work on oil rings and associated bearings. The published literature on disk-Scraper type lubrication systems appears to be much more limited. A previous paper by the author [2] on a related subject gave some brief data in this area.

One of the areas of prime interest in these tests was the operation of the disk-Scraper oil supply arrangement at turning gear speeds (normally in the range of \( \frac{1}{2} \) to \( \frac{1}{4} \) r/min for propulsion shafting, reduced from the turbine turning gear speeds by the reduction gear). Depending on the particular application, operation from these turning gear speeds up to maximum shaft speeds in the range of about 80 to 200 r/min will be experienced by the subject bearings. The lubrication system must operate satisfactorily from the lowest to highest shaft speeds, typically a ratio of 1000 to 1. Satisfactory operation implies the supply of sufficient oil to meet the requirements of the particular bearing. These requirements vary with shaft speed, as does the supply from the disk-Scraper system.

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1 Numbers in brackets designate References at end of paper.
Thus knowledge of the anticipated flow rate from this type of lube system is critical to the satisfactory design of bearings so lubricated.

Test Equipment and Procedure

Flow delivery tests were made on two sizes of disk-scraper arrangements. One had a nominal disk OD of 560 mm (22 in.) with a disk width of 70 mm (2.75 in.). This was used for lubricating a 330 mm (13 in.) dia sleeve bearing and the setup for these tests is shown in Fig. 2. The bearing housing upper half has been removed and the scraper held in place by an added bracket. The oil flow rate was measured by diverting the oil stream to a graduated beaker and checking with a stopwatch the time required for a fixed quantity of oil to flow. Tests on this setup were limited to shaft speeds below which oil is thrown from the disk by centrifugal force.

Speed variations were obtained from the hydraulic drive used, with the very low (turning gear) speeds obtained by inserting a 156:1 gear reducer in the drive line. Shaft speed was measured by means of a 60 tooth gear on the drive motor shaft, a magnetic pickup, and an electronic counter.

The other test setup used a disk with a nominal OD of 940 mm (37 in.) and a width of 127 mm (5 in.). This was designed for use with a 641 mm (25.25 in.) dia sleeve bearing. This disk-scraper arrangement was mounted to the same basic test stand as the smaller disk, using the same drive and instrumentation. This disk was encased in a sheet metal and clear plastic housing which permitted visual observation of the flow, retained the oil at the higher speeds, and provided a catch pocket similar in size and location to that on the bearing with which this lube system was used. The setup is shown in Fig. 3. The oil from the scraper is directed to the catch pocket where it flows axially out of the enclosure, falls vertically to the sloping ramp where it is guided down and to the left to be measured by beaker and stopwatch, as in the other tests, or directed back to the sump. Flow tests were made on this larger disk from 1/8 to 200 r/min.

Oil temperatures were monitored by thermocouples throughout these tests and recorded. Turbine type oils of four different viscosity grades were used. Typical viscosities for these oils are given below. All flow rates were confirmed by a minimum of one additional flow determination at each test point.

<table>
<thead>
<tr>
<th>Oil Grade</th>
<th>Viscosity, mPa-s @ 38°C (100°F)</th>
<th>Viscosity, mPa-s @ 99°C (210°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Light</td>
<td>30</td>
<td>4.0</td>
</tr>
<tr>
<td>Medium Heavy</td>
<td>62</td>
<td>6.4</td>
</tr>
<tr>
<td>Heavy</td>
<td>81</td>
<td>7.2</td>
</tr>
<tr>
<td>Extra Heavy</td>
<td>117</td>
<td>11.0</td>
</tr>
</tbody>
</table>

The OD surfaces of the test disks are not cylindrical, as might be anticipated, but are slightly conical. This is to compensate for shaft rake, which is a common practice on marine propulsion shafting, and for the additional dynamic factor of pitch, which may be present. The angle on the disk is thus generally a few degrees, but serves to insure that oil from the disk flows in the proper direction. This is further insured by the contour of the scraper, which is normally tapered or curved in its axial dimension to direct the oil to the bearing inlet.

The scrapers ride on the disks, held down only by their own weight. They are held axially and circumferentially by guides which allow a limited movement. The details of this are shown in Fig. 4 for the large disk. The large guide (on the side opposite the bearing) is simply to direct the oil scraped from this face of the disk away from the shaft seal. The smaller guide (on the bearing end of the scraper) directs the flow of oil from the disk OD into the bearing. The scraper is bored to match the disc OD. The center axial groove in the bore of the scraper helps reduce the hydrodynamic capacity. The entrance edges are not rounded or tapered but left square to improve the scraping action.

Test Results

Flow data were taken from both disk-scraper setups at a large number of test points. The plots of Figs. 5 and 6 are representative of the results at some of the lower shaft speeds. The test points generated smooth, well-defined curves. The specific viscosity values for each test are noted on the curves, along with the grade of oil.

To determine the influence of oil viscosity, disk velocity, and disk width on the flow rate, the flow values were plotted independently against these variables on log-log paper. The flow versus velocity plots resulted in slopes of approximately 0.5 while those of flow versus viscosity were very close to 0.6. The effect of disk axial width, b, was linear.

Test data from both disks, using all four oils (viscosity grades) are plotted versus this resulting flow parameter, b(DN)°, in Fig. 7. The plotted test points are well represented by the straight line shown. This relationship between flow and the factors of oil viscosity, disk velocity and disk width is quite similar to that given by equation (1) of reference [1] for the delivery of oil from a ring at low speeds. As in the reference [1] work, the effect of surface tension was not evaluated. At least for the oils and temperatures involved here it is a relatively constant factor.

After plotting the data in Fig. 7 and establishing the straight line through the data points it was found that this line appeared to miss the zero-zero coordinate by a small but definite amount. This implied that the flow went to zero at some positive value of shaft speed, which had not been noted in the tests even at 1/8 r/min (See Fig. 6).

The area of this plot (Fig. 7) near zero-zero was investigated by plotting additional data at the very low speeds and this is shown in Figs. 8 and 9. (The straight lines are the same in Figs. 7, 8, and 9.) In Fig. 8 it can be seen that the test data points clearly go to zero-zero, deviating from the straight line at a value of about 12 million of the flow parameter. This is probably best seen in Fig. 9 where data from one specific test run is plotted. Excellent correlation with the straight line is seen down to about the 12 million figure for the flow parameter. From here the data points curve to the zero-zero coordinate.

At higher shaft speeds, where oil is thrown from the disk by cen-
trifugal force, only the setup with the large disk permitted measurement of oil flow values. As shaft speed increases from zero, the delivery of oil follows the pattern shown previously until some oil starts to be thrown from the disk, initially in the area where the disk emerges from the surface of the oil in the reservoir. Deviation from the straight line relationship of Fig. 7 begins, but an increase in flow rate with speed continues. As the shaft speed continues to increase, the upper boundary of this oil throw-off action moves up from the reservoir oil level.

Fig. 10 shows the flow rates obtained from the large disk up to the maximum speeds tested, using two different oils. The loss of oil delivery to the bearing due to centrifugal action is readily apparent. Data from Fig. 10 are plotted versus the flow parameter in Fig. 11. The straight line from the earlier low speed plots is shown for comparison.

Discussion

One item of interest found in this test work was the excellent correlation of the flow rate and a flow parameter at shaft speeds which
did not result in centrifugal throw-off of oil from the disk. At the very low end of this flow parameter (approaching zero r/min) deviation from the relationship was exhibited. This is believed to be associated with the relative thicknesses of the oil film developed between the disk and the scraper, and the oil layer carried up on the disk. At higher speeds, the oil layer is many times thicker than the oil film and the amount of oil carried under the scraper by this film is not significant compared to the amount scraped off. However, this film thickness decreases toward zero at very low shaft speeds and some of the flow previously carried through by the film is now part of the oil scraped off. The layer thickness decreases also as the shaft speed is reduced, with the oil represented by the film thickness at higher speeds now being a significant part of the flow. This prevents the occurrence of
zero flow at a positive shaft speed, as would be predicted by the straight line relationship between flow and the flow parameter, found at high shaft speeds.

This would indicate that one means of obtaining somewhat higher flows at low shaft speeds would be to weight or spring load the scraper to the disk to reduce the film thickness. This has not, however, been found necessary in practice.

Although no systematic study of the effect of disk immersion depth was made, no significant change in the flow rates was found with variations as would be expected in normal operation. This range was from a minimum immersion of about 25 mm (1 in.) to a maximum of about 75 mm (3 in.).

**Summary**

Tests were conducted on two sizes of disk-scraper lubrication systems as used in conjunction with marine main propulsion lineshaft bearings. Emphasis was placed on obtaining flow data at the lower speed ranges to establish aids for designing lube systems to insure sufficient oil for these conditions. The tests provided consistent flow data which correlated well with a flow parameter at the lower shaft speeds.

**References**


**Fig. 10** Flow rate for full range of shaft speeds tested

**Fig. 11** Flow rate data versus flow parameter for full range of test conditions