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Testing of Friction and Wear

14.1 INTRODUCTION

There is an increasing requirement for testing the performance of bearing materials, lubricants, lubricant additives, and solid lubricants. For bearings running on ideal full oil films, the viscosity is the only important lubricant property that affects the friction. However, in practice most machines are subjected to variable conditions, vibrations and disturbances and occasional oil starvation. For these reasons, even bearings designed to operate with a full fluid film will have occasional contact, resulting in a rubbing of surfaces under boundary lubrication conditions and, under certain circumstances, even under dry friction conditions. Many types of oil additives, greases, and solid lubricants have been developed to reduce friction and wear under boundary friction. Users require effective tests to compare the effectiveness of boundary lubricants as well as of bearing materials for their specific purpose.

It is already known that the best test is one conducted on the actual machine at normal operating conditions. However, a field test can take a very long time, particularly for testing and comparing bearing life for various lubricants or bearing materials. An additional problem in field testing is that the operation conditions of the machines vary over time, and there are always doubts as to whether a comparison is being made under identical operating conditions. For example, manufacturers of engine oils compare various lubricants by the average miles the car travels between engine overhauls (for expediting the field test, taxi service cars are used). It is obvious that the cars are driven by various drivers; and most probably, the cars are not driven under identical conditions. Field tests can be expensive if the bearings are periodically inspected for wear or any other damage. Concerning the measurement of friction-energy losses, in most cases it is impossible to test friction losses on an actual machine. Friction losses in a car engine can be estimated only by changes in the total fuel consumption. Obviously, this is a rough estimate because friction-energy loss is only a portion of the total energy consumption of the machine.

For all these reasons, various testing machines with accelerated tests have been developed and are used in laboratory simulations that are as close as possible to the actual conditions. The common commercial testing machines are intended for measuring friction and wear for various lubricants under boundary lubrication conditions or for comparing various solid lubricants under dry friction conditions. Most commercial testing machines operate under steady conditions of sliding speed and load.

14.2 TESTING MACHINES FOR DRY AND BOUNDARY LUBRICATION

Most commercial testing machines are for measuring friction and wear under high-pressure-contact conditions of point or line contact (nonconformal sliding contacts) (Fig. 14-1). These tests are primarily for rolling bearings and gear lubricants. In addition, there are many testing machines for journal bearings and thrust bearings (conformal contacts). For nonconformal contacts, a widely used test is the four-ball apparatus, where one ball rotates against three stationary balls at constant speed and under steady load. The operating parameters of wear, friction, and life to failure by seizure (when the balls weld together) are compared for various materials and lubricants. The friction torque is measured and the friction coefficient is calculated. In addition to friction, the time or number of revolutions to seizure can be measured as a function of load. Wear can also be compared by intermediate measurements of weight loss or changes in ball diameters, for various ball materials and lubricants.

The following are examples of friction and wear tests of various nonconformal contacts that have been introduced by various companies.

Four-ball machine (introduced by Shell Co.)
Pin on a disk (point contact because the edge of the pin is spherical)
Block on rotating ring (introduced by Timken Co.)
Reciprocating pad on a rotating ring
Shaft rotating between two V-shaped surfaces (introduced by Falex Co.)
SAE test of two rotating rings in line contact

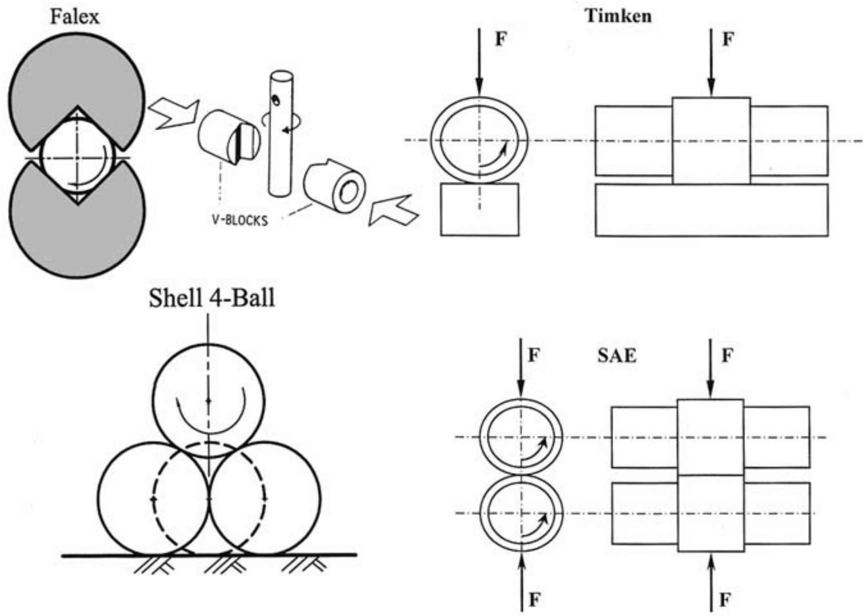

FIG. 14-1 Friction and wear tests of nonconformal contacts.

Although these testing machines are useful for evaluating the performance of solid lubricants and comparing bearing materials for dry friction, there are serious reservations concerning the testing accuracy of liquid lubricants for boundary friction or comparing various boundary friction lubricant additives. These reservations concern the basic assumption of boundary lubrication tests: that there is only one boundary lubrication friction coefficient, independent of sliding speed, that can be compared for different lubricants. However, measurements indicated that, in many cases, the friction coefficient is very sensitive to the viscosity or sliding speed. For example, certain additives can increase the viscosity, which will result in higher hydrodynamic load capacity and, in turn, reduction of the boundary friction.

The friction force has a hydrodynamic component in addition to the contact friction (adhesion friction). Therefore, it is impossible to completely separate the magnitude of the two friction components. Certain boundary additives to mineral oils may reduce the friction coefficient, only because they slightly increase the viscosity. Even for line or point contact, there is an EHD effect that increases with velocity and sliding speed. The hydrodynamic effect would reduce the boundary friction because it generates a thin film that separates the surfaces. This argument has practical consequences on the testing of boundary layer lubricants. These tests are intended to measure only the adhesion friction of boundary lubrication; however, there is an additional viscous component.

Currently, boundary lubricants are evaluated by measuring the friction at an arbitrary constant sliding speed (e.g., four-ball tester operating at constant speed). The current testing methods of liquid boundary lubricants should be reevaluated. Apparently, better tests would be obtained by measuring the complete friction versus velocity, *f-U*, curve. In Sections 14.6 and 14.7, dynamic testing machines are described that are better able to evaluate separately the contact friction at the start-up and the mixed and hydrodynamic friction. The friction is a function of speed, which can be measured by dynamic tests.

14.3 FRICTION TESTING UNDER HIGH-FREQUENCY OSCILLATIONS

It has already been mentioned that in real machines the contact stresses of mating parts in relative motion are not completely constant. There are always vibrations and time-variable conditions. Even when the load is constant, there are friction-induced vibrations, resulting in small high-frequency oscillations in the tangential direction (parallel to the surface). For these reasons, it was realized that testing machines with high-frequency oscillations would offer a better simulation of the actual conditions in machinery.

It is well known that rolling-element bearings operate under high-frequency oscillations, and there has been a need for testing machines that simulate these dynamic conditions. Tests under high-frequency oscillations have been adopted as standard tests, such as ASTM D 5706 EP and ASTM D 5707 EP for greases and oils for rolling bearings.

In the testing machine shown in Fig. 14-2, there is friction between the upper specimen and the lower disk. The upper specimen can be a ball or a cylinder, for point or line contact, or a ring, for area contact. The material and size can be adapted to the user's requirement (equivalent to the material used in the actual machine). During the friction test, the upper specimen has horizontal oscillations (parallel to the disk area). Force is applied mechanically to the upper specimen in a vertical direction (normal to the disk area). The friction force is measured by a piezoelectric sensor that is placed under the lower specimen holder. The friction coefficient is calculated and recorded on-line on a chart during the test. The environment in the test chamber (temperature and humidity) can be controlled.

This test has been adopted by the American Society of Testing Materials (ASTM) for testing greases or liquid lubricants operating under high contact pressure, such as point or line contact in rolling bearings and gears. The

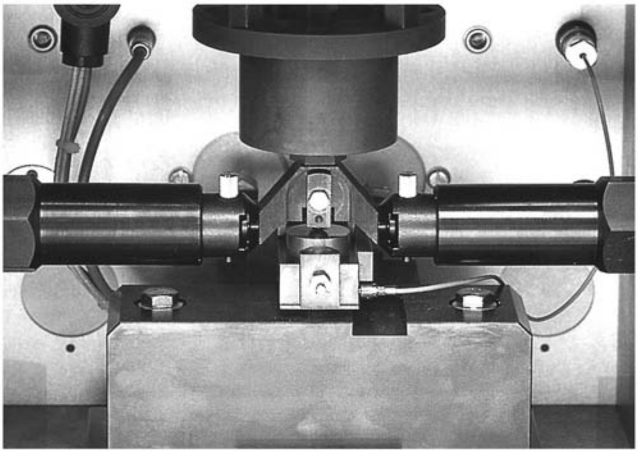

FIG. 14-2 Testing apparatus for friction and wear under high-frequency oscillations (from SRV Catalogue, with permission from Optimal Instruments).

manufacturers of the dynamic testing apparatus claim that actual comparisons with the performance of the same greases in the field indicated that dynamic tests are more reliable than static tests for comparing the performance of various greases.

During the ASTM D 5706 EP standard friction test, the upper part has horizontal oscillations of 1-mm amplitude and a frequency of 50 Hz. The test is run with a very small amount of grease, 0.1–0.2 g grease. After 30 seconds break-in under a load of 50 N, the load is raised by increments of 100 N at 2-minute intervals until failure occurs. Failure is determined by seizure or by a significant sudden rise in the friction force. The tests are run at elevated temperature to simulate the actual operating conditions of rolling-element bearings.

A similar standard test is ASTM D 5707 EP. In this test, however, the friction coefficient, as well as wear, versus time is recorded on a chart. The test is run with a very small amount of grease, 0.1–0.2 g grease; the frequency of horizontal oscillations is 50 Hz and 1-mm amplitude. After 30 seconds break-in under a load of 50 N, the load is raised to 200 N for 2 hours. The lowest and highest values of friction on the chart are reported. In addition, after the test, the average wear scar diameter on the test ball is measured with the aid of a microscope and on the lower specimen with a profilometer. These readings are reported as wear test results. The test can be applied for comparing various liquid lubricants. The wear can be measured on-line during the test by measuring the depth of the wear scar, as shown in Fig. 14-3. Larger-amplitude vibrations can be applied to better simulate the conditions in an actual machine.

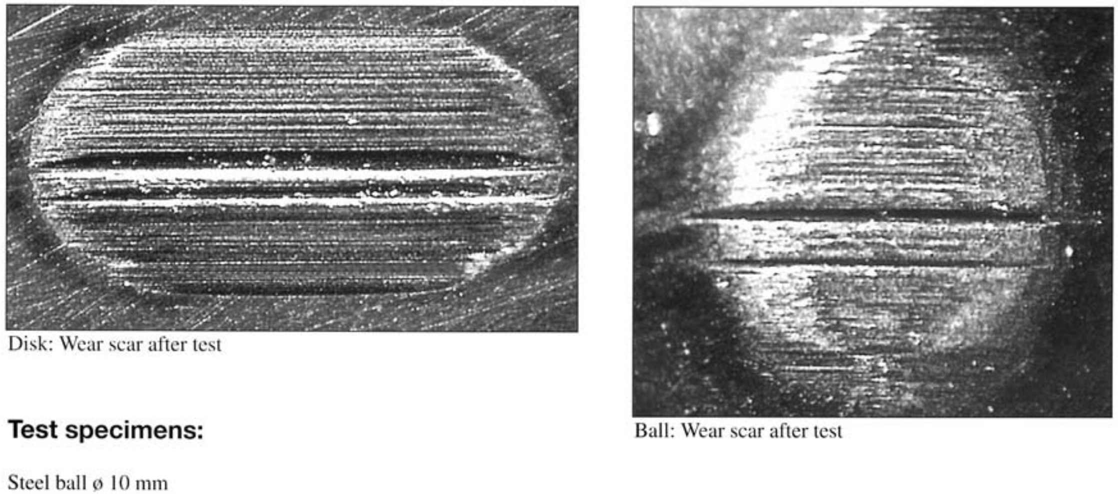

FIG. 14-3 Wear scars after a standard vibratory friction and wear test (from SRV Catalogue, with permission from Optimal Instruments).

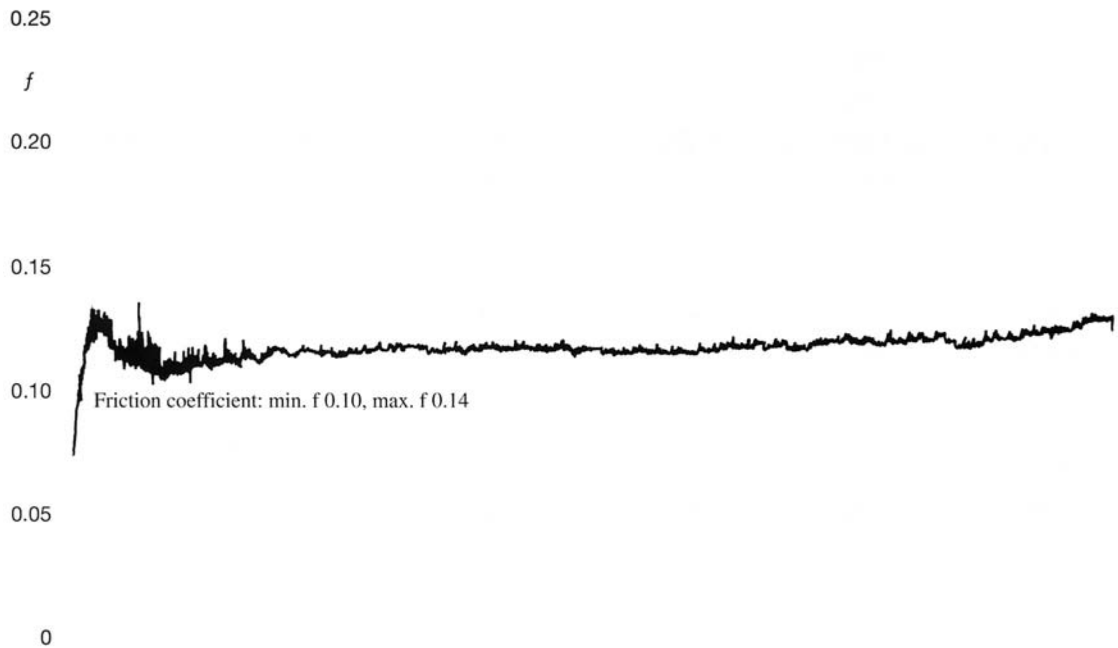

FIG. 14-4 Friction coefficient versus time (from SRV Catalogue, with permission from Optimal Instruments).

In Fig. 14-4, a test result is shown for a steel ball on a steel plate lubricated by synthetic oil. The result is a curve of friction coefficient versus time. The test time is 2 hours and the specimens are 10-mm steel balls on a lapped steel disk at a temperature of 200°C. The frequency of horizontal oscillations is 50 Hz and 1.5-mm amplitude. The reported friction coefficients are $f_{min} = 0.1$ and $f_{max} = 0.14$. The maximum wear measured during the test is 21 μm. The wear scars after the test on the disk and ball are shown in Fig. 14-3.

The reservations that have been raised for the steady tests are still valid for this vibratory test. Although these dynamic tests are effective in simulating the overall performance of real machines, a problem with the high-frequency test is that it does not test the pure effect of lubricant additives, such as antifriction and antiwear additives. The friction and wear are the combined effect of the viscosity of the lubricant as well as of the additives. In other words, there is no way to distinguish between the hydrodynamic and adhesion friction effects. Therefore, this would not be a good method to compare the effectiveness of various boundary lubrication additives.

In Sec. 14.4, a testing machine is described for testing the complete Stribeck curve. It offers a better distinction of the contact and viscous friction and the friction at each region. Therefore, the Stribeck curve can be a more useful test in developing and selecting lubricants. Nevertheless, the foregoing high-frequency test is very useful in testing solid lubricants and greases. For liquid lubricants, the test is useful for evaluating the combined effect under identical conditions of a specific application in the field.

14.4 MEASUREMENT OF JOURNAL BEARING FRICTION

The purpose of friction-testing machines is to measure the friction torque of a journal test-bearing friction or rolling-element test-bearing friction in isolation from any other source of friction in the system. There are several methods by which to measure the friction in bearings. The first method is based on the concept of the hydrostatic pad. It is designed for measuring the friction torque on the bearing housing by a load cell, while the bearing load is transferred to the bearing housing through a hydrostatic pad. Friction-testing machines with a hydrostatic pad can be designed for the measurement of static or dynamic friction. Dynamic friction is under time-variable conditions, such as oscillating velocity and time-variable load. Dynamic friction measurements require continuous recording or on-line data acquisition by a computer. All friction-testing machines for dynamic friction are universal, in the sense that they can be used under steady conditions as well as dynamic conditions. In most cases, however, machines for testing steady friction cannot be adapted for dynamic friction.

A relatively simple friction-testing machine is the pendulum tester. It can be applied for testing the friction coefficient of a journal bearing under steady conditions only.

The concept of this pendulum friction tester is to apply a load on the bearing by means of weight. The weights are placed on a rod connected to the bearing. During a steady operation under constant speed, the pendulum is tilted to an angle equal to the friction coefficient. An example of a pendulum tester is shown in Fig. 14-5. The angle is small, and the angle is measured by a dial gauge as shown.

This is a simple and low-cost tester. However, it has relatively low measurement precision in comparison to other machines. There are always small vibrations of the pendulum that make it difficult to get an average reading. This can be improved by damping the vibrations via a viscous damper. A second drawback that reduces the precision is that there is always some friction and it is

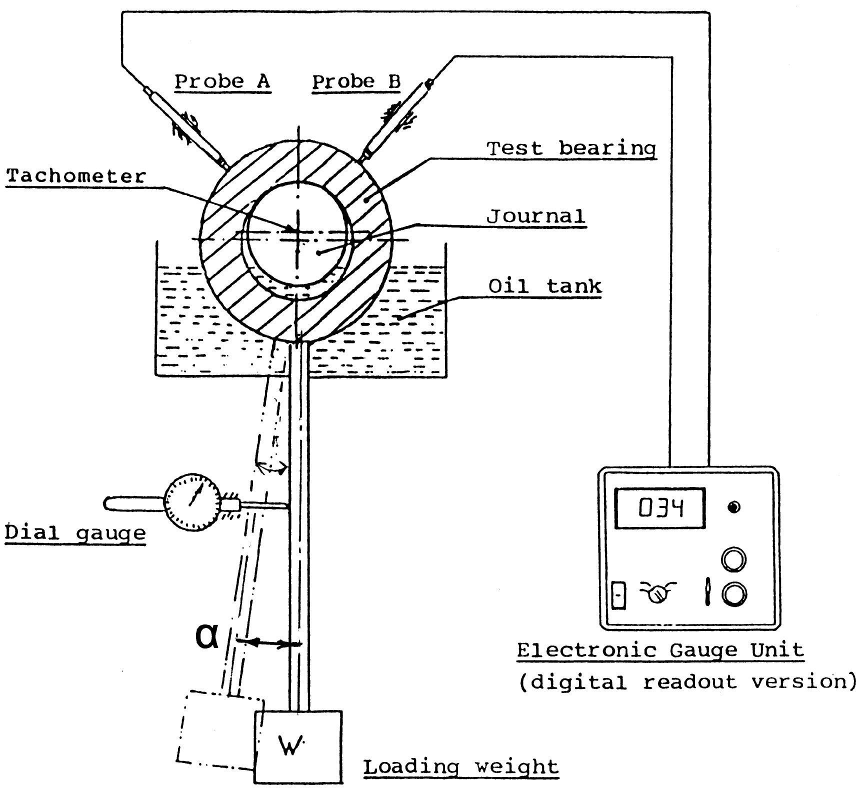

FIG. 14-5 Pendulum-type friction tester for a journal bearing.

impossible to adjust the zero position of the pendulum. A solution to this problem is to test in the two directions; namely, for each measurement the shaft is rotating in two directions. The pendulum-swing angle is measured for each direction and the average calculated. This is a relatively time-consuming test.

This tester is limited to friction measurements under steady speed. A variable-speed motor is used for measuring the f–U (friction versus velocity) curve. However, each point in this curve is measured under steady-state conditions. Since this tester is not for high-precision measurement, it is not suitable for comparing lubricants where the difference in friction is expected to be within a few percentage points.

14.5 TESTING OF DYNAMIC FRICTION

Most of the commercially available friction-testing machines have been designed for measurements under steady conditions. For the measurement of dynamic friction, under time-variable conditions, a unique design of the testing equipment, with strict requirements, is called for. In addition to a rigid design, on-line recording of the data and its processing is essential for time-variable conditions.

The most important principles in dynamic friction measurement are as follows.

1. The machine as well as the support for the test bearing must be very rigid. In addition, the load cell for measuring the friction force must be as rigid as possible.
2. Relative sliding is obtained by means of a stationary part and a moving part. The load cell for the friction measurement must always be connected to the stationary side. If the load cell were to be connected to the moving part, it would not read the correct friction force because it would read inertial forces as friction force, and the result would be a combined reading of friction and inertial forces.
3. The design must provide for a clear method of separating the measured friction in the test bearing from any other sources of friction in the system.
4. The testing system must provide the means for accurate measurement of the velocity and displacement of the sliding part relative to the stationary part.
5. The system must provide the means for on-line recording of the friction versus time and versus sliding velocity. This is currently done by a computer with a data-acquisition system. In addition, there is a requirement to measure friction versus a small displacement during the start-up.

6. The system must include the means to control the desired time-dependent sliding motion and load. This can be achieved by using a computer with direct current output and an amplifier that controls a servomotor for the required motions. The controller in the computer includes the algorithm for the control of motion and velocity. The motion and velocity are measured on-line to provide feedback to the computer controller for precision motion.

If the support of the steady part is not sufficiently rigid (including the load cell), there are several types of errors that are encountered in the measurements. Under dynamic operation, the stationary part will have a small variable displacement due to the elasticity in the system. This would result in reading errors in the load cell because small inertial forces would be added to the friction reading. This means that due to a variable elastic displacement there is a small acceleration, and the load cell will read inertial forces as friction force.

Moreover, if the system were not rigid, there would be friction-induced vibrations (stick-slip friction, see Sec. 16.1) at low velocity. In conclusion, the dynamics of the system can affect the friction measurement, and we are interested in a clean experiment where the bearing friction is measured in isolation from any other effect.

The examples in the following sections are of several universal testing machines for measuring rolling-element bearing friction or journal bearing friction under dynamic conditions. Although other designs of friction-testing machines are often used, all are based on similar concepts. The first two friction-testing machines can be applied for a journal bearing or a rolling-element bearing; the third machine is for friction in linear sliding motion.

14.6 FRICTION-TESTING MACHINE WITH A HYDROSTATIC PAD

A friction-testing machine with hydrostatic pad is shown in Fig. 14-6. It has a main shaft supported by two conical rolling bearings. The two bearings form an adjustable arrangement to eliminate undesirable clearance in these bearings. The shaft is driven by a variable-speed motor. In Fig. 14-6, the test bearing is a rolling-element bearing on the right side of the shaft, but it can be a journal bearing as well. The test bearing is housed in a cylindrical casing containing lubricant at a constant level.

The main shaft ends with a cone, on which a conical bore sleeve is mounted. The conical sleeve can by tightened by a nut, and in this way the outside diameter of the sleeve is slightly varied by elastic deformation. The test bearing, a journal bearing or rolling bearing, is mounted on this sleeve, and the clearance

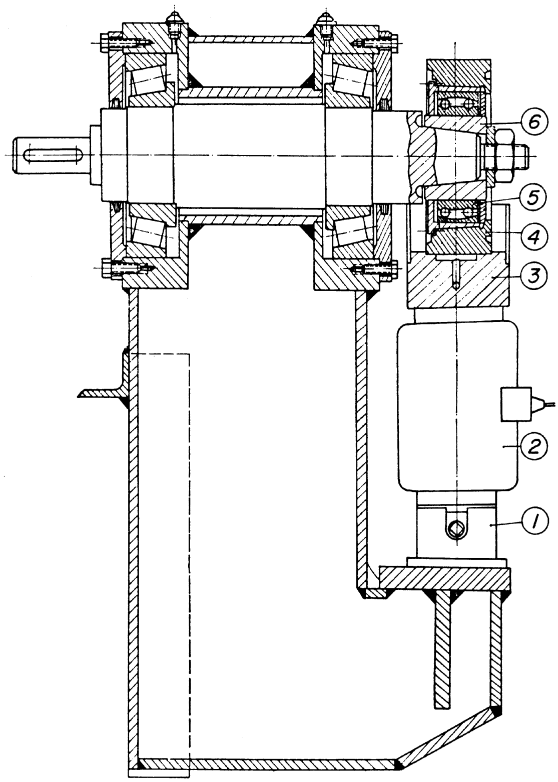

FIG. 14-6 Friction-testing machine with a hydrostatic pad. (From Harnoy, 1966).

(for a journal bearing) or the tight fit (for a rolling bearing) can be adjusted by tightening the nut on the conical sleeve.

The load is applied by means of a jack (1) and measured by a load cell (2). The bearing cylindrical casing is mounted on a hydrostatic pad (3). In this way, the load is transmitted through the hydrostatic fluid film. When the cylindrical casing is not turning, there are no shear stresses in the fluid film, and there is no additional viscous friction on the casing.

There are two symmetrical radial arms that are attached to the casing, on each side, and connected to load cells. The friction torque is measured by two calibrated load cells, which are connected to two symmetrical radial arms, thus preventing the casing from turning. Since there is no friction torque due to the hydrostatic pad, the torque on the casing that is read by the two load cells is equal to the friction torque of the test bearing.

This apparatus measures only the friction in the test bearing and not any other source of friction, such as the two conical bearings that support the shaft. This friction-testing machine is suitable for dynamic friction measurements as well as friction under steady conditions. For more details of this testing machine, see Lowey, Harnoy, and Bar-Nefi (1972).

The operation of this friction-testing machine under dynamic conditions requires a servomotor controlled by a computer and data-acquisition system, as described in Sections 14.7 and 14.8.

14.7 FOUR-BEARINGS MEASUREMENT APPARATUS

An apparatus for dynamic friction measurement has been designed, developed, and constructed in the bearing and lubrication laboratory of the Department of Mechanical Engineering at the New Jersey Institute of Technology. This apparatus can continuously measure the average dynamic friction of four equally loaded sleeve bearings in isolation from any other source of friction in the system, and the errors caused by inertial forces can be reduced to a negligible magnitude. In Fig. 14-7 a cross section of the apparatus is shown, and a photograph is shown in Fig. 14-8.

The design concept is to apply an internal load, action and reaction, between the inner housing (N) and the outer housing (K) by tightening the nut (P) on the bolt (R) to apply preload by deformation of the elastic steel ring (E). There are four equal test sleeve bearings (H), two bearings inside each housing. In this way, all four test bearings have approximately equal radial load, but in the opposite direction for each two of the four bearings, due to the preload in the elastic ring. The load on the bearings is measured by a calibrated, full strain gauge bridge bonded to the elastic ring. The total friction torque of all four bearings is measured by a calibrated rigid piezoelectric load cell, which prevents the rotation of the outer bearing housing (K). The load is transferred to the load cell by a radial arm attached to the external housing, as shown in the apparatus photograph in Fig. 14-8. Thus, the measured friction torque of the four bearings is isolated from any other sources of friction, such as friction in the ball bearings supporting the shaft. The time-variable friction measured by the load cell is stored in a computer with a data-acquisition system.

A lubricating oil reservoir is mounted above the mechanical apparatus in order to supply oil by gravity into the four bearings through four segments of flexible tubing. The oil is drained from the bearings through a hole in the external housing into a collecting vessel.

The shaft (C) is supported by two ball bearings (A) attached to the main support frame (B), and is driven by a computer-controlled DC servomotor. The

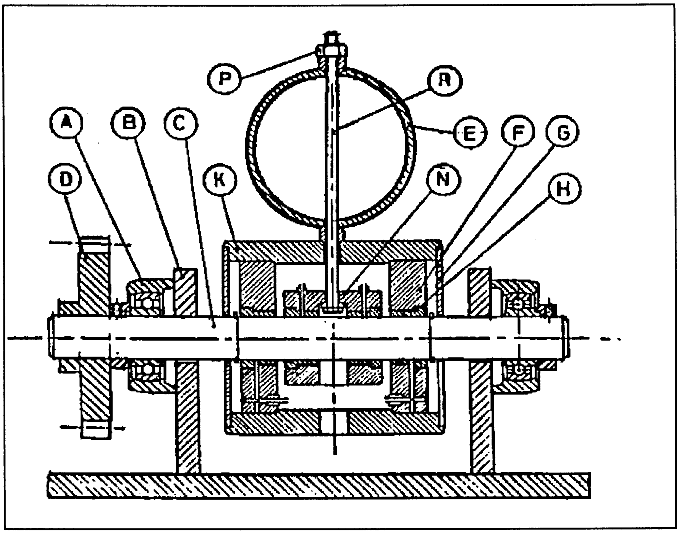

FIG. 14-7 Cross-sectional view of friction measurement apparatus. (From Bearing and Lubrication Laboratory, Department of Mechanical Engineering, New Jersey Institute of Technology.)

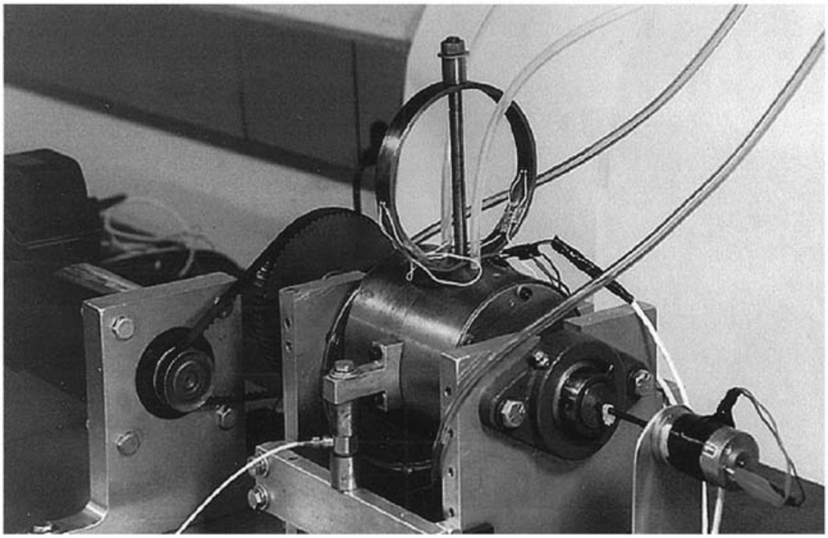

FIG. 14-8 Photograph of friction-testing apparatus. (From Bearing and Lubrication Laboratory, Department of Mechanical Engineering, New Jersey Institute of Technology.)

drive consists of a DC servomotor connected to the shaft through a timing belt and two pulleys (D). The rotational speed of the shaft is measured by an encoder, and this on-line measurement is fed into the computer, where the data is stored and analyzed. This arrangement forms a closed-loop control of the rotation of the shaft. In fact, the control algorithm includes a friction-compensation algorithm to generate the precise sinusoidal velocity or any other desired periodic velocity.

It is interesting to note that the measurement principle of four bearings was used by Mckee and Mckee as early as 1929. However, the early friction-testing apparatus used sliding weights for measuring the friction torque; and of course, this apparatus has been limited to bearings under steady conditions.

An improved version of the four-bearings friction-testing machine for bearings that require self-aligning is shown in Fig. 14-9. The self-aligning property is achieved by means of four self-aligning ball bearings. The self-aligning bearings are held from rotating by thin metal strips. At the same time, elastic bending of the strips allows a small angular rotation for self-aligning. In addition, this design can easily be adapted for measurement of steady and dynamic friction in rolling-element bearings.

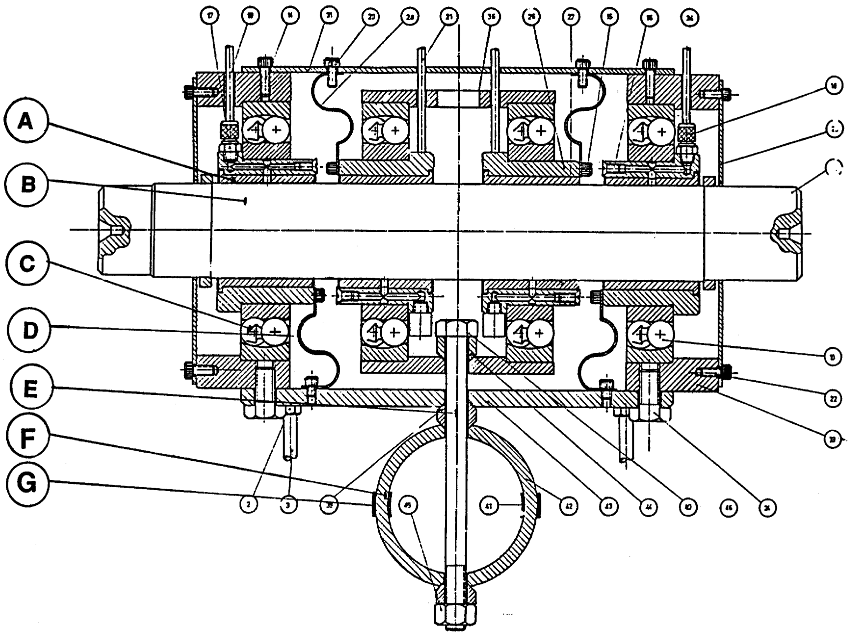

FIG. 14-9 Friction-testing machine with self-alignment arrangement.

14.7.1 Measurement Error Under Dynamic Conditions

It has already been mentioned that under dynamic conditions, there will always be a small, unsteady angular rotation of the bearing housing and sleeve due to the elasticity of its support (including elasticity of the load cell). This is the case in all the testing machines, because there is always a certain elasticity in the system. However, it is possible to minimize this elastic rotation of the bearing sleeve by a rigid design of the frame of the machine and by using a rigid load cell. As a result of this small angular elastic rotation of the housing, there will be a small angular acceleration, and the load cell, which keeps the bearing housing from rotating, has a small error because it reads inertial forces (or torque) as friction force.

For friction measurement under dynamic conditions, the magnitude of measurement error due to inertial forces has been evaluated by Harnoy et al. (1994). The condition for a negligible error is

$$\omega^2 \ll \frac{k_h}{I_h} \tag{14-1}$$

Here, k_h is the angular stiffness of the bearing housing support, ω is the freqency of oscillations under dynamic conditions, such as periodic load, and I_h is the moment of inertia of the bearing housing together with the bearing sleeve.

Piezoelectric load cells are much more rigid than strain gage, beam-type load-cells. However, for low-frequency tests, the piezoelectric cells have the disadvantage of an output drift. At the same time, at low frequency, our error analysis indicated that the elasticity of the load cell does not cause any significant measurement error. This discussion indicates that best results can be achieved by using a piezoelectric load cell for high-frequency tests and strain gage beam-type load cells for low-frequency oscillations.

14.8 APPARATUS FOR MEASURING FRICTION IN LINEAR MOTION

A cross-sectional view of the linear-motion friction measurement apparatus is shown in Fig. 14-10. The apparatus comprises of a linear motion sliding table, driven by a servomotor and a ball screw drive. A closed-loop control system is provided via personal computer. Also, the computer system can store the experimental data for analysis. For regular precision, the sliding motion is measured by an encoder, which measures the rotation of the screw that drives the table. For high-precision measurement of the linear motion, an LVDT motion sensor can be used.

The design concept of the apparatus is a ball screw driven linear-positioning table (1), where the backlash can be eliminated, by preloading the screw drive.

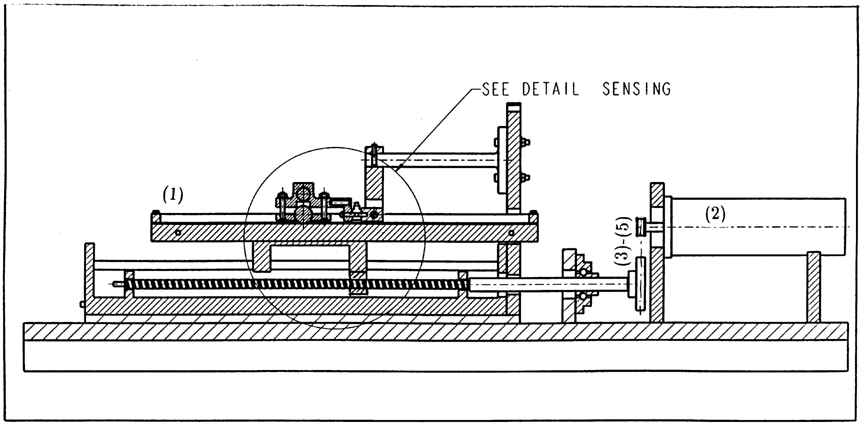

FIG. 14-10 Cross-sectional view of linear-motion friction measurement apparatus (from Bearing and Lubrication Laboratory Department of Mechanical Engineering, New Jersey Institute of Technology).

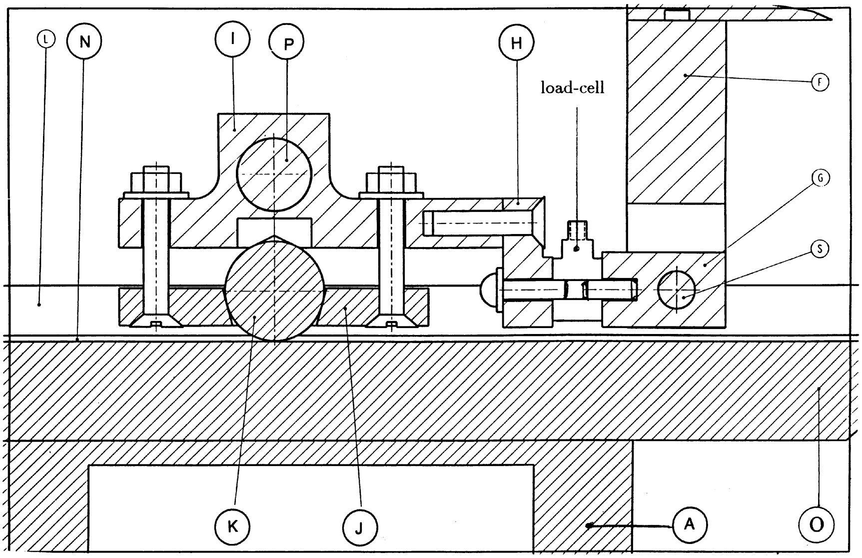

FIG. 14-11 Enlargement of cross-section contact area.

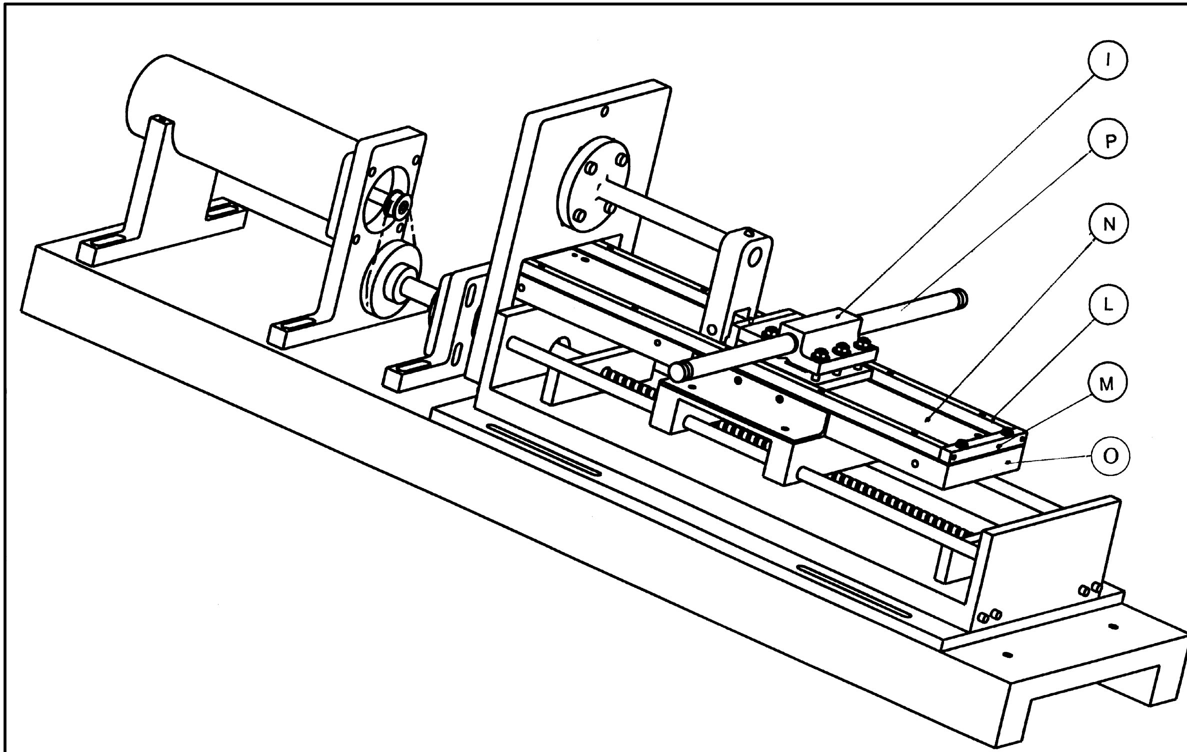

FIG. 14-12 Isometric of a linear-motion friction-measurement apparatus.

This apparatus is designed for measuring friction at very low sliding speeds. This is achieved by a speed reduction of considerable ratio by the screw drive. In addition, the speed of the motor is reduced by a set of pulleys and a timing belt (3-5). Closed-loop controlled motion is generated by a computer-controlled DC servomotor (2). Precise measurements of the motion is fed into a computer, which is equipped with a data-acquisition board.

In this linear apparatus, the contact geometry between the sliding surfaces can be replaced. Enlargement of the sensing area is shown in Fig. 14-11. It can test a sliding plane, a line or a point contact. The drawing shows a line contact between a nonrotating cylindrical shaft and a flat plate. The contact can be made of various material combinations. The line contact is created between a short, finely ground cylindrical shaft (K) and the flat friction surface (N). The shaft (K) is clamped in a housing assembly (I, J and H) designed to hold various shaft diameters. The normal load is centered above the line contact, and is supplied by a rod (P), which has weights attached (weights not shown). When the friction test surface moves, the friction force is transmitted through the housing assembly to a piezoelectric load cell. The load cell generates a voltage signal proportional to the friction force magnitude, which is fed to a data-acquisition system in a computer.

The design is shown in the isometric view in Fig. 14-12. The friction surface base (N) is attached to a moving platform (O) that is driven by the ball screw drive. The friction contact area can be dipped in lubricant, since the base has an attached railing (L and M), which serves to contain the lubricant.

For precise dynamic measurements, particularly with high-frequency oscillations, the load cell must be rigid as well as the support of the load cell. This is essential to prevent undesirable small linear displacement of the stationary cylinder (due to elastic deformation of the load-cell system and the support under the friction force). In the case of elastic displacement, the friction reading may include a small error of inertial force, acting on the load-cell (this is the major reason why most commercial friction testing devices are not suitable for dynamic tests).

Results of dynamic friction measurements performed by the last two testing machines are included in Chapter 17. The tests were conducted for oscillating motion at various frequencies, and the results are in the form of dynamic f–U curves.