16

Friction Characteristics

16.1 INTRODUCTION

The first friction model was the Coulomb model, which states that the friction coefficient is constant. Recall that the friction coefficient is the ratio

$$f = \frac{F_f}{F} \tag{16-1}$$

where F_f is the friction force in the direction tangential to the sliding contact plane and F is the load in the direction normal to the contact plane. Discussion of the friction coefficient for various material combinations is found in Chapter 11.

For many decades, engineers have realized that the simplified Coulomb model of constant friction coefficient is an oversimplification. For example, static friction is usually higher than kinetic friction. This means that for two surfaces under normal load F, the tangential force F_f required for the initial breakaway from the rest is higher than that for later maintaining the sliding motion. The static friction force increases after a rest period of contact between the surfaces under load; it is referred to as *stiction force* (see an example in Sec. 16.3). Subsequent attempts were made to model the friction as two coefficients of static and kinetic friction. Since better friction models have not been available, recent analytical studies still use the model of static and kinetic friction coefficients to analyze friction-induced vibrations and stick-slip friction effects in dynamic systems. However, recent experimental studies have indicated that this model of static and kinetic friction coefficients is not accurate. In fact, a better description of the friction characteristics is that of a continuous function of friction coefficient versus sliding velocity.

The friction coefficient of a particular material combination is a function of many factors, including velocity, load, surface finish, and temperature. Nevertheless, useful tables of constant static and kinetic friction coefficients for various material combinations are currently included in engineering handbooks. Although it is well known that these values are not completely constant, the tables are still useful to design engineers. Friction coefficient tables are often used to get an idea of the approximate average values of friction coefficients under normal conditions.

Stick-slip friction: This friction motion is combined of short consecutive periods of stick and slip motions. This phenomenon can take place whenever there is a low stiffness of the elastic system that supports the stationary or sliding body, combined with a negative slope of friction coefficient, f, versus sliding velocity, U, at low speed. For example, in the linear-motion friction apparatus (Fig. 14-10), the elastic belt of the drive reduces the stiffness of the support of the moving part.

In the stick period, the motion is due to elastic displacement of the support (without any relative sliding). This is followed by a short period of relative sliding (slip). These consecutive periods are continually repeated. At the stick period, the motion requires less tangential force for a small elastic displacement than for breakaway of the stiction force. The elastic force increases linearly with the displacement (like a spring), and there is a transition from stick to slip when the elastic force exceeds the stiction force, and vice versa. The system always selects the stick or slip mode of minimum resistance force.

In the past, the explanation was based on static friction greater than the kinetic friction. It has been realized, however, that the friction is a function of the velocity, and the current explanation is based on the negative $f - U$ slope, see a simulation by Harnoy (1994).

16.2 FRICTION IN HYDRODYNAMIC AND MIXED LUBRICATION

Hydrodynamic lubrication theory was discussed in Chapters 4–9. In journal and sliding bearings, the theory indicates that the lubrication film thickness increases with the sliding speed. Full hydrodynamic lubrication occurs when the sliding velocity is above a minimum critical velocity required to generate a full lubrication film having a thickness greater than the size of the surface asperities. In full hydrodynamic lubrication, there is no direct contact between the sliding surfaces, only viscous friction, which is much lower than direct contact friction. In full fluid film lubrication, the viscous friction increases with the sliding speed, because the shear rates and shear stresses of the fluid increase with that speed.

Below a certain critical sliding velocity, there is mixed lubrication, where the thickness of the lubrication film is less than the size of the surface asperities. Under load, there is a direct contact between the surfaces, resulting in elastic as well as plastic deformation of the asperities. In the mixed lubrication region, the external load is carried partly by the pressure of the hydrodynamic fluid film and partly by the mechanical elastic reaction of the deformed asperities. The film thickness increases with sliding velocity; therefore as the velocity increases, a larger portion of the load is carried by the fluid film. The result is that the friction decreases with velocity in the mixed region, because the fluid viscous friction is lower than the mechanical friction at the contact between the asperities.

The early measurements of friction characteristics have been described by f–U curves of friction coefficient versus sliding velocity by Stribeck (1902) and by McKee and McKee (1929). These f–U curves were measured under steady conditions and are referred to as *Stribeck curves*. Each point of these curves was measured under steady-state conditions of speed and load.

The early experimental f–U curves of lubricated sliding bearings show a nearly constant friction at very low sliding speed (boundary lubrication region). However, for metal bearing materials, our recent experiments in the Bearing and Bearing Lubrication Laboratory at the New Jersey Institute of Technology, as well as experiments by others, indicated a continuous steep downward slope of friction from zero sliding velocity without any distinct friction characteristic for the boundary lubrication region. The recent experiments include friction force measurement by load cell and on-line computer data acquisition. Therefore, better precision is expected than with the early experiments, where each point was measured by a balance scale.

An example of an f–U curve is shown in Fig. 16-1. This curve was produced in our laboratory for a short journal bearing with continuous lubrication. The experiment was performed under "quasi-static" conditions; namely, it was conducted for a sinusoidal sliding velocity at very low frequency, so it is equivalent to steady conditions. The curve demonstrates high friction at zero velocity (stiction, or static friction force), a steep negative friction slope at low velocity (boundary and mixed friction region), and a positive slope at higher velocity (hydrodynamic region). There are a few empirical equations to describe this curve at steady conditions. The negative slope of the f–U curve at low velocity is used in the explanation of several friction phenomena. Under certain conditions, the negative slope can cause instability, in the form of stick-slip friction and friction-induced vibrations (Harnoy 1995, 1996).

In the boundary and mixed lubrication regions, the viscosity and boundary friction additives in the oil significantly affect the friction characteristics. In

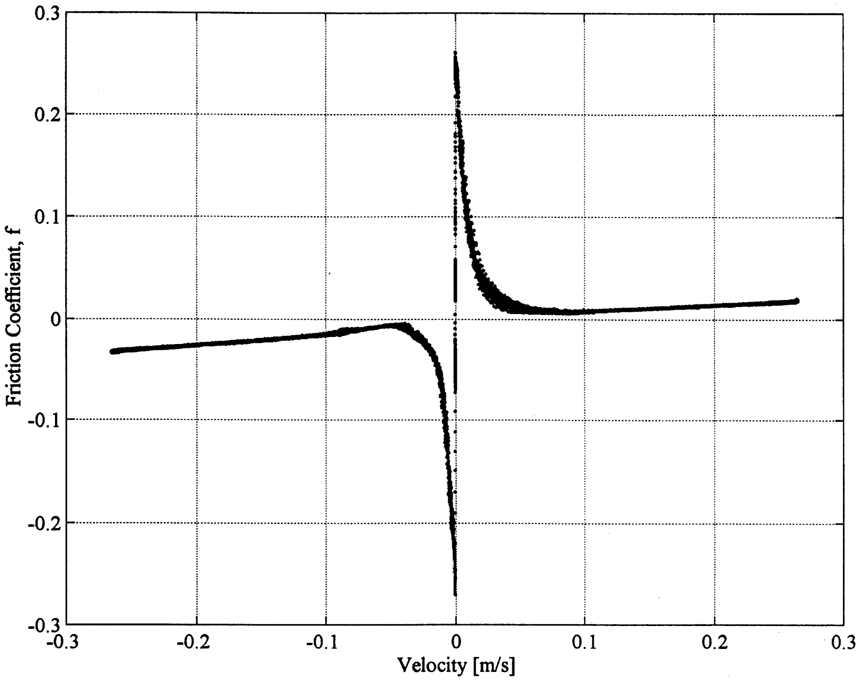

FIG. 16-1 f–U curve for sinusoidal velocity: oscillation frequency = 0.0055 rad/s, load = 104 N, 25-mm journal, $L/D = 0.75$, lubricant SAE 10W-40, steel on brass.

addition, the breakaway and boundary friction coefficients are higher with a reduced bearing load. For example, Fig. 16-2 is f–U curve for a low-viscosity lubricant without any additives for boundary friction reduction and lower bearing load. The curve indicates a higher breakaway friction coefficient than that in Fig. 16-1 lubricated with engine oil SAE 10W-40. The breakaway friction in Fig. 16-2 is about that of dry friction. However, for the two oils, the friction at the transition from mixed to full film lubrication is very low.

The steep negative slope in the mixed region has practical consequences on the accuracy of friction measurements that are widely used to determine the effectiveness of boundary layer lubricants. Currently such lubricants are evaluated by measuring the friction at an arbitrary constant sliding speed (e.g., a four-ball tester operating at constant speed). However, the f–U curve in Fig. 16-1 indicates that this measurement is very sensitive to the test speed. Apparently, a better evaluation should be obtained by testing the complete f–U curve. Similar to the journal bearing, the four-ball tester has a hydrodynamic fluid film; in turn, the

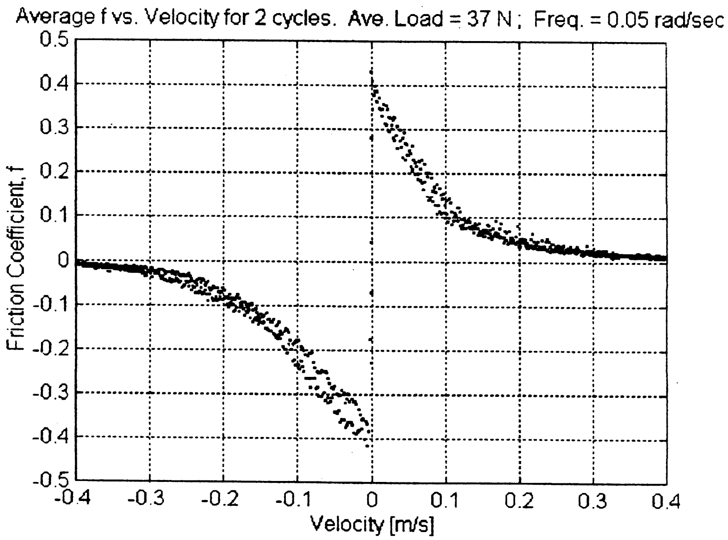

FIG. 16-2 f–U curve for sinusoidal velocity: oscillation frequency = 0.05 rad/s, load = 37 N, 25-mm journal, $L/D = 0.75$, low-viscosity oil, $\mu = 0.001$ N-s/m^2, no additives, steel on brass.

friction torque is a function of the sliding speed (or viscosity). The current testing methods for boundary lubricants should be reevaluated, because they rely on the assumption that there is one boundary-lubrication friction coefficient, independent of sliding speed.

For journal bearings in the hydrodynamic friction region, the friction coefficient f is a function not only of the sliding speed but of the Sommerfeld number. Analytical curves of $(R/C)f$ versus Sommerfeld number are presented in the charts of Raimondi and Boyd; see Fig. 8-3. These charts are for partial journal bearings of various arc angles β. These charts are only for the full hydrodynamic region and do not include the boundary, or mixed, lubrication region. For a journal bearing of given geometry, the ratio C/R is constant. Therefore, empirical charts of friction coefficient f versus the dimensionless ratio $\mu n/P$, are widely used to describe the characteristic of a specific bearing. In the early literature, the notation for viscosity is z, and charts of f versus the variable zN/P were widely used (Hershey number—see Sec. 8.7.1).

16.2.1 Friction in Rolling-Element Bearings

Stribeck measured similar f–U curves (friction coefficient versus rolling speed) for lubricated ball bearings and published these curves for the first time as early as

1902. Rolling-element bearings operating with oil lubrication have a similar curve: an initial negative slope and a subsequent rise of the friction coefficient versus speed (due to increasing viscous friction). Although there is a similarity in the shapes of the curves, the breakaway friction coefficient of rolling bearings is much lower than that of sliding bearings, such as journal bearings. This is obvious because rolling friction is lower than sliding friction. The load and the bearing type affect the friction coefficient. For example, cylindrical and tapered rolling elements have a significantly higher friction coefficient than ball bearings.

16.2.2 Dry Friction Characteristics

Dry friction characteristics are not the same as for lubricated surfaces. The f–U curve for dry friction is not similar to that of lubricated friction, even for the same material combination. For dry surfaces after the breakaway, the friction coefficient can increase or decrease with sliding speed, depending on the material combination. For most metals, the friction coefficient has negative slope after the breakaway. An example is shown in Fig. 16-3 for dry friction of a journal bearing made of a steel shaft on a brass sleeve. This curve indicates a considerably higher friction coefficient at the breakaway from zero velocity (about 0.42 in comparison to 0.26 for a lubricated journal bearing—half of the breakaway friction). In addition, a dry bearing has a significantly greater gradual reduction of friction with velocity (steeper slope).

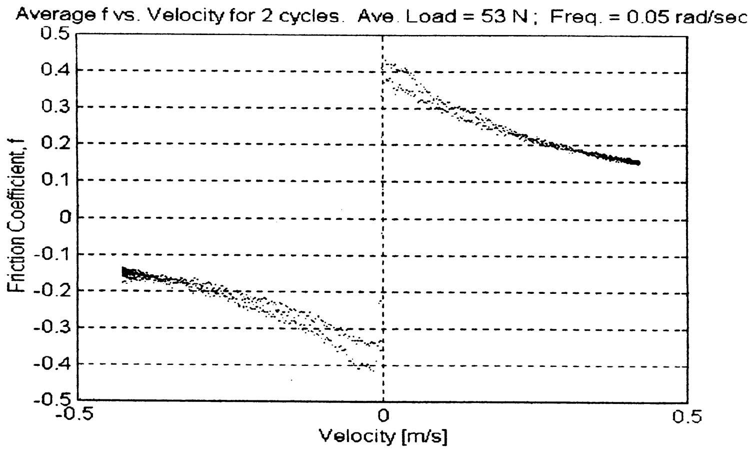

FIG. 16-3 f–U curve for sinusoidal velocity: oscillation frequency $= 0.05$ rad/s, load $= 53$ N, dry surfaces, steel on brass, 25-mm journal, $L/D = 0.75$.

16.2.3 Effects of Surface Roughness on Dry Friction

As already discussed, smooth surfaces are desirable for hydrodynamic and mixed lubrication. However, for dry friction of metals with very smooth surfaces there is adhesion on a larger contact area, in comparison to rougher surfaces. In turn, ultrasmooth surfaces adhere to each other, resulting in a higher dry friction coefficient. For very smooth surfaces, surface roughness below 0.5 μm, the friction coefficient f reduces with increasing roughness. At higher roughness, in the range of about 0.5–10 μm (20–40 microinches), the friction coefficient is nearly constant. At a higher range of roughness, above 10 μm, the friction coefficient f increases with the roughness because there is increasing interaction between the surface asperities (Rabinovitz, 1965).

16.3 FRICTION OF PLASTIC AGAINST METAL

There is a fundamental difference between dry friction of metals (Fig. 16-3) where the friction goes down with velocity, and dry friction of a metal on soft plastics (Fig. 16-4a) where the friction coefficient increases with the sliding velocity. Figure 16-4a is for sinusoidal velocity of a steel shaft on a bearing made of ultrahigh-molecular-weight polyethylene (UHMWPE). This curve indicates that there is a considerable viscous friction that involves in the rubbing of soft plastics. In fact, soft plastics are viscoelastic materials.

In contrast, for lubricated surfaces, the friction reduces with velocity (Fig. 16-4b) due to the formation of a fluid film. In Fig. 16-4b, the dots of higher friction coefficient are for the first cycle where there is an example of relatively higher *stiction* force, after a rest period of contact between the surfaces under load.

16.4 DYNAMIC FRICTION

Most of the early research in tribology was limited to steady friction. The early f–U curves were tested under steady conditions of speed and load. For example, the f–U curves measured by Stribeck (1902) and by McKee and McKee (1929) do not describe "dynamic characteristics" but "steady characteristics", because each point was measured under steady-state conditions of speed and load.

There are many applications involving friction under unsteady conditions, such as in the hip joint during walking. Variable friction under unsteady conditions is referred to as *dynamic friction*. Recently, there has been an increasing interest in dynamic friction measurements.

Dynamic tests, such as oscillating sliding motion, require on-line recording of friction. Experiments with an oscillating sliding plane by Bell and Burdekin

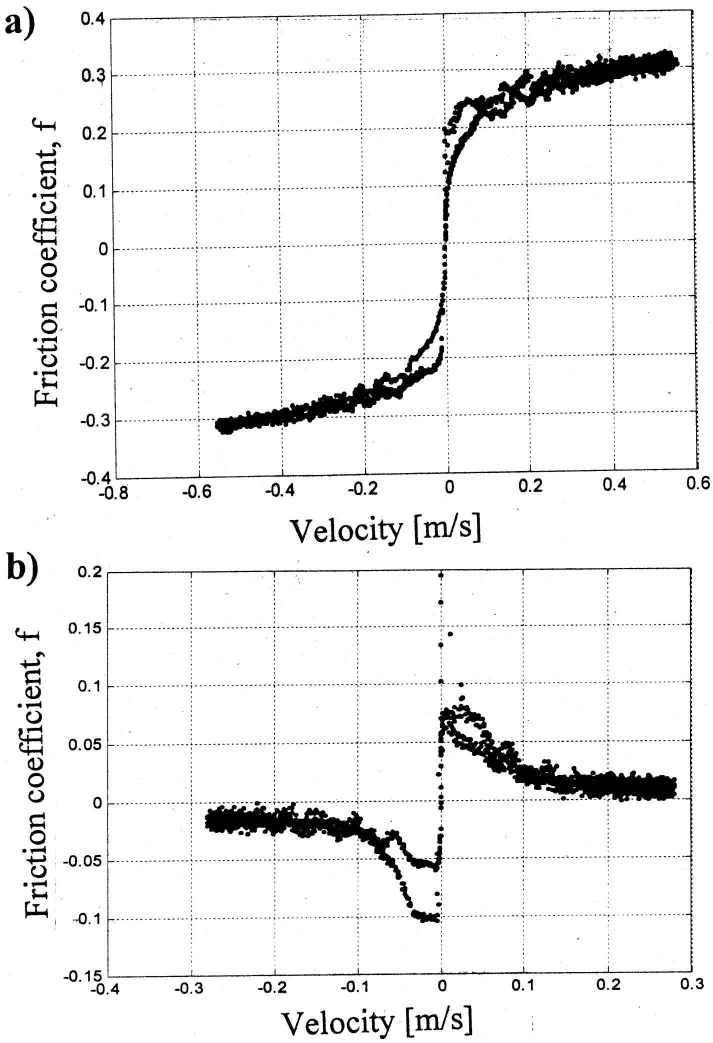

FIG. 16-4 f–U curve for sinusoidal velocity: frequency $= 0.25$ rad/s, load $= 215$ N, ultrahigh-molecular-weight polyethylene (UHMWPE) on steel, journal diameter 25 mm, $L/D = 0.75$ for rigid bearing at low frequency. (a) Dry surfaces. (b) Lubrication with SAE 5W oil.

(1969) and more recent investigations of line contact by Hess and Soom (1990), as well as recent measurements in journal bearings (Harnoy et al. 1994) revealed that the phenomenon of dynamic friction is quite complex. The f–U curves have a considerable amount of hysteresis that cannot be accounted for by any steady-friction model. The amount of hysteresis increases with the frequency of oscillations. At very low frequency, the curves are practically identical to curves produced by measurements under steady conditions.

For sinusoidal velocity, the friction is higher during acceleration than during deceleration, particularly in the mixed friction region. In the recent literature, the hysteresis effect is often referred to as *multivalued friction*, because the friction is higher during acceleration than during deceleration. For example, the friction coefficient is higher during the start-up of a machine than the friction during stopping. This means that the friction is not only a function of the instantaneous sliding velocity, but also a function of velocity history. Examples of $f-U$ curves under dynamic conditions are included in Chap. 17.