Clutches and Brakes

DESIGN AND SELECTION

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Preface

This book has two objectives. The first is to bring together the formulas for the design and selection of a variety of brakes and clutches. The second is to provide flowcharts and programs for programmable calculators and personal computers to facilitate the application of often lengthy formulas and otherwise tedious iteration procedures indigenous to the clutch and brake design and selection process.

Formulas for the torque that may be expected from each of the brake or clutch configurations and the force, pressure or current required to obtain this torque are derived and their application is demonstrated by example. Derivations are included to explicitly show the assumptions made and to delineate the role of each parameter in these governing relations so that the designer can more skillfully select these parameters to meet the demands of the problem at hand. Where appropriate, the resulting formulas are collected at the end of each chapter so that those not interested in their derivation may turn directly to the design and selection formulas.

Following the torque and force analysis for the sundry brake and clutch embodiments which dissipate heat, attention is directed to the calculation of the heat generated by these devices during the interval in which the speed is changing. Pertinent relations are derived and demonstrated for braking or accelerating of vehicles, conveyor belts, and hoists.

Calculation of the acceleration, temperature, and heat dissipation may be quite complicated and may be strongly dependent upon the location of the brake on the machine itself and upon the environment in which the machine is to operate. Discussion of acceleration, acceleration time, temperature, and heat dissipation are, therefore, limited to a common—and simple—brake configuration and to a standard environment of 20°C or 70°F, no wind, and no vibration.

Flowcharts follow the formula collection as appropriate to demonstrate their step-by-step application in arriving at the final design. They are written in the interactive mode (computer or calculator prompting for each variable and its increments) to permit use of programmable calculators and small personal computers for the comparison of several possible designs. With little modification they may be used as subprograms in larger computers having control programs to automate clutch and brake selection to whatever extent desired.

Even though the calculations may be lengthy, no flowcharts are given for those cases where branching is minimal (as in the case of acceleration or deceleration and heat dissipation calculations) where the reasoning is straightforward. It is intended that computer programs will be used for all but the simplest calculations.

William C. Orthwein

Introduction

It is the purpose of this book to briefly derive, where possible, the design formulas for the major types of clutches and brakes listed in the contents and to display an example of their use in a typical design. Some pertinent computer programs for longer formulas are listed in the references.

Each chapter is independent of the others, with the possible exception of Chapters 1 and 8, which are concerned with friction materials and with acceleration or deceleration time and heat dissipation during clutching and braking. The friction and pressure characteristic of friction materials used for brake and clutch linings and pads are discussed in Chapter 1 so that they may be available for applications in the following chapters. Chapter 8 deals with acceleration and heat dissipation considerations which apply to all chapters, and consequently draws upon the other chapters for brake types to be discussed in its examples. The logic to be delineated in that chapter is, however, contained entirely within that chapter, so that it may be read and understood without prior reading of any of the other chapters.

Since both SI and Old English units are used throughout the book, it may prove useful to have conversion values available.

To Convert	To	Multiply by
pounds/in^2 (psi)	megapascals (MPa)	0.00689476
megapascals (MPa)	pounds/in^2 (psi)	145.03774
horsepower (hp)	kilowatts (kW)	0.7457
kilowatts (kW)	horsepower (hp)	1.34102

To Convert	To	Multiply by
pounds (lb, force)	Newtons (N)	4.4482
Newtons (N)	pounds (lb, force)	0.2248
Btu	calorie	251.995
calorie	Btu	0.003968

Since force and mass are misused in both systems it is necessary to use the acceleration of gravity to convert to proper units when confronted with incorrect usage, e.g., kg/cm^2. The acceleration of gravity in the two system of units is commonly taken to be

$$g = 32.1736 \text{ ft/s} \qquad \text{Old English}$$
$$= 9.80665 \text{ m/s} \qquad \text{SI}$$

As implied by these previous numbers, we shall retain three or four places of significant digits in most calculations to minimize computational error. After all calculations are complete we shall round to the number of places that are practical for manufacture.

For those not familiar with SI stress and bearing pressure calculations, it may be well to point out that the Pascal is a rather awkward unit of stress, since

$$1 \text{ Pascal} = 1 \text{ N/m}^2$$

is an extremely small number in many applications. Two alternatives may be selected: to present pressure and stress in terms of atmospheres (atmospheric pressure at sea level) or in terms of megapascals, denoted by MPa. In the remainder of the book stress and bearing pressure in the SI system will be presented in terms of MPa because of the convenient relations

$$\text{N/mm}^2 = \text{MPa} \quad \text{and} \quad \text{MPa(mm}^2) = \text{N}$$

Since atmospheric pressure at sea level is often taken to be about 14.7 psi, it follows from the listing above that 1 MPa is approximately 10 atmospheric pressures. Conversion from MPa to atmosphere is, therefore, quite simple.

Contents

1

Friction Materials

1.1 FRICTION COEFFICIENTS FOR DRY BRAKES

The usual range of the dynamic friction coefficients for those friction materials normally used in dry brake linings and pads is given in the Society of Automotive Engineers (SAE) coding standard SAE J866a, which lists the code letters and friction coefficient ranges shown in Table 1.1 [1]. According to this code, the first letter in the lining edge code indicates the normal friction coefficient and the second letter indicates the hot friction coefficient. Thus a lining material whose normal friction coefficient is 0.29 and whose hot friction coefficient is 0.40 would be coded as follows:

```
AB - 12 - EF
|    |    |______friction code
|    |___________compound code
|________________code designation for the manufacturer
```

Temperatures for normal and hot friction coefficients are defined in SAE J661a, which also describes the measurement method to be used.

Static and dynamic coefficients of friction are usually different for most brake materials. If a brake is used to prevent shaft rotation during a particular operating phase, with its stopping torque and heat dissipation of secondary importance (i.e., a holding brake on a press), the static friction coefficient is the design parameter used. If, on the other hand, the brake is to be designed for its stopping torque and heat dissipation, the design parameters used are the dynamic friction coefficient and its change with temperature.

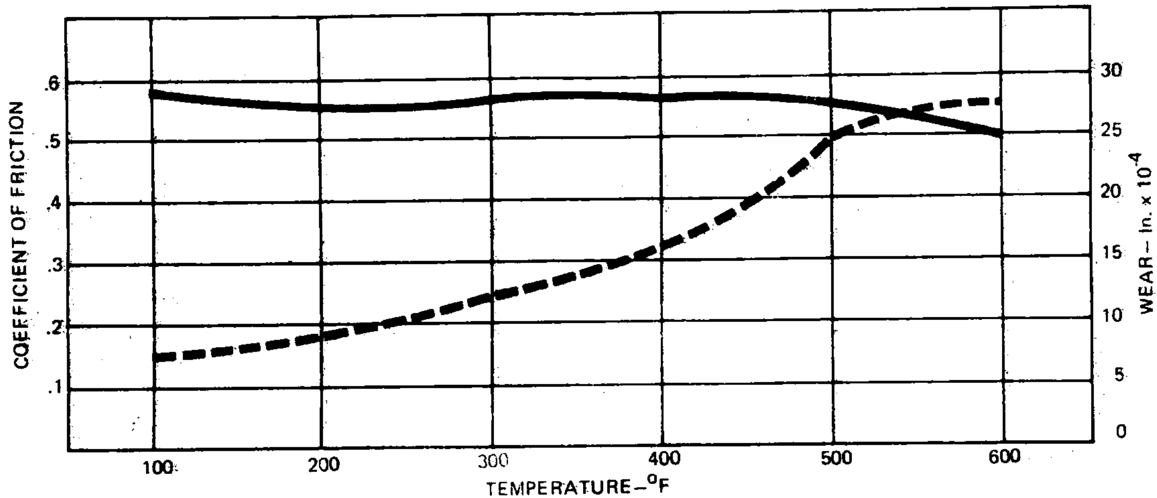
TABLE 1.1 Friction Identification System for Brake Linings and Brake Block for Motor Vehicles

Code Letter	Friction coefficient
C	Not over 0.15
D	Over 0.15 but not over 0.25
E	Over 0.25 but not over 0.35
F	Over 0.35 but not over 0.45
G	Over 0.45 but not over 0.55
H	Over 0.55
Z	Unclassified

Typical coefficients of friction between lining materials and smooth cast iron or smooth steel and their temperature dependence are displayed in Figures 1.1 through 1.5 [2]. The solid lines in these figures represent coefficients of friction, shown on the left-hand ordinate, and the dashed lines represent the wear, shown on the right-hand ordinate. Figure 1.1 is for a rigid-molded high-friction material which shows less than a 10% change in the measured dynamic friction coefficient between 100 and 600°F, but with wear increasing with increasing temperature. The brake lining shown in Figure 1.2, designed for industrial brakes and off-the-road equipment, is made from different proprietary materials which provide good face resistance but at a lower friction coefficient, to obtain better wear characteristics under 400°F. This trend for improved wear at the expense of a reduced coefficient of friction is demonstrated again in Figure 1.3 for a rigid-molded asbestos material for industrial brake and clutch applications. Note that for all three materials face resistance is obtained at the expense of greater wear at higher temperatures. This trade-off between wear and fade resistance (a large friction coefficient at high temperature) is also demonstrated in Figure 1.4, where greatly improved wear resistance may be had from this woven material designed for plate brakes and automotive clutches, but at the expense of poor fade resistance above about 400°F.

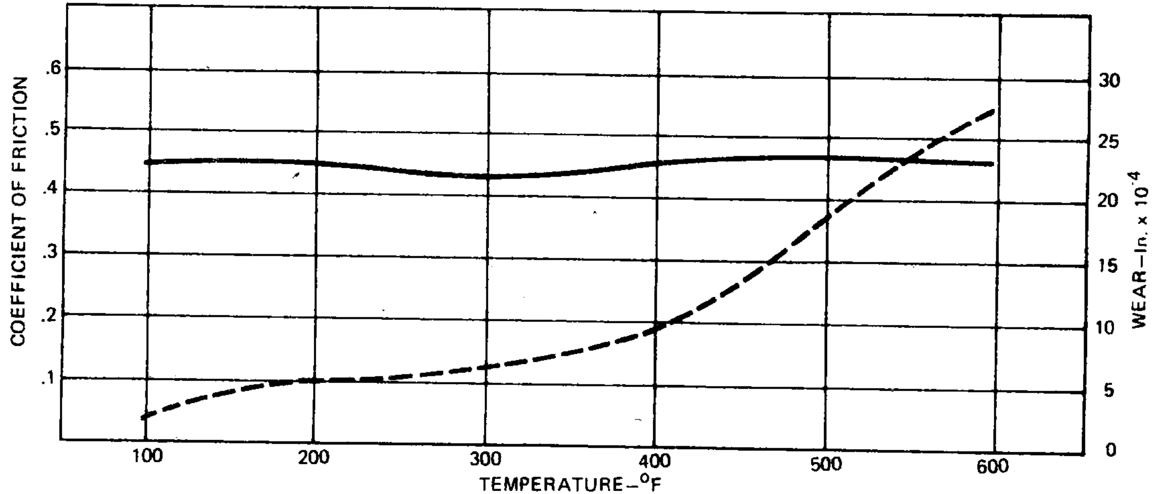
As shown in Figure 1.5, friction material may be desirable for slip clutches, a form of brake, used for tension control. In such applications the low friction coefficient offers some protection against the brake actually stopping the motion if the actuating force is increased inadvertently.

The friction data displayed above were obtained according to SAE J661a, which calls for a 100-lb force to press a 1 in. X 1 in. sample against the interior surface of a cast iron drum having an 11-in. inside diameter while the drum rotates at 417 rpm. With the drum temperature held constant the normal force is applied for 20 cycles, each cycle consisting of a 10-sec application of the force and a 20-sec rest when the lining is not in contact with the drum. The average friction coefficient



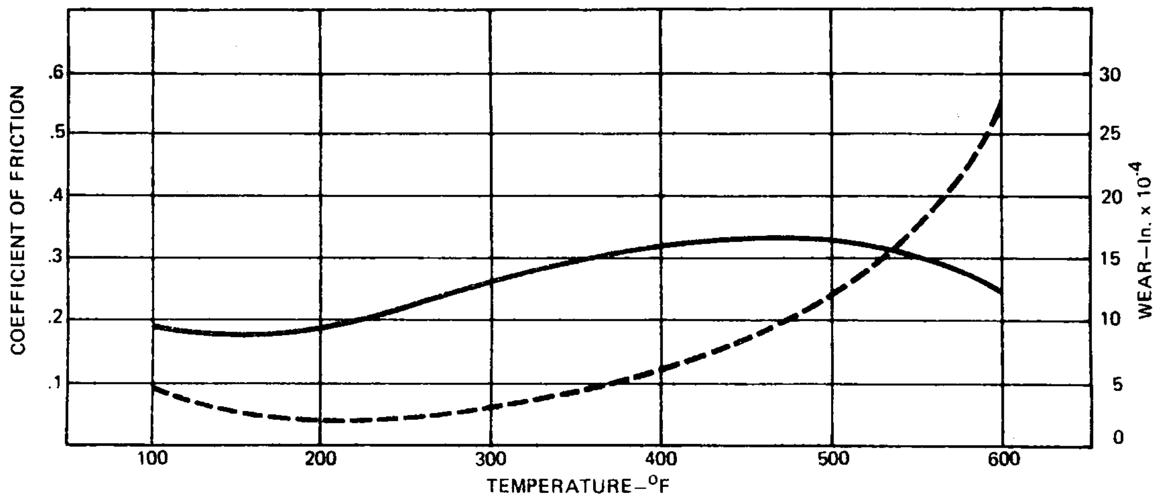
Average Friction	Average Wear	Overall Wear
100°-600°: .464 100°-300°: .480 400°-600°: .447	100°-600°: .00116 in. 100°-300°: .00048 in. 400°-600°: .00183 in.	.00697 in.

FIGURE 1.1 Rigid-molded high-friction material (static friction coefficient is 0.30 to 0.35). (Courtesy of Thermoid Division, H. K. Porter, Co., Pittsburgh, Pa.)



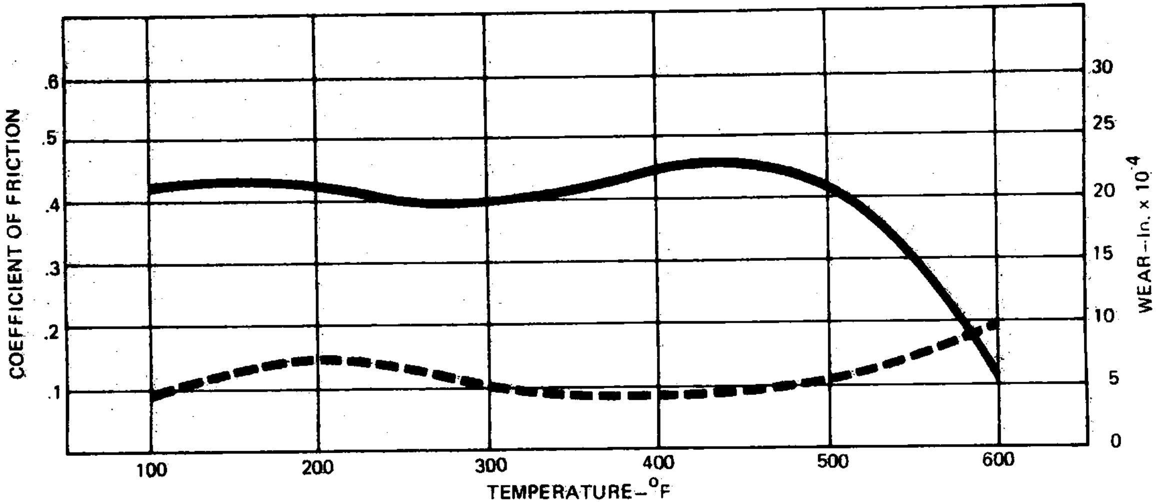
Average Friction	Average Wear	Overall Wear
100°-600°: .449 100°-300°: .439 400°-600°: .460	100°-600°: .00112 in. 100°-300°: .00043 in. 400°-600°: .00182 in.	.00674 in.

FIGURE 1.2 High-friction material for industrial brakes and off-road equipment (static friction coefficient is 0.31 to 0.36). (Courtesy of Thermoid Division, H. K. Porter Co., Pittsburgh, Pa.)



Average Friction	Average Wear	Overall Wear
100°-600°: .259 100°-300°: .213 400°-600°: .306	100°-600°: .00092 in. 100°-300°: .00031 in. 400°-600°: .00153 in.	.00554 in.

FIGURE 1.3 Rigid-molded asbestos material for industrial brakes (static friction coefficient is 0.27 to 0.31).



Average Friction	Average Wear	Overall Wear
100°-600°: .371 100°-300°: .416 400°-600°: .326	100°-600°: .06116 in. 100°-300°: .05700 in. 400°-600°: .06533 in.	.00367 in.

FIGURE 1.4 Woven material of asbestos yarn with brass wire inserts for face, or multiple-disk brakes and clutches (static friction coefficient is 0.30 to 0.34). (Courtesy of Thermoid Division, H. K. Porter, Co., Pittsburgh, Pa.)