FEARS STRUCTURAL ENGINEERING LABORATORY

Final Report

EFFECTIVE COEFFICIENT OF FRICTION OF BRIDGE BEARINGS

by

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ABSTRACT

This study was done to determine experimentally the effective coefficient of friction of four classes of steel bridge bearings used by the Oklahoma Department of Transportation and to investigate the suitability of various tetrafluoroethylene (TFE) expansion bearings for bridges. As-built, rusted and in-situ (debris at the moving surfaces) conditions were tested using full-scale bearings under normal loads to 250,000 lb. In addition, the effect of manufacturing tolerances on bearing performance were analyzed. A total of 229 steel bridge bearing tests and 228 TFE expansion bearing tests were conducted. From the tests it was found that pipe rollers exhibit the lowest effective coefficient of friction of the four rolling devices tested.

EXECUTIVE SUMMARY

Expansion and contraction caused by temperature changes, deflection, relative support settlement, creep, etc., will produce longitudinal motion in a bridge. If this motion is constrained, the resulting forces may be very large. Moveable bearings at piers or abutements are commonly used to control the magnitude of these forces. The only horizontal force transmitted to the bridge substructure is then through friction caused by relative motion of the bearing parts or eccentric loading of the bearing as found in "turned pipe" bearings. This force must be accommodated in the structural design of the substructure or damage can occur.

The purpose of this study was to determine experimentally the effective coefficient of friction of four classes of steel bridge bearings used by the Oklahoma Department of Transportation and to investigate the suitability of various tetrafluoroethylene (TFE) expansion bearings for bridges. As-built, rusted and in-situ (debris at the moving surfaces) conditions were tested using full-scale bearings under normal loads to 250,000 lb. In addition, the effect of manufacturing tolerances on bearing performance were analyzed.

For the purpose of this study the effective coefficient of friction of unturned pipe-roller, pinned rocker shoe, pintle rocker shoe and TFE bearings, the effective coefficient of friction, μ_{eff}, is defined as

$$\mu_{eff} = \frac{F}{N}$$

where F = horizontal force to overcome the resistance to allow motion, and N = normal force applied to the bearing. For turned pipe roller bearings, eccentric loading requires a horizontal force to maintain equilibrium. For comparison purposes an equivalent coefficient of friction, μ_{equiv}, was defined for turned pipe-roller bearings using the above equation.

A total of 229 steel pipe bearing tests were conducted: 38 unturned pipe rollers, 21 turned pipe rollers, 51 pinned rocker shoes, and 121 pintle rockers. In addition, 228 TFE expansion bearing tests were made using combinations of three types of TFE elements (unfilled, 25% glass filled, and woven

unfilled and glass filled fibers), two steel surfaces (stainless steel and mirror finish stainless) and two backings (carbon steel plate and 70 Durometer neoprene). A total of seven elements types in eight combinations were tested in both parallel and nonparallel (1/32 in. per ft.) conditions.

From the tests it was found that pipe rollers exhibit the lowest effective coefficient of friction of the four rolling devices tested. The effective coefficient of friction was found to be less than 0.5% for a clean 10 in. diameter pipe roller. The value increased to about 1% when tested in a rusted condition and to 5% when sand was spread over the lower bearing plate.

For turned pipe rollers it was found that the equivalent coefficient of friction is a function of the amount of horizontal movement from the center line (median line). A geometric explanation was devised and excellent agreement between predicted and measured results was achieved.

An effective coefficient of friction of 1% was found from tests using a clean pinned rocker. The value increased to 2% for a rusted condition. Both values are lower than a predicted value of 2.5% using a published criterion. The effective coefficient of friction for this rocker increased to 9% when sand was placed on the lower bearing plate.

Tests using a pintle rocker showed that fabrication inaccuracies, especially in the sole plate socket radius, can significantly affect the performance and effective coefficient of friction of the bearing. Tests with a socket plate socket radius slightly smaller than the rocker radius resulted in effective coefficient of friction values from 6.15% to 9.88%, as compared to 2.4% from published criteria. Tests with rusted bearing plates or with sand spread over the lower bearing plate showed significant increases in the effective coefficient of friction.

From test results for various TFE expansion bearings, the effective coefficient of friction was found to be higher and less consistent when both elements were TFE as opposed to one element being mirror finish stainless steel. The highest values of effective coefficient of friction were obtained for glass filled TFE versus stainless steel and the lowest for unfilled TFE versus mirror finish stainless steel. Tests using a nonparallel condition showed that the effective coefficient of friction increases about 50% for only 1/32" per foot (0.15^{o}) slope.

The effective coefficient of friction in TFE bearings was found to decrease with increasing contact pressure except for unfilled TFE versus mirror finish stainless steel. Unfilled TFE versus mirror finish stainless steel was found to have the lowest effective coefficient of friction of any combination, also, the results were the most consistent between the tests.

EFFECTIVE COEFFICIENT OF FRICTION OF BRIDGE BEARINGS

CHAPTER I

INTRODUCTION

1.1 General

Expansion and contraction caused by temperature changes, deflection, relative support settlement, creep, among others will produce motion in a bridge. The movement is very slow, but the forces involved can be tremendous and usually are accommodated by bearings at piers or abutements. If the bridge does not have the ability to move, by either not having a bearing or having a non-working one, it pushes and tears at its supports until it achieves the ability to move.

Even if the bearing is working properly, horizontal force is transmitted to the pier or abutement through friction caused by relative motion of the bearing parts or by eccentric loading of the bearing as found in certain "pipe" bearings. This force must be accommodated in the design of the supporting structure and, if not, structural damage can occur.

The purpose of this study is to determine experimentally the effective coefficient of friction of several classes of bridge bearings used by the Oklahoma Department of Transportation (ODOT). Both as-built conditions and simulated conditions, as found after several years of use, were used in the testing program. A thorough literature search revealed that very few studies of the behavior of complete bearing assemblies have been conducted and that specification provisions have been based on classic values of coefficients of friction between sliding parts without regard to effects of manufacturing tolerances or environmental effects. This study is an attempt to assess these effects and to provide guidelines to establish accurate estimates of horizontal force requirements for the class of bearings tested.

For the purpose of this study the effective coefficient of friction, μ_{eff}, is defined as:

$$\mu_{eff} = \frac{F}{N} \tag{1.1}$$

where

F = horizontal force to overcome the resistance to allow motion, and

N = normal force applied to the bearing.

The value of F was determined experimentally for the entire assembly for an applied normal force N, from which μ_{eff} is calculated.

1.2 Common Types of Bridge Bearings

Many types of bearing devices are used to accommodate bridge movement: single rollers, groups of rollers, rockers, elastomeric pads, sliding plates, sliding tetrafluoroethylene (TFE), etc. These devices often fail to function over time and some bridges are designed so that the entire structure will take up bridge movement without using bearings. This is done by designing flexible piers, accommodating for radial expansion on a curve, and other means. Only mechanical expansion devices are considered here.

In general, bridge bearings may be classed in two categories: "elastomeric" and "mechanical"(1). According to a recent synthesis on the design, fabrication, construction, and maintenance of bridge bearings published by the Transportation Research Board (TRB)(2), the elastomeric bearing pad is perhaps the best expansion bearing because it is unaffected by weather (no moving part to freeze, etc.), nothing to corrode, low cost and almost no maintenance required. However, in general, they are limited to 700 psi for vertical load capacity, 3 inches for horizontal movement and their success depends on the quality of the material. On the other hand, for mechanical bearings the movements and rotations are accommodated by rolling, rocking or sliding actions usually on metal parts which can accommodate much larger bearing pressures. Furthermore, mechanical bearing devices can be designed for virtually unlimited horizontal motion(2).

One of the simplest types of mechanical bearing is the roller or "pipe roller", simply a piece of steel pipe with a stiffener as shown in Figure 1.1(a). The load carrying capacity of the roller is a function of its radius and can be found from one of the following formulas[3]:

For diameters up to 25 inches

$$P = \frac{F_y - 13{,}000}{20{,}000} 600\ d \qquad (1.2)$$

and for diameters from 25 to 125 inches

$$P = \frac{F_y - 13{,}000}{20{,}000} 3{,}000\ \sqrt{d} \qquad (1.3)$$

where:

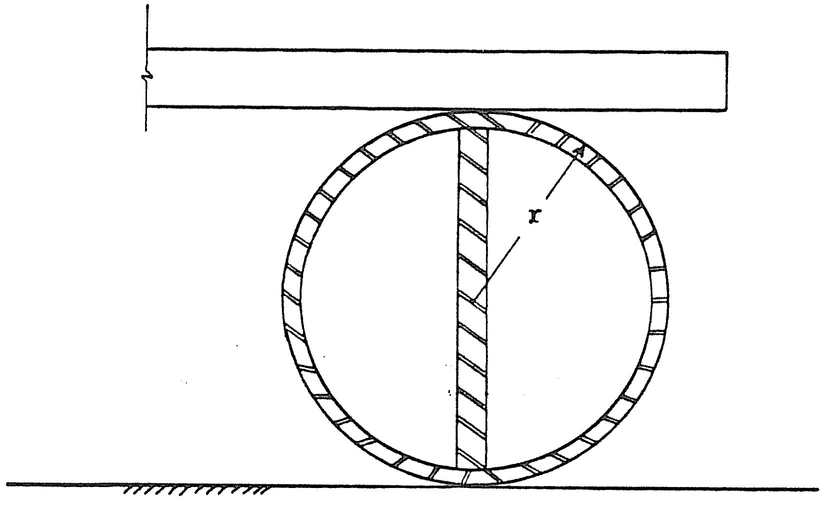
P = allowable bearing in pounds per linear inch

d = outside diameter of the roller in inches

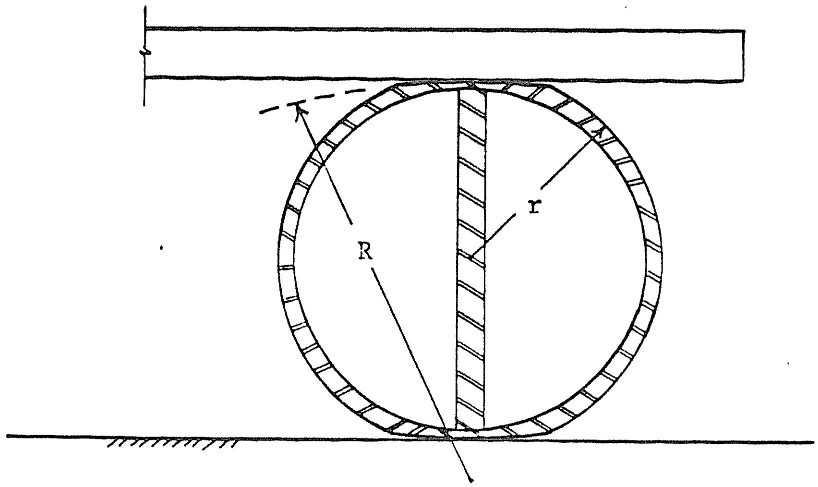
F_y = minimum yield point in tension of steel in the roller or bearing plate, whichever is the smaller in pounds per square inch.

Assuming a practical maximum diameter of 12 inches for use in small river crossings or grade separations, a yield stress of 36,000 psi, and a length of 12 inches, the capacity of a single roller is slightly less than 100,000 pounds. The principal advantage of this type of roller is the low effective coefficient of friction, in general, less than 0.01[4].

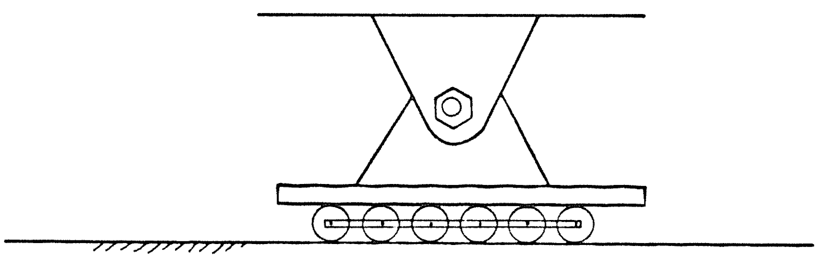
To increase load carrying capacity without increasing the diameter, a single roller can be machined (turned) to increase the radius at the contact surface as shown in Figure 1.1(b). This type of roller, which in this study is



(a) Pipe Roller (Single roller)



(b) Turned Pipe Roller



(c) Roller-nests

Figure 1.1 Roller Expansion Bearing

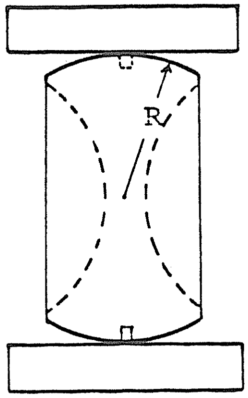
called a "turned roller", has geometrical properties which cause a high effective coefficient of friction. The effective coefficient of friction of a turned-roller is a function of the amount of movement, and is discussed in detail in Chapter III.

Rollers can be used in combination to increase load carrying capacity, as shown in Figure 1.1(c). Roller nests only work well when they are clean, which causes maintenance problems. Furthermore, this type of bearing is relatively expensive.

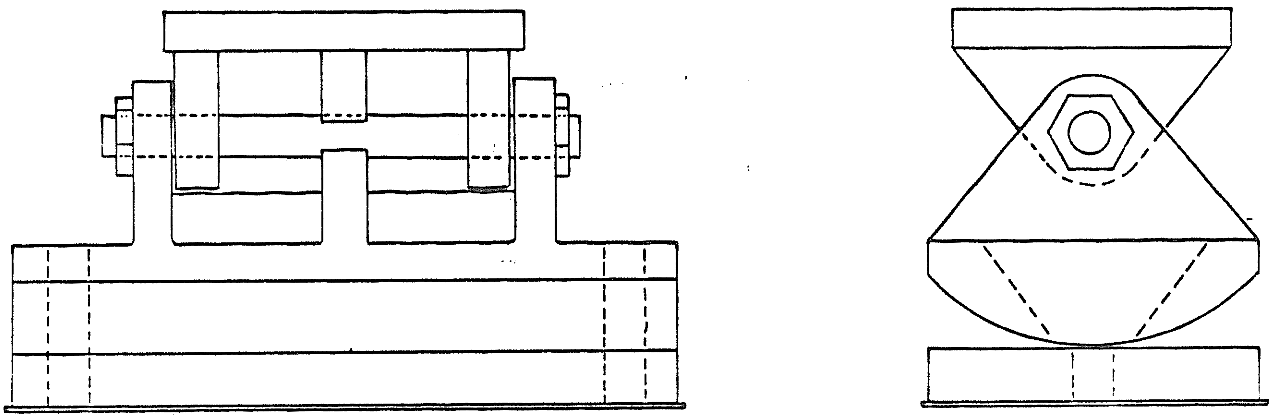
Large rollers take a lot of space and are difficult to handle. However, only a small portion of the circumference is used depending on the amount of movement. Thus, the unused portion of the roller may be cut away such that the bearing functions do not change because the radius is unchanged and space is saved. According to the TRB study[2]:

> When segmented rockers came into use, they did not have to be trimmed from a cylinder; they could be made so that the radii of the two faces would be greater than half the depth of the rocker. This should not be carried to an extreme, however, because geometry causes the bridge to rise slightly at each end of the movement range. Normally, this is not objectionable, the fact that the resisting force increases on either side of the median line may help to keep the rocker in position.

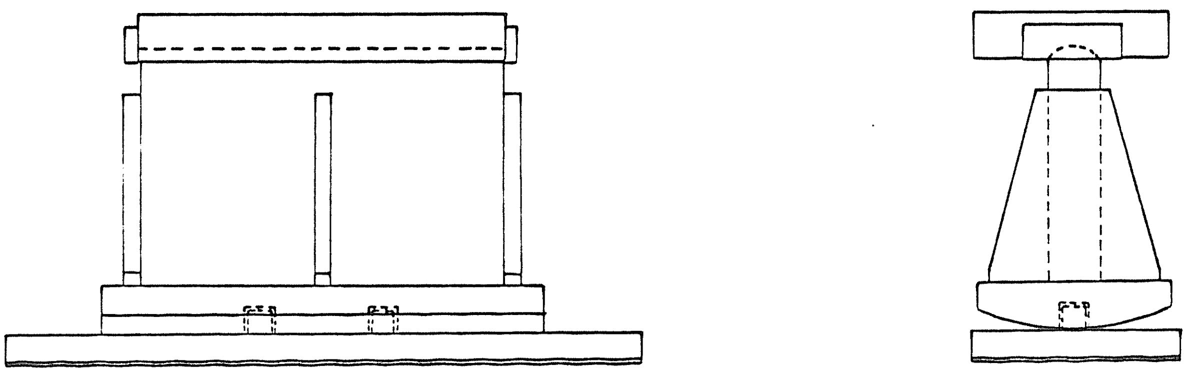
Several different types of rockers are used as expansion bearings, for instance, the segmental rocker shown in Figure 1.2(a), the pinned rocker in Figure 1.2(b) and the pintle rocker shown in Figure 1.2(c).



(a) Typical Segmented Rocker Shoe



(b) Typical Pinned Rocker Shoe



(c) Typical Pintle Rocker Shoe

Figure 1.2 Rocker Expansion Shoes

The double-segmented rocker shown in Figure 1.3, has been described by TRB[2] as a "modern rocker bearing for long steel girders". Since the radius of this rocker is greater than half of the depth, the resisting force (friction force) would be tremendous for large movements which will be discussed in detail in Chapter III.

Tetrafluoroethylene (TFE), has the lowest coefficient of friction of any solid material and has been succesfully used as sliding bearing material for bridges. The coefficient of friction is a function of load per unit area, temperature, the amount of glass fiber filler, and other factors. The sliding surfaces can consist of TFE on TFE or one surface of TFE in contact with a surface of steel, usually polished stainless steel. TFE in contact with highly polished, mirror finish stainless steel has a lower coefficient of friction than other combination. To equalize pressure at the contact surface, at least one side of the bearing should be neoprene backed.

1.3 Previous Research

As mentioned, very few experimental studies of full-scale bridge bearings were found in the literature. Specification requirements seem to have been developed from classic values of friction coefficients and from experience. Jacobson[5] has concluded that certain pin-connected details can accumulate rust between the contact surfaces of the pin and the housing which can result in major structural damage to the main supporting members of a bridge. Laboratory tests of models similar to these bearings showed that the life of the bearing can be improved by using a case hardened pin and by lubricating the bearing with a heavy duty grease. He concluded that the use of the pin-connected details subjected to large rotations and utilizing untreated, corrosive mild steels should be avoided and consideration should be given to the use of elastomeric, TFE-elastomeric and elastomeric pot-type bearings.

The TRB synthesis suggests that the following bearing types should be avoided(2):

- Roller nests: These are impossible to maintain under normal circumstances. Dirt and corrosion inevitably cause failure.

- Steel radius plates with lead sheets between: These are impossible to maintain or keep clean; the lead works out and the bearing tends to freeze and lock up.

- Bolster shoes pinned through a girder web: The pin almost always freezes and locks the joint.

- Wheel-type bearings running on smaller axles: The axles always seem to freeze and lock the joint.

- Cast-steel bearings: Generally too expensive compared with weldments.

The synthesis(2) recommended the following:

- A bridge should be designed with as few movable bearings as possible. Where allowable, movements should be absorbed within the structure.

- Bridge bearings are working, active mechanisms and should be designed and maintained as such.

- Bearings should be designed to require a minimum of maintenance.

- Bearings do fail; wherever possible, provisions should be made so that the bridge may be jacked up and the bearings adjusted or replaced.

- Material quality is of the utmost importance in elastomeric bearings. Quality must be carefully specified. In addition, an adequate inspection and testing program should be in operation.

- Inspection of bridge bearings should be an important part of a regular bridge inspection program.

- Roller and rockers are relatively trouble-free devices when properly maintained. Rollers should never be less than 4 in. in diameter and preferably should be larger.

Chang and Cohen[6] in "Long-Span Bridges: State-of-the-Art" have the following recommendations to replace the last paragraph of Article 1.2.13 of the AASHTO specification[3]:

> The longitudinal force due to friction at expansion bearings or shear resistance at elastomeric bearings shall also be provided for in the design as follows. For sliding type bearings, this force shall be based on the following percentages of the dead load supported:

Bearing type	Average static friction coefficient
Steel bearing on steel	0.2
Steel bearing on self-lubricating bronze plate	0.1
Polytetrafluorethylene (PTFE) on polytetrafluoroethylene or stainless steel	0.06

> For rocker type bearings, this force shall be based on a 20% friction coefficient using the pin, and shall be reduced in proportion to the radii of the pin and the rocker, as shown in Figure 1.4.

The British Standard BS153 [7] specifies the coefficients of friction for sliding bearing at 0.25 for steel

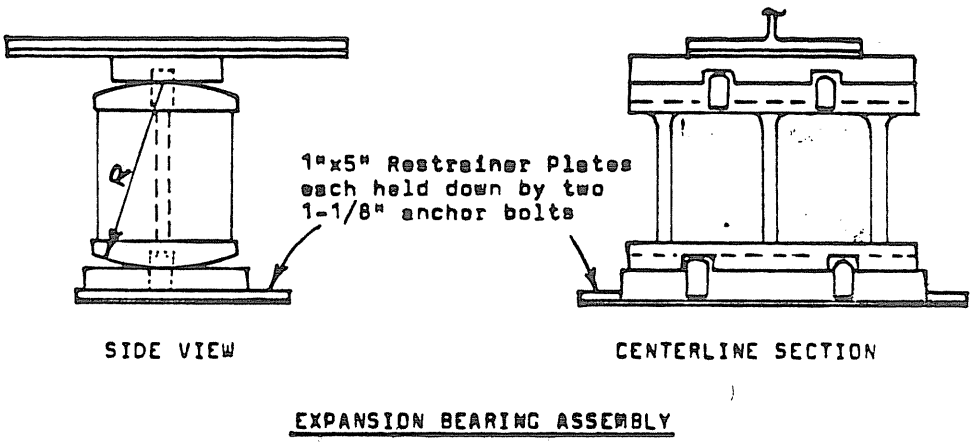
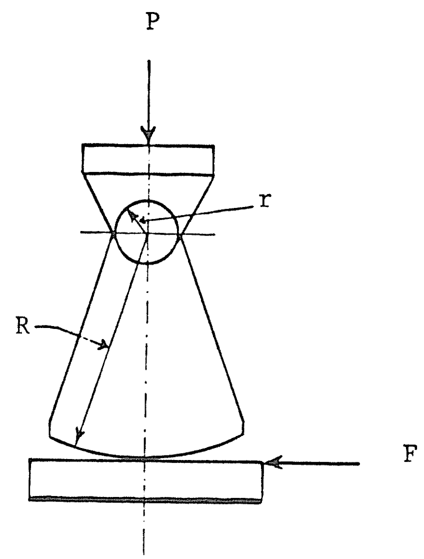


Figure 1.3 Double-Segmental Rocker
(From Bridge Bearing, TRB[2])



P = Superstructure D.L.

$F = (P \times 0.20)\,\frac{r}{R}$

r = radius of pin

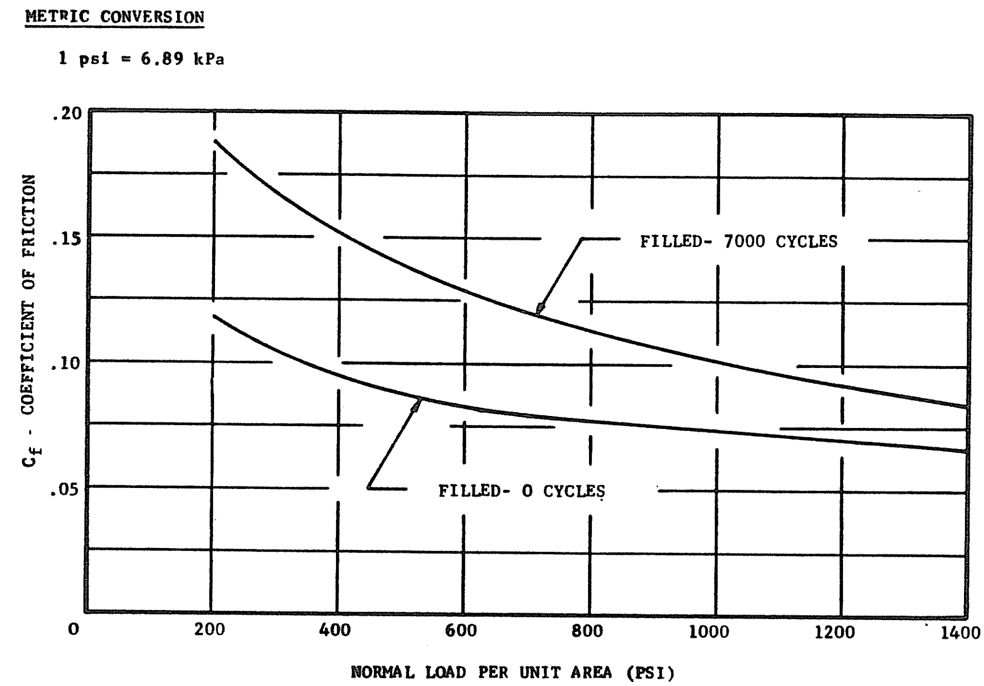
R = radius of rocker

Figure 1.4 Forces on Rocker Bearings

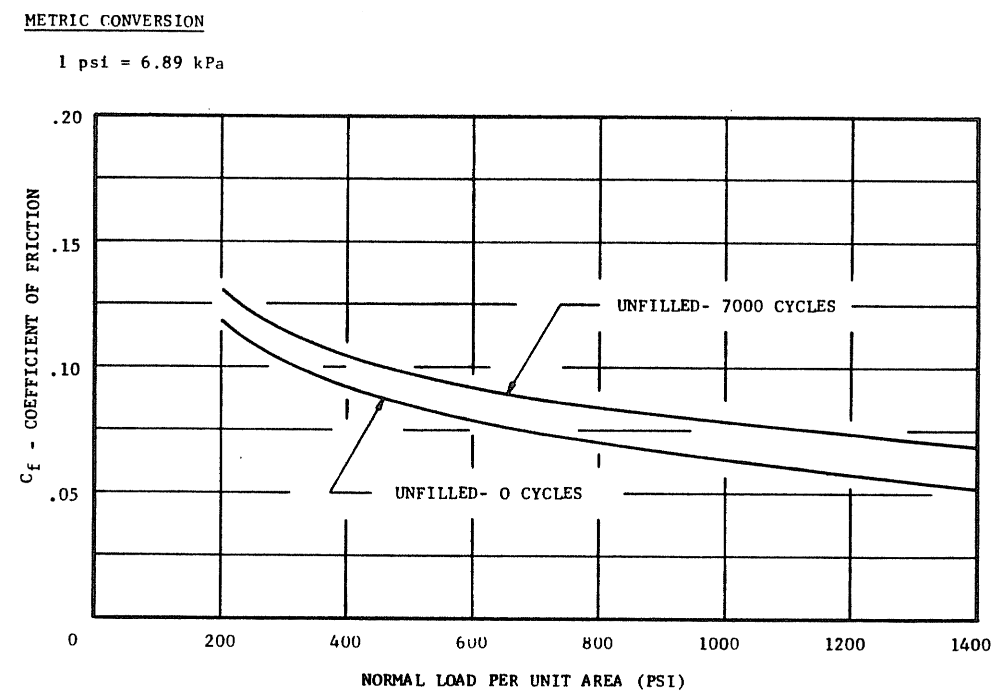
on steel or cast iron, and 0.15, for steel on copper alloy. However, friction values due to corrosion and wear are probably nearer to .50 for steel on steel and .33 for steel on copper alloy. The coefficient of friction with one or two rollers can be taken as 0.01, and due to corrosion and setting tolerances the values of 0.03 may be nearer to actual service condition.

Jacobson[8] has conducted experimental work to investigate the potential use of TFE as a sliding surface. He concludes that the TFE bearings are suitable for use in highway bridges and recommends that only unfilled TFE be used for bridge bearings. A substantial increase in the coefficient of friction for filled TFE was found after 7000 cycles of testing as shown in Figure 1.5. The use of 15 to 25% glass filler resulted in a 35 to 50% increase in the values for the coefficient of friction under applied normal loads between 200 and 800 psi. He also tested several fabric-backed specimens with filled TFE surfaces; they failed by delamination of the fabric pad. He concludes that the fabric backing materials are suitable only when used in conjunction with unfilled TFE.

Taylor[9] has found that the coefficient of friction of Polymerized Tetra-Fluoroethylene (PTFE) is influenced by a number of parameters, including pressure across sliding surfaces, rate of movement, whether lubricated or not, previous loading/movement history and temperature. Most of the



(a) Filled TFE



(b) Unfilled TFE

Figure 1.5 Coefficient of Friction vs Vertical Pressure at 0 and 7000 Cycles (From TFE Expansion Bearing For Highway Bridges, Jacobson (See Ref. 8)) tests were made on unlubricated and unfilled PTFE. The maximum value of the coefficient of friciton of all unlubricated bearings occurred during the first cycle of movement as shown in Figure 1.6. The coefficient of friction decreased with higher compressive stress across the bearing, but increased slightly at the lower temperature as shown in Figure 1.7. He discusses the theory of the real area of intimate contact between the PTFE and slider, and the shear force required to break the junctions in these areas.

1.4 Scope of Research

Since little published data is available on the effective coefficient of friction of standard bridge bearings, a testing program was undertaken to investigate the performance of types of standard ODOT bearings under several conditions. Mechanical bearings types were as follows:

- -Typical single roller bearing (Figure 1.1(a)).
- -Typical single turned-roller bearing (Figure 1.1(b)).
- -Typical pinned rocker shoe (Figure 1.2(b)).
- -Typical pintle rocker bearing (Figure 1.2(c)).

To determine the effect of environmental changes on the frictional coefficients, the following conditions were studied: 1) unlubricated (as-built condition), 2) rusted and 3) with sand.

In addition, tests of typical TFE bearings were conducted to determine the effects of varying amounts of glass

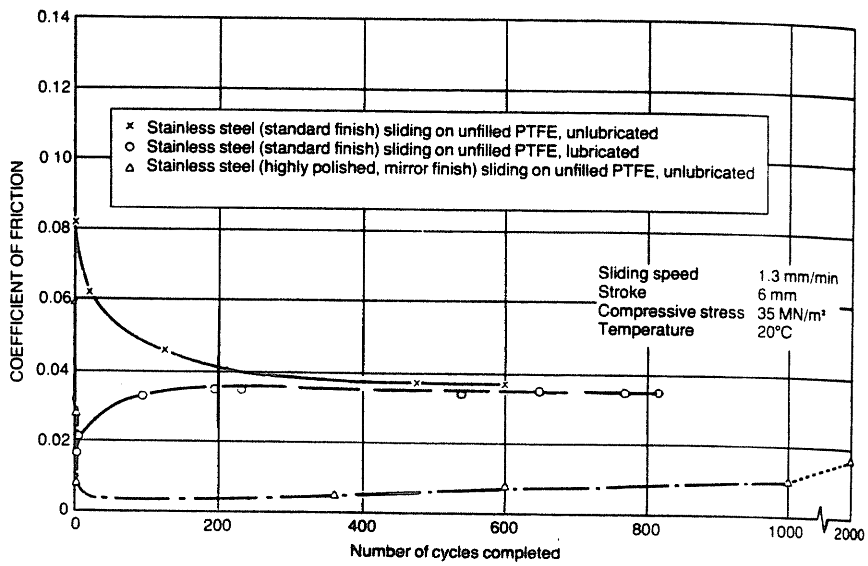


Figure 1.6 Coefficient of Friction/No. of Cycles-Unfilled PTFE (From PTFE in Highway Bridge Bearings, Taylor (See Ref. 9))

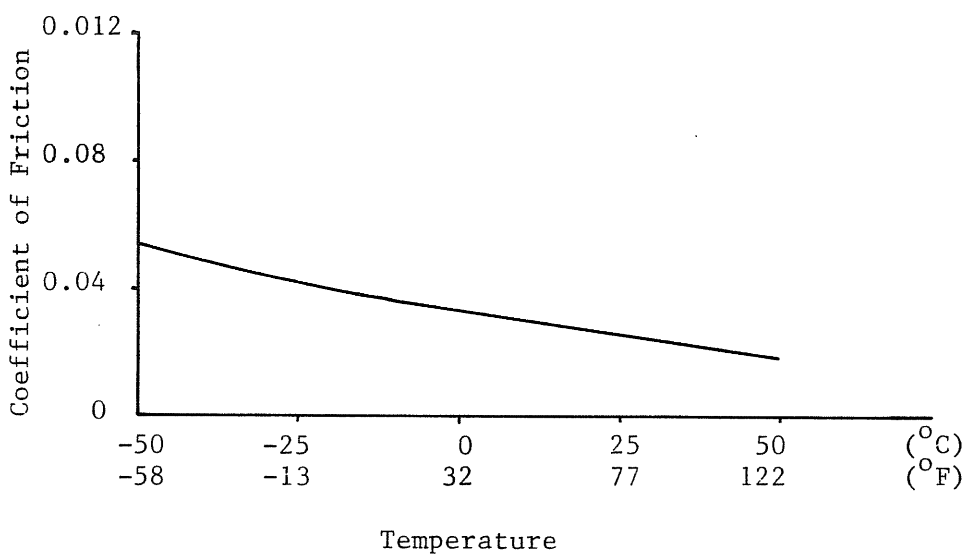


Figure 1.7 Effect of Temperature on Unlubricated PTFE

fiber filler, different size contact area, TFE vs. TFE, TFE vs. stainless steel and TFE vs. highly polished, mirror finish stainless steel on the effective coefficient of friction. All tests for TFE were done at room temperature (approximately 70°F) and no lubrication was used.

To achieve the confidence and reliability of the experimental results, several increments of loading were used and at least three tests were done at each loading for each combination.

CHAPTER II

TESTING PROCEDURE

2.1 Test Set-up

To determine the experimental coefficient of friction of bridge bearings, a test set-up, which simulates the actual bridge, was built as shown in the following photograph and schematic drawings (Figures 2.1, 2.2 and 2.3). The normal force was applied with a 750,000 lb. capacity hydraulic ram and the horizontal force with a 55,000 lb. capacity closed-loop hydraulic testing system. The data was recorded using a micro-computer system.

The test set-up was erected inside the Fears Structural Engineering Laboratory on the laboratory reaction floor. The floor is a concrete slab 30 ft. by 60 ft. by 3 ft. 6 in. deep with four W36x150 steel beams embedded in the concrete. The slab weighs one million pounds and is capable of reacting 320,000 lbs. in any one location. The set-up was erected directly over two of the embedded W36 beams spaced 8 ft. apart. The set-up consisted of three parts: 1) An H-frame (Figure 2.2) which was designed for 250,000 lbs. maximum vertical reaction and which supported

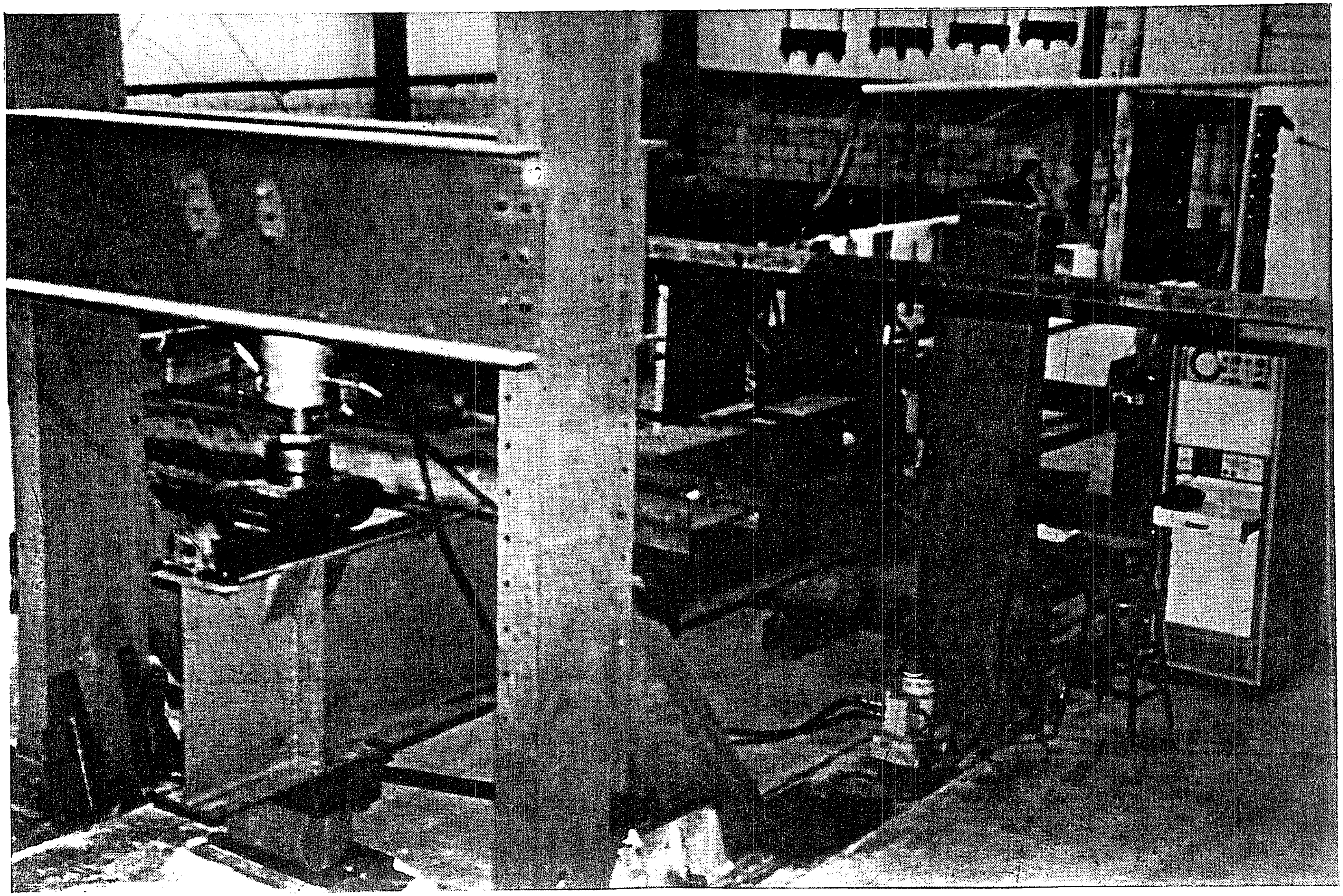


Figure 2.1 Test Set-up

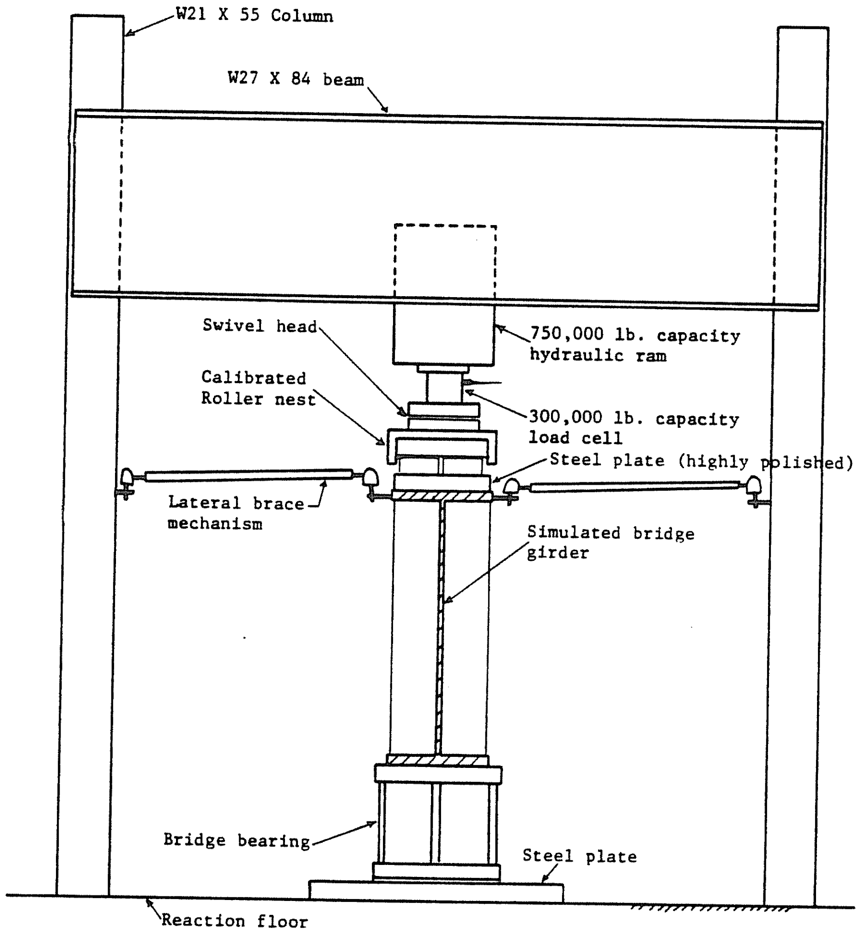


Figure 2.2 Reaction H-Frame

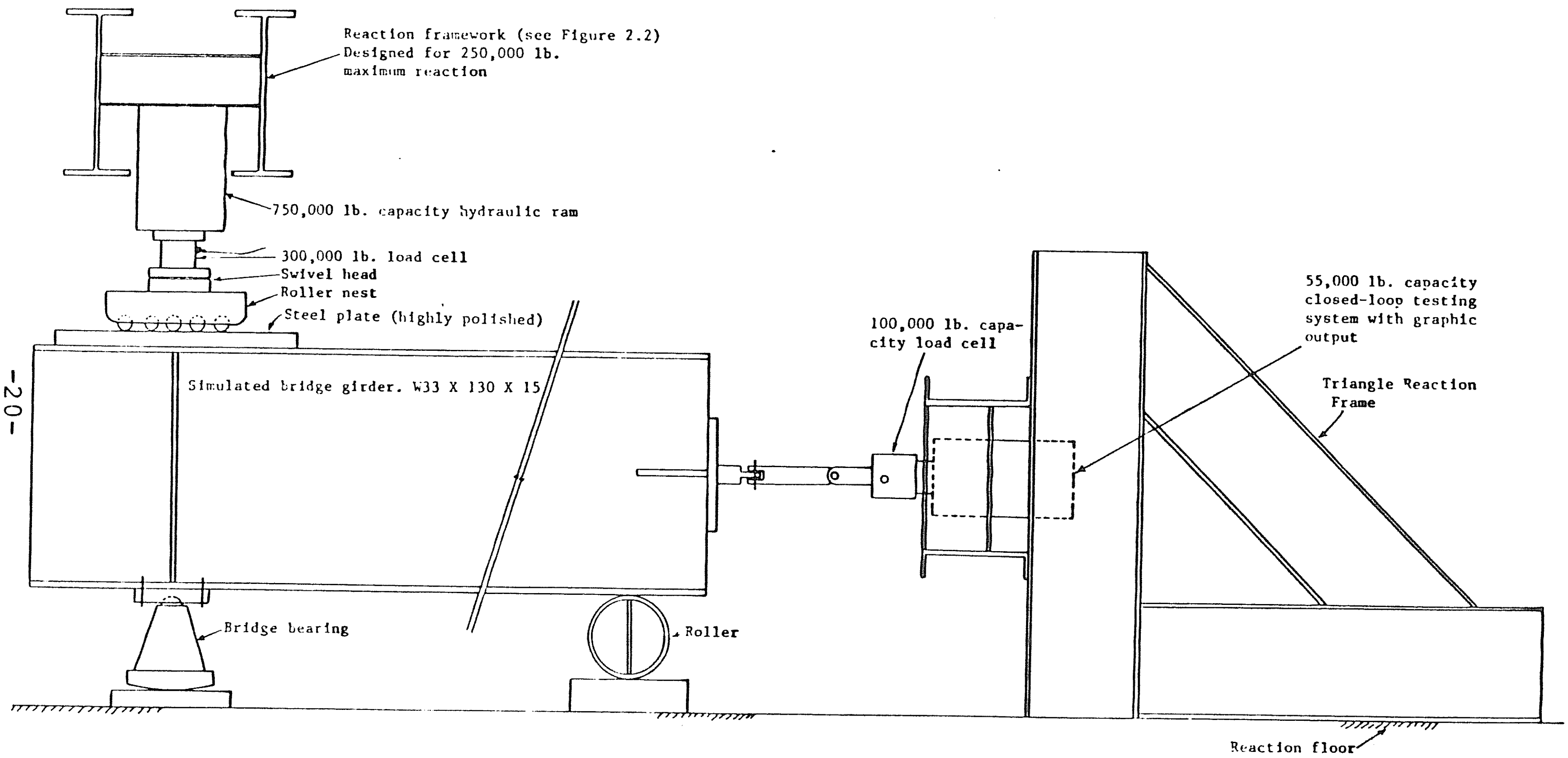


Figure 2.3 Side View of Test Set-Up

the hydraulic ram, 2) A triangle frame (Figure 2.3) which was designed for 55,000 lbs. maximum horizontal reaction and which supported the closed-loop hydraulic testing system, and 3) A W33x130x15' girder which simulated the actual bridge girder.

The vertical load chain consisted of the H-frame, hydraulic ram, load cell, swivel head, roller nest with a known effective coefficient of friction, a steel plate with a highly polished surface, the simulated bridge girder, the test bearing, a steel reaction plate and the reaction floor, as shown in Figure 2.2. The horizontal load chain consisted of the triangle frame, the actuator of the closed-loop hydraulic testing system, load cell, a loading linkage to prevent out-of-plane forces and the simulated girder, as shown in Figure 2.3. Lateral brace mechanisms were used to stabilize the girder against out-of-plane rotations and a pipe roller was used to support the unloaded end of the bridge girder.

2.2 Instrumentation

Instrumentation consisted of the two calibrated load cells, a horizontal displacement transducer, an analog to digital signal converter and a micro-processor. The applied normal force was measured using the calibrated 300,000 lb. capacity load cell; the horizontal force was measured using the calibrated 100,000 lb. capacity load cell; and the horizontal movement (girder movement) was measured by using a calibrated transducer which is part of the closed-loop hydraulic testing system.

The analog signals from the three instruments were digitized using a 16 channel differential input A/D converter with direct interface to the microprocessor. The microprocessor was used to reduce and plot the data in real time. In this manner, changes in normal force due to uncontrollable vertical movement in the vertical force chain were accounted for and the instantaneous relationship of the two force and one displacement variables was known.

2.3 Test Procedure

For each test, the centerline of the bearing was first positioned relative to a fixed vertical plane. A nominal normal force was then applied, usually in multiples of 25 kips, but not exceeding the rated capacity of the bearing. The simulated girder was then pulled at a slow rate (approximately 1 in. per minute) using the closed-loop hydraulic testing system. As mentioned, all data was recorded in real time using the micorprocessor. The length of travel and starting position varied between tests and will be described in Chapters III and IV as appropriate.

Approximately 100 data sets (each set consisted of two force and one displacement readings) were recorded for each test. The effective coefficient of friction was automatically calculated by the microprocessor taking into account the initial force on the bearing due to the weight of the system and the effective coefficient of friction of the roller nest. The graphics capabilities of the microprocessor system were used to display and plot the relationship between the horizontal forces and horizontal movement. Typical results are shown in Figure 2.4.

2.4 Rusting Procedure

To stimulate in-situ conditions, the steel bearings were subjected to an acidic environment which caused rusting of the exposed surfaces.

The bearings were kept inside a closed bucket in the acidic environment for about two months. Muriatic acid (HCl) was used to accelerate the rusting. The bearings were supported approximately 10 in. above the acid surface and the bucket was kept outside with varying temperature from about $25^{\circ}F$ to $80^{\circ}F$.

To determine the degree of corrosion the following procedure was used:

1. Five samples plus one control sample, approximately 2" x 8" x 1/8" steel plate were fabricated.
2. The samples were brushed clean and degreased with acetone.
3. Each sample was measured and weighed very accurately.
4. The samples were suspended in the same environment and for the same time as the bearings.
5. Then samples were removed and cleaned.

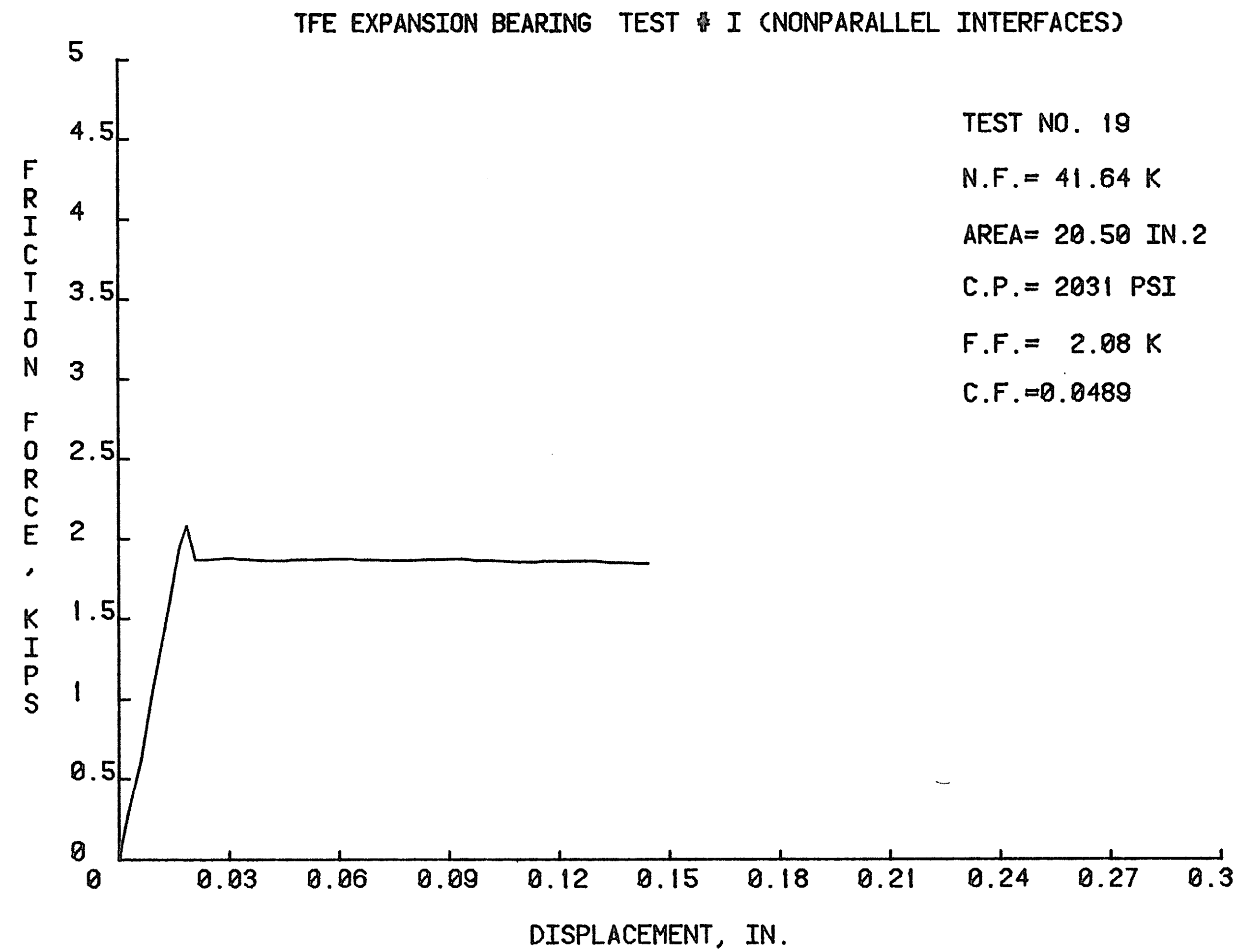


Figure 2.4 Horizontal Movement vs Friction Force

6. The samples were reweighed.

7. The corrosion rate expressed as mils penetration per year (mpy) was calculated by the following equation: (See Reference 10).

$$\text{mpy} = \frac{\text{wt. loss x 534}}{\text{(area)(time)(metal density)}} \qquad (2.1)$$

where weight loss is in milligrams, area is square inches of metal surface exposed and time is hours exposed.

Results for the five samples after two months of exposure (1440 hours) are shown in Table 2.1. No loss of weight was found in the control sample.

2.5 Sanding Procedure

After the bearing was placed in the test set-up, an approximately 1/8 in. thick layer of graded sand was spread on the lower bearing plate. The sand was obtained from ODOT and was obtained by vacuuming areas near in-place bridge bearings. Gradation is shown in Table 2.2.

Table 2.1 Degree of Rusting

Sample No.	Exposed Area (in.^{2})	Weight Loss (milligrams)	Corrosion Rate (mpy)
1	32.7	9,000	13.43
2	30.4	8,500	13.64
3	32.6	8,500	12.72
4	34.3	9,000	12.80
5	35.2	9,500	13.17

Average Corrosion Rate = 13.20 mpy

Table 2.2 Mechanical Analysis of Sand

Diameter of Particle or Sieve Number (mm)	Percent Passing by Weight
3/4 (19.05)	100
1/2 (12.7)	99
3/8 (9.525)	97
No. 4 (4.76)	78
No. 10 (2.0)	59
No. 20 (0.84)	47
No. 30	40
No. 40 (0.42)	34
No. 50	29
No. 60 (0.25)	27
No. 80	22
No. 100 (0.147)	19
No. 140	15
No. 200 (0.074)	12.5

Sp. Gr. = 2.709

PH =7.95

CHAPTER III

TEST RESULTS AND COMPARISON OF ROLLING DEVICES

3.1 Unturned Pipe-Roller (Single Roller)

A 10 in. diameter unturned, stiffened, painted pipe-roller as shown in Figure 1.1(a) was used for this phase of the study. The specimen was tested under three conditions as follows:

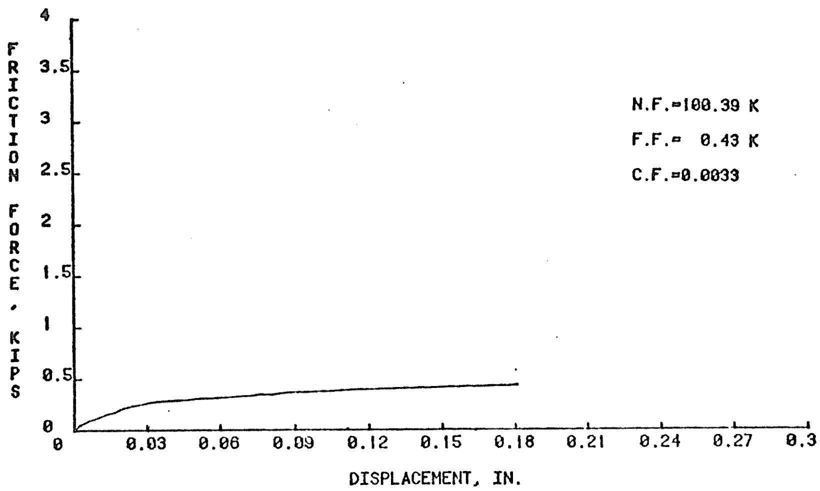
Condition 1. Clean roller and bearing plates.

Condition 2. Clean roller with rusted lower bearing plate.

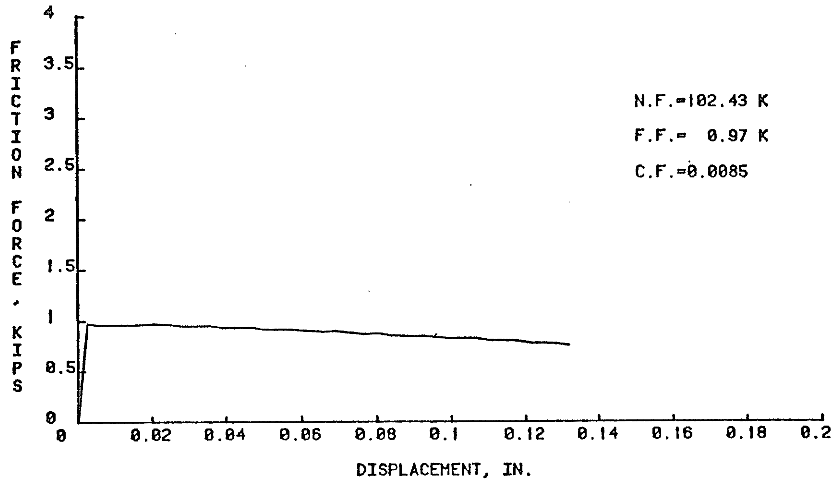
Condition 3. Roller with sand spread over the lower bearing plate.

The roller was tested at four increments of vertical loading, 25, 50, 75 and 100 kips, for each condition based on a load carrying capacity of 103.5 kips as determined from Equation 1.2.

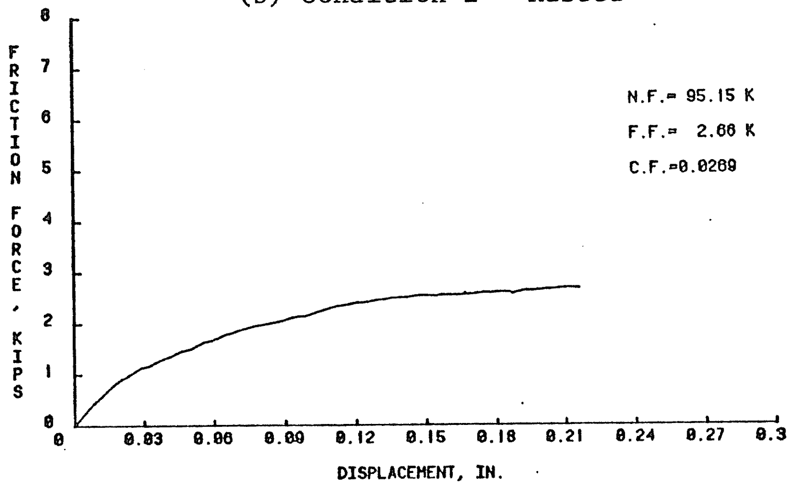
Data from the tests is shown in Tables A.1 to A.3 of Appendix A for Conditions 1 to 3, respectively. Typical horizontal force versus horizontal deflection plots are shown in Figure 3.1. For a perfectly rigid system, horizontal displacement would not take place until the rolling frictional resistance is overcome. The initial horizontal motion



(a) Condition 1 - Clean



(b) Condition 2 - Rusted



(c) Condition 3 - With Sand

Figure 3.1 Typical Displacement vs Friction Force Plots for Pipe-Roller Bearing shown in Figure 3.1 (and all subsequent similar plots) is from the elastic deformation of the test fixtures. This deformation does not affect the test results.

The average effective coefficient of friction for Condition 1 (clean roller and bearing plate) was found to be 0.33% with a standard deviation of 0.0014 over 12 tests and with a range of 0.12 to 0.58%. For Condition 2 (rusted lower bearing plate), the average effective coefficient of friction increased to 0.69% with a standard deviation of 0.0010 over 12 tests and with a range of 0.47 to 0.85%. Approximately 1/8 in. thick graded sand was placed on the lower bearing plate in front of the roller for Condition 3. In this condition, the average coefficient of friction was found to be 3.3% with a standard deviation of 0.012 for 14 tests and with a range of 2.1 to 5.8%.

The results for all tests are plotted in Figure 3.2 as effective coefficient of friction versus normal force. The straight lines shown are the result of regression analyses conducted for each condition.

From the results of the 38 tests conducted, the following are noted:

1. The effective coefficient of friction increases with increasing normal force (Figure 3.2)

2. The effective coefficient of friction increases 400-1000% if sand is placed on the lower bearing plate.

3. The effective coefficient of kinetic friction is essentially equal to the effective coefficient of static friction.

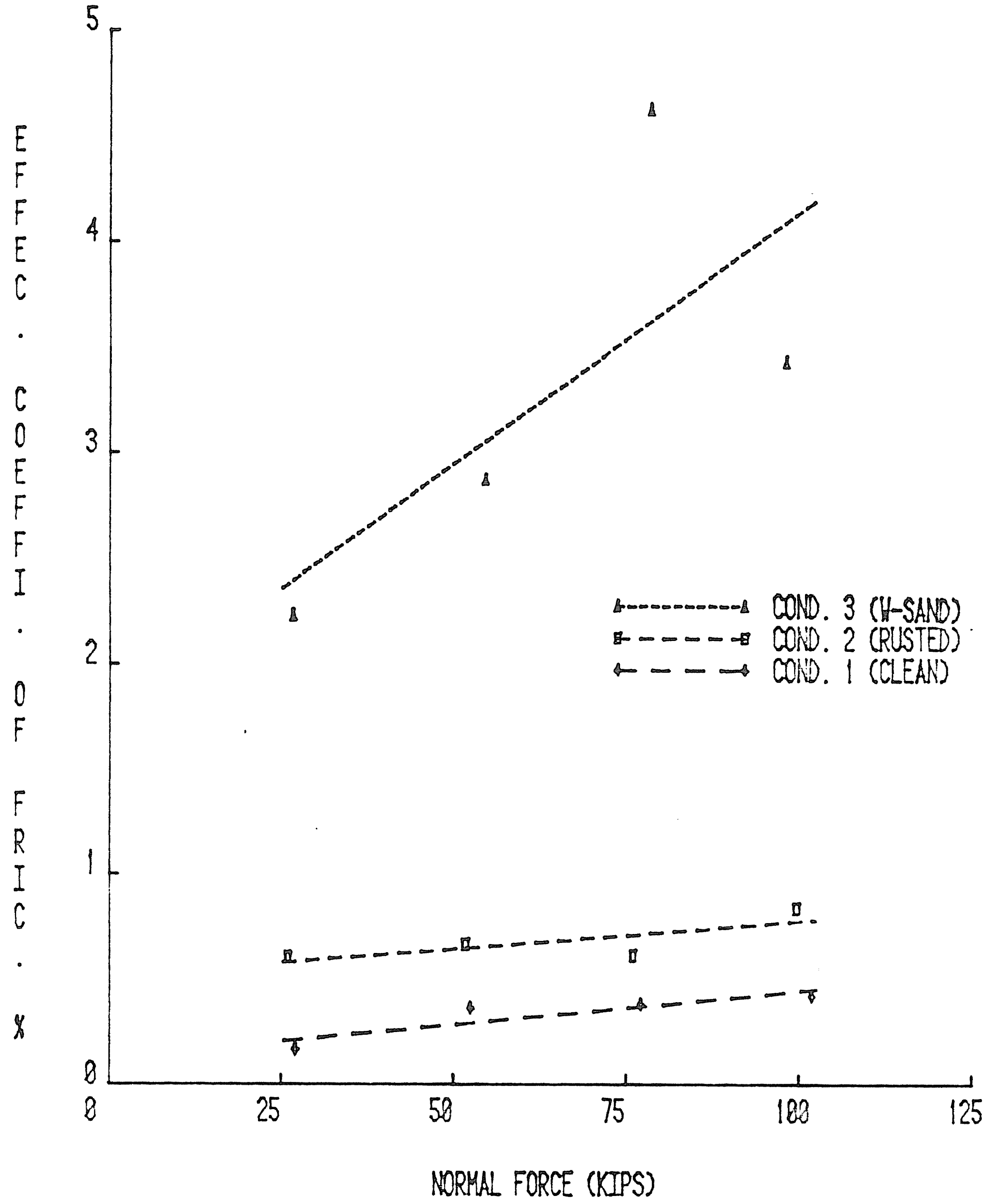


Figure 3.2 Normal Force vs Effective Coefficient of Friction of Pipe-Roller Bearing

4. The results for Condition 2 were obtained for a rusted lower bearing plate and a clean upper plate. If the upper plate had also been rusted, the increase of the effective coefficient of friction could conceivably double.

3.2 Turned Pipe-Roller

A 10 in. diameter turned, stiffened, painted pipe-roller as shown in Figure 1.1(b) was used in this phase of the study. The roller was identical to the unturned roller described in Section 3.1, except a 12 in. radius was turned on opposite sides to increase the contact surface at the upper and lower bearing plates and thus increase the load-carrying capacity. Using Equation 1.2, the allowable load is 248.4 kips.

Since the radii at the two contact surfaces is greater than half of the roller depth, the supported bridge girder rises slightly with horizontal movement. In addition, an eccentricity between the lines of action of the resultant vertical contact forces is created. A set of horizontal resisting forces is therefore needed to maintain equilibrium if the roller is moved on either side of its centerline. The magnitude of this resisting force increases with movement from the centerline as long as the turned portions of the roller are in contact with the plates. Movement beyond the turned area (usually 1-2 in. on each side of the centerline) results in a rapid decrease in horizontal force requirements, since the roller is essentially an unturned roller under this condition. For the purposes of this study, the resisting force is related to an effective coefficient of friction as defined by Equation 1.1.

The mechanics of a turned roller are best demonstrated using Figure 3.3. At the centerline position, Figure 3.3(a), the applied vertical force P and the reaction force N are colinear, and without eccentricity there is no required horizontal (friction) force for equilibrium. Once the roller turns as shown in Figure 3.3(b), the line of action of the contact forces N_1 and N_2 are no longer colinear. At point "a", the line of action of the normal component N_1 must pass through the center of the turned radius C_1 and at point "b" the line of action of N_2 must pass through the center of the radius for that surface, C_2. If the radii are equal, e.g. $R_1 = R_2 = R$, and C_1 and C_2 are at the same relative positions, the lines of action of N_1 and N_2 are parallel and vertical. The moment of the resulting couple is then

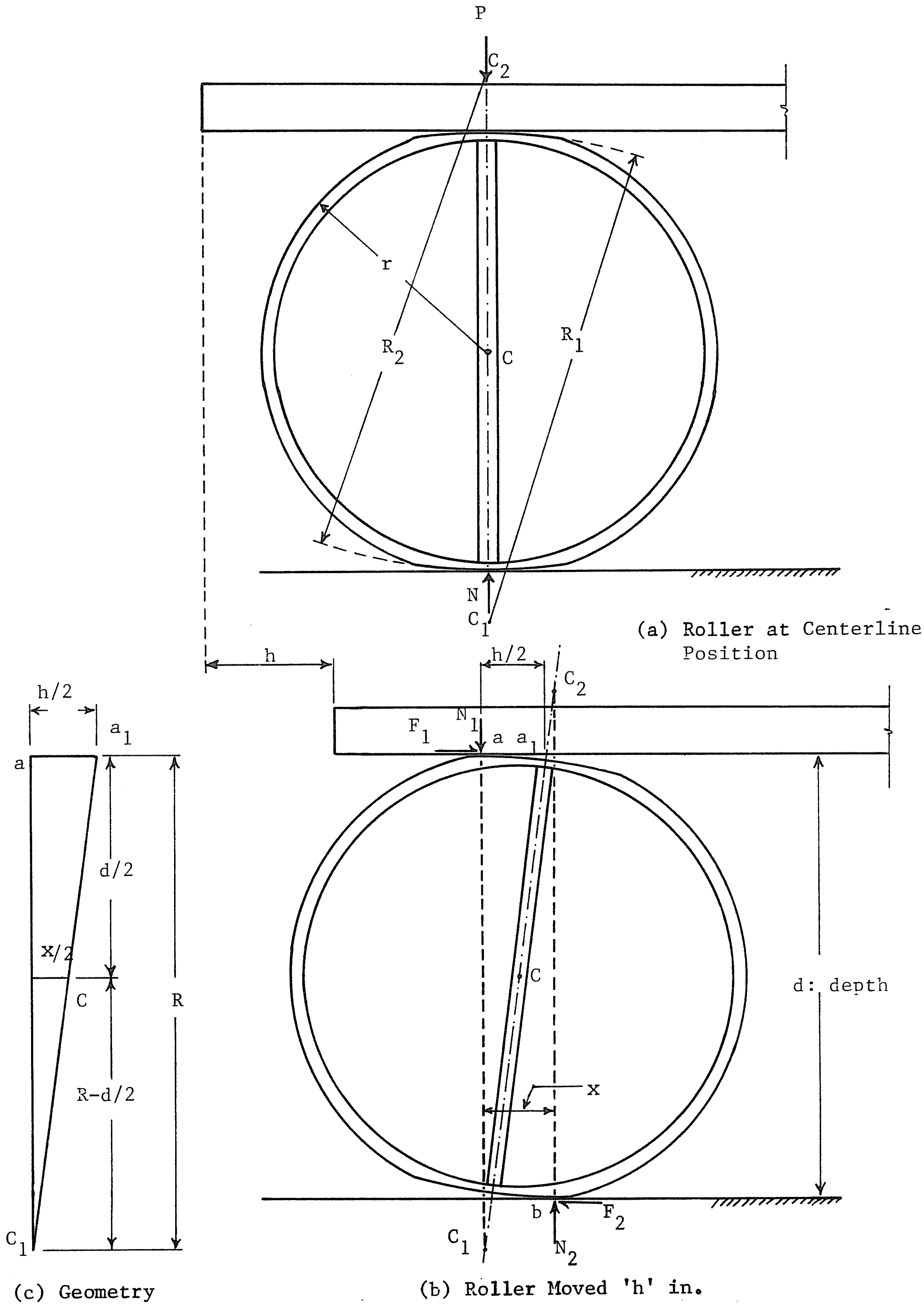
$$M = N_1 x = N_2 x = Px \tag{3.1}$$

where x = moment arm, see Figure 3.3. The moment arm x can be determined from properties of similar triangles , Figure 3.3(c), as follows

$$x = h - \frac{(R - d/2)}{R} \tag{3.2}$$

where h = total horizontal movement from either side of the centerline, R = turned radius at the contact surfaces and d = total depth of the roller.

For equilibrium, moment M must be resisted by a



(a) Roller at Centerline Position

(c) Geometry

(b) Roller Moved 'h' in.

Figure 3.3 Turned Pipe-Roller

couple associated with the friction forces F_1 and F_2. With $P = N$ and $F_1 = F_2 = F$, for equilibrium

$$P \cdot x = F \cdot d \tag{3.3}$$

Substituting Equation 3.2 into 3.3 and rearranging

$$F = \frac{(R - d/2)h}{R \cdot d} \cdot P \tag{3.4}$$

Using Equation 1.1, the equivalent coefficient of friction, as a measure of the horizontal component of force required for equilibrium, is then

$$\mu_{equiv} = \frac{P}{F} = \frac{R \cdot d}{(R - d/2)h} \tag{3.5}$$

which is independent of the applied normal force but depends on the horizontal displacement h.

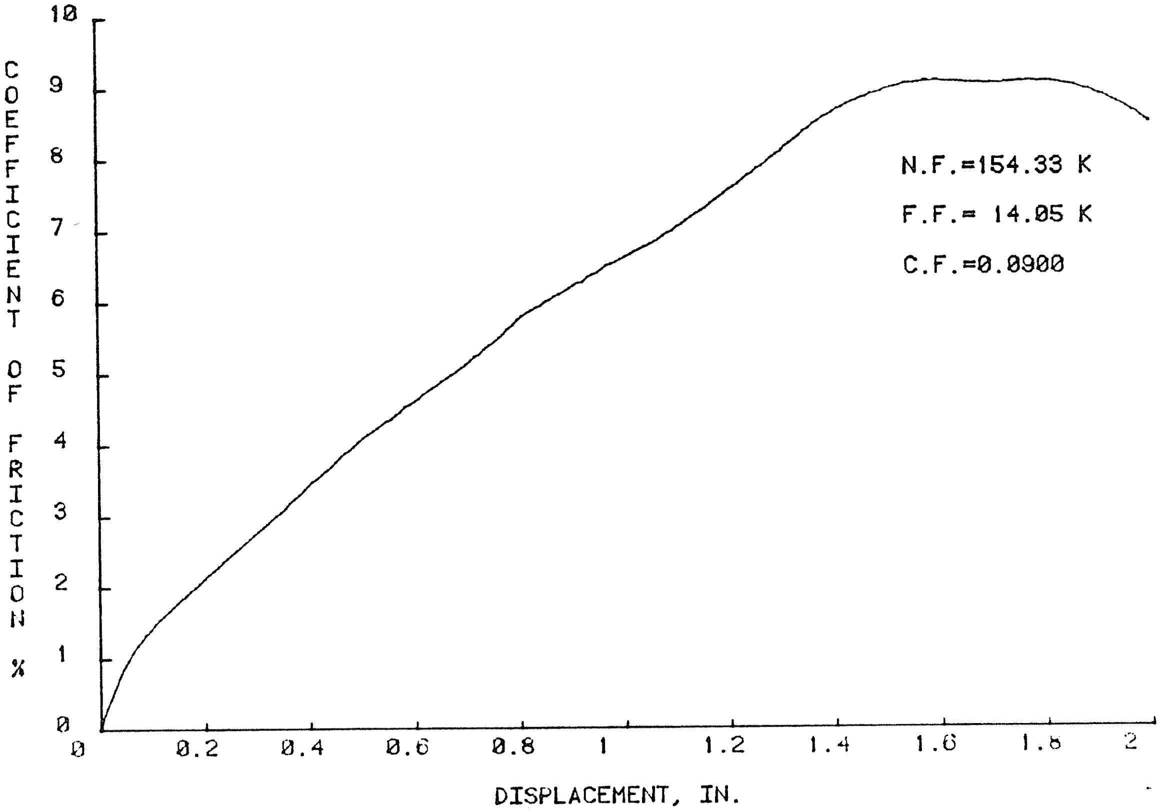
The roller was tested under two conditions as follows:

Condition 1. Clean roller and bearing plates.

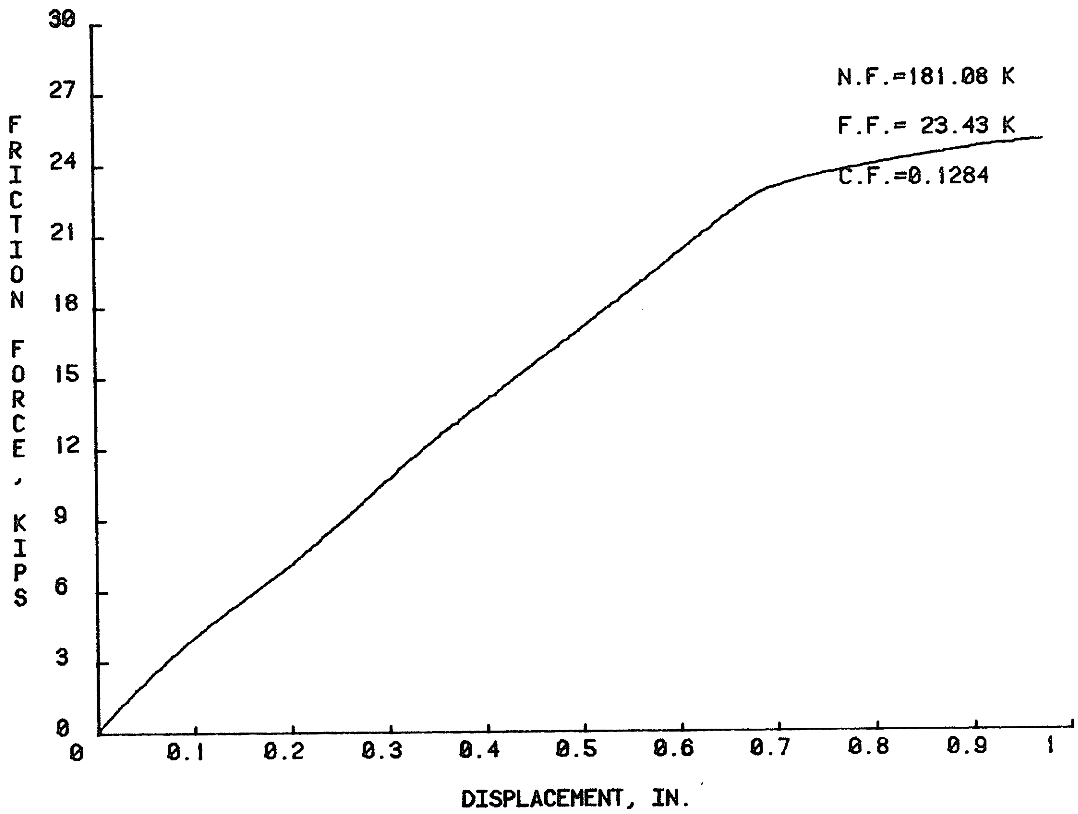
Condition 2. Roller with sand spread over the lower bearing plate.

Three increments of vertical load, 50, 100 and 150 kips, were used. Data from the tests is shown in Tables A.4 and A.5 of Appendix A for Conditions 1 and 2, respectively. Typical coefficient of friction and horizontal force versus horizontal deflection plots are shown in Figure 3.4.

Using the measured dimensions ,shown in Figure 3.5, theoretical predictions for the required resisting force (horizontal force) were calculated from Equation 3.4. Results are shown in Table A.4 together with the percent difference between measured (Condition 1) and predicted values.



(a) Condition 1 - Clean



(b) Condition 2 - With Sand

Figure 3.4 Friction vs Displacement for Turned Pipe-Roller

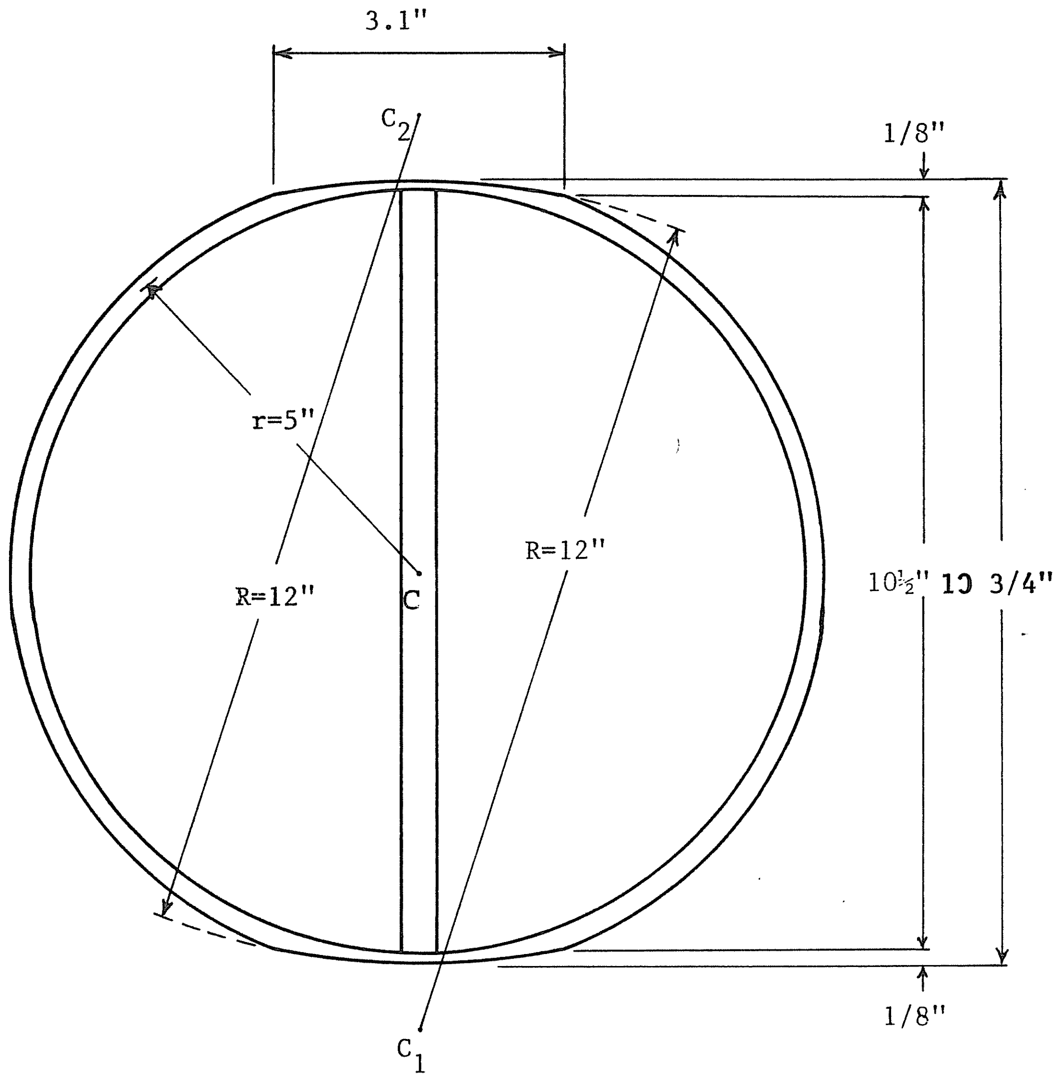


Figure 3.5 Turned Pipe-Roller Dimension

The results are also shown graphically in Figure 3.6. Close agreement was found except near 1 in. of horizontal movement. The discrepancy at this location is due to an imperfection in one or both of the turned surfaces. Close examination of the curve shown in Figure 3.4(a) shows a slight "dip" near 1 in. of displacement which indicates the length of the imperfection(s).

Table A.5 shows results for Condition 2 (with sand). Because movement of the roller over the sand effectively changes the vertical position of the roller and due to the use of hydraulic pressure to apply the normal force, magnitude of the normal force varied significantly with horizontal displacement especially at higher levels. The horizontal forces shown in Table A.5 are for displacements at the limit of the turned portion of the roller. Comparison with predicted forces (Equation 3.4) shows that the presence of sand increases the required resisting force 250% to 400%.

From the results of the 21 tests and the theoretical analyses, the following are noted:

1. The equivalent coefficient of friction is a function of horizontal displacement and increases rapidly with displacement.

2. Small imperfections in the turned surfaces can cause significant changes in equivalent coefficient of friction.

3. The presence of sand on the lower bearing plate can increase the equivalent coefficient of friction 250% to 400%.

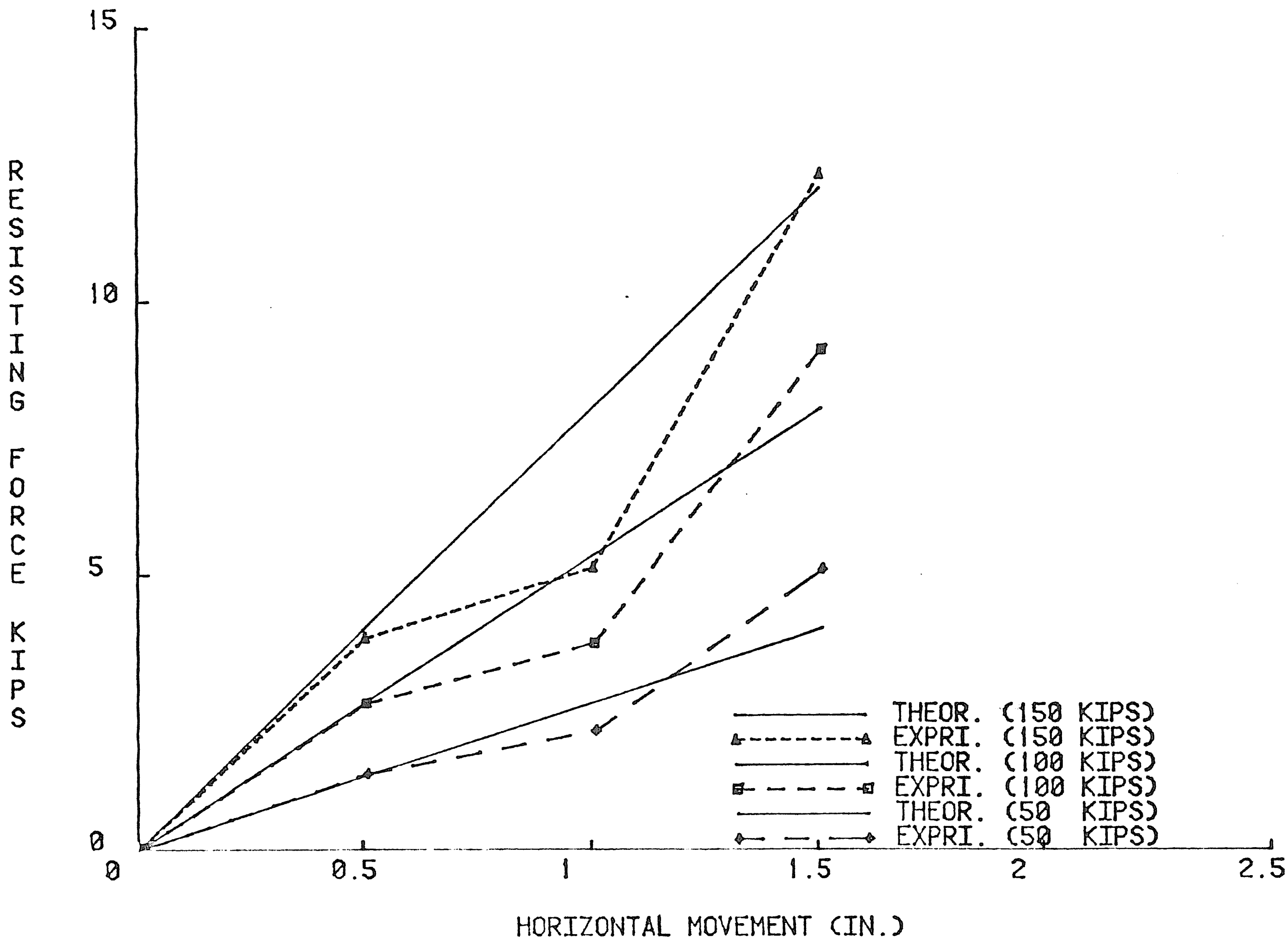


Figure 3.6 Resisting Force vs Movement for Turned Pipe-Roller

3.3 Pinned Rocker Shoe

A pinned-rocker shoe with detail dimensions shown in Figure 3.7 was tested for the following three conditions:

Condition 1. Clean and unlubricated

Condition 2. Rusted

Condition 3. Sand spread over the lower bearing plate.

The load carrying capacity was calculated as 232 kips using Equation 1.2 and the shoe was tested in approximately 25 kip increments from 50 kips to 225 kips.

Data from the tests is shown in Tables B.1 to B.3 of Appendix B for Conditions 1 to 3, respectively. Typical horizontal force versus horizontal deflection plots are shown in Figure 3.8.

The average effective coefficient of friction for Condition 1 (clean and unlubricated) was found to be 0.99% with a standard deviation of 0.00137 over 16 tests and with a range of 0.71 to 1.18%. For Condition 2 (rusted), the average effective coefficient of friction increased to 1.85% with a standard deviation of 0.00310 over 23 tests and with a range of 1.38 to 3.23%. Approximately 1/8 in. thick graded sand was placed on the lower bearing plate for Condition 3. The average effective coefficient of friction was found to be 8.95% with a standard deviation of 0.00711 over 12 tests and with a range of 4.42 to 10.40%.

The results of all tests are plotted in Figure 3.9 as friction force (horizontal force) versus normal force.

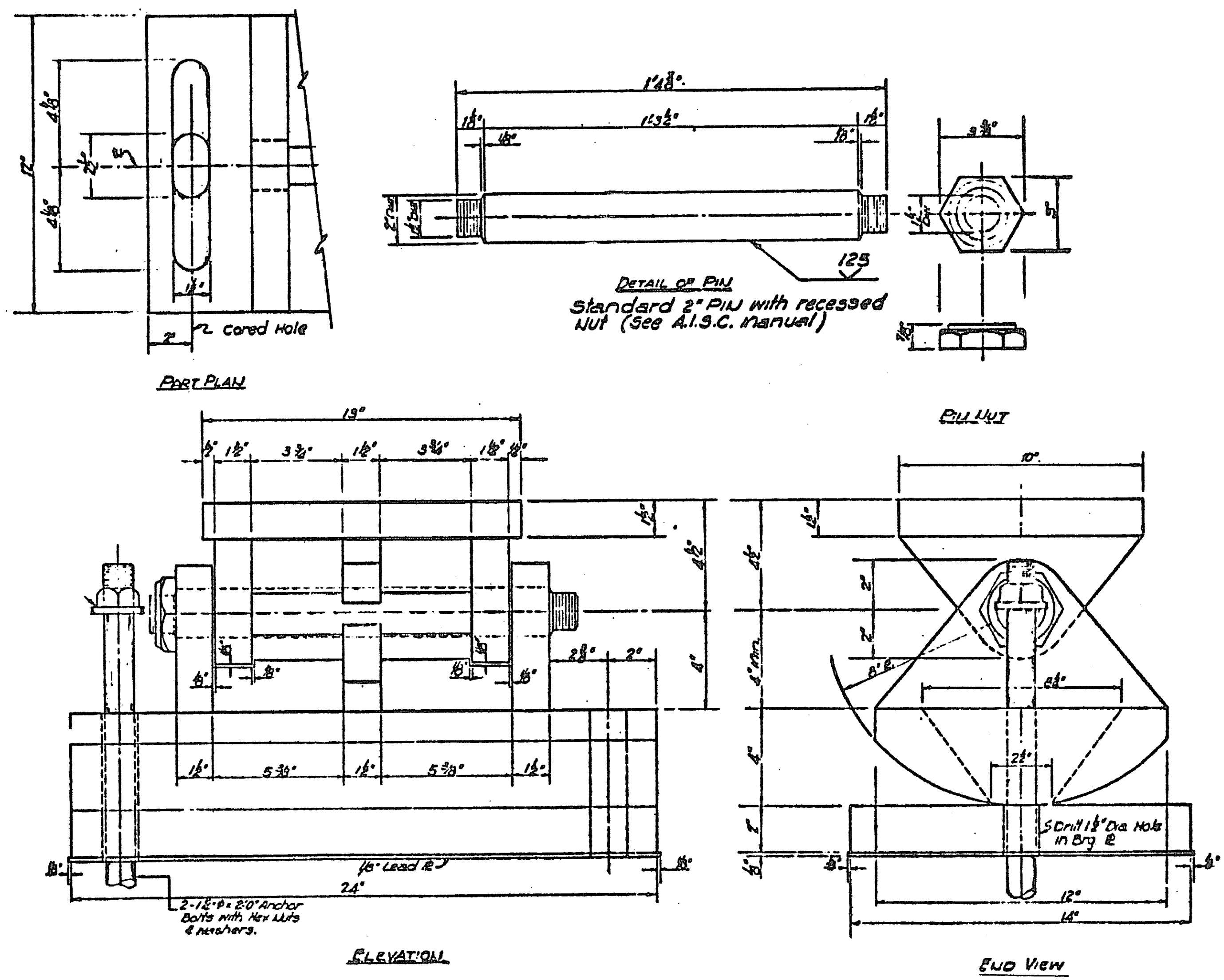
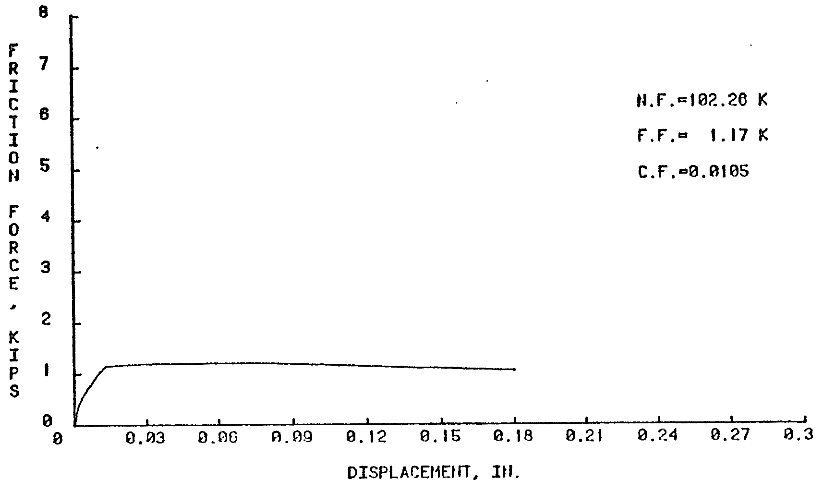
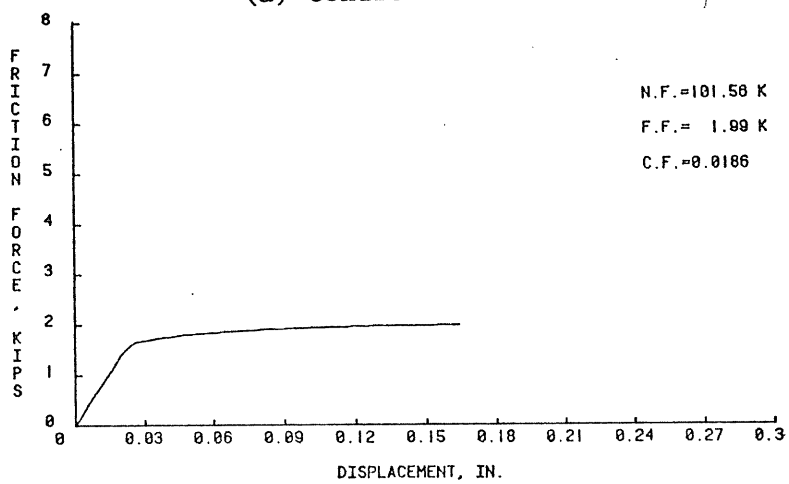


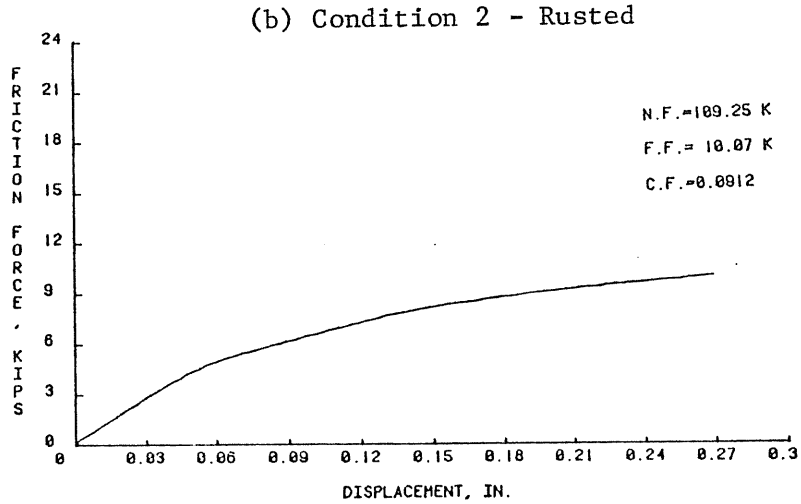
Figure 3.7 Details of Pinned Rocker Shoes



(a) Condition 1 - Clean



(b) Condition 2 - Rusted



(c) Condition 3 - With Sand

Figure 3.8 Typical Friction Force vs Displacement Plots for Pinned Rocker Bearing

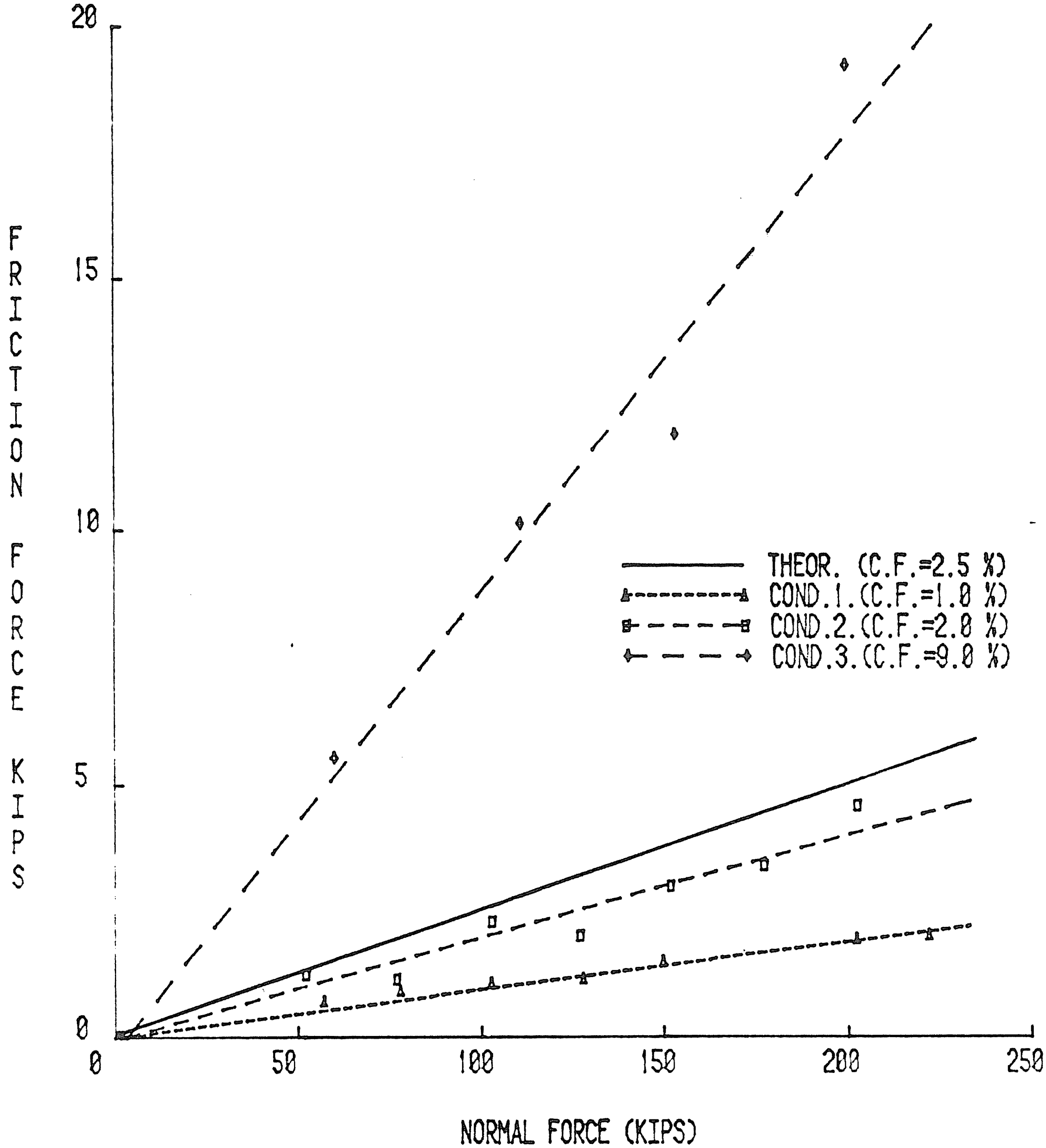


Figure 3.9 Friction Force vs Normal Force for Pinned Rocker Shoe

The straight lines shown are the result of regression analyses conducted for each condition.

The theoretical effective coefficient of friction was calculated as 2.5% using Figure 1.4. Comparison of theoretical and measured values is shown in Figure 3.9.

The following are noted from the 51 tests:

1. The effective coefficient of friction for a rusted rocker can be as high as 185% of value for a clean unlubricated rocker.
2. The presence of sand significantly alters the effective coefficient of pinned-rocker bearings.
3. As can be seen in Table B.1 to B.3, the effective coefficient of friction decreases with use. (Test numbers are in the order conducted.)

3.4 Pintle Rocker Shoe

Two pintle rocker bearings with detail dimensions as shown in Figure 3.10 were tested under three conditions:

Condition 1. As removed from a bridge site

Condition 2. Partially rusted

Condition 3. Sand spread over the lower bearing plate.

Using Equation 1.2, the load carrying capacity of the bearing was calculated to be 260 kips. Tests were conducted from 25 to 225 kips in increments of approximately 25 kips.

Results for test bearing I in Condition 1 (as received) are shown in Table C.1 and plotted in Figure 3.11. The average coefficient of friction was 7.6% with a standard deviation of 0.0111 over 24 tests with a range of 6.15 to 9.88%.

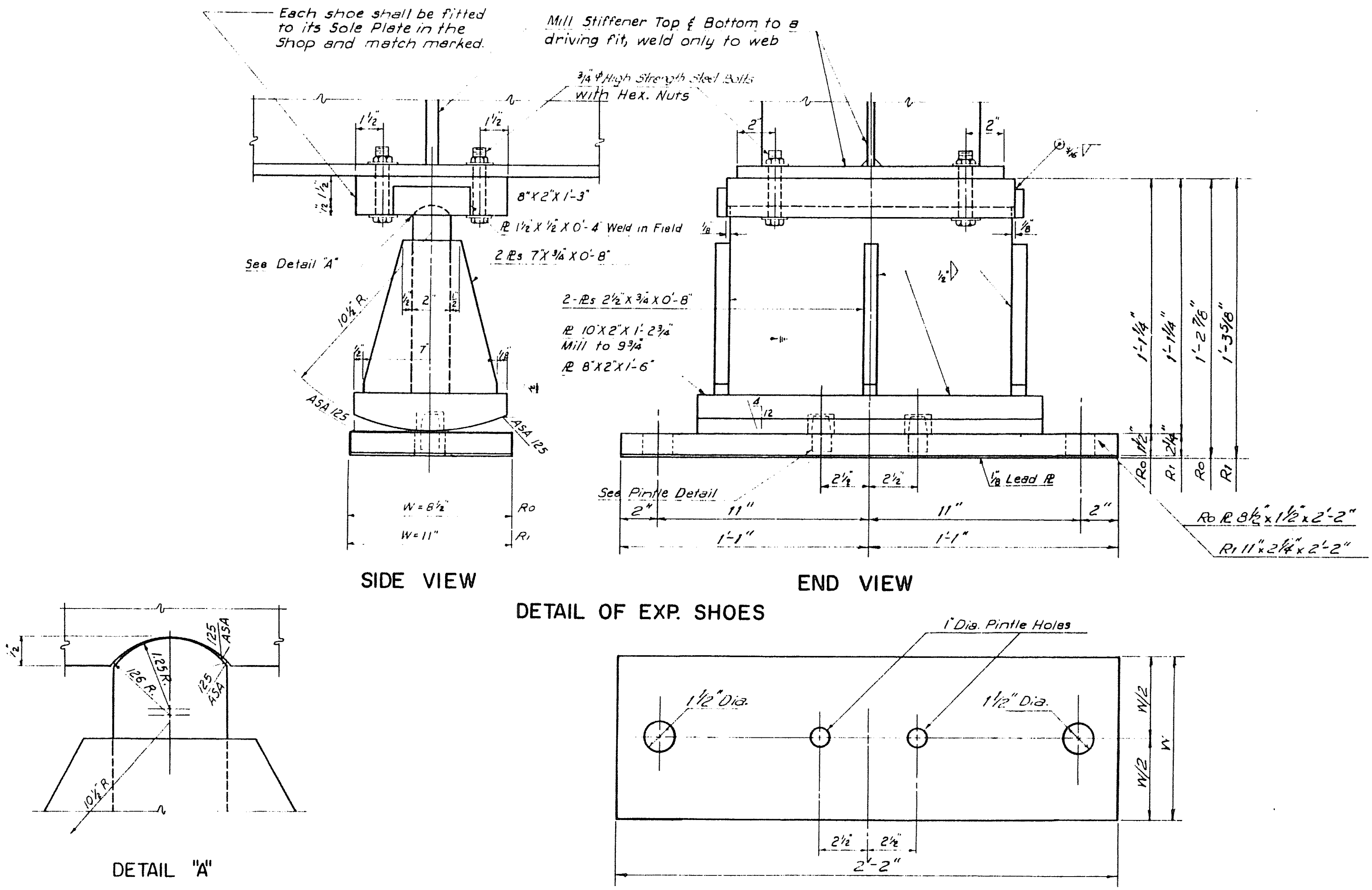


Figure 3.10 Details of Pintle Test Bearing

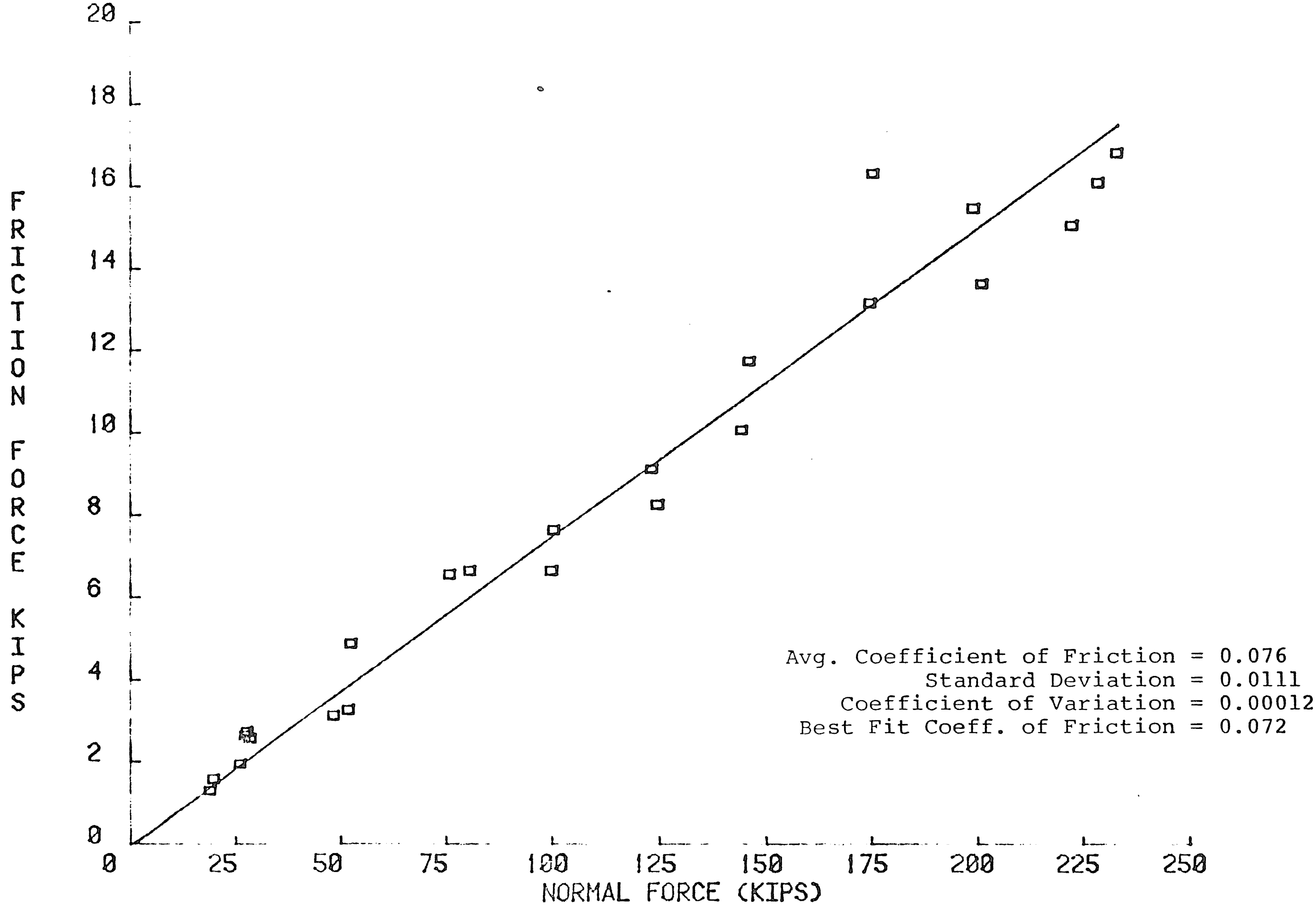


Figure 3.11 Friction Force vs Normal Force - Pintle Bearing I, Condition 1

In conducting these tests, it was noticed that the bearing exhibited significantly different effective coefficients of friction depending on the initial position of the centerline of the rocker relative to the direction of movement. Figure 3.12 shows the variation of friction force over a 3 in. travel. A series of tests for each bearing was then conducted in which the starting position was varied from before dead center (-) to after dead center (+). Results are shown in Tables C.2 and C.3 for bearings I and II, respectively. The last column of the tables indicates the initial position of the rocker. In the 55 tests conducted, the effective coefficient of friction varied from 3.13 to 7.94%, a variation not found in the other test bearings. Further, the effective coefficient of friction predicted by Figure 1.4 was 2.4%.

In an attempt to determine the cause of the discrepancy, the outside radius of the top portion of the rocker and the inside radius of the sole plate were carefully measured. Results are shown in Figures 3.13 (a) and (b). In both cases, the outside radius of the rocker was found to be larger than specified (Figure 3.10) and larger than the inside radius of the sole plate. Because of this geometry, the top part of the rocker tends to wedge inside the socket of the sole plate which causes a high effective coefficient of friction.

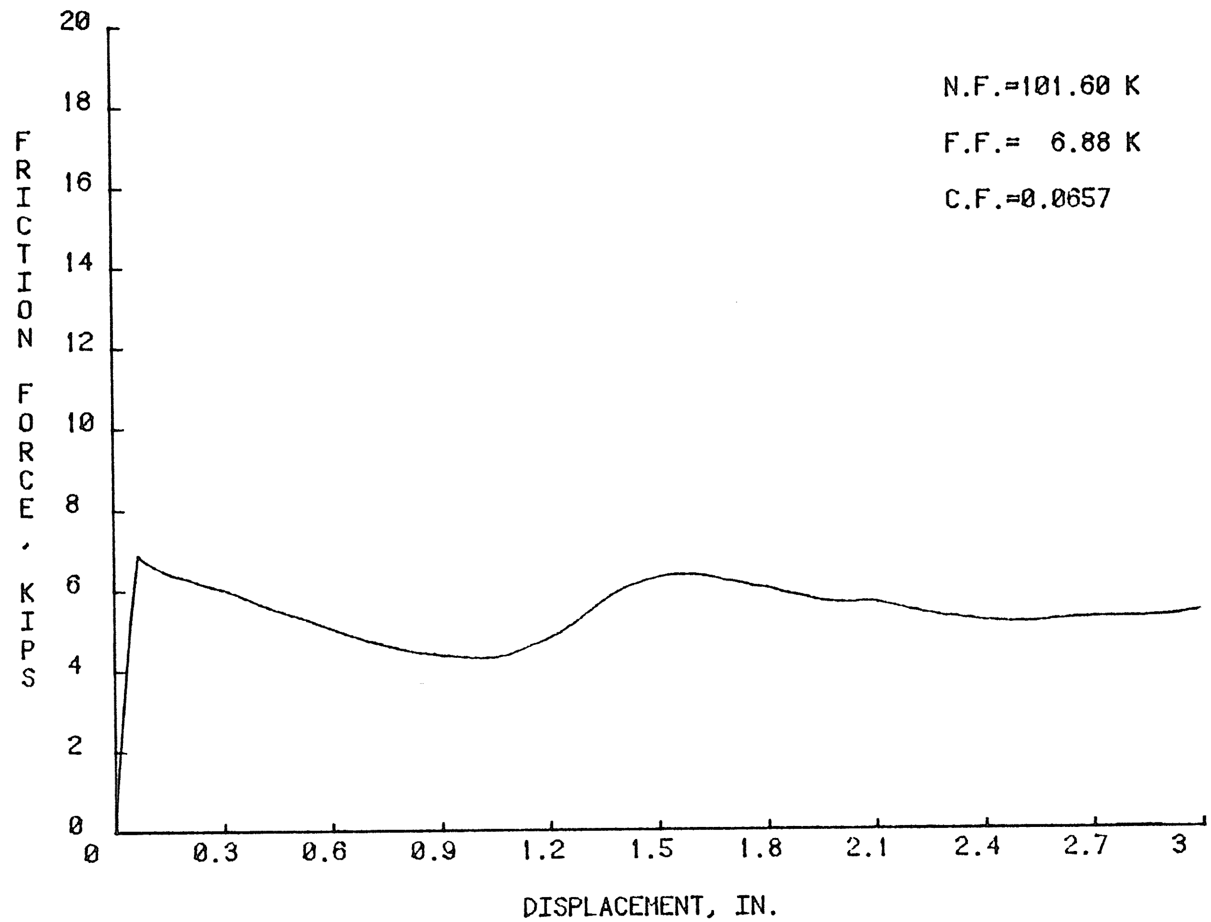
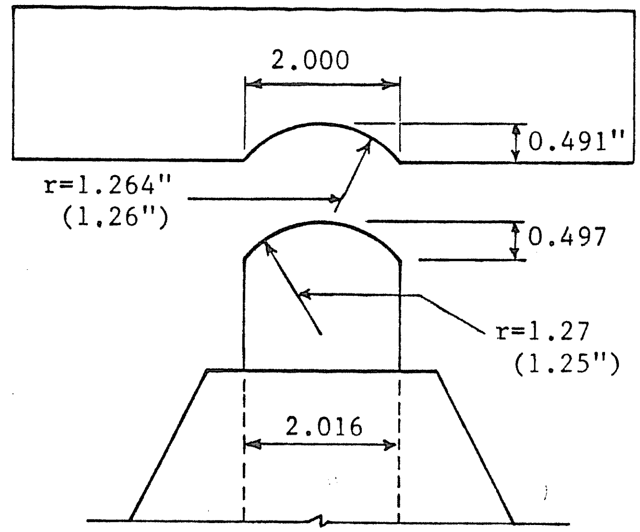
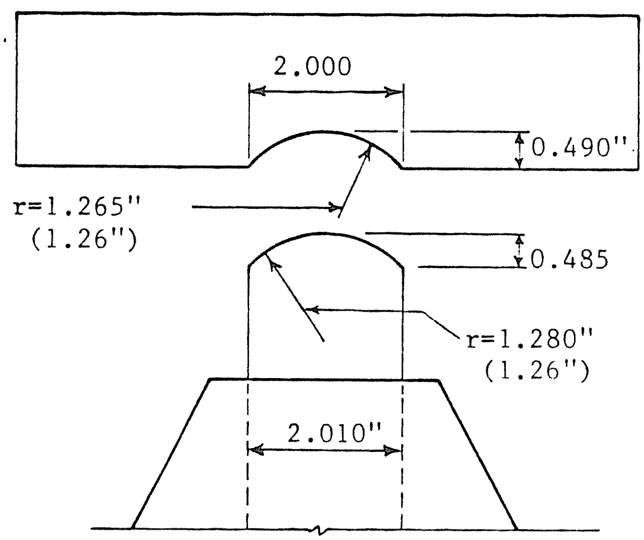


Figure 3.12 Friction Force vs Displacement - Pintle Rocker Bearing I, Condition 1



Test Bearing I



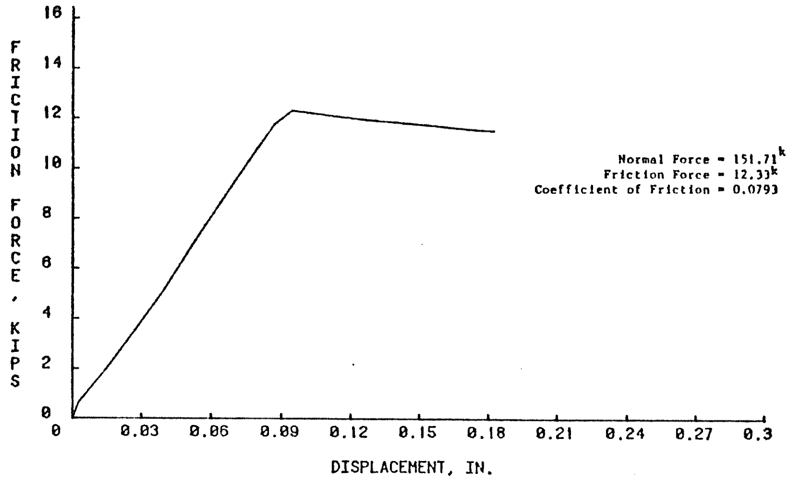
Test Bearing II

Figure 3.13 Measured Dimension of Pintle Rocker Bearings

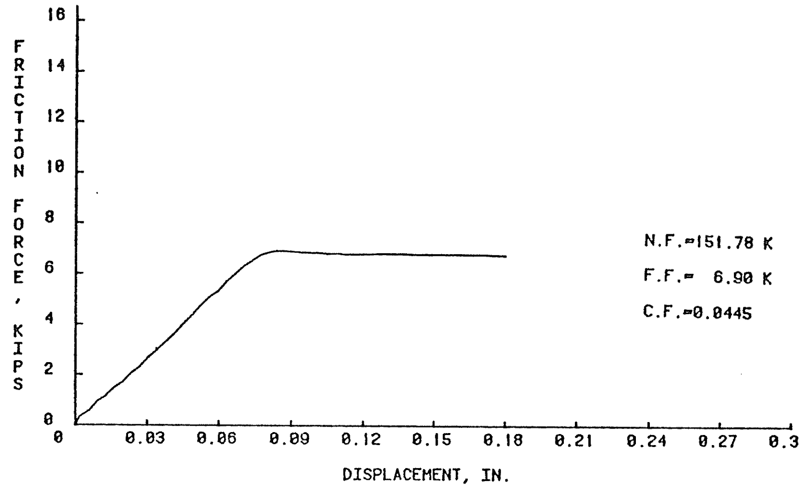
To verify this contention, sole plates with inside radii of 1.27 and 1.35 in. were used for additional testing. Table C.4 shows the results of 15 tests using the 1.25 in. radius sole plate, pintle bearing I in Condition 1. Figures 3.14(a) and (b) compare the friction force versus displacement relationships for the two sole plates (r = 1.26 in. and r = 1.27 in.). For the series with the 1.27 in. radius, the average effective coefficient of friction was 4.31% with a standard deviation of 0.0049 and a range of 2.22 to 5.45%. The average coefficient of friction decreased from 7.60% to 4.31% with an increase in inside radius of only 0.01 in.

A series of tests was also attempted with the large radius (1.35 in.) sole plate. A typical friction force versus horizontal displacement relationship is shown in Figure 3.14(c). Since the radius in the sole plate was significantly larger than the outside radius of the rocker, by 0.07 in., the rocker was rolling inside the sole plate rather than sliding. The rocker was observed to roll in the sole plate socket until the required coefficient of friction was greater than possible between the steel surfaces and then the parts suddenly "jumped" to an initial position and the process was repeated as clearly shown in Figure 3.14. Results using the large radius sole plate were too scattered for use in this report.

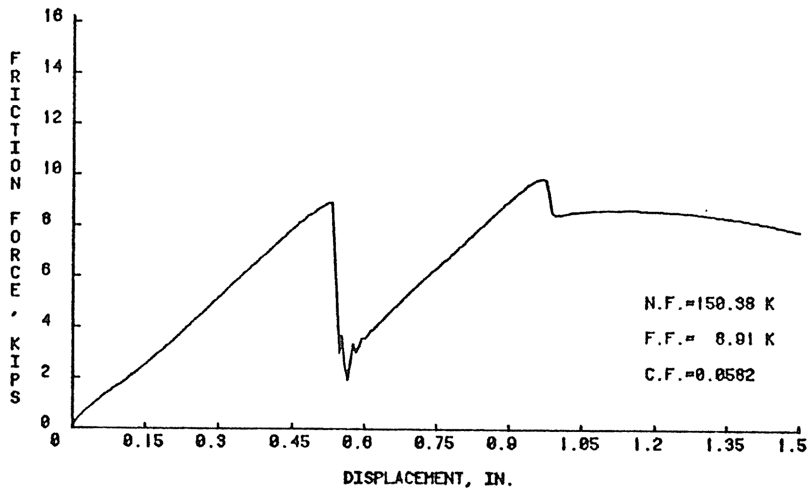
The tests were repeated using the 1.27 in. radius sole plate for Conditions 2 and with the original sole plate



(a) Original Sole Plate, r = 1.26"



(b) Sole Plate With r = 1.27"



(c) Sole Plate With r = 1.35"

Figure 3.14 Friction Force vs Displacement - Pintle Rocker Bearing I With Various Sole Plates for Condition 3 (with sand). Results are shown in Tables C.5 and C.6. The average effective coefficient of friction for the rusted condition increased to 4.8% with a standard deviation of 0.0018 over 15 tests with a range of 3.64 to 5.48% and for the sand condition to 13.13% with a standard deviation of 0.0014 and a range from 12.08 to 14.11% for 12 tests.

From the numerous tests, conditions, and configurations of this phase of the study the following are noted:

1. Fabrication accuracy is necessary if the predicted effective coefficient of friction (Figure 1.4) is used to estimate the horizontal friction force of pintel bearings.

2. Slight inaccuracies in the radii of mating parts can result in a substantial increase in the effective coefficient of friction.

3. Rust and, particularly, sand can substantially increase the effective coefficient of friction of pintle bearings.

CHAPTER IV

TEST RESULTS FOR TFE EXPANSION BEARINGS

4.1 General

A series of tests was conducted to determine the suitability of tetrafluoroethylene (TFE) expansion bearings for bridges. For the tests, the rigid stand shown in Figure 4.1 was added to the test set-up previously used for the mechanical bearing tests. In each test the bottom element was tack welded to the stand and the top element tack welded to the girder such that movement could occur only between the element surfaces. The interface was moved at least 0.15 in. horizontally in a direction parallel to the short side of the elements at a speed of 1 in. per minute.

In this study, three types of TFE elements (unfilled, glass filled 25% by weight and woven unfilled and glass filled fibers), two steel surfaces (stainless steel and mirror finish stainless) and two backings (carbon steel plate and 70 Durometer neoprene vulcanized to a steel plate) were tested in appropriate combinations. Table 4.1 lists the seven element types and Table 4.2 shows the eight combinations tested. Tests were conducted in either "parallel" or

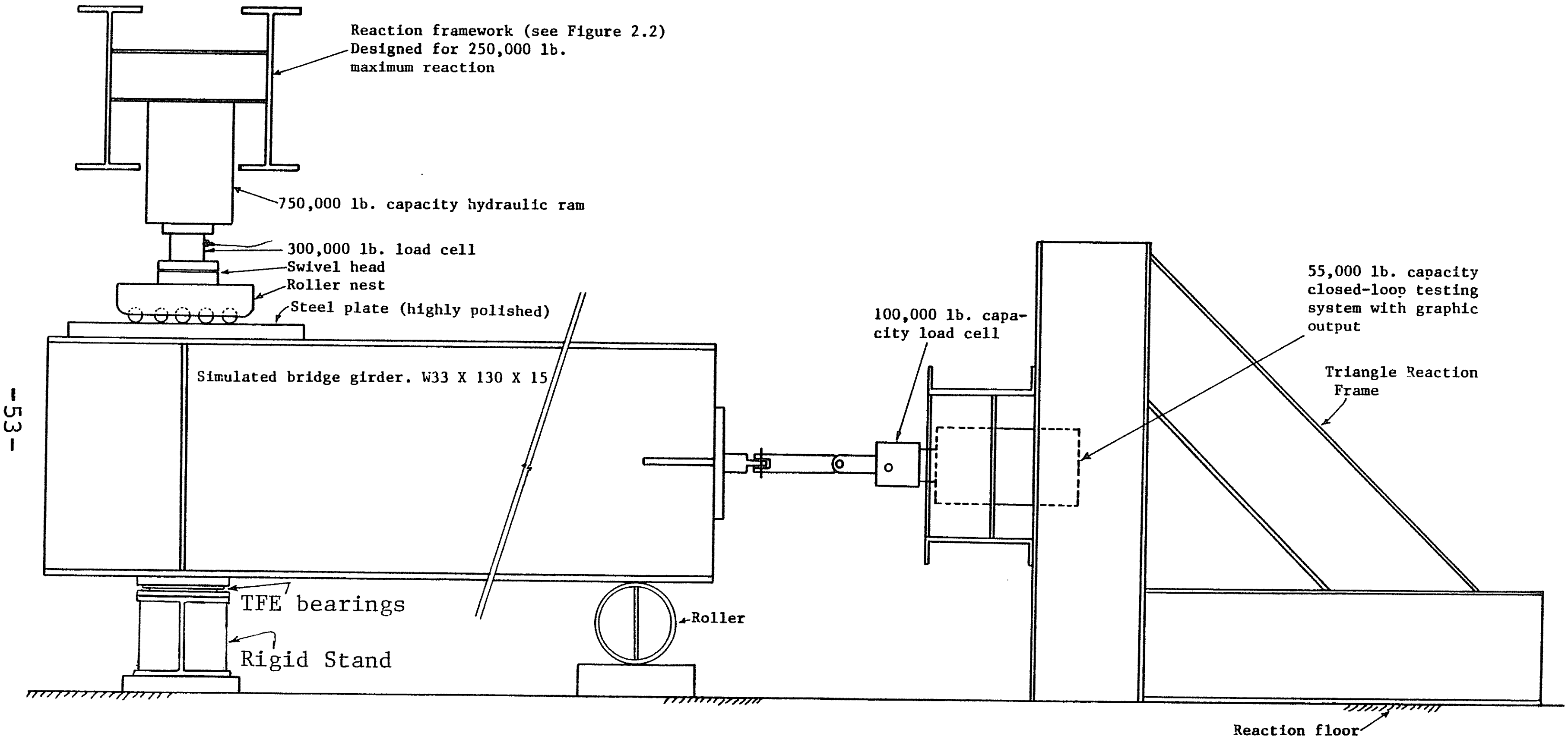


Figure 4.1 Test Set-up for Teflon Bearings

Table 4.1 TFE Test Elements

Element No.	Description
1	3/32" Glass filled* TFE bonded to 1/4" A-3 carbon steel
2	3/32" Glass filled* TFE, mechanically locked to 1/4" carbon steel
3	1/4" Mirror finish stainless steel
4	3/32" Glass filled* TFE bonded to #10 gage carbon steel hot vulcanized to 3/4" 70 Durometer AASHTO grade neoprene
5	1/16" Unfilled TFE bonded to 1/4" carbon steel
6	Unfilled TFE fibers and glass fibers woven and bonded to 1/4" carbon steel
7	1/8" Stainless steel

*Glass filled 25% by weight

Table 4.2 TFE Test Element Combinations

Test Series	Top Element	Bottom Element
I	Glass Filled TFE (#1)	Glass Filled TFE (#1)
II	Mirror Finish Stainless Steel (#3)	Glass Filled TFE (#1)
III	Glass Filled TFE (#2)	Mirror Finish Stainless Steel (#3)
III-A	Unfilled TFE (#3)	Mirror Finish Stainless Steel (#3)
IV-N	Glass Filled TFE (#1)	Glass Filled TFE w/Neoprene Backing (#4)
V	Woven TFE (#6)	Mirror Finish Stainless Steel (#3)
VI-N	Stainless Steel (#7)	Glass Filled TFE (#1)
VII-N	Unfilled TFE (#5)	Glass Filled TFE (#1)

"nonparallel" conditions. For the former the girder and rigid stand were leveled as accurately as possible. For the non-parallel condition, the girder was shimed such that a 1/32 in. per ft. slope (0.15°) was induced. Contact area between elements was varied for test combination I (glass filled TFE versus glass filled TFE) only. Table 4.3 shows the complete variation of test parameters. Typical friction force versus horizontal displacement curves are shown in Figure 4.2. As mentioned previously, the initial slope of the curves is due to the elastic deformation of the test set-up and does not affect the test results.

All tests were done at room temperature (approximately 70°F) on new elements (0 cycle). The effect of dirt or sand in the interface was not investigated.

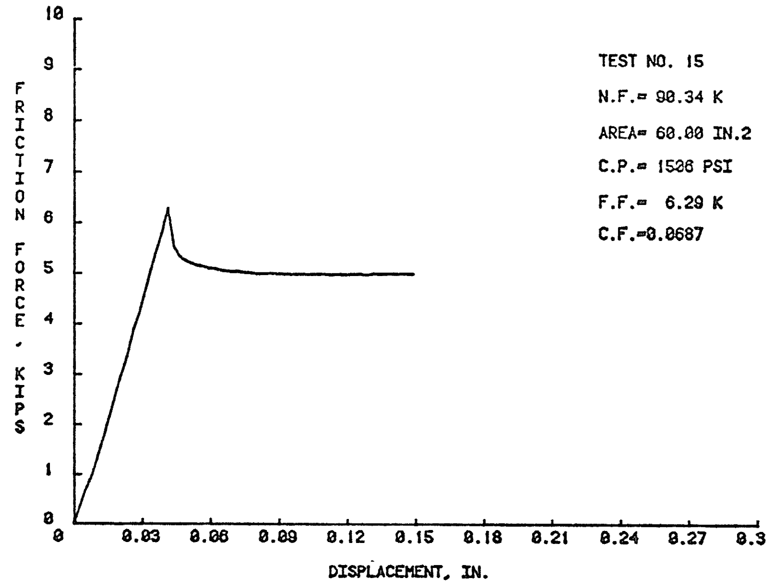
4.2 Effect of Contact Area and Contact Pressure

To determine the effect of contact area on the effective coefficient of friction, a series of tests was conducted using test combination I, glass filled TFE versus glass filled TFE. Contact area was varied from 20 sq. in., Tests I-20, to 100 sq. in., Tests I-100 and with contact pressure varying from 250 to 2000 psi. Table D.1 shows the high, low and average coefficient of friction of at least three tests for each combination of area and pressure. The average values are plotted versus contact pressure in Figure 4.3. A typical friction force versus horizontal displacement curve is shown in Figure 4.2(a).

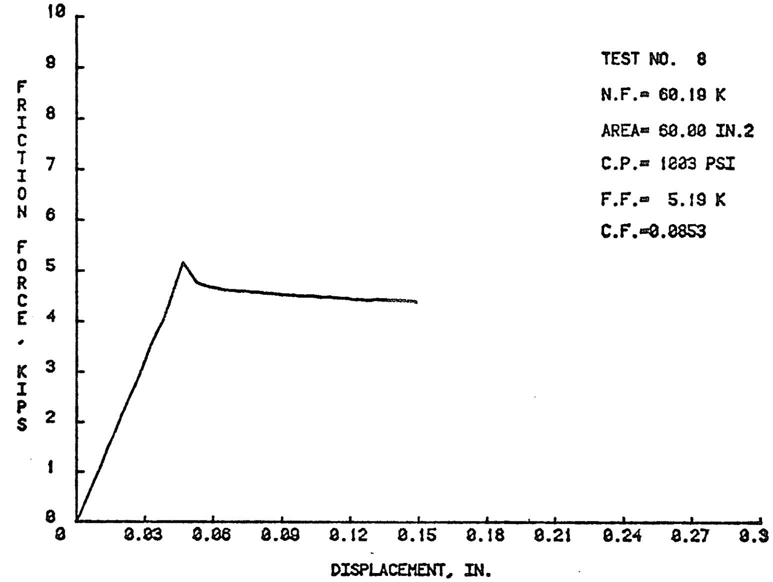
Table 4.3 Summary of Test Combinations

Series	Top Element	Bottom Element	Dimension (in)	Contact Area (in^2)	Parallel*	Test Result (Table)
I-20	#1	#1	3 x 6.6	20	no	D.1
I-40	#1	#1	5 x 8	40	no	D.1
I-60	#1	#1	6 x 10	60	no	D.1
I-100	#1	#1	8.7 x 11.5	100	no	D.1
I	#1	#1	2.93 x 7	20.5	yes	D.2
I-N	#1	#1	2.93 x 7	20.5	no	D.3
II	#3	#1	5 x 8.91	44.55	yes	D.4
II-N	#3	#1	5 x 8.91	44.55	no	D.5
III	#2	#3	5.45 x 9.4 4.2 x 7.6	51.8 31.90	yes	D.6
III-A	#5	#3	5 x 9	45	yes	D.7
IV-N	#1	#4	5 x 8.91	44.55	no	D.8
V	#6	#3	4.9 x 9	44.1	yes	D.9
VI-N	#7	#1	6 x 10	60	no	D.10
VII-N	#5	#1	4.9 x 9	44.10	no	D.11

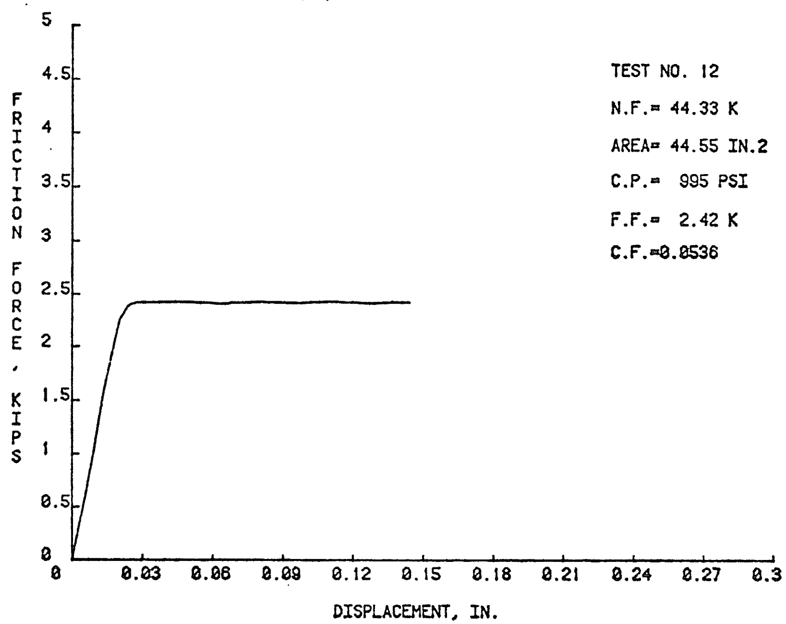
*Yes - Parallel Interface
No - Nonparallel (1/32" per 12" slope) Interface (N)



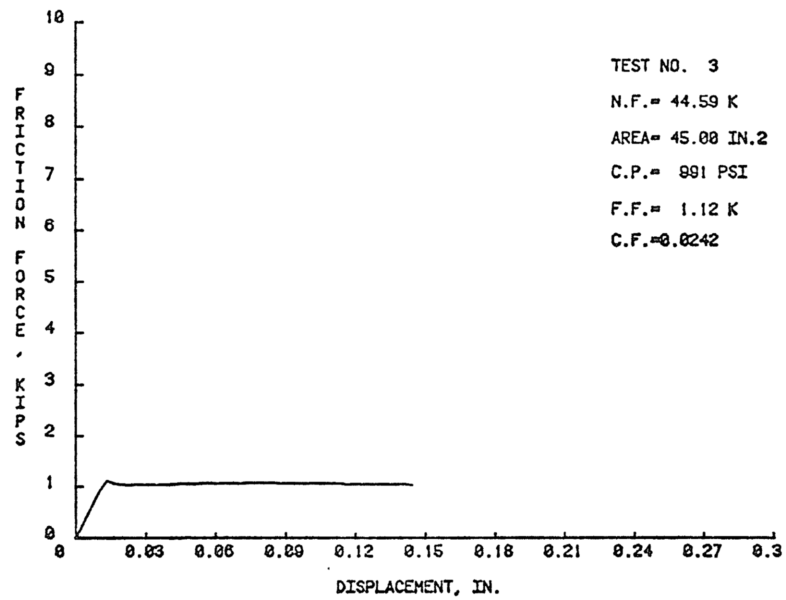
(a) Test #I-60



(b) Test #VI



(c) Test #II-N



(d) Test #III-A

Figure 4.2 Typical Friction Force vs Displacement Curves for TFE Bearings

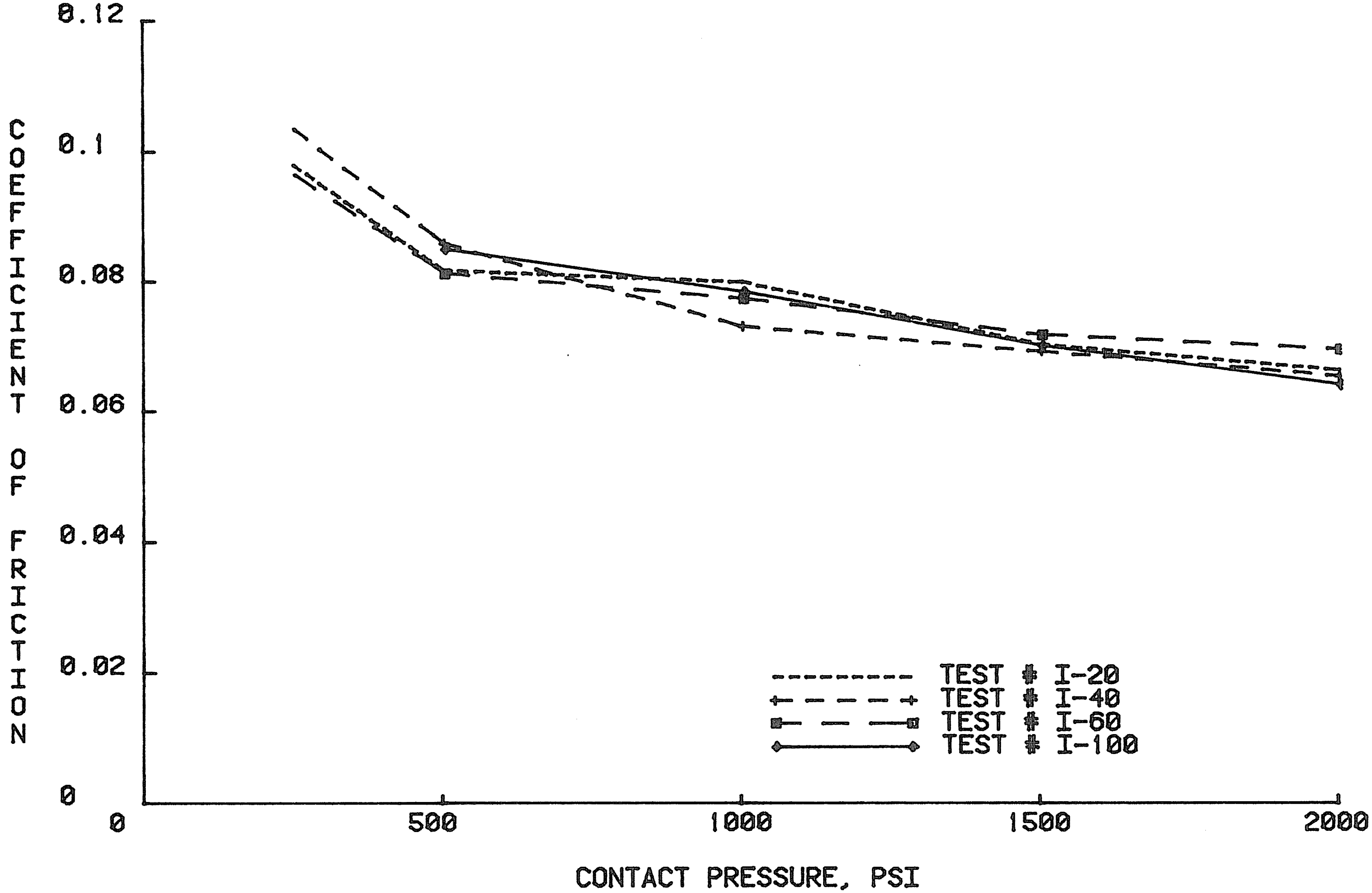


Figure 4.3 Coefficient of Friction vs Vertical Pressure, Series I

It is clear from Figure 4.3 that contact area has little effect on the effective coefficient of friction. However, the effective coefficient of friction for this combination was found to decrease with increasing contact pressure. At low contact pressure, 250 psi, the effective coefficient of friction is approximately 10%, decreasing sharply to approximately 8.25% at 500 psi and then at a slower uniform rate to approximately 6.75% at 2000 psi.

Based on the above results only contact pressure was varied in subsequent testing.

4.3 Results for Glass Filled TFE vs. Glass Filled TFE

Table D.2, Test Series I, shows the results of 22 tests conducted with glass filled TFE elements, top and bottom. The contact area for all tests was 20.5 sq. in. and the contact pressure was varied from nominally 200 psi to 2000 psi. The effective coefficient of friction was found to decrease abruptly from 5.5% to 3.6% between 500 and 1000 psi and then to increase gradually to 3.9% at 2000 psi as shown in Figure 4.4.

The tests reported in Table D.2, Test Series I, were repeated with nonparallel interfaces with results shown in Table D.3 and Figure 4.4. Both the magnitude and relationship to contact pressure of the effective coefficient of friction were influenced by the nonparallel interface. The effective coefficient of friction increased on average of

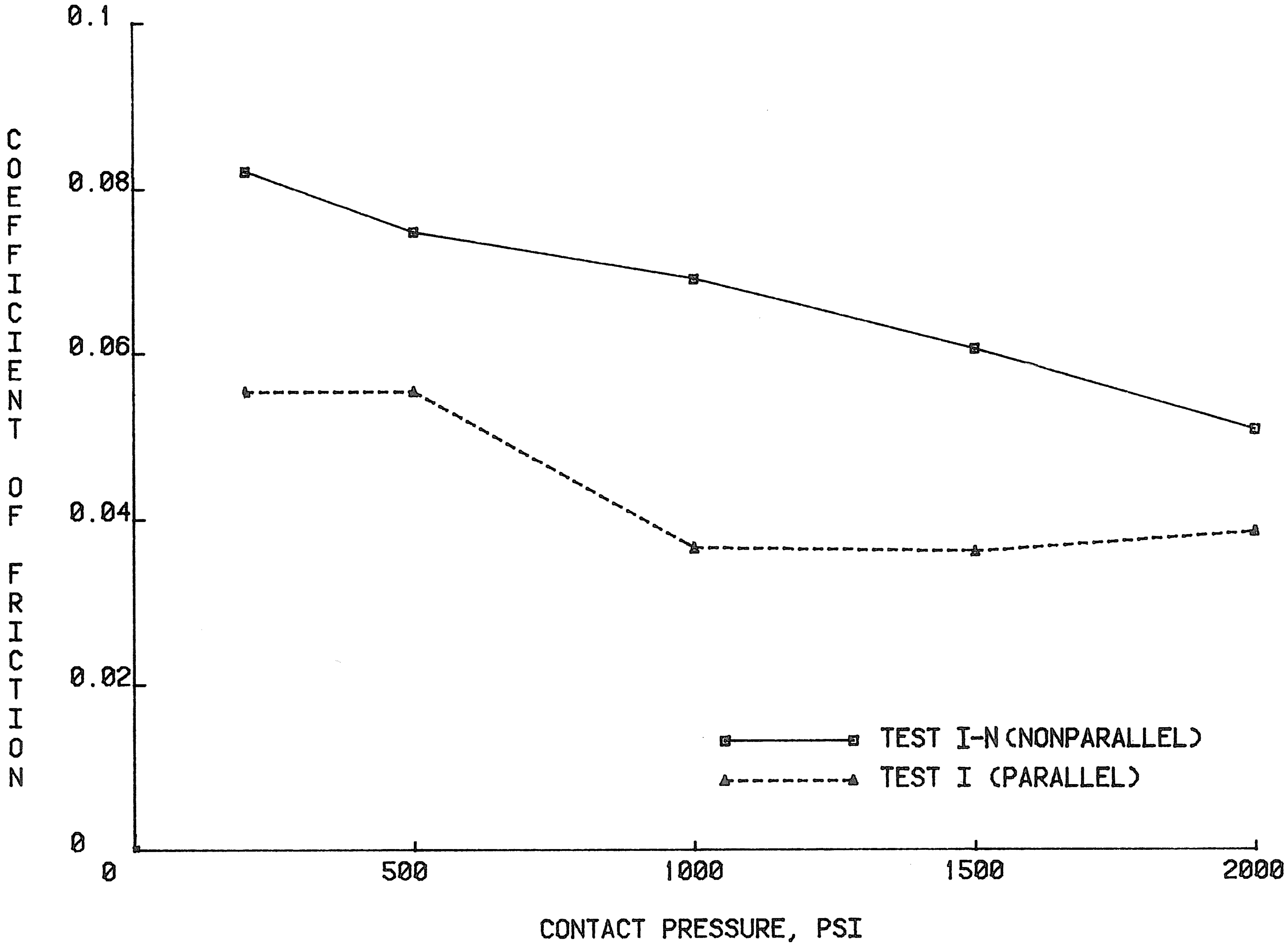


Figure 4.4 Coefficient of Friction vs Contact Pressure for Series I, Glass Filled TFE vs Glass Filled TFE

150% and the relationship became approximately linear with increasing contact pressure, Figure 4.4.

4.4 Tests with Mirror Finish Stainless Steel

Test Series II, III, III-A, and V were conducted with one TFE element and the other mirror finish stainless steel. Glass filled TFE was used for Series II and III, unfilled TFE for Series III-A and woven TFE for Series V. Both parallel and nonparallel interfaces were used in Test Series II and III. Results of all tests are shown in Tables D.4 through D.7 and D.9.

Figure 4.5 shows the effect of nonparallel interfaces on the effective coefficient of friction. The average increase is approximately 140% for a slope of 1/32 in. per ft. From Figure 4.5 it is clear that contact pressure has little effect on the coefficient of friction of glass filled sliding on mirror finish stainless steel.

Test Series III varied from Series II only in that the mirror finish stainless steel element was placed on the bottom and a glass filled TFE mechanically locked to a 1/4 in. thick stainless steel plate was used for the upper element. This type of TFE element has a significantly higher allowable contact pressure than does glass filled TFE bonded to carbon steel plates, 6000 psi versus 2000 psi. Only the parallel condition was tested. Results are shown in Table D.6 and comparison with Table D.4 shows little variation in effective coefficient of friction between Series II and Series III.

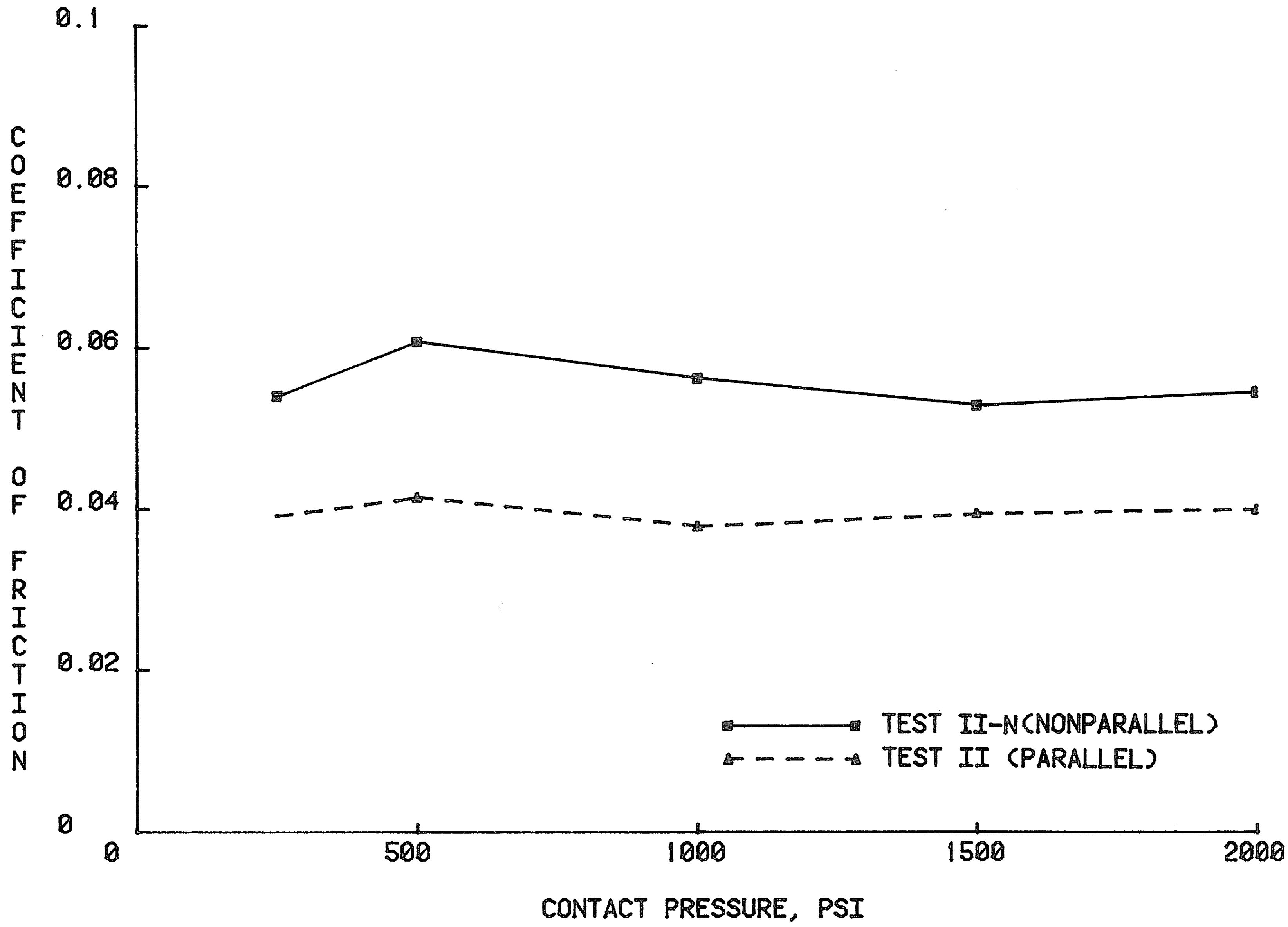


Figure 4.5 Coefficient of Friction vs Contact Pressure for Series II, Mirror Finish Stainless Steel vs Glass Filled TFE

Test Series III-A was identical to Series III except that the top element was unfilled TFE bonded to 1/4 in. thick carbon steel. The allowable contact pressure for this combination was 5000 psi. Results are shown in Table D.7 and comparison with Table D.6 shows a significant decrease in the effective coefficient of friction. The effective coefficient of friction using the unfilled TFE element is approximately 64% of that for the glass filled TFE element. Again, little effect was found when the contact pressure was varied from 1000 to 5000 psi.

Unfilled TFE fibers and glass fibers woven and bonded to 1/4 in. thick carbon steel were used as the top element in Test Series V. The bottom element was mirror finish stainless steel. The allowable contact pressure for this combination was 2000 psi. Results from 15 tests are shown in Table D.9. The effective coefficient of friction for this combination is essentially the same as for unfilled TFE versus mirror finish stainless steel, Series III-A, Table D.7.

4.5 Miscellaneous TFE Tests

Test Series IV-N was conducted using a glass filled TFE bonded to 1/4 in. thick carbon steel top element and glass filled TFE bonded to #10 gage carbon steel hot vulcanized to 3/4 in. thick 70 durometer AASHTO grade neoprene bottom element. This combination was tested in the nonparallel condition with a limiting contact pressure of 500 psi.

Results are shown in Table D.8. The effective coefficient of friction varied from 9.2% to 6.8% when the contact pressure was varied from nominally 250 psi to 500 psi.

Test Series VI used an unfinished stainless steel top element and glass filled TFE bottom element. The series was conducted in the nonparallel condition and the contact pressure was limited to 2000 psi. Results are shown in Table D.10. The effective coefficient of friction was found to be higher than for any other combination, as high as 12.3% and was found to vary considerably with contact pressure, 12.3% at 275 psi to 7.5% at 2000 psi.

Test Series VII-N was conducted with an unfilled TFE top element and a glass filled TFE bottom element in the nonparallel condition with a limiting contact pressure of 2000 psi. Results are shown in Table D.11. The effective coefficient of friction varied from 6.9% at 250 psi to 5.3% at 1500 psi to 5.6% at 2000 psi.

4.6 Summary of TFE Tests

A summary of all TFE expansion bearing tests is found in Table 4.4. The average effective coefficient of friction from at least three tests for each contact pressure in the range 250 to 2000 psi are shown. The highest values were found for the lowest contact pressure and the lowest for the highest contact pressure. Values varied from 12.3% to 2.0%.

Table 4.4 Summary of TFE Expansion Bearing Test Results

Test No.	Elements	Parallel	Average Effective Coefficient of Friction				
			250 psi	500 psi	1000 psi	1500 psi	2000 psi
I-60	Glass Filled TFE vs Glass Filled TFE	no	0.096	0.081	0.077	0.072	0.070
I	Glass Filled TFE vs Glass Filled TFE	yes	0.055	0.055	0.037	0.036	0.039
I-N	Glass Filled TFE vs Glass Filled TFE	no	0.082	0.075	0.069	0.061	0.051
II	Glass Filled TFE vs Mirror Finish Stainless Steel	yes	0.039	0.041	0.038	0.039	0.040
II-N	Glass Filled TFE vs Mirror Finish Stainless Steel	no	0.054	0.061	0.056	0.053	0.055
III	G.F. Mechanically Locked TFE vs Mirror Finish Stainless Steel	yes	-	-	0.040	-	0.042
III-A	Unfilled TFE vs Mirror Finish Stainless Steel	yes	-	-	0.026	-	0.024
IV-N	Glass Filled TFE vs Glass Filled TFE with Neoprene	no	0.092	0.068	-	-	-
V	Woven TFE and Glass vs Mirror Finish Stainless Steel	yes	0.025	0.022	0.026	0.020	0.020
VI-N	Glass Filled TFE vs Stainless Steel	no	0.123	0.102	0.082	0.077	0.075
VII-N	Unfilled TFE vs Glass Filled TFE	no	0.069	0.062	0.059	0.053	0.056

*Minimum of 3 tests.

A comparison of the results for the four most commonly used element combinations is shown in Figure 4.6: glass filled TFE versus stainless steel, glass filled TFE versus glass filled TFE, glass-filled TFE versus mirror finish stainless steel and unfilled TFE versus mirror finish stainless steel. The effective coefficient of friction decreases with increasing contact pressure for all combinations. The highest values were obtained for glass filled TFE versus stainless steel and the lowest for unfilled TFE versus mirror finish stainless steel. For contact pressure greater than 500 psi, the effective coefficient of friction varies linearly with contact pressure. It is noted that Figure 4.6 shows results for parallel and nonparallel conditions.

From Figure 4.2, it can be seen that when mirror finish stainless steel is used (Figure 4.2(a) and 4.2(b)) there is little difference in static and dynamic coefficients of friction. In other cases (Figures 4.2(c) and 4.2(d)), the static coefficient of friction is higher than the dynamic coefficient of friction.

Of all tests, the lowest effective coefficient of friction was found for the combination of unfilled TFE fibers and glass fibers woven and bonded to carbon steel versus stainless steel. However, when tests using the nonparallel condition were attempted, the woven element tended to "dig" into the opposite element causing damage and a very

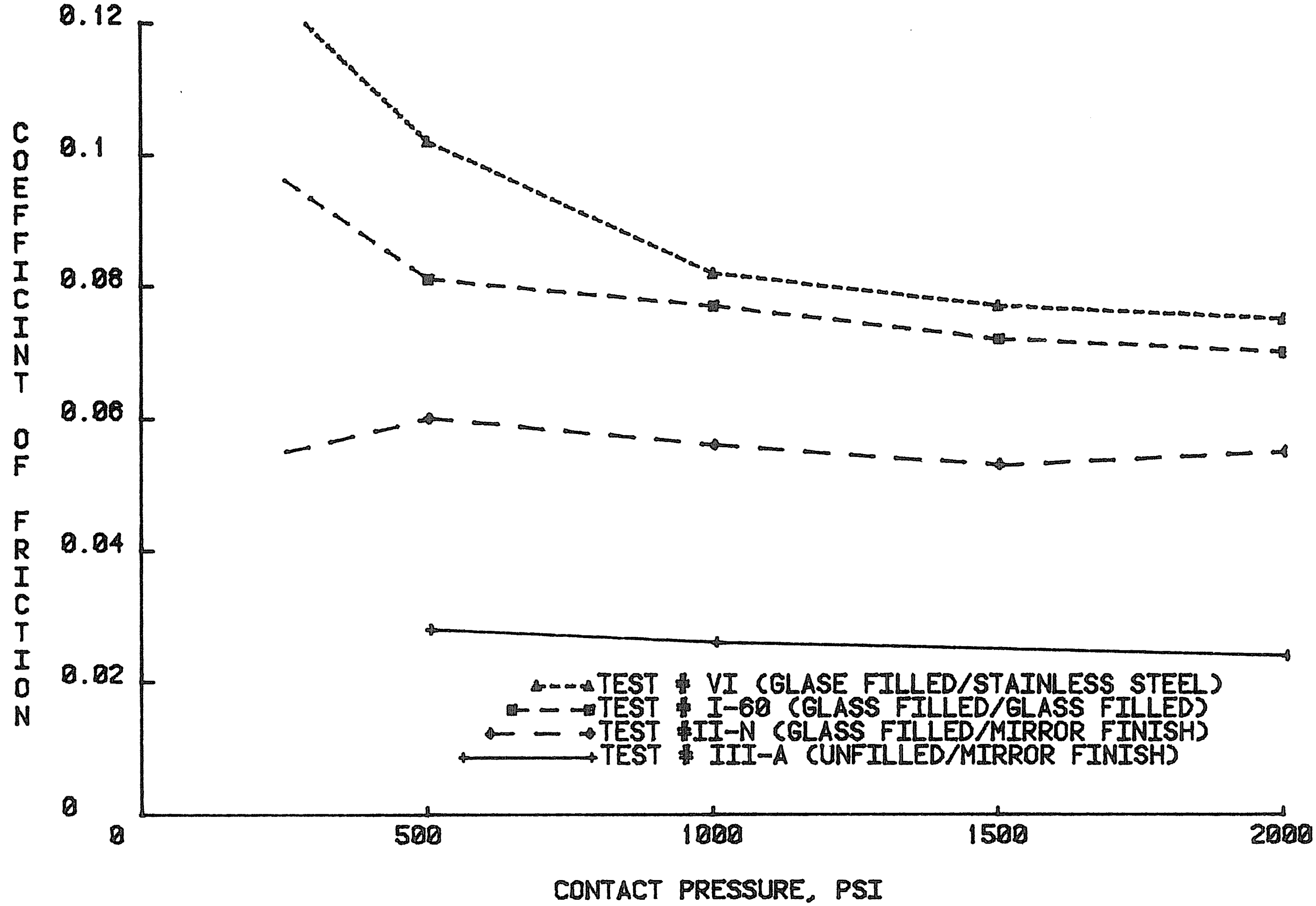
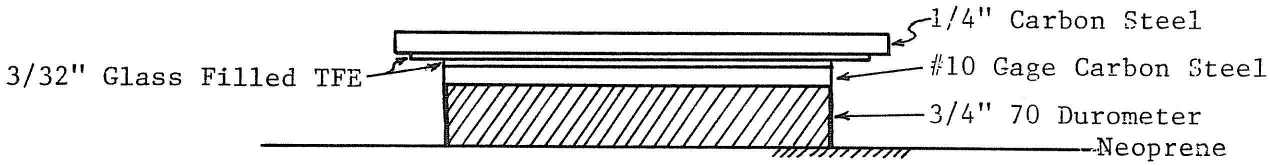


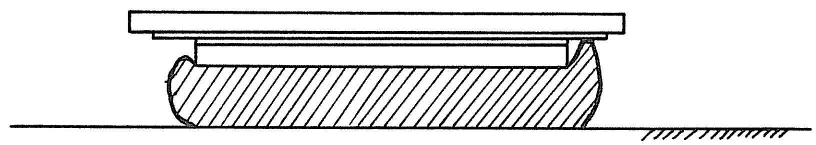
Figure 4.6 Comparison of Results for Various TFE Elements

high effective coefficient of friction. Consequently, this combination is not recommended unless a perfectly parallel interface can be guaranteed.

TFE bearings backed with rubber (neoprene) are commonly recommended for nonparallel surfaces. Test Series VI was conducted using 3/32 in. thick glass filled TFE bonded to #10 gage carbon steel which in turn was hot vulcanized to 3/4 in. thick 70 durometer AASHTO grade neoprene versus 3/32 in. thick glass filled TFE bonded to 1/4 in. thick carbon steel (Figure 4.7(a)). The bearing was tested at 250, 500 and 700 psi contact pressure. At 700 psi, the allowable contact pressure, the neoprene failed as shown in Figure 4.5(b), with a substantial increase in effective coefficient of friction. A possible cause was poor quality neoprene, which emphasizes a need for quality control procedures if this type of bearing is used.



(a) Before the Test



(b) At 700 psi Vertical Load

Figure 4.7 Detail of TFE Test #VI

CHAPTER V

SUMMARY

The results of this study show that the unturned pipe roller exhibits the lowest effective coefficient of friction of the four rolling devices tested. The effective coefficient of friction was found to be less than 0.5% for a clean 10 in. diameter pipe roller. The value increased to about 1% when tested in a rusted condition and to 5% when sand was spread over the lower bearing plate.

Tests using a turned roller showed the equivalent coefficient of friction to be a function of the amount of horizontal movement from the center line (median line). A geometric explanation was found and excellent agreement between predicted and measured results was achieved.

An effective coefficient of friction of 1% was found from tests using a clean pinned rocker. The value increased to 2% for a rusted condition. Both values are lower than a predicted value of 2.5% using a published criterion. The effective coefficient of friction for this rocker increased to 9% when sand was placed on the lower bearing plate.

Tests using a pintel rocker showed that fabrication inaccuracies, especially in the sole plate socket radius, can significantly affect the performance and effective coefficient of friction of the bearing. Tests with a socket plate socket radius slightly smaller than the rocker radius resulted in effective coefficient of friction values from 6.15% to 9.88%, as compared to 2.4% from published criteria. Tests with rusted bearing plates or with sand spread over the lower bearing plate showed significant increases in the effective coefficient of friction.

From test results from various TFE expansion bearings, the effective coefficient of friction was found to be higher and less consistent when both elements were TFE as opposed to one element being mirror finish stainless steel. The highest values of effective coefficient of friction were obtained for glass filled TFE versus stainless steel and the lowest for unfilled TFE versus mirror finish stainless steel. Tests using a nonparallel condition showed that the effective coefficient of friction increases about 50% for only 1/32" per foot (0.15°) slope.

The effective coefficient of friction, in general, was found to decrease with increasing contact pressure. It is noted that for unfilled TFE versus mirror finish stainless steel: 1) the lowest effective coefficient of friction of any combination was found, 2) the effective coefficient of friction did not significantly vary with contact pressure,

3) the same value was found for both static and dynamic conditions and 4) values were found to be consistent between tests.

REFERENCES

1. Long, J.E., "Bridge Bearings and Joints", Highways and Public Works, Vol. 46, No. 1825, December, 1978, pp. 9-20.

2. "Bridge Bearings", Bulletin No. 41, Transportation Research Board, Washington, D.C., 1977.

3. "Standard Specifications for Highway Bridges", American Association of State Highway Officials, Washington, D.C.

4. Long, J.E., "Bearings in Structural Engineering", Newnes-Butterworths, 1974.

5. Jacobson, Floyd K., "Investigation of Bridge Approach Spans to Poplar Street Bridge", A Preliminary Study, Illinois Department of Transportation, Bureau of Materials and Physical Research, October, 1975.

6. Chang, Fu-Kuel and Edward Cohen, "Long-Span Bridges: State-of-the-Art", Journal of the Structural Division, ASCE, Vol. 107, No. ST7, July 1981, pp. 1145-1213.

7. BS 153: Specification for Steel Girder Bridges, British Standards Institution, London.

8. Jacobson, Floyd K., "TFE Expansion Bearings for Highway Bridges", Physical Research Report No. 71, Illinois Department of Transportation, Bureau of Materials and Physical Research, April, 1977.

9. Taylor, M.E., "PTEF in Highway Bridge Bearings", Transport and Road Research Laboratory, Report LR491, 1975.

10. "Laboratory Corrosion Testing of Metals for the Process Industries", NACE Standard TM-01-69, National Association of Corrosion Engineers, 1969.

11. McDermott, Robert J., "Longitudinal Forces on Bridge Bearings", Special Report 58, Engineering Research and Development Bureau, New York State Department of Transportation, September, 1978.

12. Reynolds, Joseph C. and Jack H. Emanuel, "Thermal Stresses and Movements in Bridges", Journal of the Structural Division, ASCE, Vol. 100, No. ST1, January, 1974, pp. 63-78.

13. Loe, J.A., "Bridge Bearings: The Influence of Research", Highways and Public Works, Vol. 48, No. 1847, October, 1980, pp. 14-19.

14. Sestak, J.J., Jr., W.W. Aldridge and F.K. Fears, "A Comparative Study of Elastomeric Materials for Bridge Bearing Pads", University of Oklahoma Research Institute, Norman, Oklahoma, March 1967.

15. "Design of Neoprene Bearing Pads", E.I. Dupont Nemours and Company, Inc., Wilmington, Delaware, 1959.

APPENDIX A

PIPE-ROLLER TEST RESULTS

Table A.1 Pipe Roller - Unturned

Condition 1 - Clean Roller and Bearing Plate

Test No.	Normal Force (Kips)	Horizontal Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	27.14	0.08	0.0021	
2	26.0	0.06	0.0012	
3	27.29	0.08	0.0019	0.0017
4	52.35	0.31	0.0049	
5	51.92	0.18	0.0025	0.0037
6	77.08	0.32	0.0032	
7	76.73	0.43	0.0046	0.0039
8	101.96	0.69	0.0058	
9	102.64	0.50	0.0039	
10	100.39	0.43	0.0033	0.0043

Average Effective Coefficient of Friction = 0.0033

Standard Deviation = 0.0014

Coefficient of Variation = 0.000002

Table A.2 Pipe Roller - Unturned

Condition 2 - Rusted Bearing Plate

Test No.	Normal Force (Kips)	Horizontal Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	95.90	0.91	0.0085	
2	99.82	0.92	0.0082	
3	102.43	0.97	0.0085	0.0084
4	72.63	0.54	0.0065	
5	76.78	0.51	0.0056	
6	77.86	0.58	0.0065	0.0062
7	52.57	0.35	0.0057	
8	51.39	0.43	0.0073	
9	50.54	0.41	0.0072	0.0067
10	26.94	0.20	0.0065	
11	26.10	0.17	0.0056	
12	24.00	0.18	0.0063	0.0061

Average Effective Coefficient of Friction = 0.0069

Standard Deviation = 0.0010

Coefficient of Variation = 0.0000011

Table A.3 Pipe Roller - Unturned

Condition 3 - Sand on Bearing Plate

Test No.	Normal Force (Kips)	Horizontal Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	95.15	2.66	0.0269	
2	99.08	3.79	0.0373	
3	99.49	3.98	0.0390	0.0344
4	78.74	4.10	0.0511	
5	78.89	3.85	0.0478	
6	79.69	4.18	0.0515	
7	76.34	2.72	0.0347	0.0463
8	52.82	1.83	0.0336	
9	60.42	3.55	0.0578	
10	52.33	1.34	0.0244	
11	52.16	1.28	0.0235	0.0287
12	26.64	0.65	0.0236	
13	26.43	0.62	0.0224	
14	26.40	0.58	0.0210	0.0223

Average Effective Coefficient of Friction = 0.0329

Standard Deviation = 0.0120

Coefficient of Variation = 0.0001

Table A.4 Pipe Roller - Turned

Condition 1 - Clean Roller and Bearing Plate

Test No.	Distance from Centerline (in)	Normal Force (Kips)	Measured Resisting Force (Kips)	Predicted Resisting Force (Kips)	Percent Difference
1	0.0	50	0.0	0	0
2	0.5	50	1.36	1.34	+1.5
3	1.0	50	2.15	2.68	-19.8
4	1.5	50	5.09	4.02	+26.6
5	0.0	100	0.0	0	0
6	0.5	100	2.65	2.68	-1.1
7	1.0	100	3.75	5.36	-30.0
8	1.5	100	9.11	8.04	+13.3
9	0.0	150	0.0	0	0
10	0.5	150	3.84	4.02	-4.5
11	1.0	150	5.12	8.04	-36.3
12	1.5	150	12.31	12.05	+2.2

Table A.5 Pipe Roller - Turned

Condition 2 - Sand on Bearing Plate

Test No.	Distance from Centerline (in)	Normal Force (Kips)	Measured Horizontal Force (Kips)	Predicted Resisting Force (Kips)	Percent Difference
1	0.70	181.08	23.43	6.79	245.1
2	0.97	185.32	24.14	9.63	150.7
3	0.43	161.86	11.04	3.73	196.0
4	0.24	100.55	4.88	1.29	278.3
5	0.22	100.12	4.35	1.18	268.6
6	0.21	99.97	4.15	1.12	270.5
7	0.30	57.01	3.12	0.92	239.1
8	0.30	56.52	3.10	0.91	240.7
9	0.29	56.16	3.10	0.87	256.3

APPENDIX B

PINNED ROCKER SHOE TEST RESULTS

Table B.1 Pinned Rocker Shoe

Condition 1 - Clean and Unlubricated

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient	Average Coeff. of Friction
1	201.85	2.73	0.0115	
2	200.58	2.78	0.0118	0.0117
3	55.37	0.69	0.0114	
4	57.22	0.72	0.0116	0.0115
5	77.94	0.95	0.0112	
6	76.14	0.88	0.0107	0.0109
7	101.05	0.95	0.0084	
8	102.26	1.17	0.0105	0.0095
9	128.58	1.27	0.0088	
10	125.47	1.02	0.0071	0.0080
11	147.65	1.54	0.0095	
12	150.08	1.43	0.0085	0.0090
13	199.69	1.78	0.0078	
14	203.12	2.10	0.0093	0.0086
15	222.17	2.08	0.0084	
16	220.01	1.94	0.0078	0.0081

Average Effective Coefficient of Friction = 0.0099

Standard Deviation = 0.00137

Coefficient of Variation = 0.000002

Table B.2 Pinned Rocker Shoe

Condition 2 - Rusted

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient of Friction	Average Coeff. of Friction
1	202.32	4.35	0.0205	
2	201.58	4.64	0.0220	
3	200.15	4.71	0.0225	0.0217
4	177.16	3.58	0.0192	
5	176.24	3.73	0.0201	
6	174.53	2.85	0.0153	0.0182
7	151.45	2.81	0.0176	
8	150.72	2.74	0.0172	
9	150.15	3.35	0.0214	0.0187
10	126.14	2.12	0.0158	
11	125.97	1.86	0.0138	
12	125.84	2.01	0.0150	0.0149
13	101.56	1.99	0.0186	
14	101.19	1.95	0.0182	
15	102.15	2.90	0.0274	0.0214
16	75.65	1.18	0.0146	
17	75.45	0.91	0.0110	
18	76.17	1.31	0.162	0.0139
19	51.14	0.99	0.0184	
20	50.55	0.96	0.0181	
21	52.04	1.73	0.0323	0.0229
22	199.20	3.33	0.0157	
23	200.84	3.57	0.0168	0.0163

Average Effective Coefficient of Friction = 0.0185

Standard Deviation = 0.00310

Coefficient of Variation = 0.00001

Table B.3 Pinned Rocker Shoe

Condition 3 - Sand on Bearing Plate

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient of Friction	Average Coeff. of Friction
1	197.41	20.79	0.1043	
2	199.17	18.37	0.0912	
3	198.60	18.50	0.0921	0.0959
4	147.25	6.66	0.0442	
5	153.18	13.21	0.0852	
6	155.09	15.82	0.1010	0.0768
7	109.25	10.17	0.0912	
8	109.08	10.10	0.0916	
9	109.21	10.21	0.0925	0.0918
10	58.96	5.88	0.0988	
11	58.42	5.36	0.0908	
12	58.38	5.32	0.0902	0.0933

Average Effective Coefficient of Friction = 0.0895

Standard Deviation = 0.0074

Coefficient of Variation = 0.000055

APPENDIX C

PINTLE ROCKER TEST RESULTS

Table C.1 Pintle Rocker Bearing I

Condition 1 - As Received

Test No.	Normal Force Kips	Friction Force Kips	Effective Coefficient of Friction	Remarks
1	26.80	2.70	0.0988	
2	25.11	1.93	0.0747	
3	26.29	2.61	0.0971	
4	27.50	2.53	0.0898	
5	17.84	1.27	0.0690	
6	18.67	1.56	0.0815	
7	47.36	3.10	0.0635	
8	51.46	4.86	0.0924	
9	50.77	3.23	0.0615	
10	74.76	6.54	0.0854	
11	79.55	6.63	0.0813	
12	98.80	6.63	0.0651	
13	99.44	7.61	0.0746	
14	123.47	8.24	0.0647	
15	122.03	9.11	0.0727	
16	143.15	10.06	0.0683	
17	144.93	11.74	0.0790	
18	173.46	13.15	0.0736	
19	174.09	16.31	0.0917	
20	199.82	13.63	0.0662	
21	197.73	15.47	0.0762	
22	231.70	16.82	0.0706	
23	227.16	16.09	0.0688	
24	221.04	15.05	0.0661	

Table C.2 Effect on Initial Position - Pintle Rocker Bearing I

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient of Friction	Distance from Center-line of Rocker (in.)
1	53.59	3.52	0.0637	-1.15
2	52.00	3.59	0.0670	+0.05
3	53.78	3.92	0.0709	+0.87
4	75.40	4.69	0.0601	-1.00
5	62.29	4.13	0.0643	+0.05
6	78.45	5.71	0.0708	+0.40
7	102.4	-	-	-2.10
8	104.28	7.66	0.0714	+0.04
9	103.42	7.19	0.0676	+1.11
10	125.31	8.81	0.0683	-1.22
11	126.06	9.10	0.0702	+0.07
12	124.55	8.18	0.0637	+1.5
13	149.12	10.01	0.0651	-1.31
14	150.40	11.71	0.0758	+0.19
15	151.29	10.50	0.0674	+0.67
16	174.69	10.15	0.0561	-0.90
17	175.58	12.45	0.0689	_0.04
18	174.38	13.86	0.0775	+0.32
19	202.90	12.97	0.0619	-1.08
20	202.12	11.92	0.0579	-0.11
21	201.30	17.26	0.0838	+0.32
22	227.59	17.11	0.0732	-1.64
23	229.85	17.89	0.0758	+0.08
24	228.29	14.12	0.0598	+0.80

Table C.3 Effect on Initial Position -~~Pintle~~ Rocker Bearing II

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient of Friction	Distance from Center-line of Rocker (in.)
1	201.97	14.28	0.0727	+0.23
2	203.83	16.60	0.0794	+0.38
3	204.55	16.29	0.0776	+0.54
4	52.28	1.74	0.0313	-0.53
5	49.30	2.43	0.0474	-0.27
6	48.94	3.05	0.0602	+0.01
7	54.84	3.01	0.0529	+0.20
8	54.46	3.74	0.0666	+0.39
9	55.44	4.01	0.0703	+0.61
10	75.28	2.94	0.0371	-0.54
11	70.09	4.27	0.0590	+0.21
12	77.21	5.45	0.0686	+0.46
13	99.80	4.50	0.0431	-0.53
14	102.15	6.55	0.0622	+0.03
15	100.52	6.10	0.0586	+0.02
16	105.00	7.57	0.0701	+0.87
17	124.44	5.76	0.0443	-0.43
18	125.88	8.24	0.0639	+0.03
19	130.82	10.38	0.0773	+0.68
20	145.84	8.71	0.0577	-0.27
21	149.89	10.49	0.0680	+0.02
22	151.71	12.33	0.0793	+0.36
23	172.70	9.11	0.0508	-0.46
24	170.09	12.24	0.0700	+0.01
25	174.86	12.45	0.0692	+0.84
26	196.18	10.34	0.0502	-0.52
27	195.27	14.43	0.0719	+0.20
28	202.38	16.36	0.0789	+0.46
29	224.52	12.21	0.0524	-0.46
30	227.75	16.33	0.0697	+0.13
31	230.00	17.14	0.0725	+0.48

Table C.4 Modified Sole Plate Test - Pintle Bearing I

Condition 1

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient	Average Effective Coeff. of Friction for Each Loading
1	199.25	9.81	0.0482	
2	201.73	9.82	0.0477	
3	200.82	11.14	0.0545	0.0501
4	174.96	9.11	0.0511	
5	173.98	7.82	0.0439	
6	177.91	7.50	0.0412	0.0454
7	151.84	7.16	0.0461	
8	150.02	5.36	0.0348	
9	149.58	6.61	0.0432	0.0414
10	100.19	2.33	0.0222	
11	98.07	3.70	0.0367	
12	98.83	4.73	0.0469	0.0353
13	75.40	2.72	0.0351	
14	75.25	3.45	0.0449	
15	76.30	3.92	0.0504	0.0435

Average Effective Coefficient of Friction = 0.0431

Standard Deviation = 0.0049

Coefficient of Variation = 0.000024

APPENDIX D

TFE EXPANSION BEARING TEST RESULTS

Table C.6 Pintle Rocker Bearing I

Condition 3 - Sand on Bearing Plate

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction for Each Loading
1	190.58	23.22	0.1208	
2	188.45	26.79	0.1411	
3	205.76	26.65	0.1285	0.1301
4	147.18	19.12	0.1289	
5	149.32	19.76	0.1313	
6	148.98	19.63	0.1307	0.1303
7	101.91	13.56	0.1321	
8	101.74	13.38	0.1305	
9	101.64	13.35	0.1304	0.1310
10	53.87	7.32	0.1349	
11	53.40	7.15	0.1329	
12	53.69	7.19	0.1329	0.1336

Average Effective Coefficient of Friction = 0.1313

Standard Deviation = 0.0014

Coefficient of Variation = 0.000002

Table C.5 Modified Sole Plate Test-Pintle Bearing II

Condition 2 - Rusted

Test No.	Normal Force (Kips)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction for Each Loading
1	199.96	9.60	0.0470	
2	201.06	9.08	0.0441	
3	200.53	10.10	0.0494	0.0468
4	179.08	8.38	0.0458	
5	178.89	8.67	0.0475	
6	179.12	10.00	0.0548	0.0494
7	151.78	6.90	0.0445	
8	151.36	7.77	0.0503	
9	152.08	7.90	0.0510	0.0486
10	100.06	4.82	0.0471	
11	100.23	5.10	0.0499	
12	100.83	5.52	0.0537	0.0502
13	76.02	2.84	0.0364	
14	75.25	3.74	0.0487	
15	75.81	3.89	0.0503	0.0451

Average Effective Coefficient of Friction = 0.0480

Standard Deviation = 0.0018

Coefficient of Variation = 0.0000039

Table D.1 TFE Expansion Bearing Test Series I-20, I-40, I-60 and I-100

Glass Filled TFE vs. Glass Filled TFE

(Nonparallel Interface)

Test Series	Contact Area (in^2)	Effective Coefficient of Friction				
		250 psi	500 psi	1000 psi	1500 psi	2000 psi
I-20	20	0.1123[1]	0.0852	0.0868	0.0757	0.0677
		0.0901[2]	0.0765	0.0688	0.0672	0.0643
		0.0979[3]	0.0818	0.0800	0.0703	0.0664
I-40	40	0.1205	0.0963	0.0818	0.0794	0.0739
		0.0922	0.0779	0.0671	0.0641	0.0581
		0.1033	0.0859	0.0731	0.0692	0.0655
I-60	60	0.1254	0.0865	0.0897	0.0772	0.0826
		0.0796	0.0768	0.0713	0.0676	0.0635
		0.0964	0.0813	0.0774	0.0718	0.0696
I-100	100		0.0877	0.0892	0.0815	0.0699
			0.0817	0.0731	0.0636	0.0587
			0.0850	0.0785	0.0701	0.0641

1= High value

2= Low value

3= Average

Table D.2 TFE Expansion Bearing Test Series I

Glass Filled TFE vs Glass Filled TFE

(Parallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	4.22	20.5	206	0.24	0.0550	
2	3.83	20.5	187	0.15	0.0384	
3	4.00	20.5	195	0.37	0.0916	
4	4.03	20.5	197	0.15	0.0365	0.0554
5	11.97	20.5	584	0.97	0.0796	
6	12.04	20.5	587	0.52	0.0419	
7	11.21	20.5	547	0.51	0.0442	
8	11.04	20.5	539	0.63	0.0559	0.0554
9	20.99	20.5	1024	0.77	0.0355	
10	20.29	20.5	990	0.76	0.0362	
11	20.29	20.5	990	0.63	0.0303	
12	20.73	20.5	1011	0.93	0.0438	0.0365
13	32.20	20.5	1571	1.17	0.0353	
14	31.78	20.5	1550	1.02	0.0310	
15	31.25	20.5	1524	1.02	0.0316	
16	31.27	20.5	1525	1.48	0.0464	0.0361
17	41.90	20.5	2044	1.55	0.0360	
18	42.00	20.5	2049	1.28	0.0294	
19	41.50	20.5	2024	1.59	0.0373	
20	41.38	20.5	2018	1.50	0.0353	
21	40.84	20.5	1992	1.70	0.0407	
22	41.38	20.5	2018	2.21	0.0523	0.0385

Table D.3 TFE Expansion Bearing Test Series I-N

Glass Filled TFE vs. Glass Filled TFE

(Nonparallel Interfaces)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	5.66	20.5	276	0.43	0.0750	
2	5.55	20.5	271	0.45	0.0809	
3	5.51	20.5	269	0.43	0.0771	
4	6.09	20.5	297	0.59	0.0953	0.0821
5	11.37	20.5	554	0.76	0.0655	
6	10.99	20.5	536	0.73	0.0656	
7	11.11	20.5	542	1.09	0.0968	
8	11.05	20.5	539	0.79	0.0707	0.0747
9	22.03	20.5	1075	1.50	0.0670	
10	16.95	20.5	827	1.29	0.0750	
11	16.82	20.5	820	1.19	0.0696	
12	16.79	20.5	819	1.10	0.0644	0.0690
13	32.23	20.5	1572	2.20	0.0671	
14	31.85	20.5	1554	1.67	0.0516	
15	31.84	20.5	1553	2.18	0.0674	
16	31.67	20.5	1545	1.80	0.0557	0.0605
17	42.01	20.5	2049	2.37	0.0555	
18	41.45	20.5	2022	2.03	0.0479	
19	41.64	20.5	2031	2.08	0.0489	
20	41.30	20.5	2014	2.14	0.0508	0.0508

Table D.4 TFE Expansion Bearing Test Series II

Mirror Finish Stainless Steel (Top) vs Glass Filled TFE (Bottom)

(Parallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	9.36	44.55	210	0.39	0.0408	
2	10.04	44.55	225	0.42	0.0410	
3	9.77	44.55	219	0.38	0.0381	
4	9.73	44.55	218	0.36	0.0365	0.0391
5	22.71	44.55	510	1.16	0.0502	
6	22.39	44.55	503	0.91	0.0395	
7	23.03	44.55	517	0.87	0.0369	
8	23.49	44.55	527	0.94	0.0391	0.0414
9	44.85	44.55	1007	1.78	0.0388	
10	44.96	44.55	1009	1.72	0.0373	
11	44.94	44.55	1009	1.71	0.0370	
12	44.70	44.55	1003	1.75	0.0381	0.0378
13	67.96	44.55	1526	2.86	0.0411	
14	67.62	44.55	1518	2.63	0.0378	
15	67.47	44.55	1515	2.72	0.0393	
16	67.17	44.55	1508	2.71	0.0393	0.0394
17	89.61	44.55	2011	3.59	0.0391	
18	89.00	44.55	1998	3.64	0.0399	
19	90.10	44.55	2022	3.77	0.0408	
20	89.82	44.55	2016	3.65	0.0396	0.0399

Table D.5 TFE Expansion Bearing Test Series II-N

Mirror Finish Stainless Steel (Top) vs Glass Filled TFE (Bottom)

(Nonparallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	9.79	44.5	220	0.65	0.0654	
2	9.88	44.5	222	0.47	0.0468	
3	9.39	44.5	211	0.46	0.0476	
4	9.56	44.5	215	0.55	0.0560	0.0540
5	22.59	44.5	507	1.61	0.0702	
6	22.79	44.5	512	1.29	0.0554	
7	22.79	44.5	512	1.25	0.0540	
8	22.41	44.5	503	1.44	0.0634	0.0608
9	44.29	44.5	994	2.94	0.0654	
10	44.15	44.5	991	2.19	0.0486	
11	43.62	44.5	979	2.54	0.0573	
12	44.33	44.5	995	2.42	0.0536	0.0562
13	66.86	44.5	1501	4.04	0.0595	
14	66.65	44.5	1496	3.45	0.0508	
15	67.68	44.5	1519	3.74	0.0543	
16	66.69	44.5	1497	3.20	0.0469	0.0529
17	88.79	44.5	1993	4.66	0.0514	
18	89.10	44.5	2000	4.40	0.0484	
19	89.21	44.5	2003	5.93	0.0655	
20	88.79	44.5	1993	4.76	0.0526	0.0545

Table D.6 TFE Expansion Bearing Test Series III

Glass Filled, Mechanically Locked TFE (Top) vs Mirror Finish Stainless Steel (Bottom)

(Parallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	51.83	51.80	1001	2.33	0.0440	
2	53.43	51.80	1031	2.96	0.0375	
3	53.05	51.80	1024	2.16	0.0398	0.0404
4	102.38	51.80	1976	5.01	0.0479	
5	101.44	51.80	1958	4.22	0.0406	
6	101.04	51.80	1951	3.92	0.0378	0.0421
7	153.58	51.80	2967	6.08	0.0386	
8	155.68	51.80	3005	7.01	0.0441	
9	155.27	51.80	2997	6.08	0.0382	0.0403
10	200.06	51.80	3862	7.93	0.0386	
11	202.11	51.80	3902	7.88	0.0380	
12	202.52	51.80	39.10	7.93	0.0382	0.0383
10-1	128.05	31.90	4014	4.26	0.0322	
11-1	128.28	31.90	4021	4.24	0.0321	
12-1	128.68	31.90	4034	3.94	0.0296	0.0313
13	158.92	31.90	4982	5.24	0.0320	
14	160.03	31.90	5017	5.00	0.0302	
15	159.95	31.90	5014	4.64	0.0280	0.0301
16	190.34	31.90	5967	6.00	0.0305	
17	189.50	31.90	5940	6.31	0.0323	
18	189.49	31.90	5940	5.81	0.0297	0.0308

Table D.7 TFE Expansion Bearing Test Series III-A
Unfilled TFE (Top) vs Mirror Finish Stainless Steel (Bottom)
(Parallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	45.93	45.00	1021	1.30	0.0274	
2	45.42	45.00	1009	1.19	0.0251	
3	44.59	45.00	991	1.12	0.0242	0.0256
4	90.58	45.00	2013	2.26	0.0239	
5	89.58	45.00	1991	2.38	0.0256	
6	89.48	45.00	1989	2.03	0.0217	0.0237
7	135.50	45.00	3011	4.23	0.0302	
8	135.22	45.00	3005	3.26	0.0232	
9	136.03	45.00	3023	2.60	0.0181	0.0238
10	180.21	45.00	4005	3.72	0.0197	
11	180.59	45.00	4013	3.88	0.0205	
12	180.27	45.00	4006	4.15	0.0220	0.0207
13	225.11	45.00	5003	5.46	0.0232	
14	225.94	45.00	5021	4.96	0.0209	
15	225.03	45.00	5001	4.92	0.0209	0.0217

Table D.8 TFE Expansion Bearing Test Series IV-N

Glass Filled TFE (Top) vs Glass Filled TFE with 3/4" 70 D. Neoprene

(Nonparallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	11.27	44.55	253	1.10	0.0971	
2	11.92	44.55	267	1.07	0.0890	
3	11.97	44.55	269	0.93	0.0765	
4	11.85	44.55	266	1.24	0.1034	0.0915
5	23.12	44.55	519	1.59	0.0688	
6	22.40	44.55	503	1.57	0.0693	
7	22.86	44.55	513	1.57	0.0677	
8	23.07	44.55	518	1.59	0.0679	0.0684

Table D.9 TFE Expansion Bearing Test Series V

Woven TFE (Top) vs Mirror Finish (Bottom)

(Parallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	12.5	44.10	284	0.45	0.0347	
2	12.48	44.10	283	0.27	0.0208	
3	12.45	44.10	282	0.27	0.0207	0.0254
4	23.33	44.10	529	0.60	0.0248	
5	22.97	44.10	521	0.50	0.0209	
6	22.95	44.10	520	0.47	0.0196	0.0218
7	44.62	44.10	1012	1.34	0.0289	
8	43.62	44.10	989	0.93	0.0204	
9	43.33	44.10	983	1.28	0.0284	0.0259
10	66.78	44.10	1514	1.82	0.0262	
11	66.93	44.10	1518	1.23	0.0174	
12	66.75	44.10	1514	1.24	0.0175	0.0204
13	89.32	44.10	2025	2.14	0.0229	
14	88.51	44.10	2007	1.80	0.0193	
15	88.41	44.10	2005	1.54	0.0164	0.0195

Table D.10 TFE Expansion Bearing Test Series VI-N
Stainless Steel (Top) vs Glass Filled TFE (Bottom)
(Nonparallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	16.98	60.00	283	-	-	
2	16.23	60.00	270	2.03	0.1241	
3	16.95	60.00	282	2.08	0.1219	0.1230
4	30.23	60.00	504	5.24	0.1061	
5	31.03	60.00	517	3.59	0.1146	
6	30.41	60.00	507	2.58	0.0837	0.1015
7	60.09	60.00	1002	5.10	0.0839	
8	60.19	60.00	1003	5.19	0.0853	
9	60.14	60.00	1002	4.64	0.0761	0.0818
10	89.41	60.00	1490	6.92	0.0764	
11	89.71	60.00	1495	7.08	0.0779	
12	89.45	60.00	1491	7.07	0.0780	0.0774
13	119.81	60.00	1997	9.15	0.0754	
14	119.35	60.00	1989	9.17	0.0759	
15	119.10	60.00	1985	8.72	0.0722	0.0745

Table D.11 TFE Expansion Bearing Test Series VII-N
Unfilled TFE (Top) vs Glass Filled (Bottom)
(Nonparallel Interface)

Test No.	Normal Force (Kips)	Contact Area (in^2)	Contact Pressure (Psi)	Friction Force (Kips)	Effective Coefficient of Friction	Average Effective Coeff. of Friction
1	11.68	44.10	265	0.75	0.0634	
2	11.56	44.10	262	0.75	0.0638	
3	11.67	44.10	265	0.93	0.0785	0.0685
4	22.76	44.10	516	1.39	0.0603	
5	22.29	44.10	506	1.40	0.0618	
6	22.16	44.10	503	1.42	0.0631	0.0617
7	43.62	44.10	989	2.66	0.0601	
8	44.50	44.10	1009	2.62	0.0579	
9	43.81	44.10	993	2.56	0.0574	0.0585
10	66.29	44.10	1503	4.48	0.0666	
11	66.89	44.10	1517	3.23	0.0473	
12	66.33	44.10	1504	2.99	0.0440	0.0526
13	88.15	44.10	1999	5.95	0.0665	
14	89.64	44.10	2033	4.31	0.0470	
15	89.95	44.10	2040	5.02	0.0548	0.0561