# FEARS STRUCTURAL ENGINEERING LABORATORY

## EFFECTIVE COEFFICIENT OF FRICTION OF STEEL BRIDGE BEARINGS


by

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# ABSTRACT


## EFFECTIVE COEFFICIENT OF FRICTION OF BRIDGE BEARINGS

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. 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.

From the tests it was found that unturned pipe rollers exhibit the lowest effective coefficient of friction of the four rolling devices tested. 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. A geometric explanation was devised and excellent agreement between predicted and measured results was achieved.

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.

In all cases, tests with rusted bearing plates or with sand spread over the lower bearing plate showed significant increases in the effective coefficient of friction.

## INTRODUCTION

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 was 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, $\mu_{eff}$, is defined as:

$$\mu_{eff} = \frac{F}{N} \tag{1}$$

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 $\mu_{eff}$ is calculated.

## BACKGROUND

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. 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 parts to freeze, etc.), nothing to corrode, low cost and almost no maintenance is required. However, 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 accomodate 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(a). The load carrying capacity of the roller is a function of its radius and can be found from the following formula [3]:

For diameters up to 25 inches

$$P = \frac{F_y - 13{,}000}{20{,}000} \; 600 \; d \tag{2}$$

and for diameters from 25 to 125 inches

$$P = \frac{F_y - 13{,}000}{20{,}000} \; 3{,}000 \; \sqrt{d} \tag{3}$$

where P = allowable bearing in pounds per linear inch, d = outside diameter of the roller in inches, and $F_y$ = minimum yield point in tension of steel in the roller or bearing plate, whichever is the smaller in pounds per square inch. For a roller diameter of 12 in. and a length of 12 in., 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(b). This type of roller, which in this paper is called a "turned-roller", has geometrical properties which cause a high horizontal resistance. The equivalent effective coefficient of friction of a turned-roller is a function of the amount of movement.

Rollers can be used in combination to increase load carrying capacity, as shown in Figure 1(c). Roller nests only work well when they are clean, hence, maintenance is required. Furthermore, this type of bearing is relatively expensive.

Several different types of rockers are used as expansion bearings, for instance, the segmental rocker shown in Figure 2(a), the pinned rocker in Figure 2(b) and the pintle rocker shown in Figure 2(c). The double-segmented rocker shown in Figure 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 (equivalent friction force) would be tremendous for large movements.

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-connection details can accumulate rust between the contact surfaces of the pin and the housing. Resulting increased horizontal forces can cause 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. Jacobson concluded that the use of pin-connected details subjected to large rotations and utilizing untreated, corrosive mild steels should be avoided.

Chang and Cohen [6] in "Long-Span Bridges: State-of-the-Art" have suggested coefficients of friction of 0.2 for steel bearing on steel, 0.1 for steel bearing on self-lubricating bronze plate and 0.06 for polytetrafluoroethylene (PTFE) on PTFE or stainless steel. For rocker type bearings, they suggest that the force be calculated based on a 20% friction coefficient but reduced in proportion to the radii of the pin and rocker as shown in Figure 4.

The British Standard BS153 (7) specifies the coefficients of friction for sliding bearing as 0.25 for steel on steel or cast iron, and 0.15, for steel on copper alloy. The coefficient of friction with one or two rollers is taken as 0.01.

Jacobson (8) has conducted experimental work to investigate the potential use of TFE as a sliding surface. He concluded that the TFE bearings are suitable for use as highway bridge bearings. A substantial increase in the coefficient of friction for filled TFE was found after 7000 cycles of testing.

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. The coefficient of friction decreased with higher compressive stress across the bearing, but increased slightly at lower temperatures.

## 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 several types of standard ODOT bearings under several conditions. Mechanical bearings types were as follows:

-Typical single roller bearing (Figure 1(a)).

-Typical single turned-roller bearing (Figure 1(b)).

-Typical pinned rocker shoe (Figure 2(b)).

-Typical pintle rocker bearing (Figure 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 debris on the lower bearing plates. The unturned pipe-roller, turned pipe roller and pinned rocker shoe bearings used in the study were new bearings. The pintle rocker bearings were removed from a bridge prior to testing.

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

## 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 Figure 5. 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 microcomputer system.

The test set-up was erected on the reaction floor inside the Fears Structural Engineering Laboratory at the University of Oklahoma. The set-up was erected directly over two W36 beams spaced 8 ft. apart and consisted of three parts: 1) An H-frame which was designed for 250,000 lbs. maximum vertical reaction and which supported the hydraulic ram, 2) A triangle frame which was designed for 55,000 lbs. maximum horizontal reaction and which supported the closed-loop hydraulic testing system, and 3) A W33x130x15 ft. 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. 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 5. 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.

## 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 a calibrated 300,000 lb. capacity load cell; the horizontal force was measured using a calibrated 100,000 lb. capacity load cell; and the horizontal movement (girder movement) was measured 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 micro-processor. 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.

## TEST PROCEDURES

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 microprocessor.

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 force and horizontal movement.

To stimulate in-situ conditions, the steel bearings were subjected to rusting and debris environments. To achieve the rusting condition, the bearings were placed inside a closed bucket in an 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 where temperatures varied from $25^{o}F$ to $80^{o}F$.

To achieve the debris environment, an approximately 1/8 in. thick layer of graded sand was spread on the lower bearing plate. The sand, supplied by ODOT, was obtained by vacuuming areas near in-place bridge bearings.

## TEST RESULTS

The details of the test data for this project have been given in Reference 10 and will not be repeated herein. Essential results and conclusions follow.

Unturned Pipe-Roller (Single Roller). A 10 in. diameter unturned, stiffened, painted pipe-roller as shown in Figure 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 2. Typical horizontal force versus horizontal deflection plots for Conditions 1 to 3, are shown in Figure 6. For a perfectly rigid system, horizontal displacement would not take place until the rolling frictional resistance is overcome. The initial horizontal motion shown in Figure 6 (and all subsequent similar plots) is from the elastic deformation of the test fixtures.

The results for all tests are plotted in Figure 7 as effective coefficient of friction versus normal force. The straight lines shown are the result of regression analyses conducted for each condition.

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 0.14% 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 0.10% 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 3.38% with a standard deviation of 1.2% for 14 tests and with a range of 2.1% to 5.8%.

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

1. The effective coefficient of friction seems to increase with increasing normal force (Figure 7). It is more pronounced for the condition with sand.

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.

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.

Turned Pipe-Roller. A 10 in. diameter turned, stiffened, painted pipe-roller as shown in Figure 1(b) was used in this phase of the study. The roller was identical to the unturned roller 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 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 equivalent effective coefficient of friction as defined

$$\mu_{equiv} = \frac{F}{N} = \frac{R \cdot d}{(R - d/2)h} \qquad (4)$$

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

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.

Typical coefficient of friction and horizontal force versus horizontal deflection plots are shown in Figure 8. Figure 9 compares measured and theoretical results. Correlation is good except at a horizontal movement of approximately 1 in. Close inspection of the bearing showed an imperfection in the turned surface which is believed to account for the discrepancy.

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 the equivalent coefficient of friction.

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

Pinned Rocker Shoe. A pinned-rocker shoe, similar to that shown in Figure 2(b), 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 2 and the shoe was tested in approximately 25 kips increments from 50 kips to 225 kips.

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 0.31% 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 0.071% over 12 tests and with a range of 4.42% to 10.40%.

The results of all tests are plotted in Figure 10 as friction force (horizontal force) versus normal force. The straight lines shown are the result of regression analyses conducted for each condition.

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 the value for a clean unlubricated rocker.

2. The presence of sand significantly alters the effective coefficient of pinned-rocker bearings.

Pintle Rocker Shoe. Two pintle rocker bearings similar to that shown in Figure 2(c) 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 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 plotted in Figure 11. The average coefficient of friction was 7.6% with a standard deviation of 0.111% over 24 tests with a range of 6.15% to 9.88%.

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. 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. In the 55 tests conducted, the effective coefficient of friction varied from 3.13% to 7.94%, a variation not found in tests using other bearings. Further, the effective coefficient of friction predicted by the equation shown in Figure 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. Actual and specified dimensions are shown in Figure 12. In both cases, the outside radius of the rocker was found to be larger than specified 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.

To verify this contention, sole plates with inside radii of 1.27 and 1.35 in. were used for additional testing. For the series with the 1.27 in. radius, the average effective coefficient of friction was 4.31% with a standard deviation of 0.49% 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. Typical results are shown in Figure 13.

A series of tests was also attempted with a large radius (1.35 in.) sole plate. 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. Results using the large radius sole plate were too scattered to be of use.

The tests were repeated using the 1.27 in radius sole plate for Condition 2 and with the original sole plate for Condition 3 (with sand). The average effective coefficient of friction for the rusted condition increased to 4.8% with a standard deviation of 0.18% 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 0.14% 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 is noted:

1. Fabrication accuracy is necessary if the predicted effective coefficient of friction (Figure 4) is used to estimate the horizontal friction force of pintle 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.

## SUMMARY

The results of this study are summarized in Table 1 and show that an 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.

## ACKNOWLEDGEMENT

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## LIST OF FIGURES



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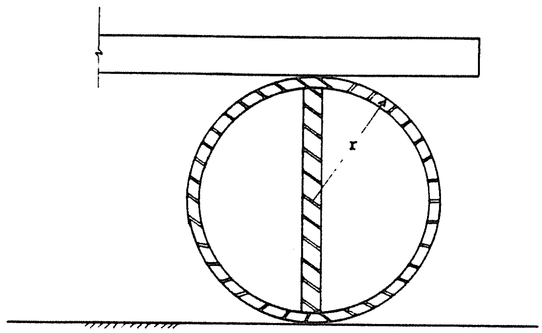
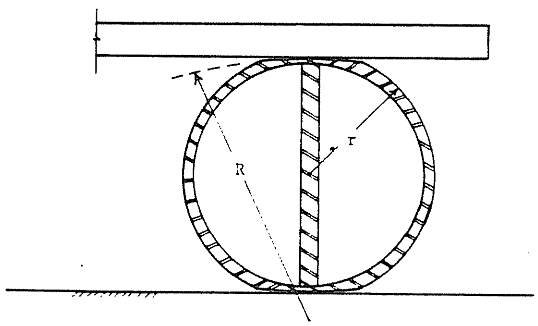
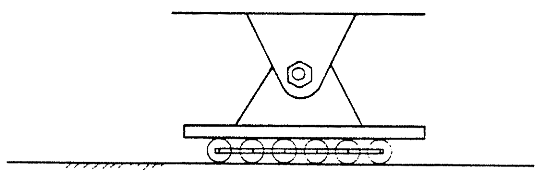



(a) Pipe Roller (Single roller)

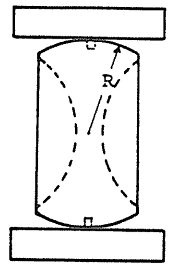



(b) Turned Pipe Roller

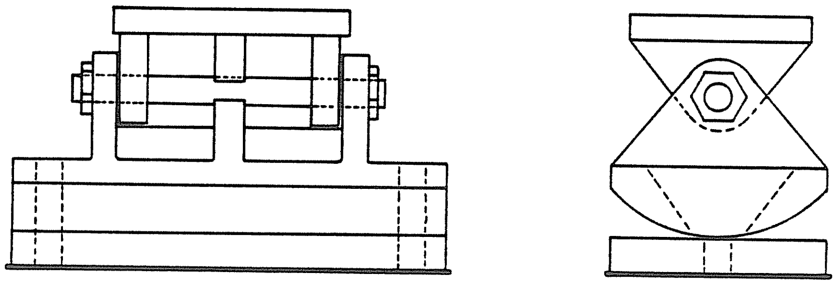


(c) Roller-nests

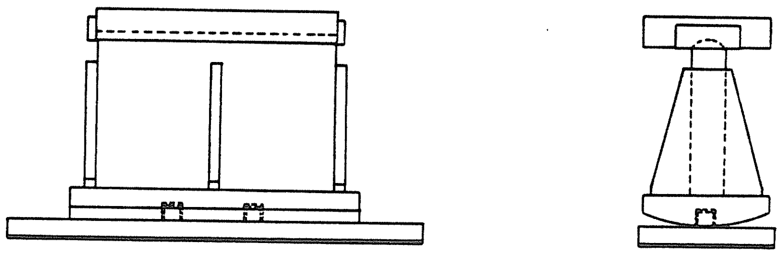
Figure 1     Roller Expansion Bearing




(a) Typical Segmented Rocker Shoe



(b) Typical Pinned Rocker Shoe



(c) Typical Pintle Rocker Shoe

Figure 2.    Rocker Expansion Shoes

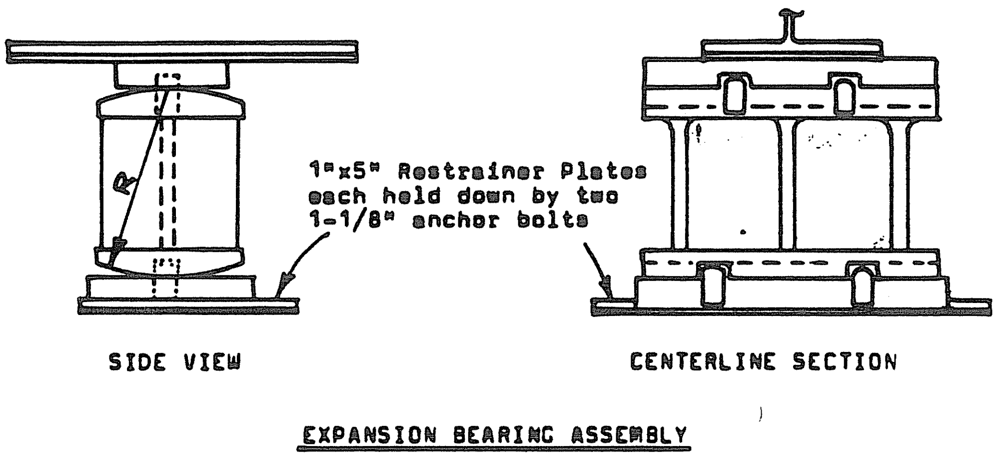



Figure 3. Double-Segmental Rocker

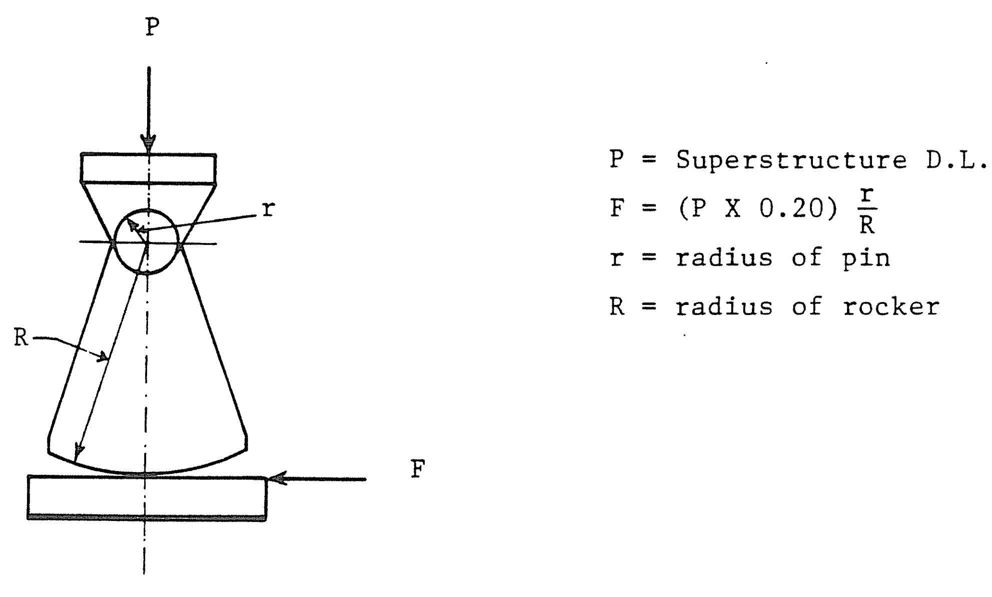



Figure 4. Forces on Rocker Bearings

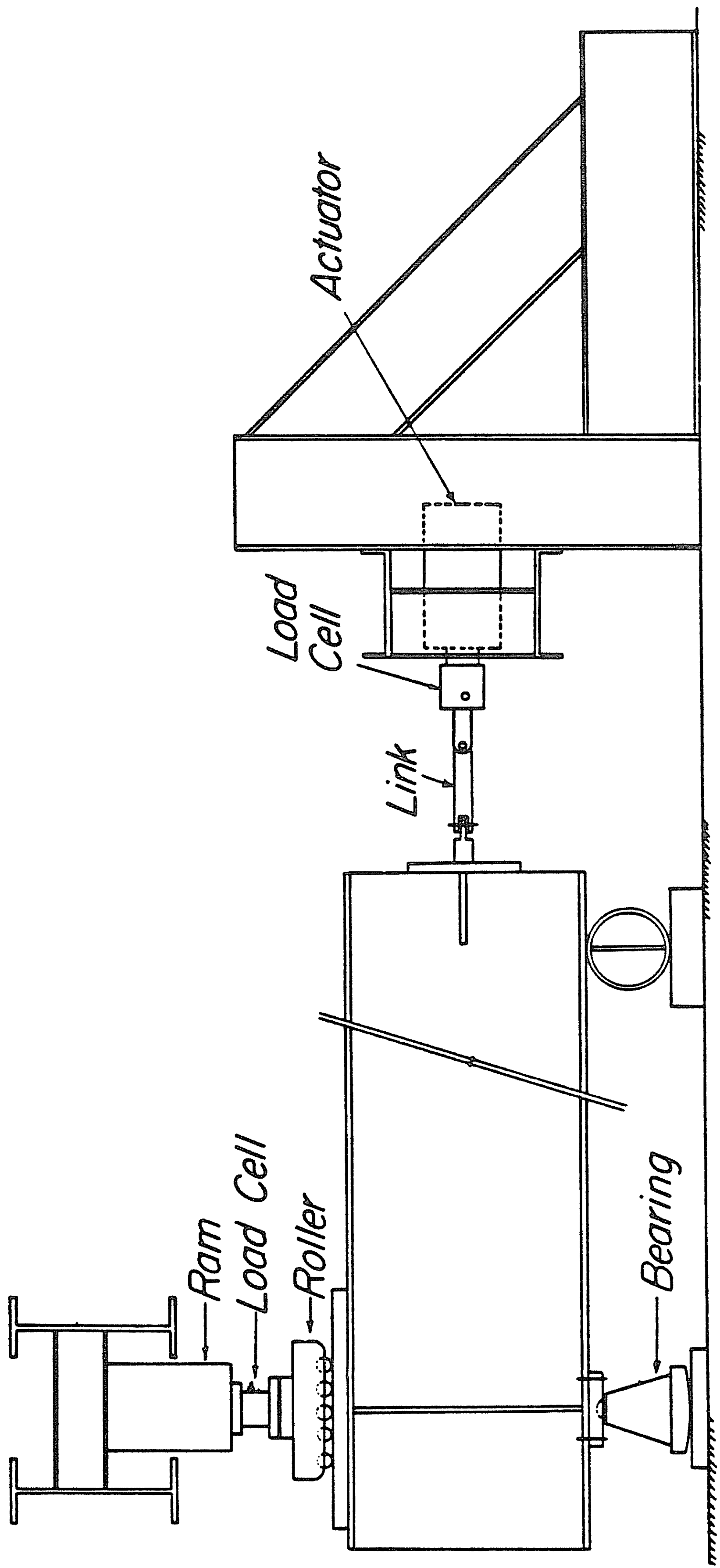



Figure 5. Side View of Test Set-up

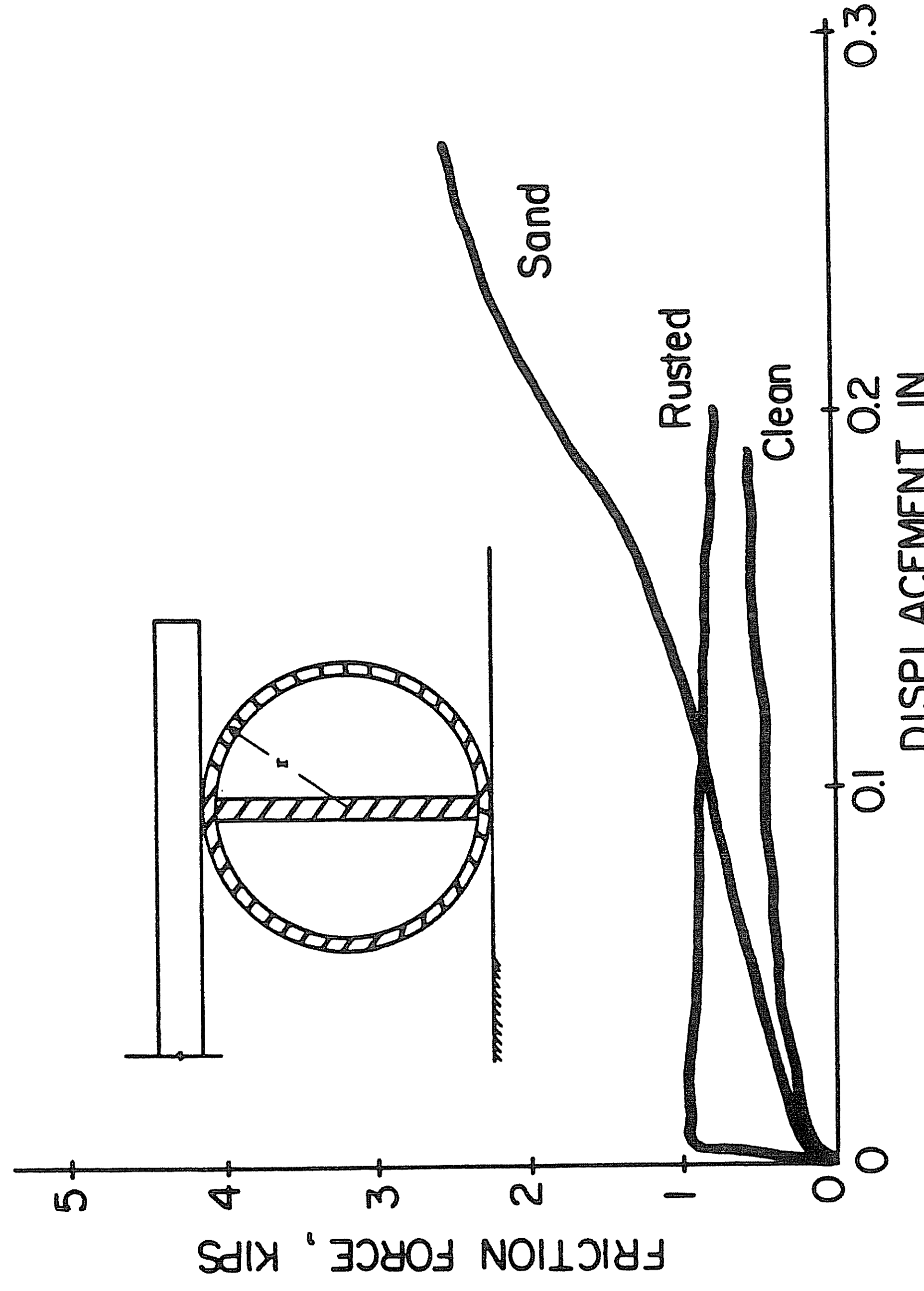



Figure 6. Typical Displacement vs. Friction Force Plots for Pipe Roller Bearing (100 kips Normal Force)

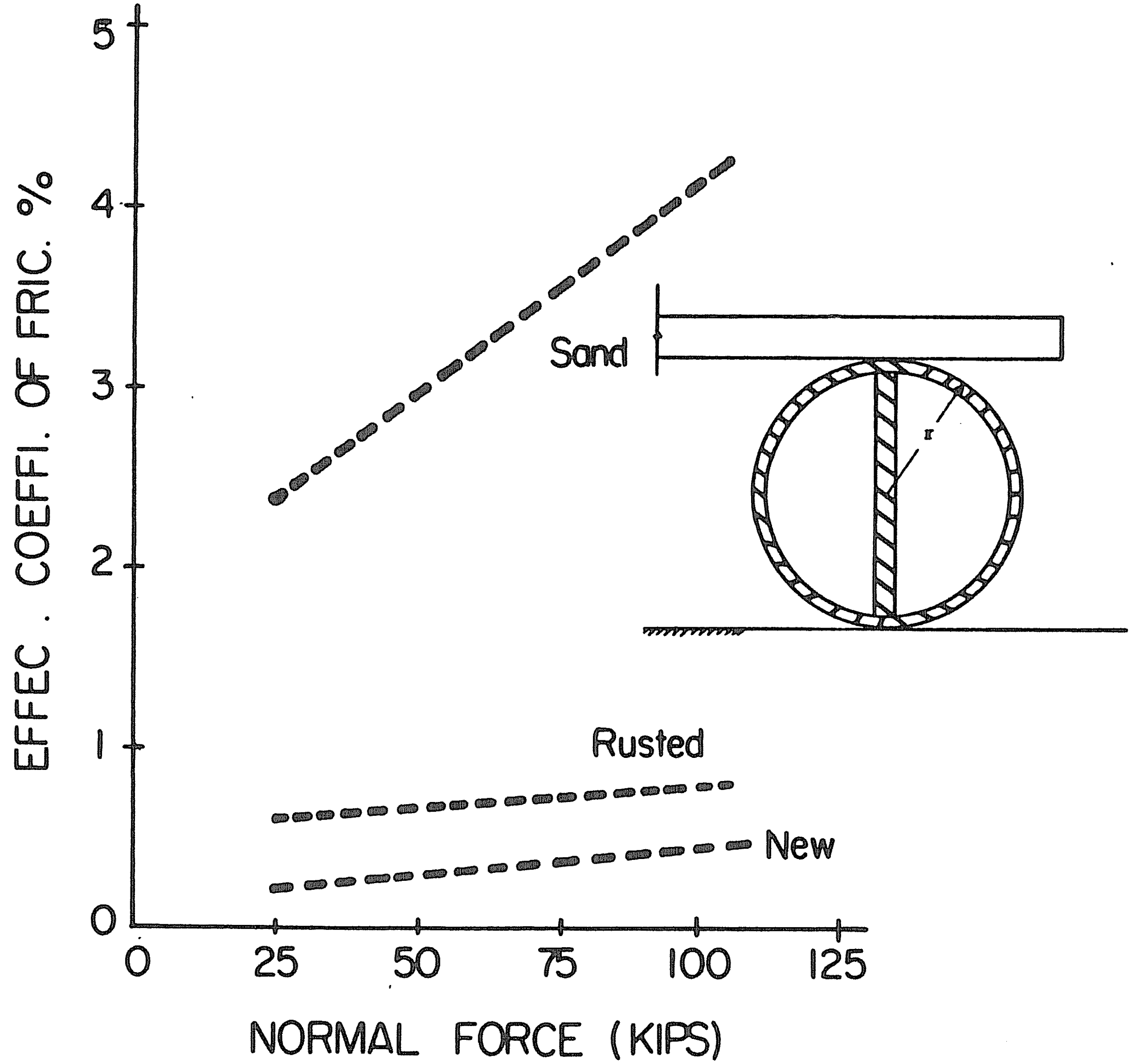



Figure 7. Normal Force vs. Effective Coefficient of Friction of Pipe-Roller Bearing

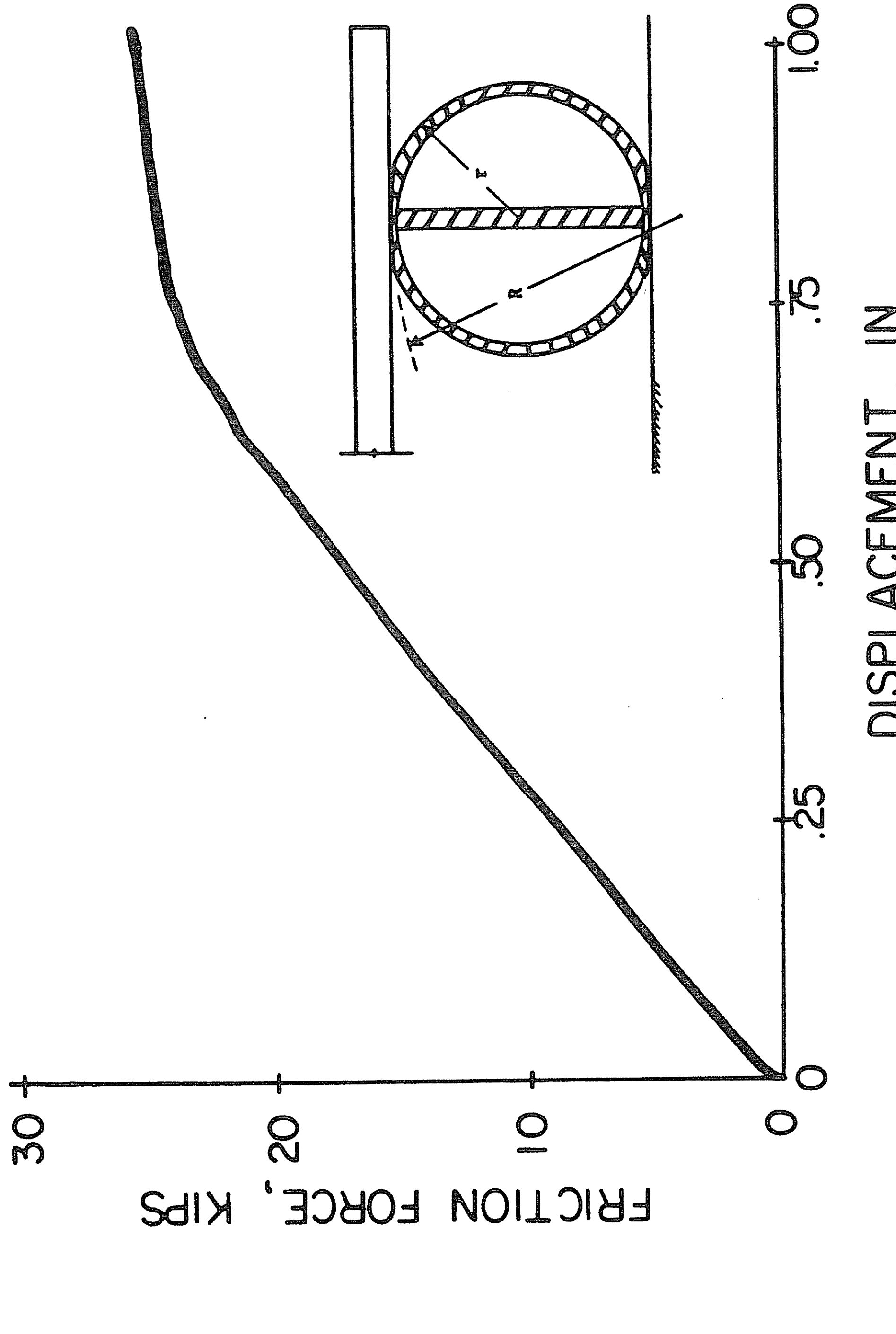



Figure 8. Friction vs. Displacement for Turned Pipe-Roller

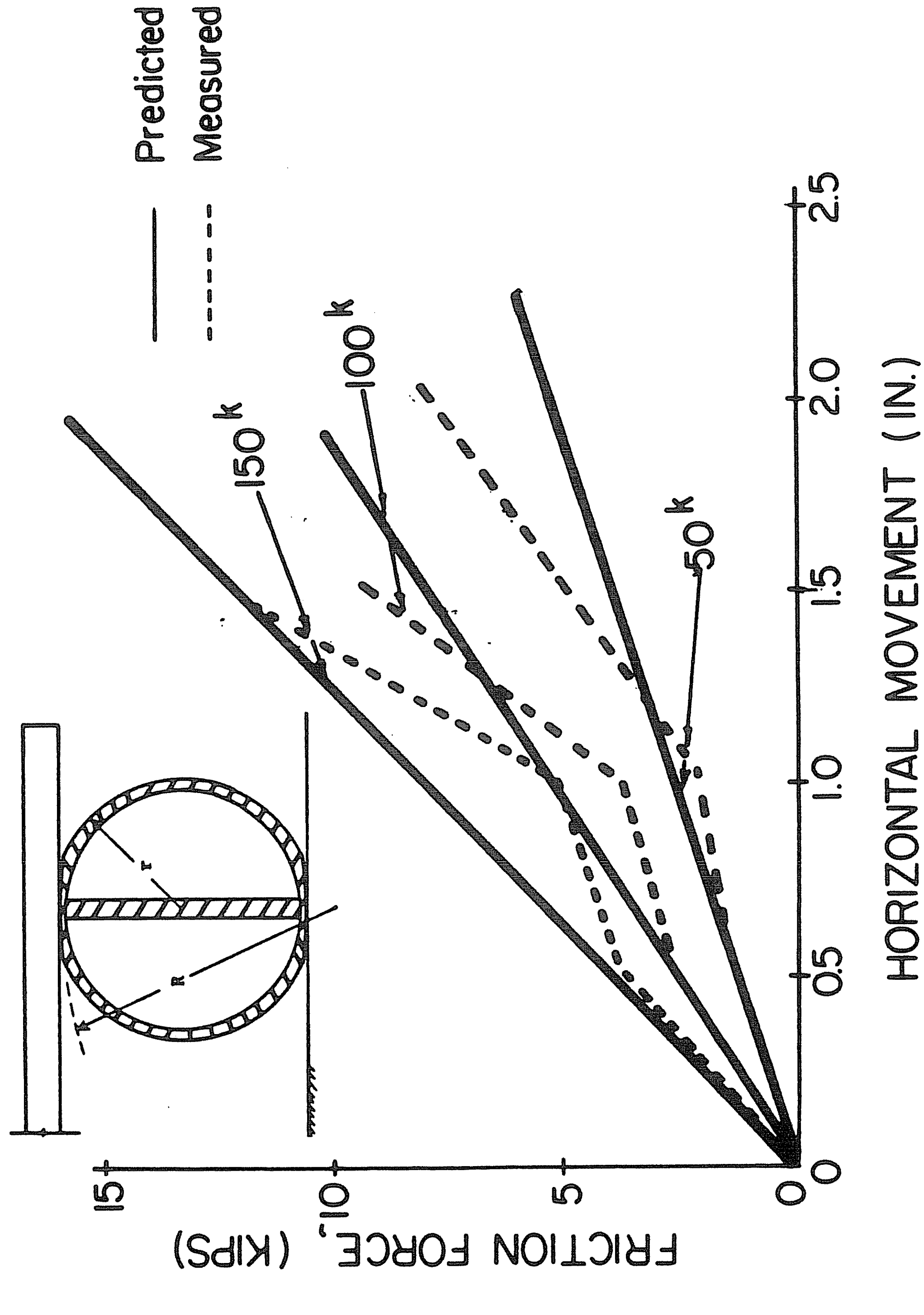



Figure 9. Resisting Force vs. Movement for Turned Pipe-Roller

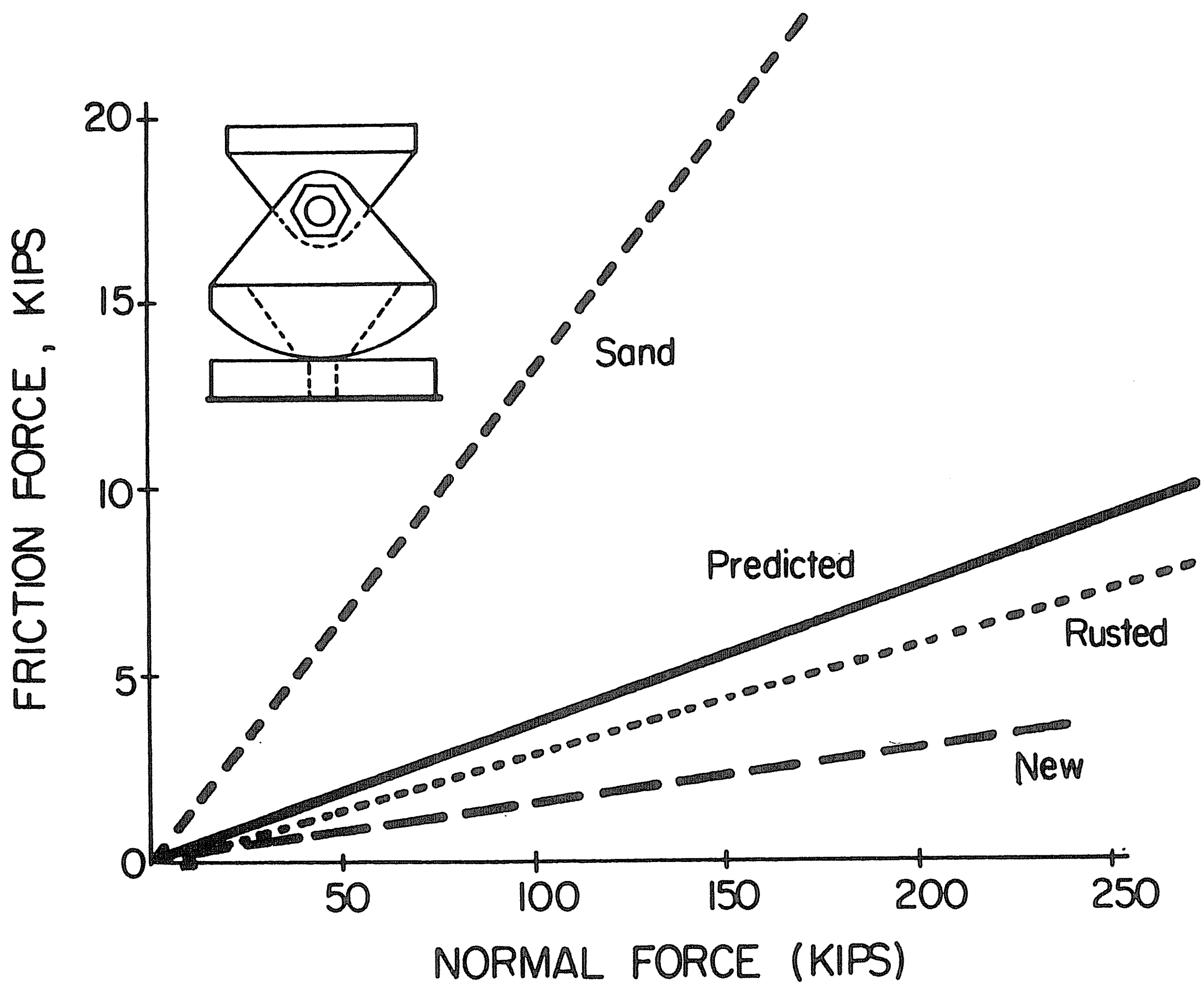



Figure 10. Typical Friction Force vs. Displacement Plots for Pinned Rocker Bearing

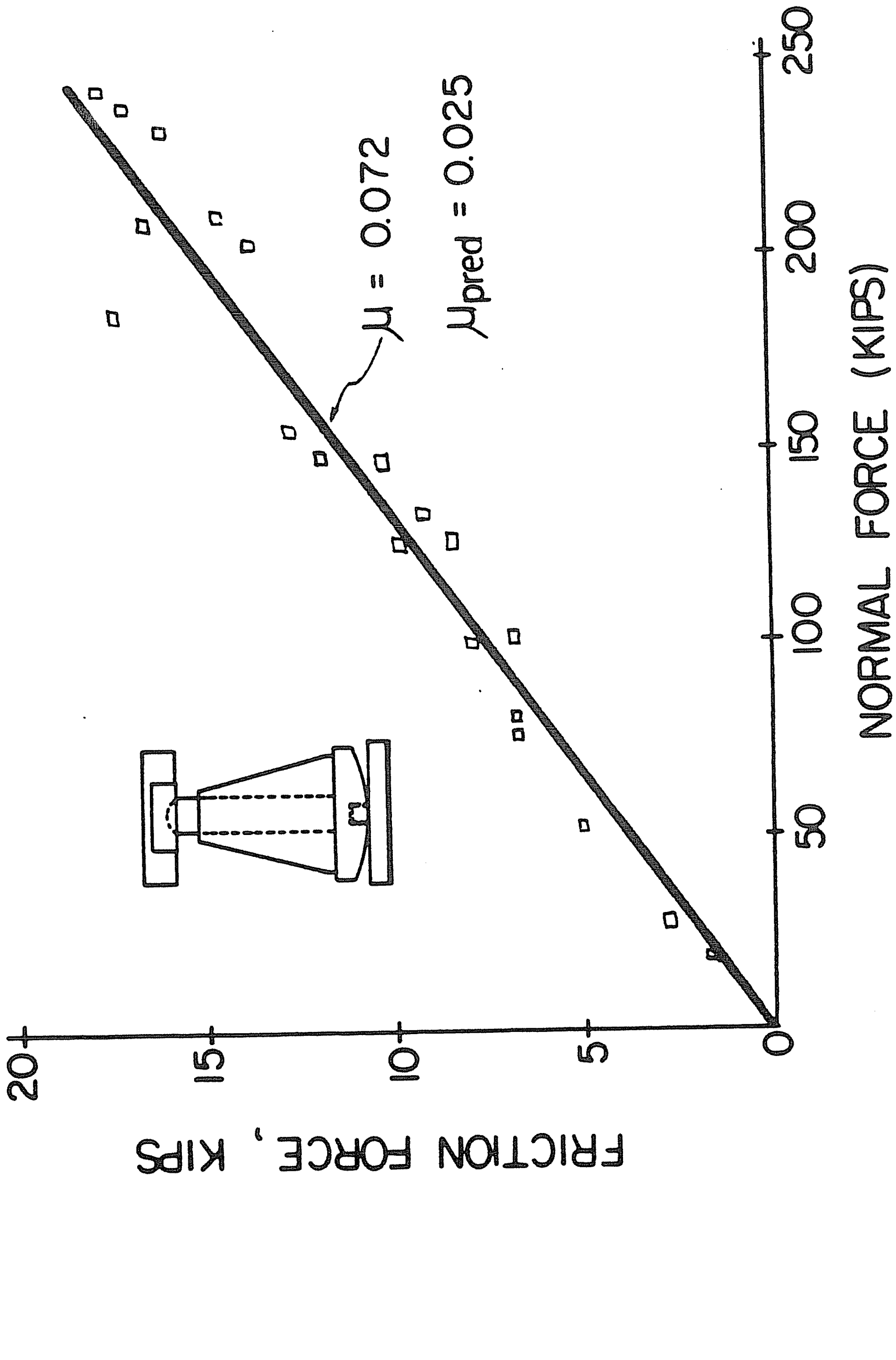
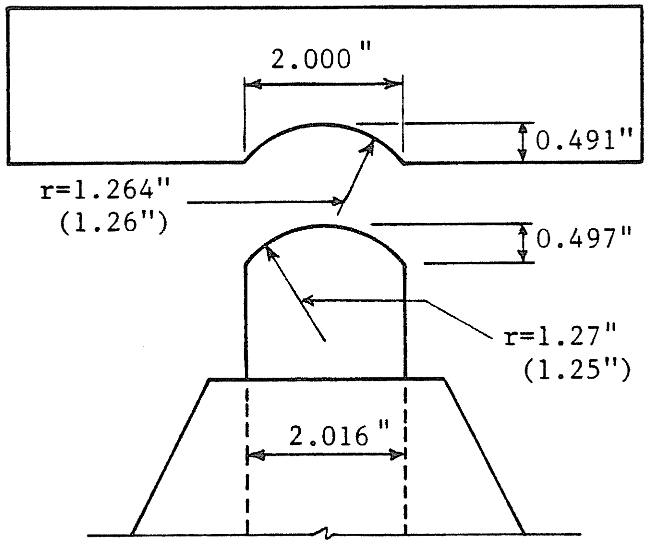
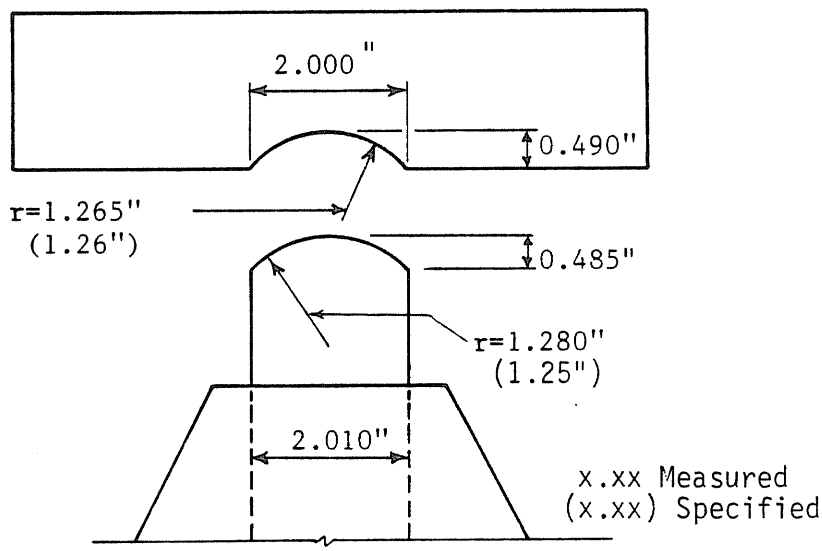



Figure 11. Friction Force vs. Normal Force - Pintle Bearing I, Condition 1




(a) Test Bearing I




(b) Test Bearing II

Figure 12. Measured and Specified Dimensions of the Pintle Rocker Bearings

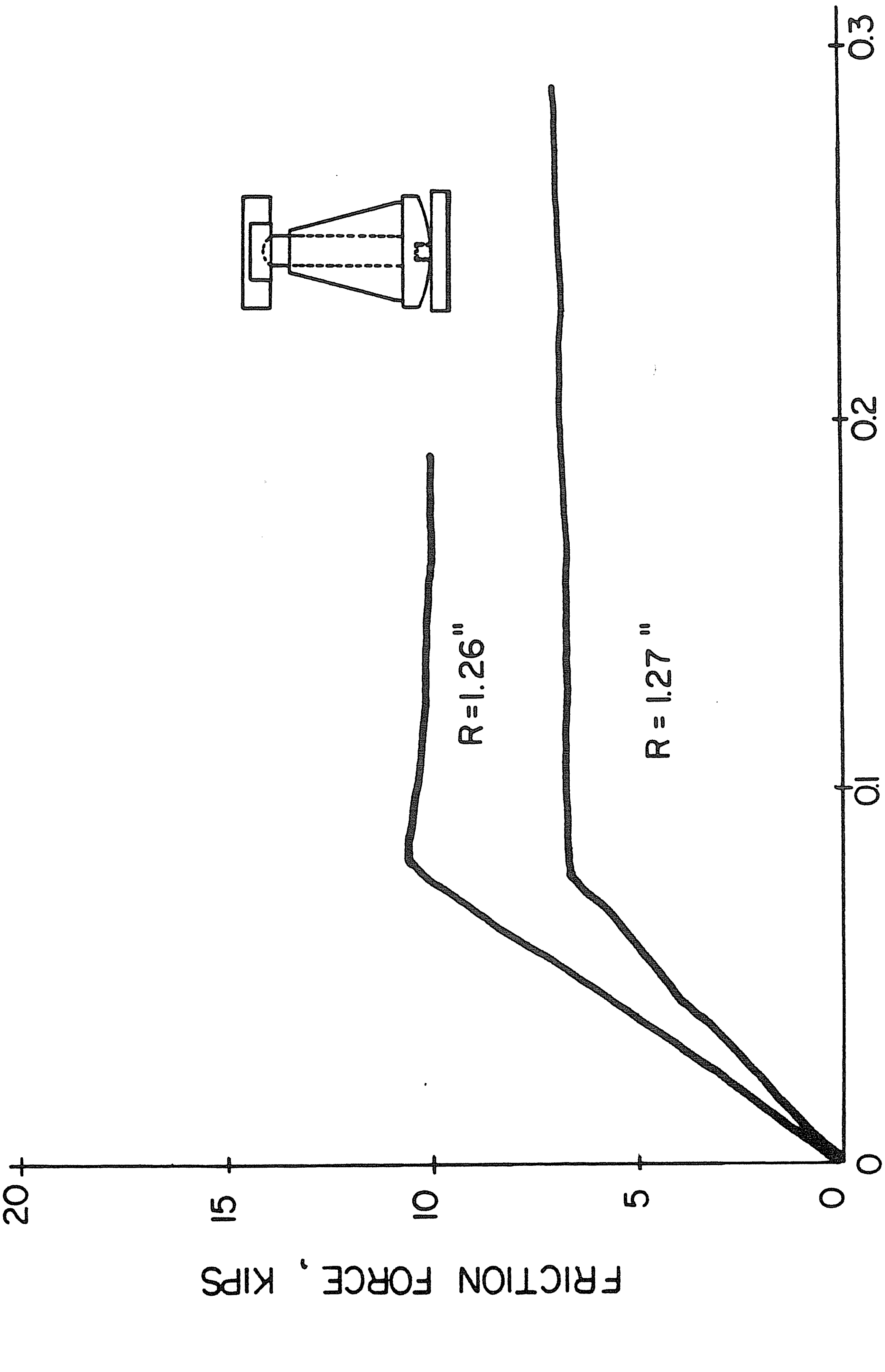



Figure 13. Friction Force vs. Displacement for Pintle Rocker Bearing

Table 1
Summary of Results

| Bearing Type | Condition | Effective Coefficient of Friction | | Remarks |
|---|---|---|---|---|
| | | Predicted (%) | Measured (%) | |
| Single Roller Bearing | Clean<br>Rusted<br>w/Sand | | 0.5<br>1.0<br>5.0 | |
| Turned Roller Bearing | | | | Geometric relationship<br>Sand increases<br>250-400% |
| Pinned Rocker Shoe | Clean<br>Rusted<br>w/Sand | 2.5 | 1.0<br>2.0<br>9.0 | |
| Pintel Rocker Bearing | Clean<br>Clean<br>Rusted<br>w/Sand | 2.5*<br>2.5*<br>2.5*<br>2.5* | 6.2-9.9<br>2.2-5.5<br>3.6-5.5<br>12.1-14.1 | 1.26"/1.27"R**<br>1.27"/1.27"R<br>1.27"/1.27"R<br>1.27"/1.27"R |

*Radius of sole plate = 1.26 in.; radius of rocker = 1.25 in.

**Radius of sole plate/Radius of rocker