UNITED STATES EXPERIMENTAL MODEL BASIN

NAVY YARD, WASHINGTON, D.C.

AN INVESTIGATION OF FRICTIONAL VALUES FOR LINE AND PROPELLER SHAFT BEARINGS AND STUFFING BOXES

BY GRANT A DeSHAZER

EXPERIMENTAL MODEL BASIN
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CONSTRUCTION AND REPAIR
NAVY DEPARTMENT

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Summary

Tests were conducted with two types of apparatus to determine values of the coefficient of friction for lignum-vitae propeller shaft bearings as used in the stern tubes and struts of ships. Tests were also made to determine friction losses in propeller shaft stuffing boxes. A full-scale test was made on the starboard shaft of the U.S.S. HAMILTON.

Values for the coefficient of friction are given for rubbing velocities from 275 to 1600 ft. per min. and for bearing pressures from 10 to 40 lb. per sq.in. The lowest practicable coefficient of friction for average conditions was found to be 0.0035 at a frictional surface velocity of approximately 700 ft. per min., but values of the coefficient of friction as low as 0.0020 were obtained during model tests.

Frictional values for stuffing boxes are given for rubbing surface speeds from 150 to 2400 ft. per min. The packing offers an average resistance between 0.48 and 0.70 pounds per square inch of rubbing surface.

The loss of power due to bearing friction, stuffing box friction, and "windage" on the shaft and propeller hub was quite low, approximating 0.20 per cent of the power transmitted under normal operation.

Introduction

Reports of ship trials generally give only the powers recorded or indicated by torsionmeters, indicators or similar instruments. While by far the greater portion of this power is absorbed by the propellers, some of it is expended in friction in the line shaft and propeller shaft bearings abaft the torsionmeters, in friction in the various bulkhead and stern tube stuffing boxes, and in "windage" in the stern tube and on propeller shafts, propeller hub and hub cap. It has been customary in the past either to neglect all of these effects or at best to make an approximation of their value since they were generally within the limits of error of trials. Increasing refinements in the methods of measuring power and more persistent demands for satisfactory correlation of data for model tests and ship trials make it necessary to use more reliable methods in the determination of shaft friction.

It is customary, in self-propelled model tests conducted at the U.S. Experimental Model Basin, to take the so-called "zero" power readings with the model traveling at a given speed, the shaft making the required revolutions for that speed, with a dummy propeller hub of lead in place of the model propeller. The final or net power readings, therefore, represent apparently only the power absorbed by the propeller <u>blades</u>.

It is to be expected that there will be some difference when comparing the power results of self-propelled model tests with those of full size ship trials.

Since it would not be practicable to tow the full size vessel with a dummy propeller, even though it were possible to record the "zero" power readings accurately with existing torsionmeters, the obvious remedy is to obtain data upon which a reasonably accurate estimate of the power loss in the shafting can be obtained.

The present investigation was undertaken with this purpose in view.

History

While there have been attempts from time to time to measure the mechanical efficiency of reciprocating steam and Diesel machinery, these attempts have fallen so far short of success as to leave entirely out of consideration any such small item as shaft bearing and stuffing box friction.

It has been the practice, therefore, to estimate the loss of power due to the latter as a certain small percentage of the full power, <u>at</u> full power. Estimates vary from 1-1/2 to 3 per cent.

While there are some data available in handbooks upon which to estimate frictional constants for line shaft bearings, (steel running on white metal), nothing whatever could be found for the type of wood bearing, (bronze upon lignum-vitae), so universally used for stern tube and propeller shaft bearings.

Theory

Primarily, the study of lignum-vitae bearings was made to ascertain the frictional loss; but in order to give a better interpretation of data obtained, some concise general theory, see Appendix I, will be given, as well as a few assumptions pertaining to theory used in this particular test.

The characteristics of friction in bearings depend not only upon the bearing surfaces but also upon the lubricant used. Friction approximates that of solid friction when a journal is run dry, and that of fluid friction when it is flooded with lubricant. In a lubricated bearing, the friction is affected by the viscosity of the lubricant. Over a limited range of perfect film lubrication, the coefficient of friction decreases with decrease of viscosity, but the viscosity cannot be reduced indefinitely to produce lower and lower friction in bearings. As the viscosity of a lubricant is reduced, the lubricating film will become so thin as to allow contact between the bearing and journal, thus increasing the friction.

The following notation is used:

W = load on bearing, lb.

N = normal reaction, lb.

F = frictional force at surface of journal, lb.

P = scale reading or balancing force, lb.

d = diameter of journal, in.

r = radius of journal, in.

a = lever arm at which P acts, in.

l = length of bearing, in.

p = unit bearing pressure, lb. per sq. in.

f = coefficient of friction.

n = number of revolutions per minute.

Frictional values for lignum-vitae bearings were determined on a specially designed machine resembling in principle the one on which Towers[5] made his tests from 1883 to 1888. The load was applied to an I-beam and was transmitted to a knife edge directly above the center line of the journal, see Figs. 4 and 5. The frictional force was measured through a lever mechanism. This apparatus will be designated as the lever type.

The frictional force, F, is given by the equation

$$F = Pa/r \quad \ldots\ldots\ldots\ldots \quad (1)$$

which is developed in Appendix II.

Frictional values were determined also on a special apparatus in which a Ford torsionmeter was mounted on a direct drive shaft, see Figs. 17 and 18. This apparatus will be designated as the torsionmeter type. The scale reading gave an indication of the torque transmitted by the shaft. The actual torque was read from a calibration curve.

The coefficient of friction, f, is defined by the equation

$$f = F/W \quad \ldots\ldots\ldots\ldots \quad (2)$$

in which F is the frictional resisting force at the surface of the journal and W is the total load on the bearing perpendicular to its axis.

The actual bearing pressure over the journal is not easily obtained. The bearing pressure, p, has been taken as the load, W, divided by the projected area

$$p = W/ld \quad \ldots\ldots\ldots\ldots \quad (3)$$

The frictional horsepower can be calculated by the equation

$$\text{Frict. HP} = \frac{2\pi Pa\, n}{(12)(33{,}000)} \quad \ldots\ldots\ldots \quad (4)$$

Four different complete tests were made to determine friction loss abaft the torsionmeters. These tests were made on the special apparatus designated as (1) the lever type, (2) the torsionmeter type, (3) stuffing box, and (4) a full size test on the U.S.S. HAMILTON.

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*Indices refer to the Bibliography.

Tests on Lever Type Apparatus

Apparatus. The bearings used for test were made to duplicate actual bearings as closely as possible. A detailed description is given in Appendix V. Two bearings, 4-1/2 in. and 6 in., were tested. Each bearing was made up of four half sections, three of which are shown in Fig. 1, making it possible to vary the length.

The bearing sections were fastened securely in a housing forming one continuous bearing, see Fig. 2, which rested on the upper half of a composition G (bronze) journal. The journal was supported at both ends by metal bearings, leaving the bearing to be tested able to float freely on its journal. The lignum-vitae bearings were bored out to give a clearance of 0.023 and 0.026 inches respectively for the 4-1/2-in. and 6-in. bearings.

Fig. 3 shows the original experimental set-up. The bearing was loaded with lead weights applied to the I-beam. The power to turn the shaft was transmitted by belt from a D.C. motor with Ward Leonard method of speed control. The unbalanced force on the bearing was measured by a spring scale and weights added to the scale pan.

In order to increase the sensitiveness of the apparatus, part of the machine was redesigned so that the load could be applied through a knife edge directly over the center line of the shaft, see Figs. 4 and 5. The center line of the knife seat, A Fig. 5, was projected over the edge of the tank by an arm. At either end of the shaft was a mechanism, B which when leveled by the gunners quadrant, C, located the center line of the shaft relative to the knife edge with an accuracy of 0.001 in. by means of the dial micrometer, D.

Friction on the bearing was measured by weights applied to a lever system, stabilized by a spring scale. The application of the correct weight would bring the knife edge directly over the center line of the shaft. The sensitiveness was 0.2 lb. at a three foot lever arm or 0.6 lb.-ft. torque.

Method of Test. All tests, on the lever type apparatus, were made with fresh water in the tank. Water was circulated to keep constant temperature on the bearing. The load was applied to the I-beam and held in a balanced condition by the spring scale, H Fig. 4. The measuring lever arm was then adjusted so that the load was applied directly over the center of the journal. In this position a zero reading was established.

A reading was made by first obtaining the desired speed. Weights were then added or subtracted from the scale pan, E Fig. 5, until the lever arm returned to the zero position while the gunners quadrant, C, and dial micrometer, D, indicated the correct position of the load.

Several runs were made with the same direction of rotation and the same load. Then the direction of rotation was reversed and run with the bearing in zero position until the friction became constant with constant speed. Similar

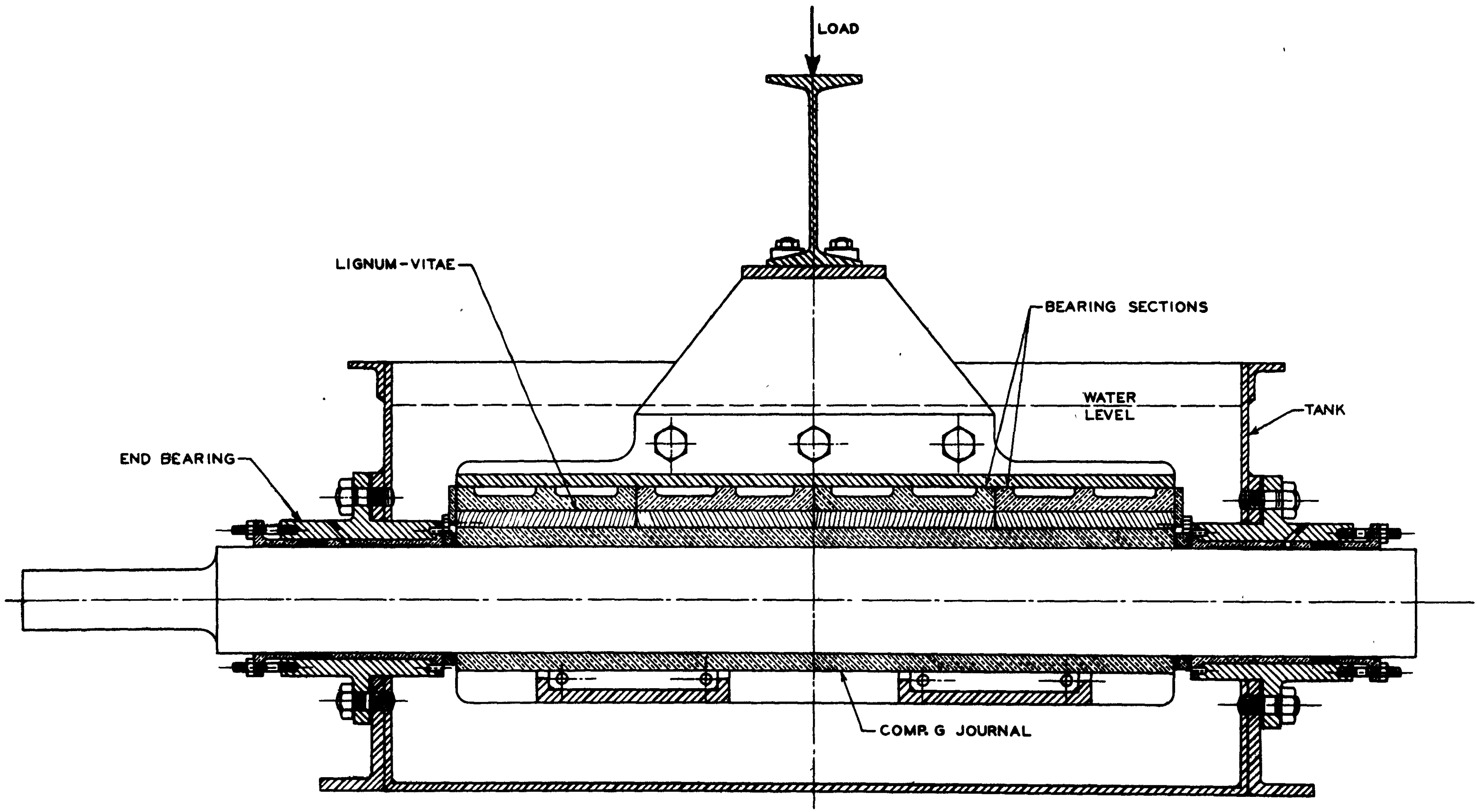

FIG. 2. Sectional View of Bearing Assembly

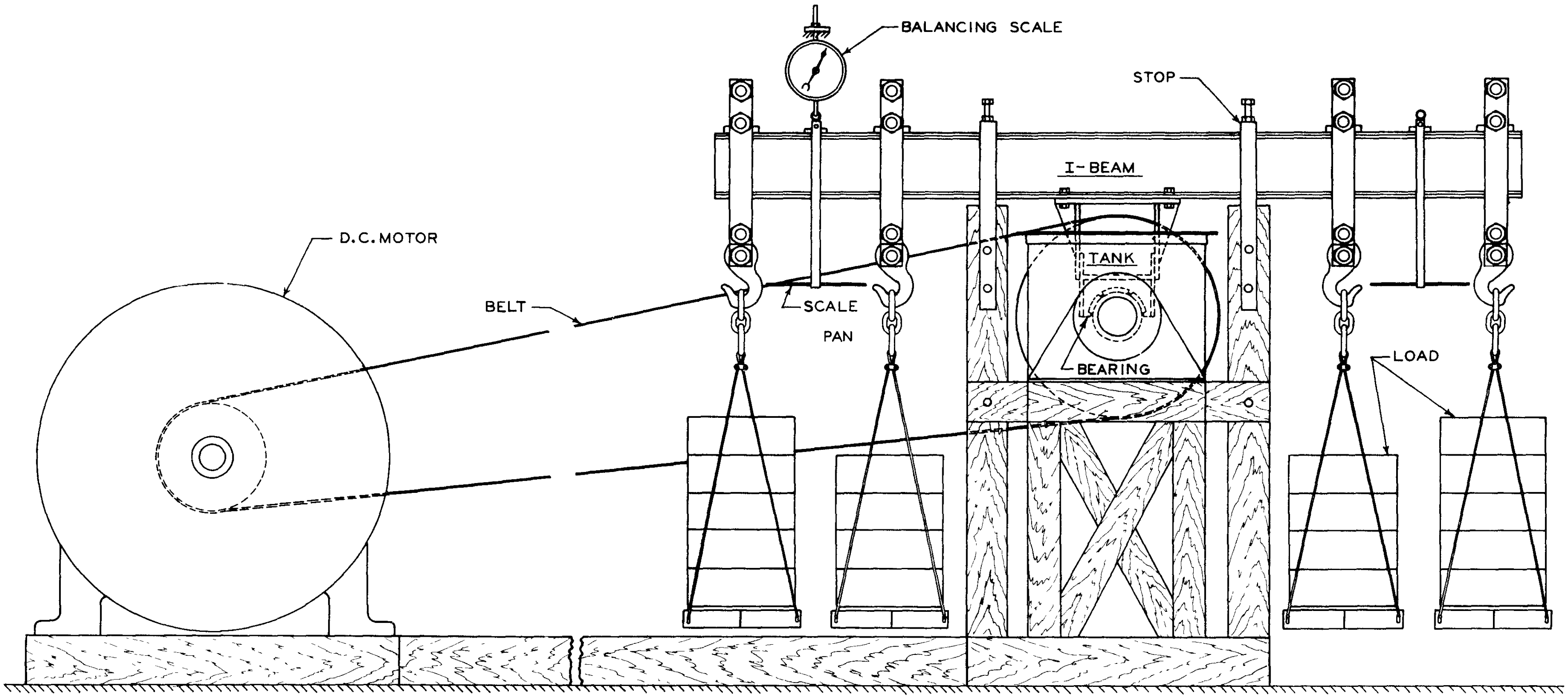

FIG. 3. General Assembly Plan

runs were repeated with the reversed rotation. The frictional values plotted on the curves are the average of equal number of clockwise and counterclockwise readings. The two directions of rotation did away with tare readings.

The load on the bearing was then changed but the direction of rotation was maintained until the desired number of runs was made. After each change of load, the bearing was run until the friction became constant with constant speed before making recorded readings.

This particular procedure was adhered to because it saved time by limiting the number of adjustments, and more consistent results were obtained. However, this method necessitated a great deal of time for running in the bearings before satisfactory readings were obtained.

From the results of tests, it will be found that care had to be exercised in taking readings under the right conditions. After starting up, after reversing rotation, and after a change of load, the bearing had to be run a certain time before consistent readings could be had.

Results. Test values for the coefficient of friction for four lengths of bearings with a diameter of 4-1/2 in. are given in Table I. In part, these values are shown in the curves Figs. 6 to 9 inclusive. Figs. 6 and 7 show the effect of speed on friction at different bearing pressures for three sizes of bearings. The friction was quite high at the lower speeds but at the higher speeds the frictional values were exceptionally low. Test values at the higher speeds were quite consistent but for the low speeds the values were slightly erratic.

Figs. 8 and 9 show the effect of bearing pressure on friction at different speeds. Except at the lower speeds, the values were fairly consistent.

Curve 1, Fig. 10 indicates that at the higher speeds the friction gradually increases with speed. This curve compares favorably with the results shown in Figs. 6 and 7 but the values for the curves 2 and 3 are much higher. While making tests for these two curves the fast rotating shaft threw grease from the end bearings into the water. This grease circulated with the water and caused a thin film of grease to form over the journal and bearing surfaces. As the bearing pressures were low, the higher values were no doubt due to the higher viscosity lubricant.

The effect of varying the length of a 4-1/2 in. bearing is shown in Fig. 11. Although the points are somewhat scattered, they clearly show that an increase in length of bearing resulted in a slight increase in coefficient of friction.

Running in a bearing decreased the friction. Some typical curves are shown in Fig. 12. Until at least 40 hours of running in, quite a noticeable change in friction was evident.

When first starting a bearing the frictional values were considerably higher than after running for a length of time, see Fig. 13. It took longer for the friction to become constant with the higher bearing pressures. Relative values are given due to rotating the shaft in one direction only. With one direction of rotation, zero torque was not established for the lever type apparatus.

Directly after reversing rotation on a bearing the friction was quite high. Curve 1, Fig. 14, shows a typical curve. As in starting up, the higher bearing pressures required a longer time for the friction to become constant with constant speed than for the lower bearing pressures, see curve 2, Fig. 14. For a bearing pressure of 43 lb. per sq. in., it required approximately 4-1/2 hours for the friction to become constant with constant speed after reversing.

At low speeds with the higher bearing pressures, groaning or chattering was noticeable in the bearing. The effect of bearing length and bearing pressures on this chattering is shown in Fig. 15. The presence of this chattering was detected by ear only.

TABLE I

COEFFICIENT OF FRICTION FOR LIGNUM-VITAE BEARINGS

$4\frac{1}{2} \times 22\frac{1}{2}$-IN. BEARING TEMP. 65°F

SPEED FT. PER MIN.	BEARING PRESSURE, LB. PER SQ. IN.				
	7.1	16.3	24	32	40
118	0.0138	0.0165	0.0171	0.0255	0.0466
177	0.0056	0.0063	0.0065	0.0095	0.0152
236	0.0031	0.0036	0.0040	0.0060	0.0081
295	0.0024	0.0026	0.0038	0.0046	0.0065
353	0.0022	0.0025	0.0033	0.0034	0.0045
412	0.0022	0.0025	0.0028	0.0034	0.0034
471	0.0022	0.0025	0.0028	0.0034	0.0034
530	0.0022	0.0025	0.0028	0.0034	0.0034
589	0.0022	0.0025	0.0028	0.0034	0.0034
649	0.0022	0.0025	0.0028	0.0034	0.0034

$4\frac{1}{2} \times 16\frac{7}{8}$-IN. BEARING TEMP. 60°F

SPEED FT. PER MIN.	BEARING PRESSURE, LB. PER SQ. IN.				
	8.4	16.1	23.2	32.3	43
118	0.0090	0.0098	0.0153	0.0336	0.0732
177	0.0041	0.0044	0.0052	0.0094	0.0315
236	0.0023	0.0025	0.0038	0.0043	0.0163
295	0.0020	0.0022	0.0033	0.0033	0.0106
353	0.0020	0.0022	0.0028	0.0030	0.0082
412	0.0020	0.0022	0.0026	0.0030	0.0061
471	0.0020	0.0022	0.0026	0.0030	0.0045
530	0.0020	0.0022	0.0026	0.0030	0.0039
589	0.0020	0.0022	0.0026	0.0030	0.0033
649	0.0020	0.0022	0.0026	0.0030	0.0033

$4\frac{1}{2} \times 11\frac{1}{4}$-IN. BEARING TEMP. 60°F

SPEED FT. PER MIN.	BEARING PRESSURE, LB. PER SQ. IN.						
	10.6	16	24	32	40	50	58
118	0.0088	0.0093	0.0139	0.0137	0.0341	0.0587	0.0744
177	0.0035	0.0038	0.0059	0.0054	0.0098	0.0174	0.0228
236	0.0025	0.0023	0.0041	0.0039	0.0055	0.0095	0.0137
295	0.0020	0.0022	0.0026	0.0033	0.0040	0.0060	0.0098
353	0.0020	0.0022	0.0026	0.0031	0.0034	0.0047	0.0071
412	0.0020	0.0022	0.0026	0.0031	0.0034	0.0038	0.0050
471	0.0020	0.0022	0.0026	0.0031	0.0034	0.0036	0.0040
530	0.0020	0.0022	0.0026	0.0031	0.0034	0.0036	0.0038
589	0.0020	0.0022	0.0026	0.0031	0.0034	0.0036	0.0038
649	0.0020	0.0022	0.0026	0.0031	0.0034	0.0036	0.0038

$4\frac{1}{2} \times 5\frac{5}{8}$-IN. BEARING TEMP. 60°F

SPEED FT. PER MIN.	BEARING PRESSURE, LB. PER SQ. IN.			
	20	25.3	32	40
118	0.0067	0.0140	0.0445	0.0710
177	0.0017	0.0043	0.0064	0.0180
236	0.0017	0.0020	0.0034	0.0067
295	0.0017	0.0020	0.0027	0.0042
353	0.0017	0.0020	0.0027	0.0035
412	0.0017	0.0020	0.0027	0.0035
471	0.0017	0.0020	0.0027	0.0035
530	0.0017	0.0020	0.0027	0.0035
589	0.0017	0.0020	0.0027	0.0035
649	0.0017	0.0020	0.0027	0.0035

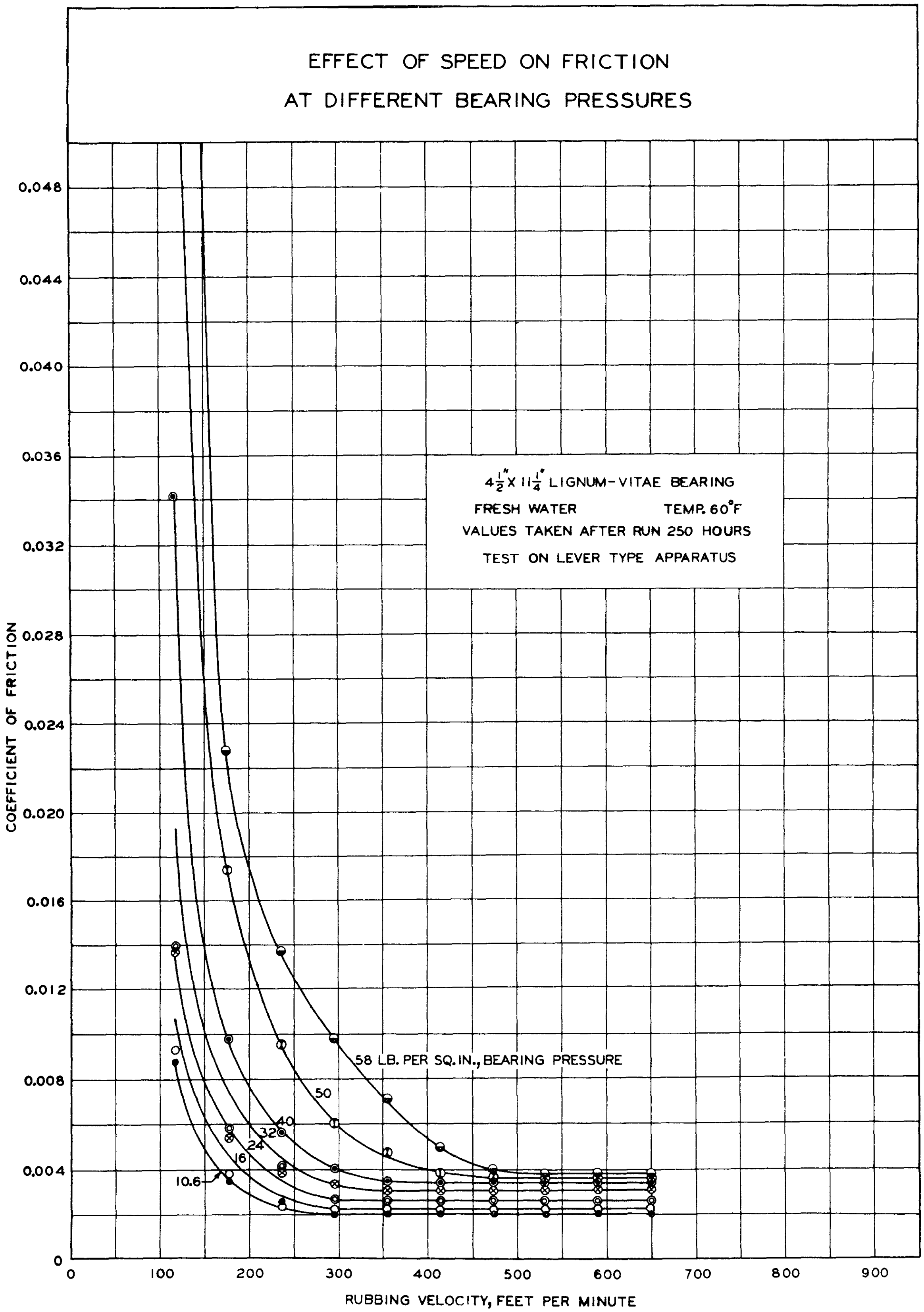
EFFECT OF SPEED ON FRICTION
AT DIFFERENT BEARING PRESSURES
4½" X 11¼" LIGNUM-VITAE BEARING
FRESH WATER TEMP. 60°F
VALUES TAKEN AFTER RUN 250 HOURS
TEST ON LEVER TYPE APPARATUS
58 LB. PER SQ. IN., BEARING PRESSURE
50
40
32
24
16
10.6
COEFFICIENT OF FRICTION
0.048
0.044
0.040
0.036
0.032
0.028
0.024
0.020
0.016
0.012
0.008
0.004
0
100
200
300
400
500
600
700
800
900
RUBBING VELOCITY, FEET PER MINUTE

FIG. 6.

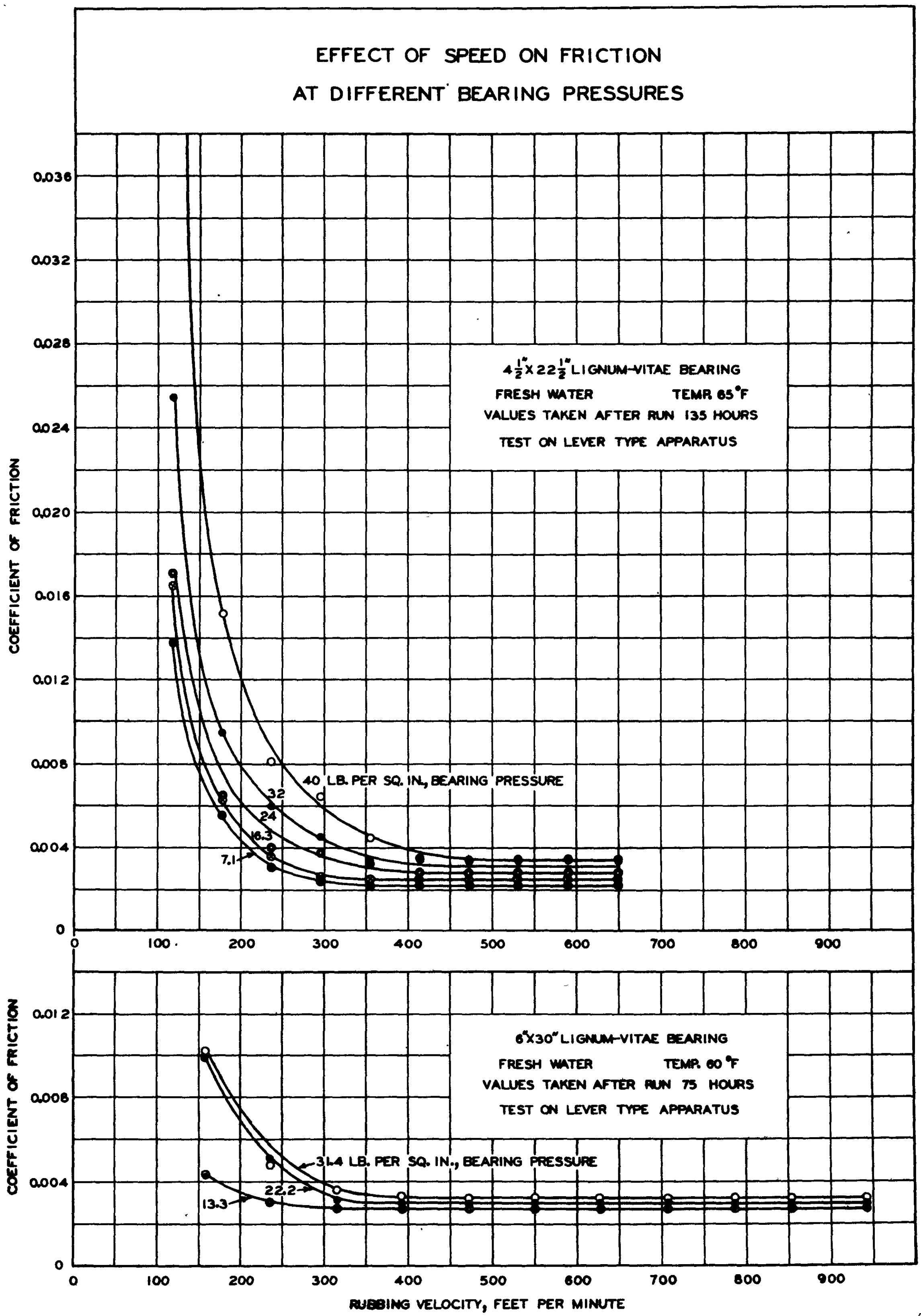
EFFECT OF SPEED ON FRICTION
AT DIFFERENT BEARING PRESSURES
$4\frac{1}{2}$" X $22\frac{1}{2}$" LIGNUM-VITAE BEARING
FRESH WATER TEMP. 65°F
VALUES TAKEN AFTER RUN 135 HOURS
TEST ON LEVER TYPE APPARATUS
40 LB. PER SQ. IN., BEARING PRESSURE
32
24
16.3
7.1
COEFFICIENT OF FRICTION
0.036
0.032
0.028
0.024
0.020
0.016
0.012
0.008
0.004
0
0
100
200
300
400
500
600
700
800
900
6" X 30" LIGNUM-VITAE BEARING
FRESH WATER TEMP. 60°F
VALUES TAKEN AFTER RUN 75 HOURS
TEST ON LEVER TYPE APPARATUS
31.4 LB. PER SQ. IN., BEARING PRESSURE
22.2
13.3
COEFFICIENT OF FRICTION
0.012
0.008
0.004
0
RUBBING VELOCITY, FEET PER MINUTE

FIG. 7.

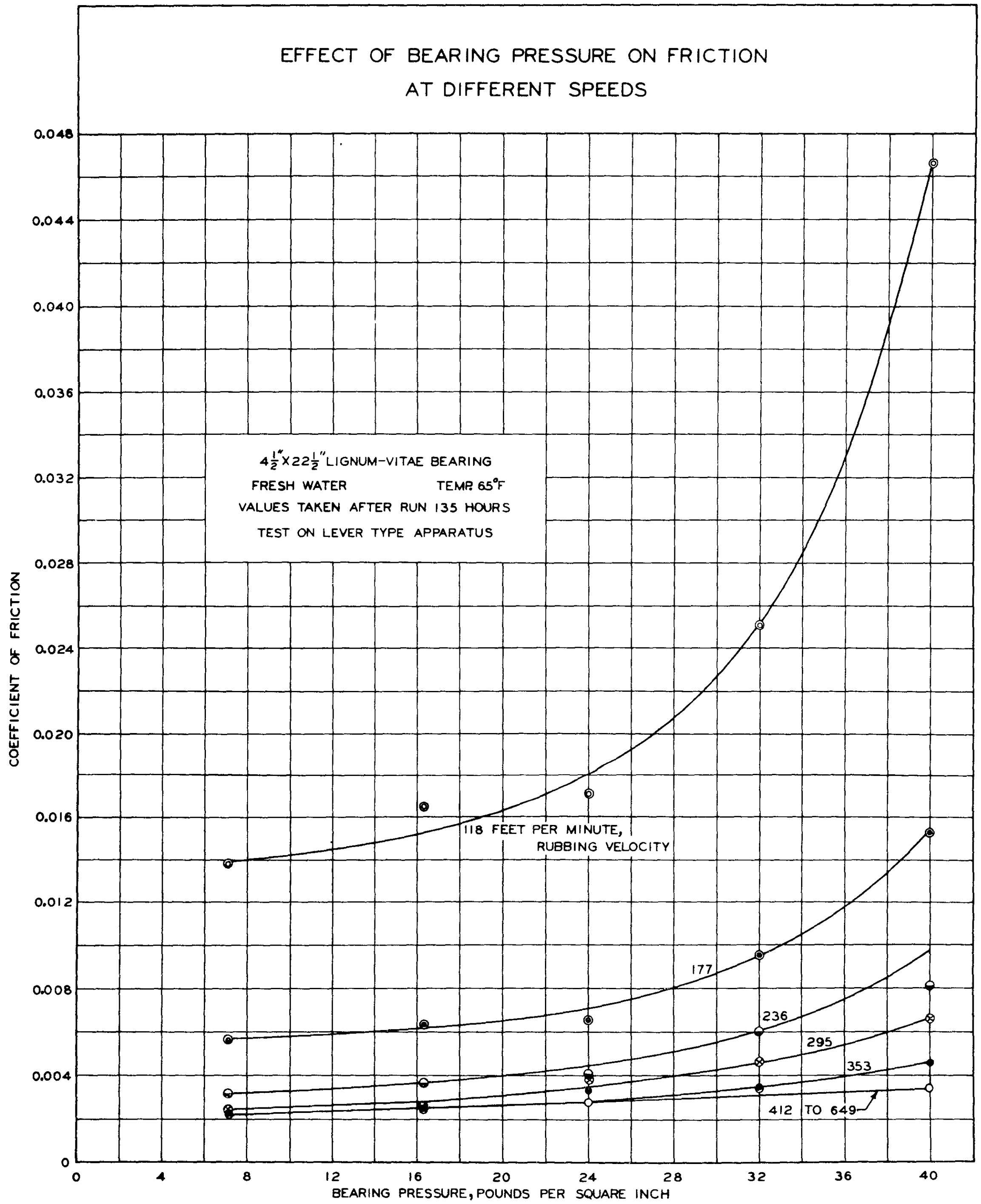
EFFECT OF BEARING PRESSURE ON FRICTION
AT DIFFERENT SPEEDS
4½" X 22½" LIGNUM-VITAE BEARING
FRESH WATER
TEMP. 65°F
VALUES TAKEN AFTER RUN 135 HOURS
TEST ON LEVER TYPE APPARATUS
118 FEET PER MINUTE,
RUBBING VELOCITY
177
236
295
353
412 TO 649
COEFFICIENT OF FRICTION
0.048
0.044
0.040
0.036
0.032
0.028
0.024
0.020
0.016
0.012
0.008
0.004
0
0
4
8
12
16
20
24
28
32
36
40
BEARING PRESSURE, POUNDS PER SQUARE INCH

FIG. 8.

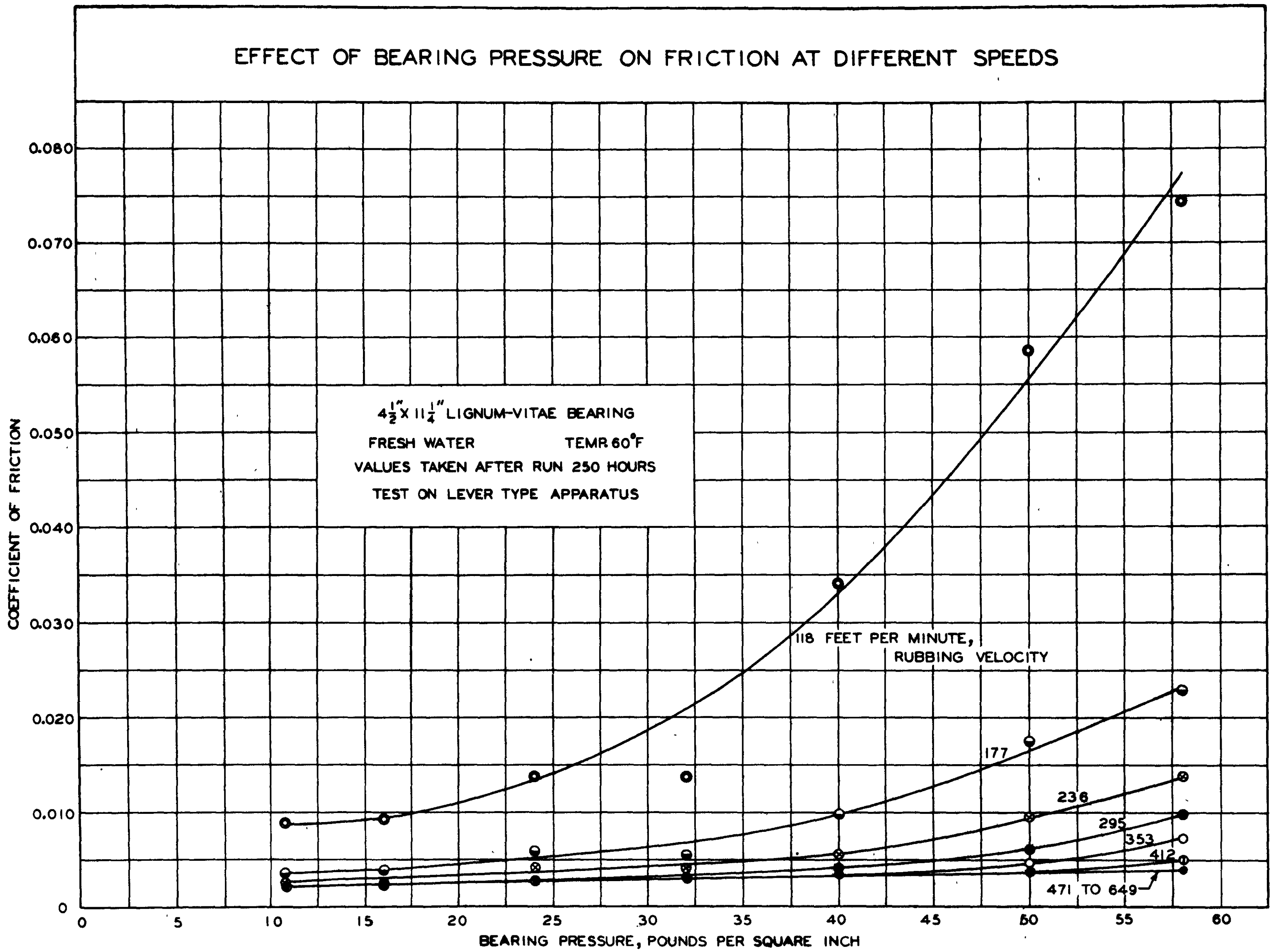
EFFECT OF BEARING PRESSURE ON FRICTION AT DIFFERENT SPEEDS
4½" X 11¼" LIGNUM-VITAE BEARING
FRESH WATER
TEMP. 60°F
VALUES TAKEN AFTER RUN 250 HOURS
TEST ON LEVER TYPE APPARATUS
118 FEET PER MINUTE, RUBBING VELOCITY
177
236
295
353
412
471 TO 649
COEFFICIENT OF FRICTION
0.080
0.070
0.060
0.050
0.040
0.030
0.020
0.010
0
5
10
15
20
25
30
35
40
45
50
55
60
BEARING PRESSURE, POUNDS PER SQUARE INCH

FIG. 9.

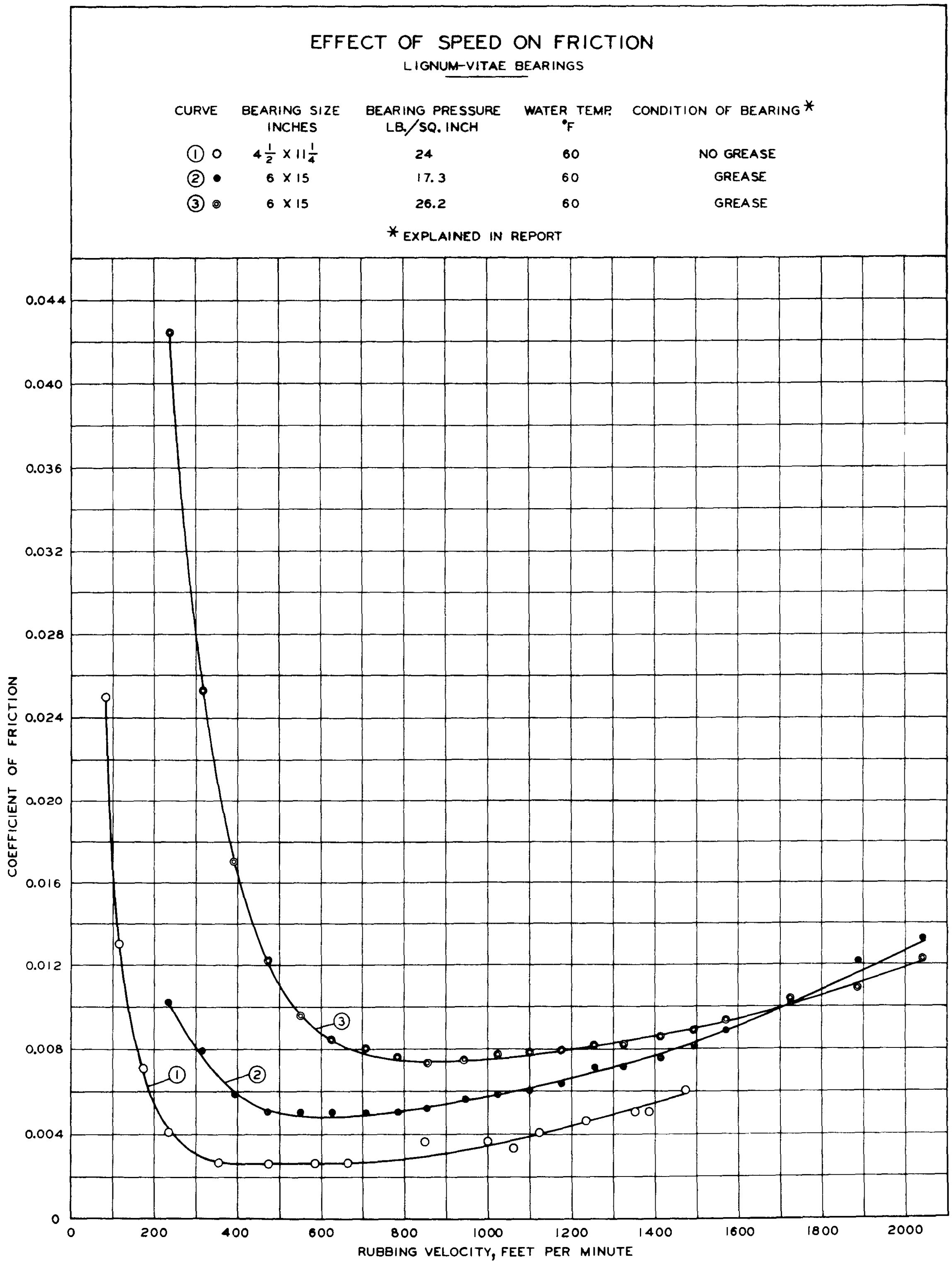
EFFECT OF SPEED ON FRICTION
LIGNUM-VITAE BEARINGS
CURVE
BEARING SIZE INCHES
BEARING PRESSURE LB./SQ. INCH
WATER TEMP. °F
CONDITION OF BEARING *
① ○ 4 1/2 X 11 1/4 24 60 NO GREASE
② ● 6 X 15 17.3 60 GREASE
③ ◎ 6 X 15 26.2 60 GREASE
* EXPLAINED IN REPORT
COEFFICIENT OF FRICTION
0.044
0.040
0.036
0.032
0.028
0.024
0.020
0.016
0.012
0.008
0.004
0
0
200
400
600
800
1000
1200
1400
1600
1800
2000
RUBBING VELOCITY, FEET PER MINUTE
1
2
3

FIG. 10.

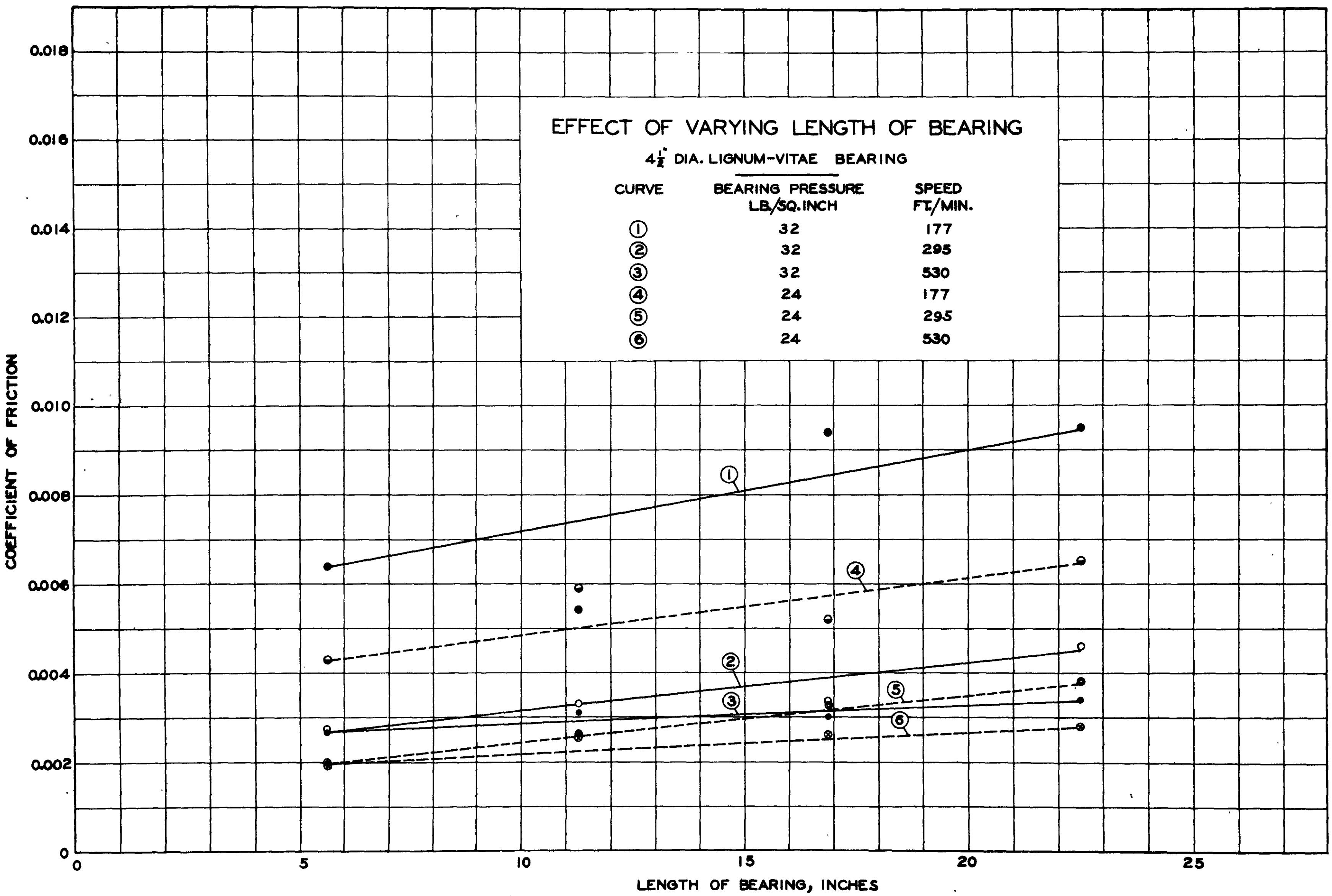
EFFECT OF VARYING LENGTH OF BEARING
4½" DIA. LIGNUM-VITAE BEARING
CURVE
BEARING PRESSURE LB/SQ. INCH
SPEED FT/MIN.
① 32 177
② 32 295
③ 32 530
④ 24 177
⑤ 24 295
⑥ 24 530
COEFFICIENT OF FRICTION
0.018
0.016
0.014
0.012
0.010
0.008
0.006
0.004
0.002
0
LENGTH OF BEARING, INCHES
0
5
10
15
20
25

FIG. 11.

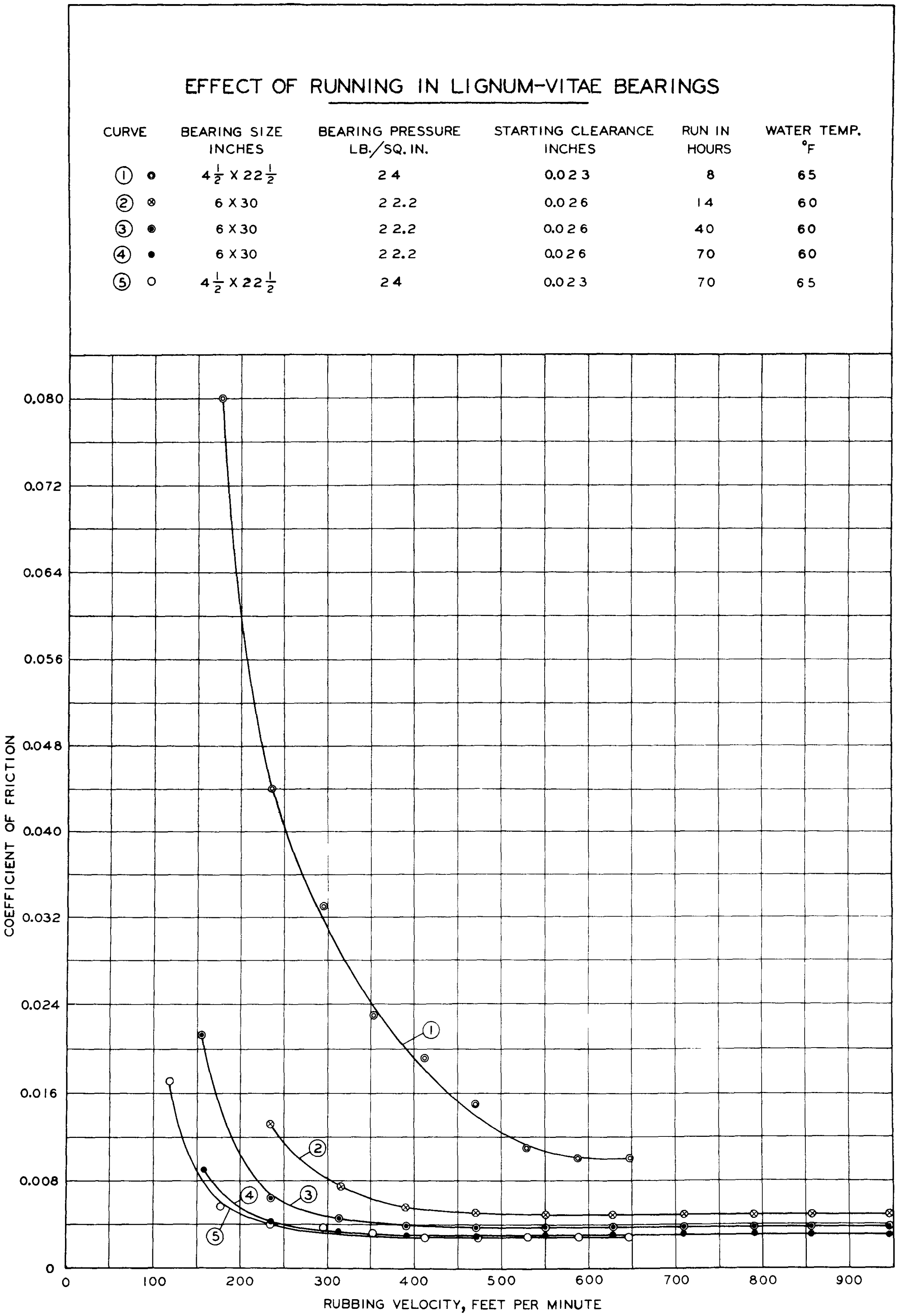
EFFECT OF RUNNING IN LIGNUM-VITAE BEARINGS
CURVE
BEARING SIZE INCHES
BEARING PRESSURE LB./SQ. IN.
STARTING CLEARANCE INCHES
RUN IN HOURS
WATER TEMP. °F
1 4 1/2 X 22 1/2 24 0.023 8 65
2 6 X 30 22.2 0.026 14 60
3 6 X 30 22.2 0.026 40 60
4 6 X 30 22.2 0.026 70 60
5 4 1/2 X 22 1/2 24 0.023 70 65
COEFFICIENT OF FRICTION
0.080
0.072
0.064
0.056
0.048
0.040
0.032
0.024
0.016
0.008
0
RUBBING VELOCITY, FEET PER MINUTE
0
100
200
300
400
500
600
700
800
900

FIG. 12.

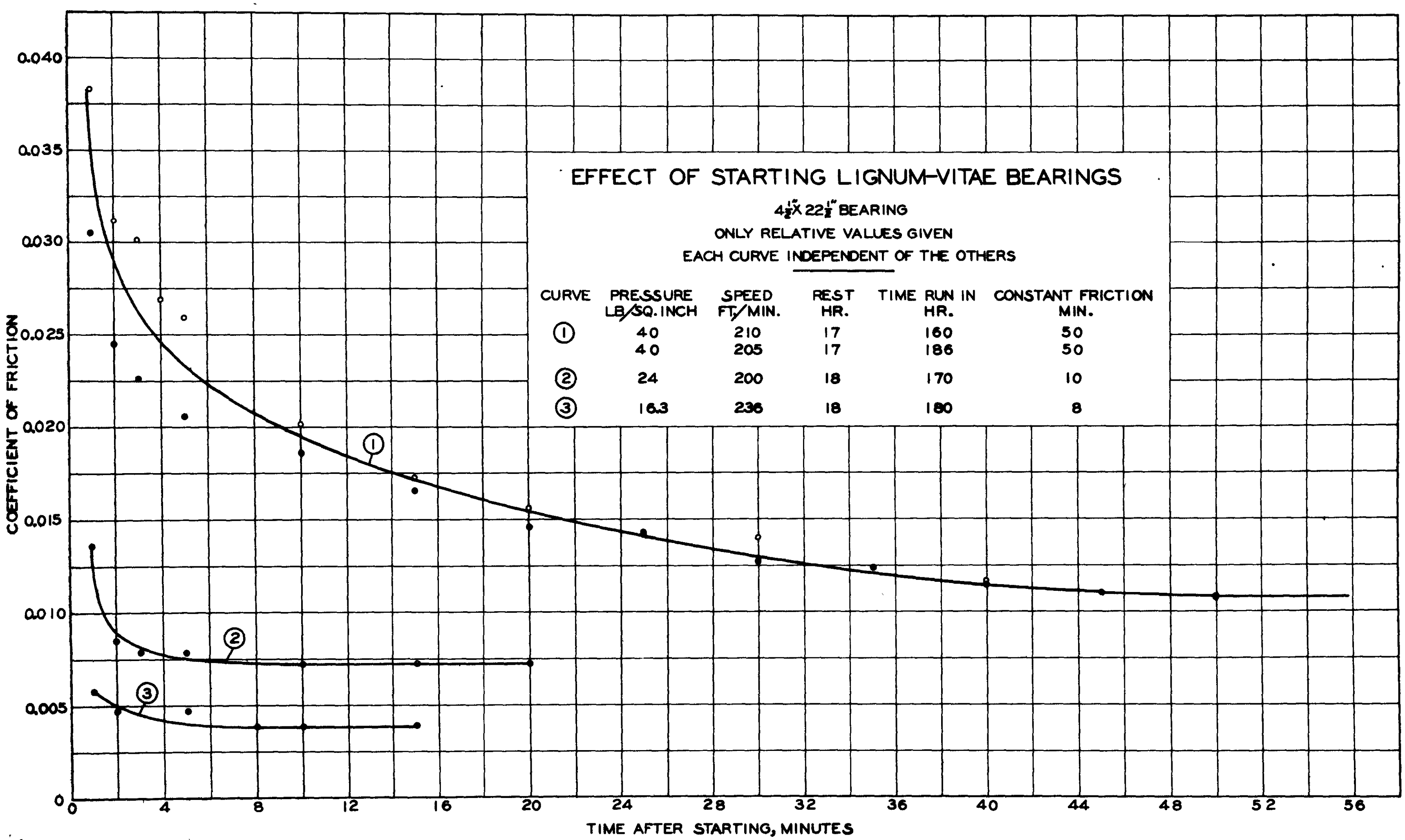
EFFECT OF STARTING LIGNUM-VITAE BEARINGS
4½" X 22½" BEARING
ONLY RELATIVE VALUES GIVEN
EACH CURVE INDEPENDENT OF THE OTHERS
CURVE | PRESSURE LB/SQ. INCH | SPEED FT/MIN. | REST HR. | TIME RUN IN HR. | CONSTANT FRICTION MIN.
1 | 40 | 210 | 17 | 160 | 50
| 40 | 205 | 17 | 186 | 50
2 | 24 | 200 | 18 | 170 | 10
3 | 16.3 | 236 | 18 | 180 | 8
COEFFICIENT OF FRICTION
0.040
0.035
0.030
0.025
0.020
0.015
0.010
0.005
0
0 4 8 12 16 20 24 28 32 36 40 44 48 52 56
TIME AFTER STARTING, MINUTES
1
2
3

FIG. 13.

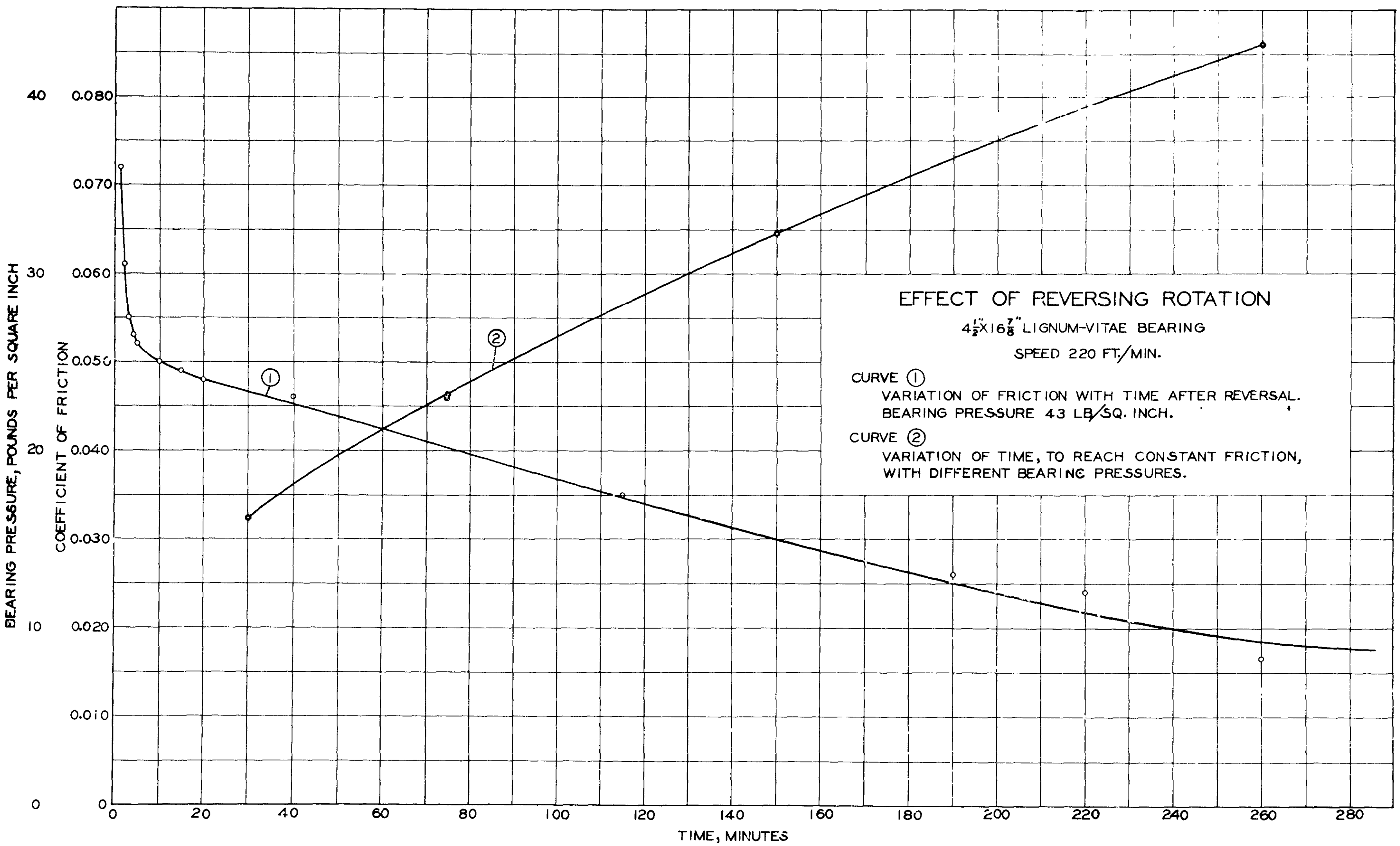
EFFECT OF REVERSING ROTATION
4½"X16⅞" LIGNUM-VITAE BEARING
SPEED 220 FT./MIN.
CURVE 1
VARIATION OF FRICTION WITH TIME AFTER REVERSAL.
BEARING PRESSURE 43 LB/SQ. INCH.
CURVE 2
VARIATION OF TIME, TO REACH CONSTANT FRICTION, WITH DIFFERENT BEARING PRESSURES.
BEARING PRESSURE, POUNDS PER SQUARE INCH
COEFFICIENT OF FRICTION
TIME, MINUTES
0
10
20
30
40
0.010
0.020
0.030
0.040
0.050
0.060
0.070
0.080
20
40
60
80
100
120
140
160
180
200
220
240
260
280

FIG. 14.

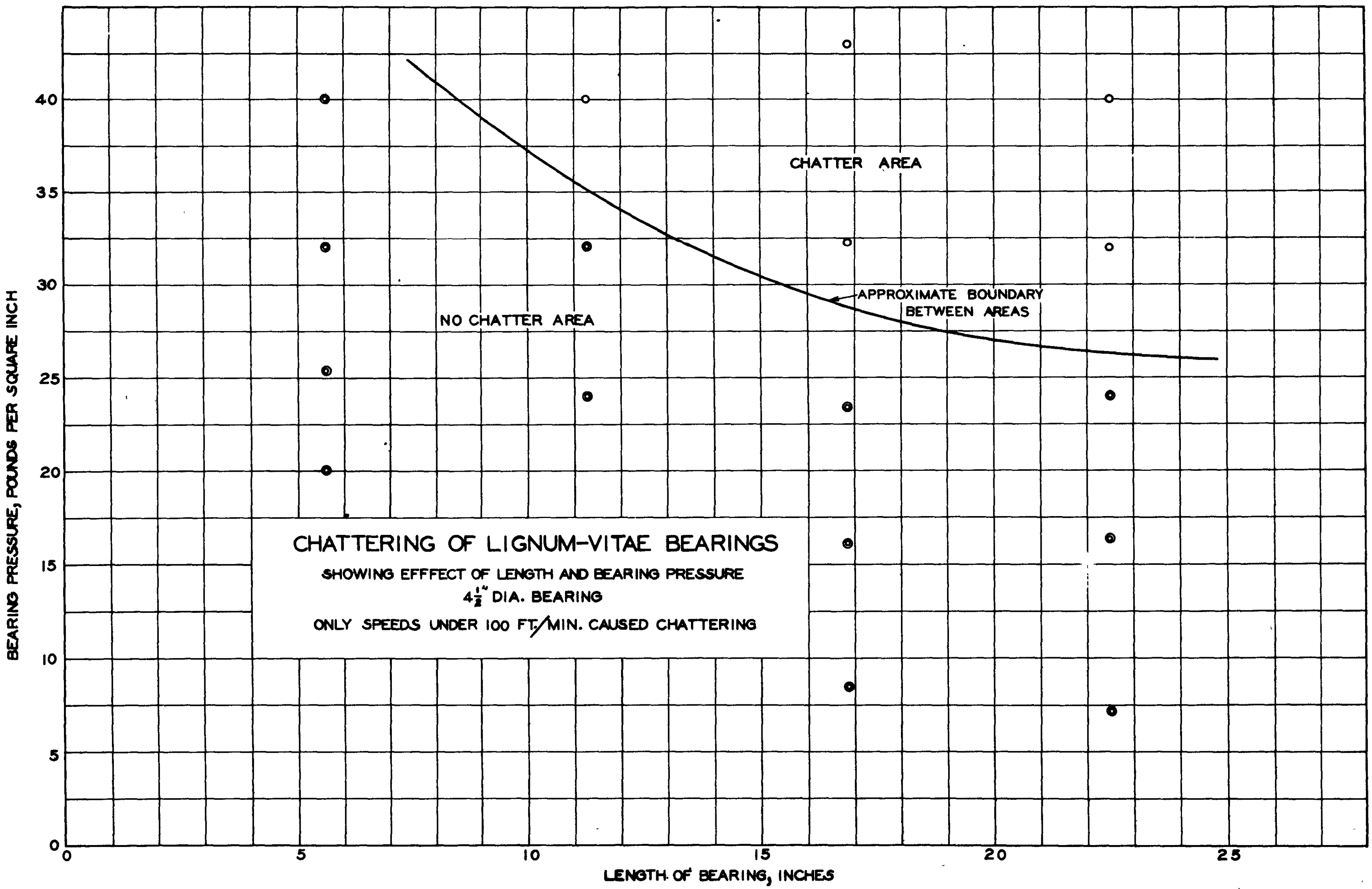
CHATTERING OF LIGNUM-VITAE BEARINGS
SHOWING EFFFECT OF LENGTH AND BEARING PRESSURE
4½" DIA. BEARING
ONLY SPEEDS UNDER 100 FT./MIN. CAUSED CHATTERING
CHATTER AREA
NO CHATTER AREA
APPROXIMATE BOUNDARY BETWEEN AREAS
BEARING PRESSURE, POUNDS PER SQUARE INCH
LENGTH OF BEARING, INCHES
0
5
10
15
20
25
30
35
40

FIG. 15.

Tests on Torsionmeter Type Apparatus

Apparatus. The frictional values, as determined by the lever type apparatus, were considerably lower than expected; therefore a second set-up, designated as the torsionmeter type, was made in order to check the results of the lever type apparatus and to further determine the frictional values under varying conditions.

Figs. 17 and 18 show the set-up for the torsionmeter type apparatus making use of the various parts used in the earlier tests. The journal was directly driven through a special shaft section on which was mounted a 2-1/2 in. Ford torsionmeter, A. An indication of the torque was electrically transmitted to the indicator, B. Two bearing sections for the 6-in. bearing were firmly fastened in the bottom of the tank, see Fig. 16, while the other two sections were fastened in a housing supporting the I-beam. This gave the equivalent of two 6-in. bearings each 15 in. long. The journal was supported by the lower bearing sections.

A 1/16-in. rubber diaphragm was used to check the flow of water from the tank at the motor end. A galvanized steel trough was placed to catch the flow and return it to a storage tank as shown in Fig. 18. Either fresh water could be added or the same water could be returned by a pump.

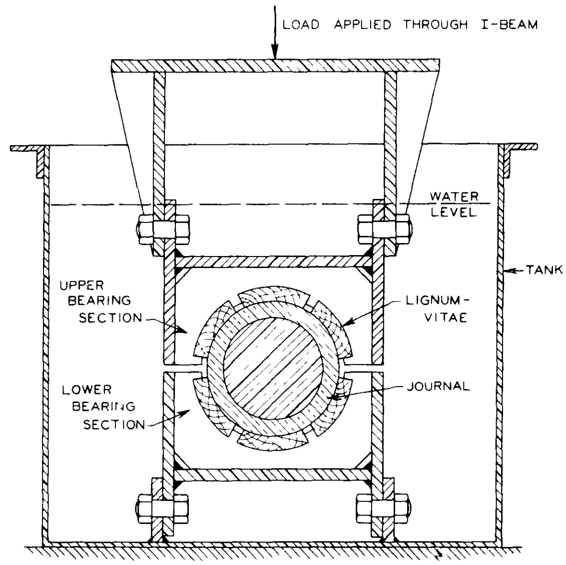

FIG. 16, Arrangement of Bearing for Torsionmeter Method.

Method of Test. The Ford torsionmeter was mounted on the special shaft section and calibrated in the shop. The calibration is shown in Fig. 19. As the special section and the torsionmeter were light, they were assembled as one unit. The zero established during the calibration was checked after assembly.

The desired load was then placed on the weight pans but the load was left off the bearing until a speed of at least 100 R.P.M. was attained on the shaft. Then by means of a chain fall the load was gradually applied.

Readings were taken for various conditions on the bearing only after the friction became constant with constant speed. This condition on starting up, sometimes was not attained until about one hour of running.

The load was always removed from the bearing before stopping the motor.

For this method of test, the shaft was turned in one direction of rotation, only.

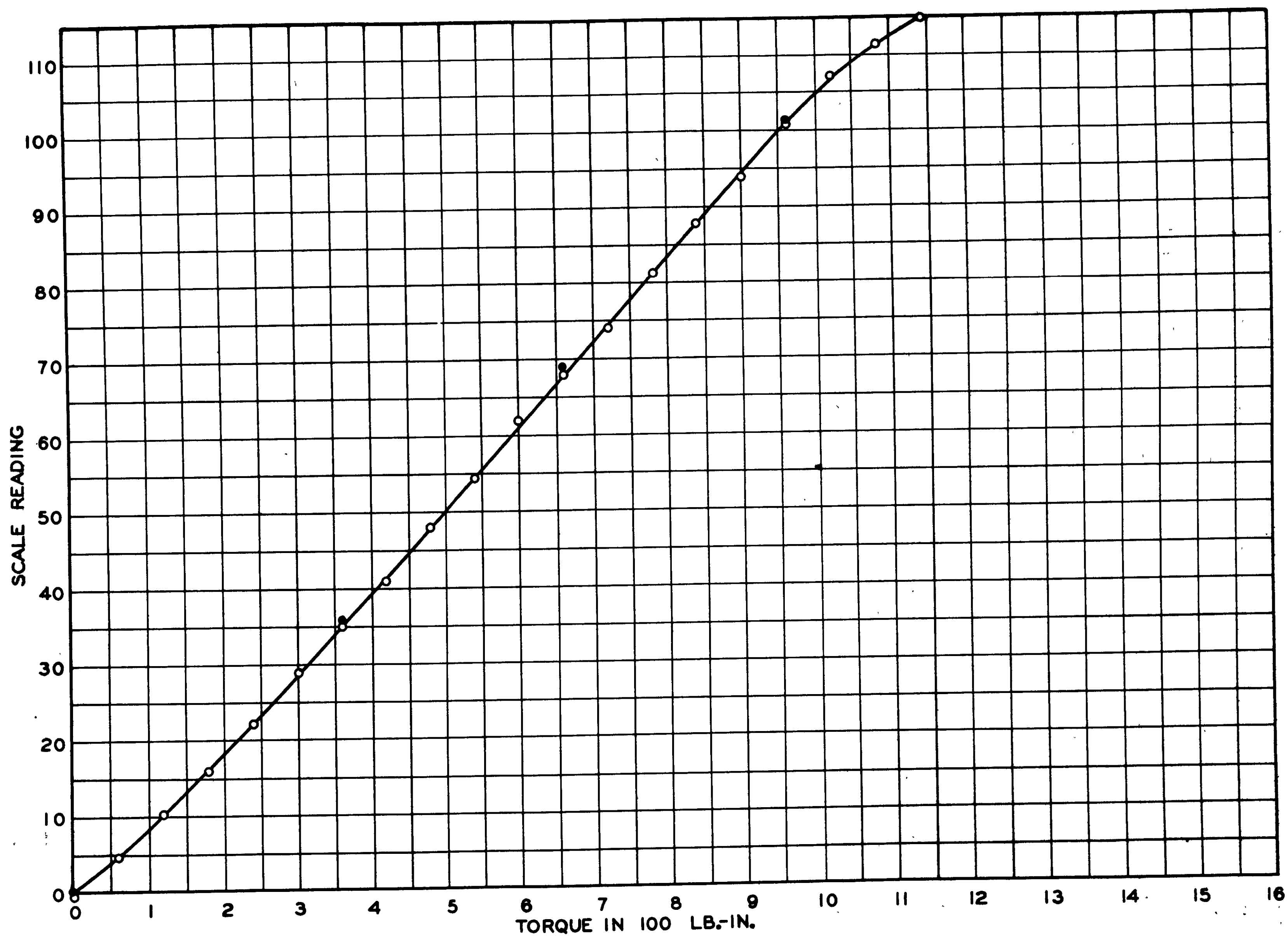

FIG. 19. Calibration for 2-1/2-in. Ford Torsionmeter

Results. Under like conditions, the torsionmeter type apparatus gave frictional values slightly higher than the lever type apparatus.* At the lower speeds, however, the values given by the torsionmeter method were considerably higher than by the lever method.

Temperature had a very decided effect upon the bearing friction. The curves, Fig. 20, (except for the 70° curve) indicate that the friction is higher for increased temperature at low speeds and lower for increased temperature at high speeds. As the temperature of the water would increase, the viscosity would decrease allowing more contact between the bearing and the journal at low speeds, thus increasing the friction, but as the speed increased and the journal became supported by the lubricant, the friction would decrease with decreased viscosity of the lubricant.

No reason can be given for the erratic position of the 70° F. curve.

The gripping of the bearing at low speeds made the torsionmeter on the special shaft chatter to such an extent that the load had to be applied after obtaining a fairly high speed and removed before stopping the motor.

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*See Appendix III.

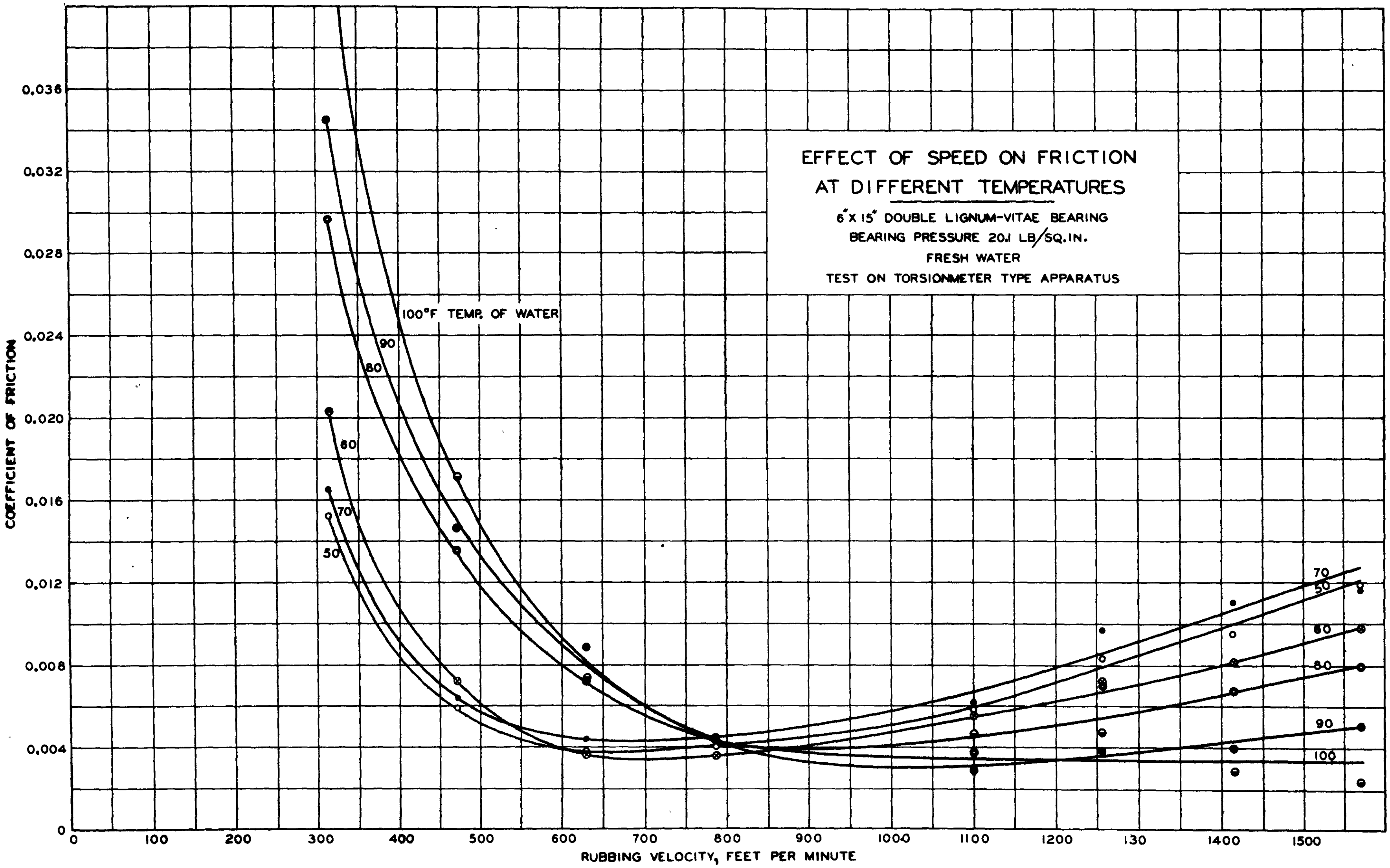
EFFECT OF SPEED ON FRICTION
AT DIFFERENT TEMPERATURES
6" X 15" DOUBLE LIGNUM-VITAE BEARING
BEARING PRESSURE 20.1 LB/SQ.IN.
FRESH WATER
TEST ON TORSIONMETER TYPE APPARATUS
100°F TEMP. OF WATER
90
80
60
70
50
COEFFICIENT OF FRICTION
0.036
0.032
0.028
0.024
0.020
0.016
0.012
0.008
0.004
0
RUBBING VELOCITY, FEET PER MINUTE
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
130
1400
1500
70
50
60
80
90
100

FIG. 20.

Tests on Stuffing Boxes

Apparatus. Stuffing boxes as used on board ship were tested to determine the friction loss. Three types were tested to determine, if possible, any effect of stuffing box design on friction. Fig. 21 shows a stuffing box of standard dimensions. Fig. 22 shows the stuffing box as altered in accord with the American Marine Standard E No. 8–1927. Fig. 23 shows the same box made over to be used with a lantern ring so that a lubricating grease may be used, supposedly for the reduction of friction.

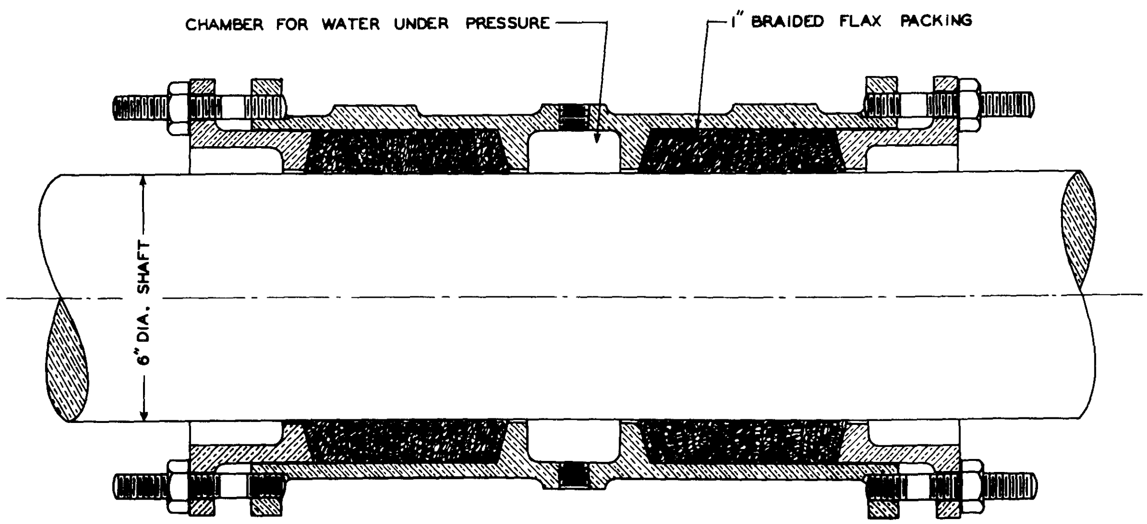

FIG. 21, CROSS–SECTION OF STUFFING BOX AS FIRST TESTED.

The stuffing boxes were mounted on a 6–in. shaft as shown in Fig. 24. The stuffing boxes and measuring mechanism were counterbalanced by weights, the reactions of which were transmitted through the flexible wires, A, at right angles to the measuring arm. The set–up resembled that of a Prony brake, the frictional resistance measured being that between the packing and the shaft. The desired water pressure at the packing was transmitted through the hose, B.

Method of Test. The stuffing box, measuring devices, and hose connections were placed on knife edges, and perfectly balanced before being assembled on the shaft. This made it necessary to take readings in one direction of rotation, only.

First, 4 rings of one inch flax packing were placed in each stuffing box. The glands were set up only finger tight and allowed to set 15 hours or more under water pressure before any testing was done. The packing was then run in for several hours until most of the tallow ran out and the friction reached a fairly constant value with a constant speed. The water system was then cleaned out, as it was found that dirty and greasy water raised the frictional values, and was re–

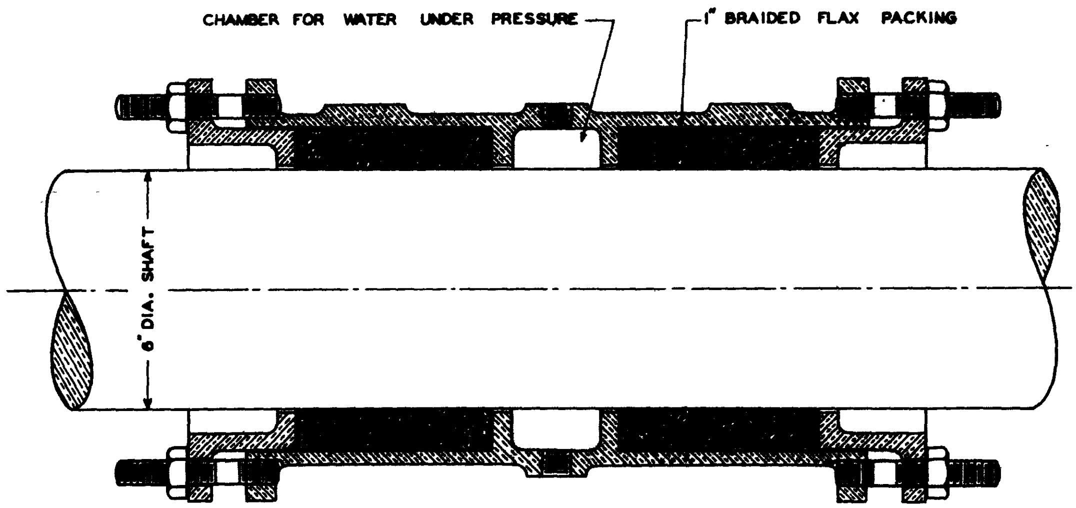

FIG. 22, CROSS-SECTION OF STUFFING BOX AS ALTERED IN ACCORD WITH AMERICAN MARINE STANDARD E NO. 8-1927

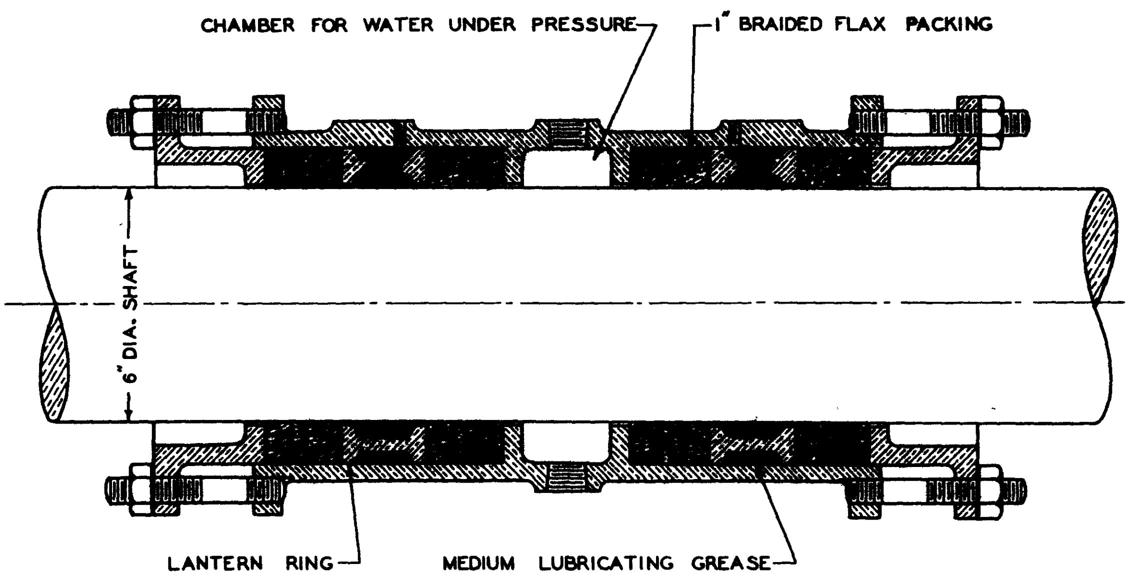

FIG.23, CROSS-SECTION OF STUFFING BOX AS USED WITH LANTERN RING.

filled with clean salt water (density 1.025). Several runs were made with variation of speeds and with various amounts of setting up on the glands, also the packing was replaced by new packing for each type of stuffing box.

The above procedure was followed with but few exceptions; because the stuffing boxes could not easily accommodate 6 rings, the glands were set up tight when the packing was first placed in the boxes; in the tests with two rings of packing each side of the lantern rings, the glands were set up tight when first placed in the boxes; because grease from the lantern rings kept continuously seeping through with the water, tests in this case were made with dirty and greasy water.

Results. Figs. 25 to 30 inclusive give the results of tests on stuffing boxes.

No difference in friction was detected between the stuffing box of standard dimensions and that altered in accord with the American Marine standard E No. 8-1927.

Tests made on the stuffing boxes with the lantern rings were unsatisfactory. The friction was quite low when one ring of packing was placed each side the lantern ring but this was not sufficient packing to hold the grease. There was not sufficient space to efficiently accommodate two rings of packing each side the lantern rings without setting the glands up tightly. Grease still continued to seep through with the water making it very greasy. Values measured in this condition were quite high as shown in Fig. 28.

The friction was erratic and was higher just after the glands were tightened than after the packing was run an hour or more. Curve 1, Fig. 25, is the result of a run being made just after the glands were tightened.

Dirty or greasy water passing through the packing seemed to raise the frictional values. Curve 3, Fig. 25, is a representative run.

Because sufficient means of cooling were not available, the stuffing boxes attained fairly high temperatures. At times steam was generated within the stuffing boxes.

A peculiar characteristic of the friction of packing is shown in curve 6, Fig. 27. When runs were made with increasing speeds, the friction decreased slightly with increased speeds; but at a speed between 400 and 600 feet per minute, the friction rose quite rapidly with constant speed but if left at that speed for 5 or 10 minutes the friction dropped back to its original value. This was more noticeable when the glands were set up tight, but could be detected in other runs. The stuffing box usually increased in temperature with time and naturally increased more rapidly with the higher speeds.

Three representative curves for running in packing are shown in Fig. 30. The friction increased quite rapidly right after starting even after a constant speed was attained. After being run a short time, the friction dropped to a fairly constant value. This happened even after a short shut down.

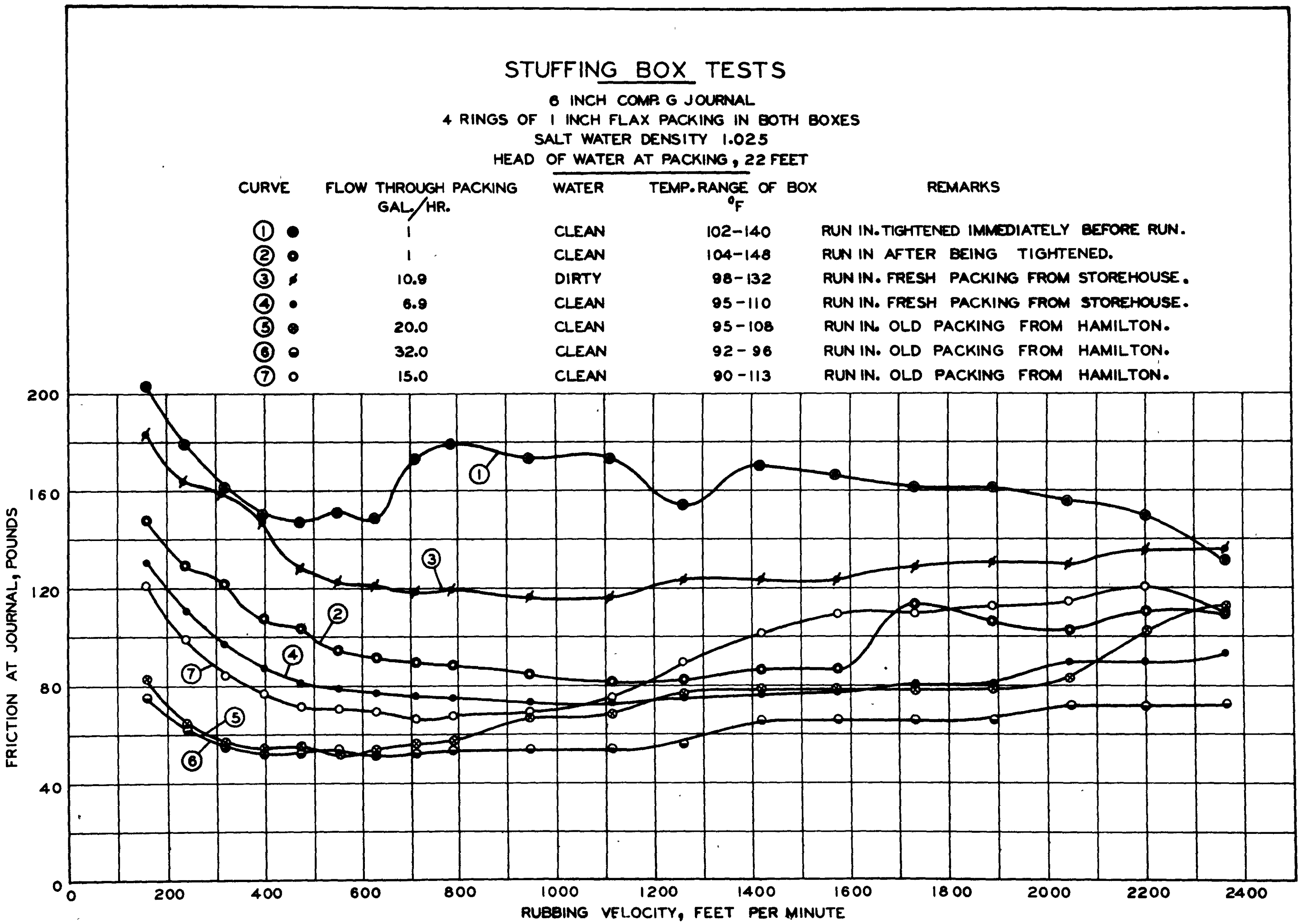
STUFFING BOX TESTS
6 INCH COMP. G JOURNAL
4 RINGS OF 1 INCH FLAX PACKING IN BOTH BOXES
SALT WATER DENSITY 1.025
HEAD OF WATER AT PACKING, 22 FEET
CURVE
FLOW THROUGH PACKING GAL./HR.
WATER
TEMP. RANGE OF BOX °F
REMARKS
1 1 CLEAN 102-140 RUN IN. TIGHTENED IMMEDIATELY BEFORE RUN.
2 1 CLEAN 104-148 RUN IN AFTER BEING TIGHTENED.
3 10.9 DIRTY 98-132 RUN IN. FRESH PACKING FROM STOREHOUSE.
4 6.9 CLEAN 95-110 RUN IN. FRESH PACKING FROM STOREHOUSE.
5 20.0 CLEAN 95-108 RUN IN. OLD PACKING FROM HAMILTON.
6 32.0 CLEAN 92-96 RUN IN. OLD PACKING FROM HAMILTON.
7 15.0 CLEAN 90-113 RUN IN. OLD PACKING FROM HAMILTON.
FRICTION AT JOURNAL, POUNDS
0
40
80
120
160
200
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
RUBBING VELOCITY, FEET PER MINUTE

FIG. 25.

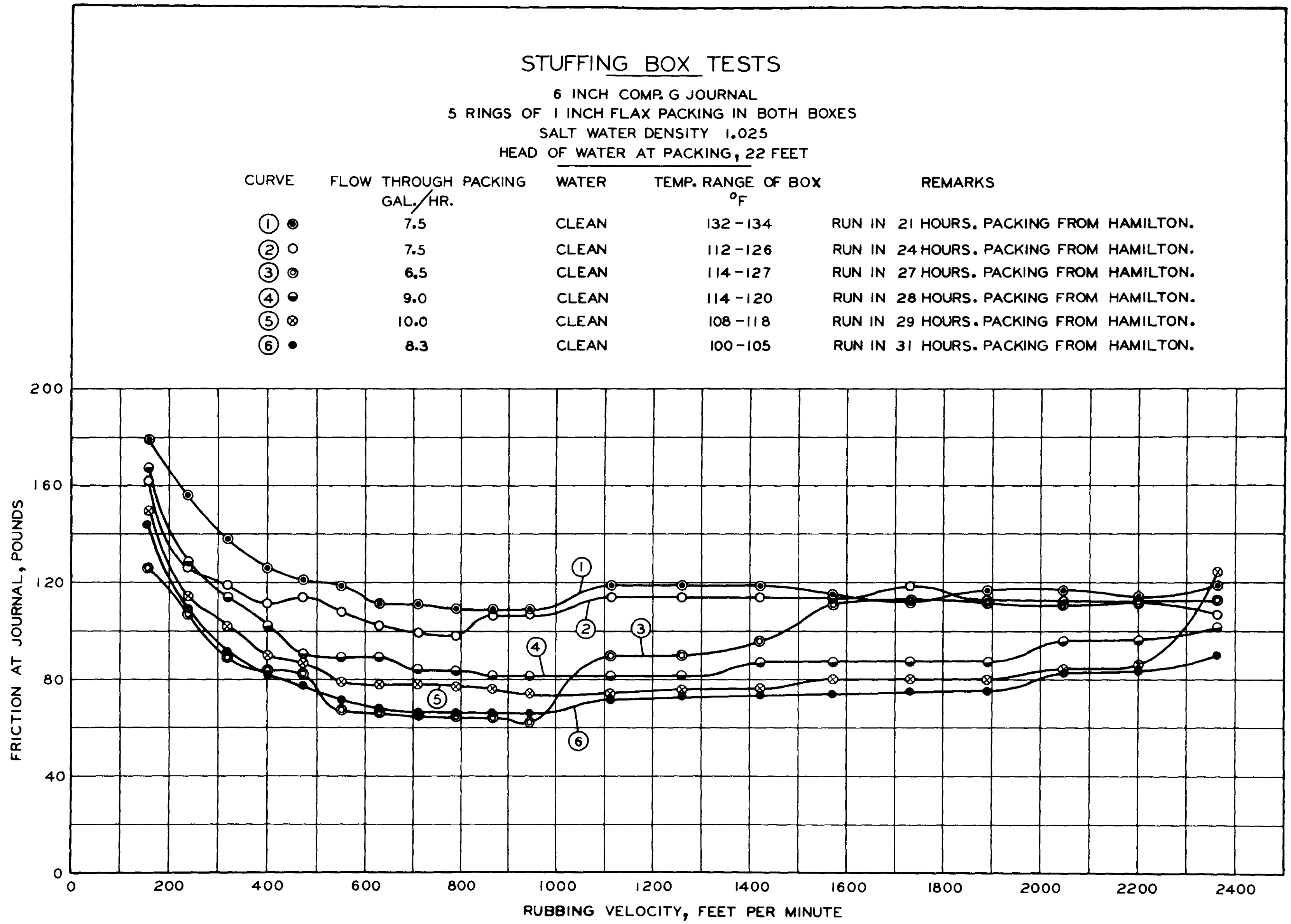
STUFFING BOX TESTS
6 INCH COMP. G JOURNAL
5 RINGS OF 1 INCH FLAX PACKING IN BOTH BOXES
SALT WATER DENSITY 1.025
HEAD OF WATER AT PACKING, 22 FEET
CURVE | FLOW THROUGH PACKING GAL./HR. | WATER | TEMP. RANGE OF BOX °F | REMARKS
1 | 7.5 | CLEAN | 132-134 | RUN IN 21 HOURS. PACKING FROM HAMILTON.
2 | 7.5 | CLEAN | 112-126 | RUN IN 24 HOURS. PACKING FROM HAMILTON.
3 | 6.5 | CLEAN | 114-127 | RUN IN 27 HOURS. PACKING FROM HAMILTON.
4 | 9.0 | CLEAN | 114-120 | RUN IN 28 HOURS. PACKING FROM HAMILTON.
5 | 10.0 | CLEAN | 108-118 | RUN IN 29 HOURS. PACKING FROM HAMILTON.
6 | 8.3 | CLEAN | 100-105 | RUN IN 31 HOURS. PACKING FROM HAMILTON.
FRICTION AT JOURNAL, POUNDS
0
40
80
120
160
200
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
RUBBING VELOCITY, FEET PER MINUTE
1
2
3
4
5
6

FIG. 26.

STUFFING BOX TESTS

6 INCH COMP. G JOURNAL
6 RINGS OF 1 INCH FLAX PACKING IN BOTH BOXES
SALT WATER DENSITY 1.025
HEAD OF WATER AT PACKING, 22 FEET

CURVE	FLOW THROUGH PACKING	WATER	TEMP. RANGE OF BOX °F	REMARKS
1	DRIP ONLY	CLEAN	176 - 180	AFTER 32 HR. RUNNING. PACKING FROM STOREHOUSE.
2	DRIP ONLY	CLEAN	163 - 172	COMPLETE RUN MADE IN 5 MINUTES
3	DRIP ONLY	CLEAN	166 - 178	COMPLETE RUN MADE IN 5.5 MINUTES
4	DRIP & SMALL STREAM	CLEAN	164 - 168	COMPLETE RUN MADE IN 5 MINUTES
5	DRIP & SMALL STREAM	CLEAN	172 - 175	COMPLETE RUN MADE IN 5 MINUTES
6	DRIP & SMALL STREAM	CLEAN	155 - 185	EACH READING MADE AFTER 2.5 MINUTES RUNNING AT CONSTANT SPEED

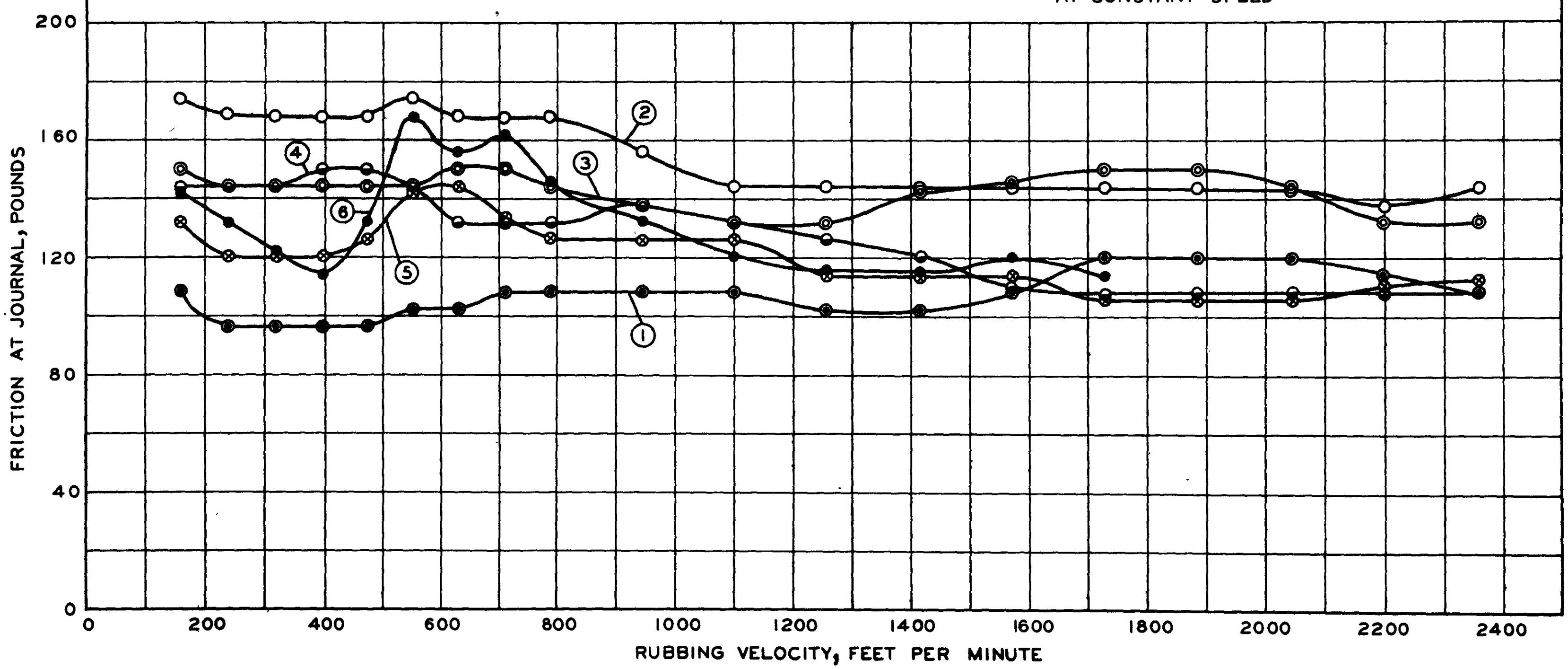

FIG. 27.

STUFFING BOX TESTS

6 INCH COMP. G JOURNAL

2 RINGS OF 1" PACKING, EACH SIDE OF LANTERN RING, IN BOTH BOXES

MEDIUM LUBRICATING GREASE IN LANTERN RINGS

SALT WATER DENSITY 1.020

HEAD OF WATER AT PACKING, 22 FEET

CURVE	FLOW THROUGH PACKING	WATER	TEMP. RANGE OF BOX °F	REMARKS
1 ⊙	DRIP ONLY	DIRTY	115-140	RUN IN 9.0 HOURS. SET UP TIGHT.
2 ○	DRIP ONLY	DIRTY	150-170	RUN IN 9.5 HOURS. SET UP TIGHT.
3 ◎	DRIP ONLY	DIRTY	165-176	RUN IN 10.5 HOURS. SET UP TIGHT.
4 ●	DRIP ONLY	DIRTY	176-180	RUN IN 11.0 HOURS. SET UP TIGHT.
5 ⊗	DRIP ONLY	DIRTY	170-182	RUN IN 12.0 HOURS. SET UP TIGHT.

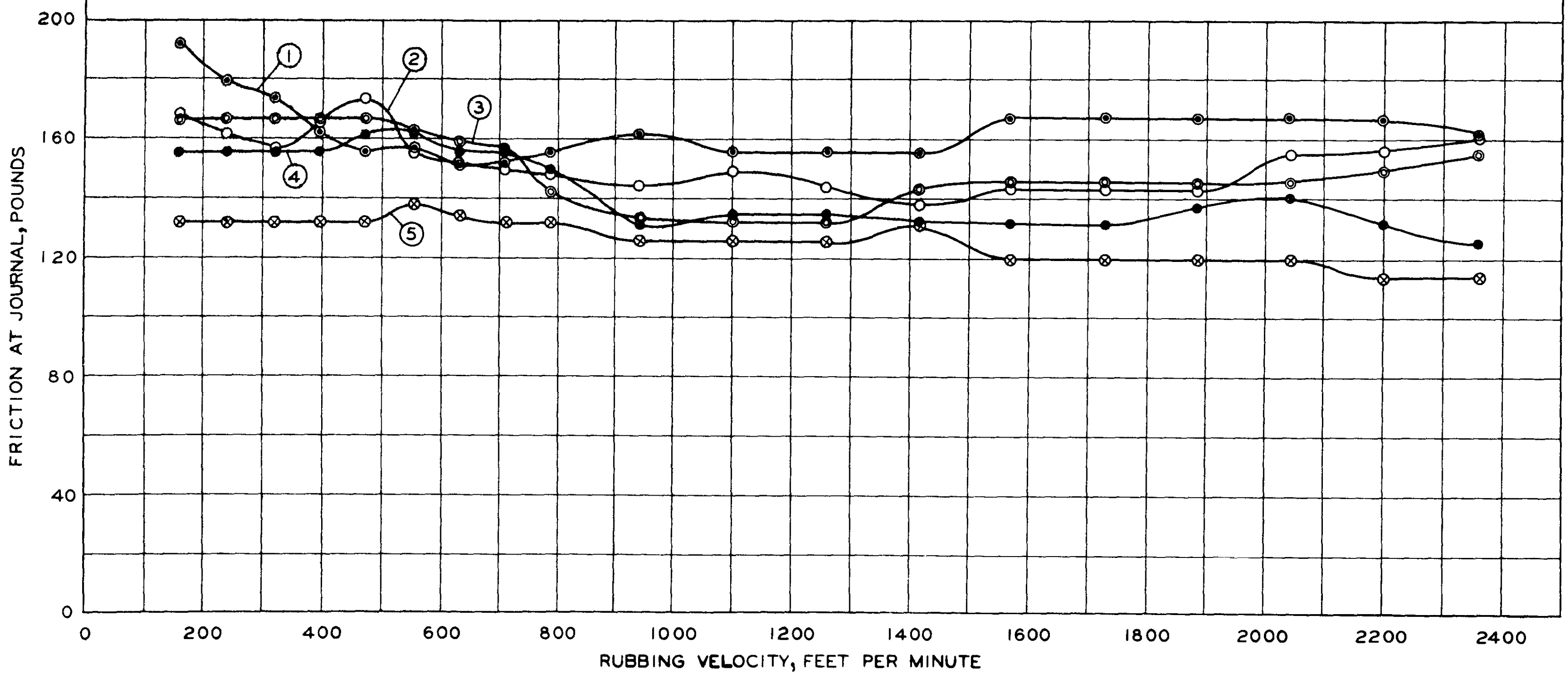

FIG. 28.

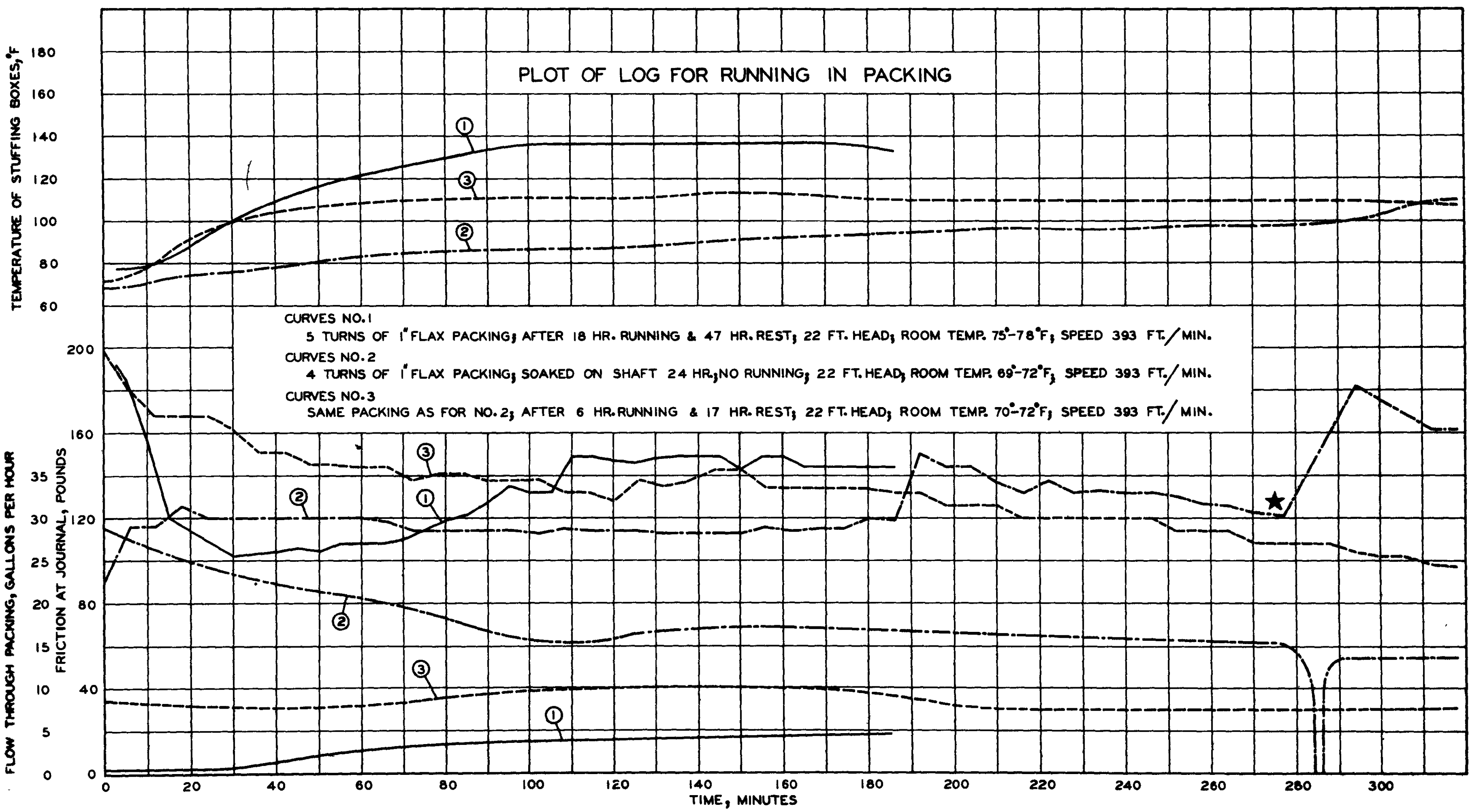
PLOT OF LOG FOR RUNNING IN PACKING
CURVES NO. 1
5 TURNS OF 1" FLAX PACKING; AFTER 18 HR. RUNNING & 47 HR. REST; 22 FT. HEAD; ROOM TEMP. 75°-78°F; SPEED 393 FT./MIN.
CURVES NO. 2
4 TURNS OF 1" FLAX PACKING; SOAKED ON SHAFT 24 HR.; NO RUNNING; 22 FT. HEAD; ROOM TEMP. 69°-72°F; SPEED 393 FT./MIN.
CURVES NO. 3
SAME PACKING AS FOR NO. 2; AFTER 6 HR. RUNNING & 17 HR. REST; 22 FT. HEAD; ROOM TEMP. 70°-72°F; SPEED 393 FT./MIN.
TEMPERATURE OF STUFFING BOXES, °F
FLOW THROUGH PACKING, GALLONS PER HOUR
FRICTION AT JOURNAL, POUNDS
TIME, MINUTES

FIG. 29.

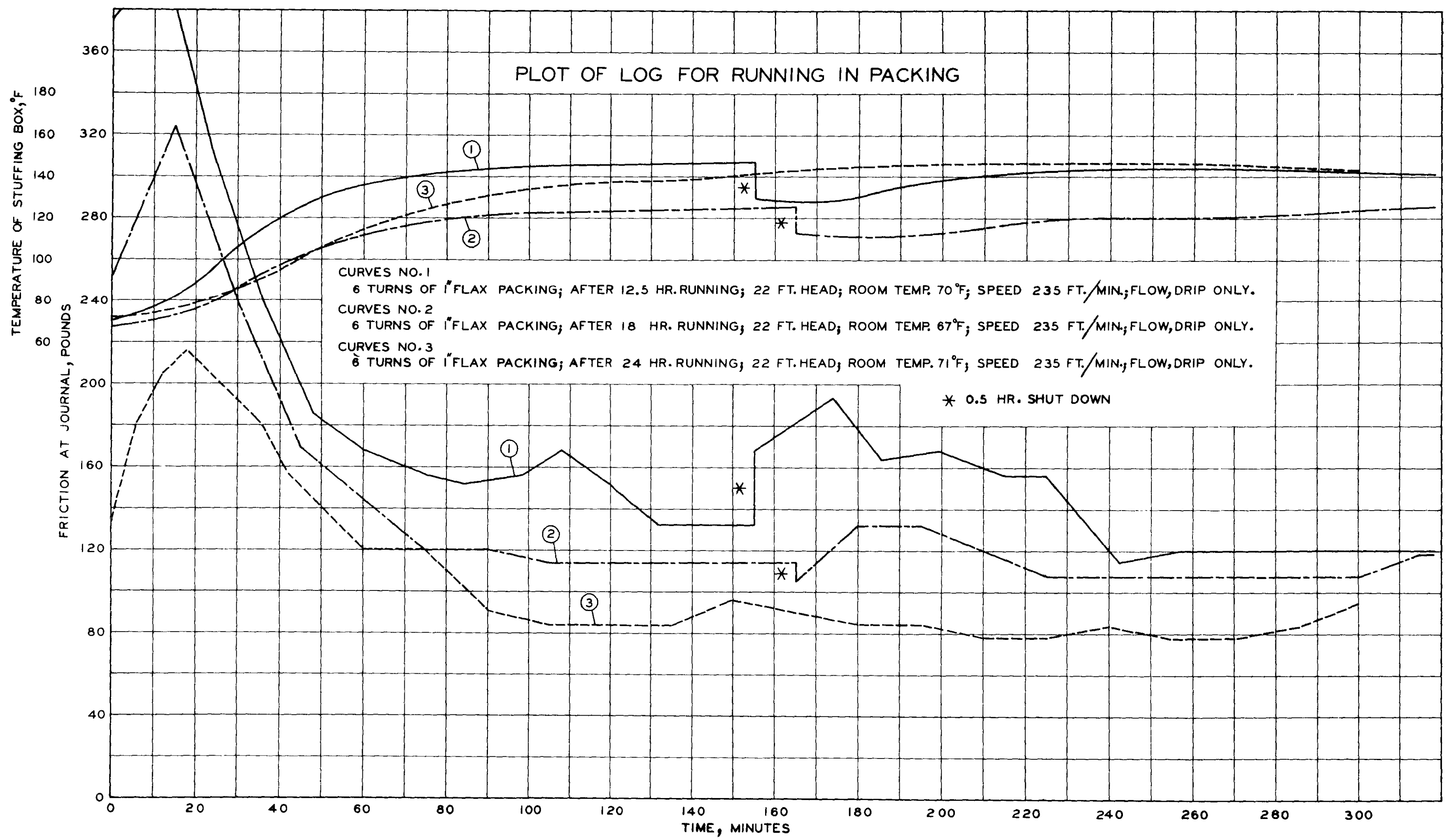
PLOT OF LOG FOR RUNNING IN PACKING
CURVES NO. 1
6 TURNS OF 1" FLAX PACKING; AFTER 12.5 HR. RUNNING; 22 FT. HEAD; ROOM TEMP. 70°F; SPEED 235 FT./MIN.; FLOW, DRIP ONLY.
CURVES NO. 2
6 TURNS OF 1" FLAX PACKING; AFTER 18 HR. RUNNING; 22 FT. HEAD; ROOM TEMP. 67°F; SPEED 235 FT./MIN.; FLOW, DRIP ONLY.
CURVES NO. 3
6 TURNS OF 1" FLAX PACKING; AFTER 24 HR. RUNNING; 22 FT. HEAD; ROOM TEMP. 71°F; SPEED 235 FT./MIN.; FLOW, DRIP ONLY.
* 0.5 HR. SHUT DOWN
TEMPERATURE OF STUFFING BOX, °F
FRICTION AT JOURNAL, POUNDS
TIME, MINUTES
1
2
3
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
360

FIG. 30.

Full Scale Tests on U.S.S. HAMILTON

Apparatus. Tests were made on the U.S.S. HAMILTON starboard shaft. The bearings and their position are shown in Fig. 33. The friction measured was that abaft the torsionmeter section.

The bearings were checked for clearance with the propeller removed. The results are as follows:

BEARING	SHAFT CLEARANCE, inches			
	Outboard	Top	Bottom	Inboard
Inboard stern tube	0.091	0.141	0.000	0.091
Outboard stern tube	0.060	0.117	0.000	0.056
Strut .... Forward	0.128	0.140	0.040	0.120
Strut .... Aft	0.062	0.109	0.008	0.062

The diameter of all journals was 12.250 in.

The inboard and outboard stern tube bearings were not rewooded since recommissioning. The strut bearing was rewooded May 30, 1930. The subsequent mileage was 45,127.

The line shaft bearing was new as was its journal.

The stuffing box contained 6 rings of one inch flax packing well run in.

A section of the regular line shaft was removed and replaced by a special torsionmeter shaft section and special shaft. The running torque on the shaft was expected to be comparatively low while the starting torque was expect to be high; therefore a special torsionmeter shaft section was made for mounting the ship's regular Ford torsionmeter, see Fig. 31. The stops on the sleeves were set to give a definite clearance between the sleeves. The running torque was transmitted by the 2-1/2 in. shaft but if the torque became excessive, the clearance between the sleeves closed so that the sleeves would help transmit the torque.

The propeller was removed and replaced by a dummy hub of the same weight. It was desired to keep the same size as the propeller hub but in order to keep the weight, the hub was larger. The dimensions of the hub are given in Fig. 41, Appendix IV.

Method of Test. The calibration of the ship's Ford torsionmeter and special shaft section was made on board with the shafting in place. The ship with even keel was resting on blocks in dry dock. All shaft sections were moved 1/4 in. aft of the regular position so that the forward face of the flange on the special shaft would clear the after face of the flange on the reduction gear shaft. The special shaft was a shaft section, made of heavy pipe, placed between the torsionmeter shaft section and the reduction gear shaft.

The line shaft was clamped in the after handling room which was aft of the

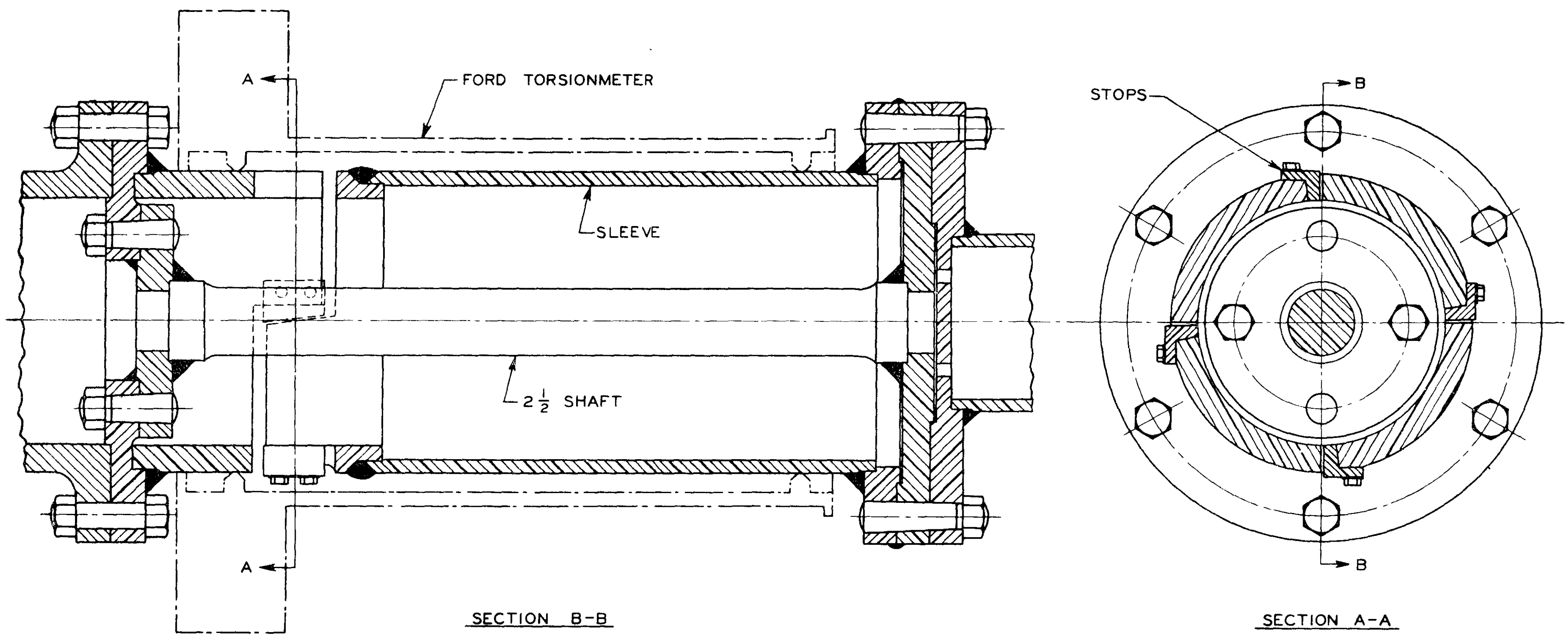

FIG. 31. Special Torsionmeter Shaft Section

torsionmeter section. Calibration arms were bolted on the forward flange of the special shaft with a weight carrying device on the outboard arm and a force-measuring device on the inboard arm. The arms were nearly horizontal. The special shaft was supported by rollers.

After the first preliminary calibration and checking of the set-up, a preliminary run was made to stress the special torsionmeter section. The shaft was run from zero to full speed for about 1-1/4 hours, stopping and starting three times at intervals.

After this preliminary run, a careful calibration was made, see Fig. 32.

The dry dock was flooded until the water was approximately 4 feet above the center of the shaft at the dummy hub. This covered all wood bearings and sea connections but did not lift the ship from the blocks.

The stuffing box was set up to typical running condition.

The shaft was run for one hour before any recorded data were taken. After one hour of running the bearings for the reduction gears were at running temperature. Readings were taken at various speeds.

After this test the gland of the stuffing box was slacked off. Readings were taken at two different speeds.

The gland of the stuffing box was then tightened excessively. Readings were taken at several speeds.

The gland was then slacked off to running conditions.

After shutting down the turbine, the shaft was jacked over by hand so that the torque could be determined for starting and at low speeds.

Results. The results of the friction test on the U.S.S. HAMILTON starboard shaft are given in Fig. 33. The friction measured is that due to the three lignum-vitae bearings, the line shaft bearing, the stuffing box, the fluid friction on the dummy hub, and any windage that may occur abaft the torsionmeter section. As in the laboratory friction tests, the starting torque was quite high but reduced rapidly with increased speed. The instant that the shaft began to turn, an increased torque was noticeable as indicated by the large scale plotting.

Calculations for the shaft friction, see Appendix IV, compare quite favorably with the actual test values.

When starting up and at low speeds, the armature of the Ford torsionmeter was seen to move back and forth with a period easily detected by the eye. This indicated a gripping and letting loose of the bearings the same as noticed in the laboratory tests.

The center of the shaft was approximately 4 feet under water at the dummy hub, but at the higher speeds, around 500 R.P.M., the resistance of the dummy hub greatly agitated the surface of the water.

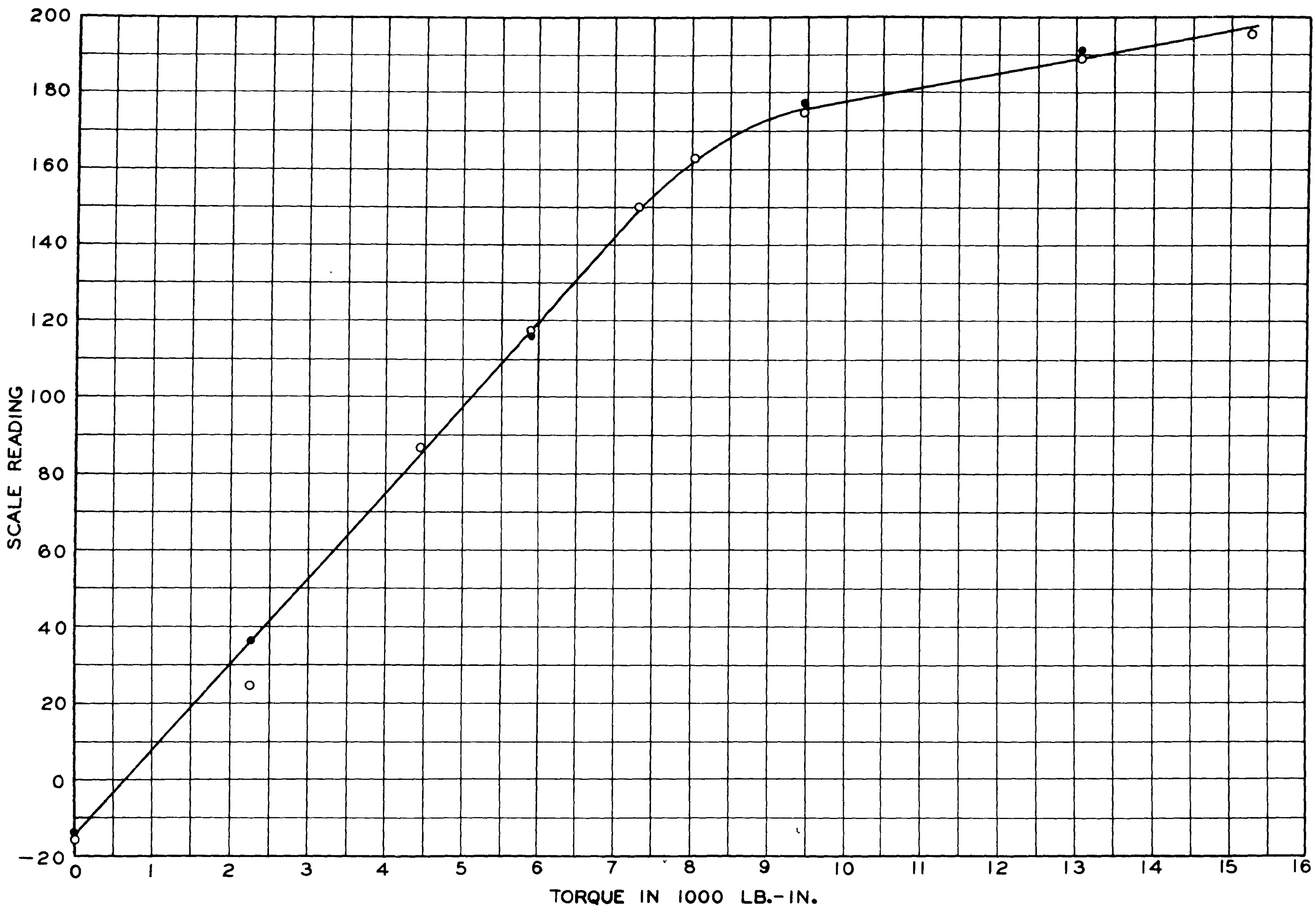

FIG. 32. Calibration of Hamilton Ford Torsionmeter on Special Shaft Section

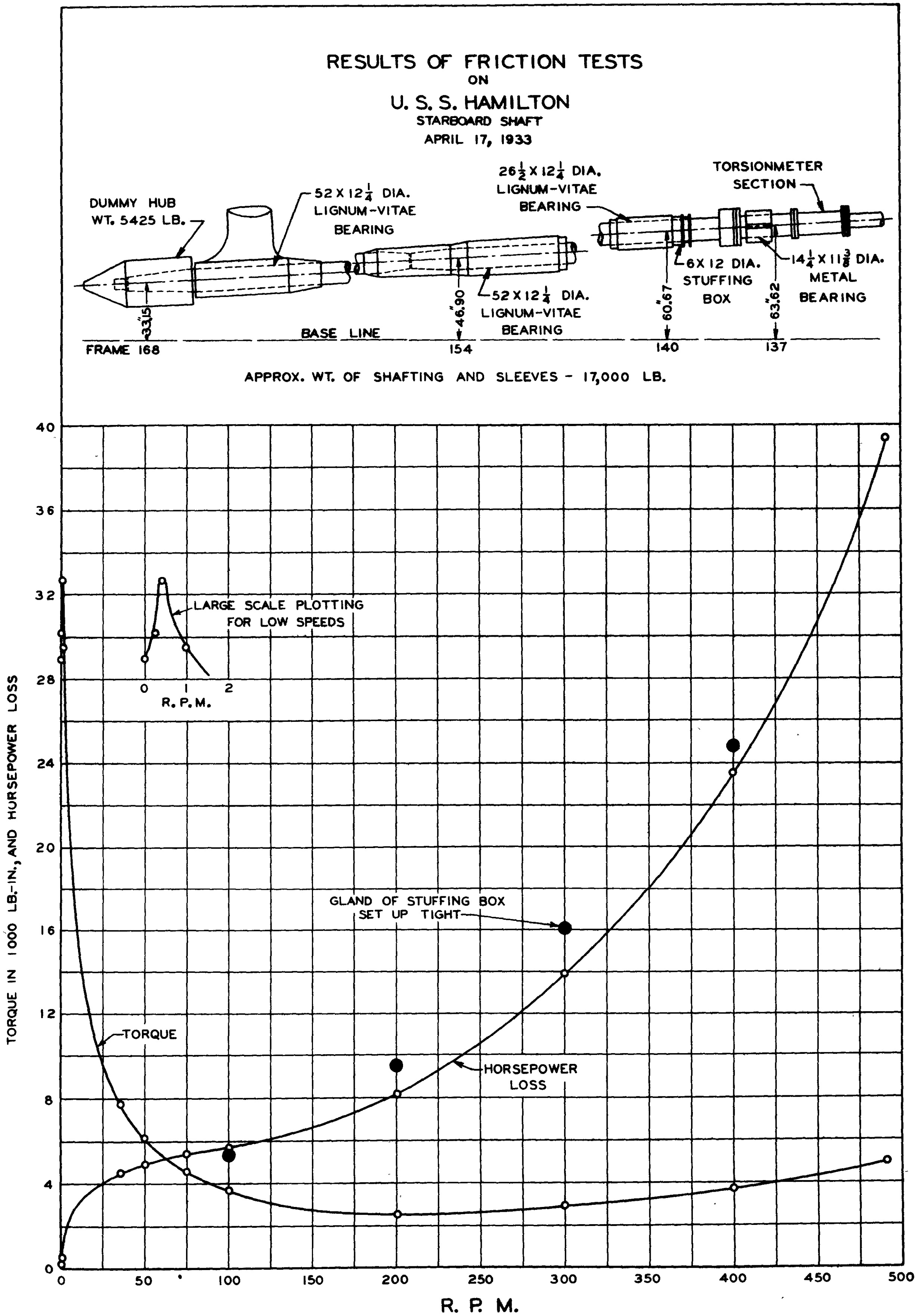
RESULTS OF FRICTION TESTS
ON
U. S. S. HAMILTON
STARBOARD SHAFT
APRIL 17, 1933
DUMMY HUB
WT. 5425 LB.
52 X 12¼ DIA.
LIGNUM-VITAE
BEARING
26½ X 12¼ DIA.
LIGNUM-VITAE
BEARING
TORSIONMETER
SECTION
6 X 12 DIA.
STUFFING
BOX
14¼ X 11⅜ DIA.
METAL
BEARING
52 X 12¼ DIA.
LIGNUM-VITAE
BEARING
33.15"
46.90"
60.67"
63.62"
BASE LINE
FRAME 168
154
140
137
APPROX. WT. OF SHAFTING AND SLEEVES - 17,000 LB.
LARGE SCALE PLOTTING
FOR LOW SPEEDS
R. P. M.
GLAND OF STUFFING BOX
SET UP TIGHT
TORQUE
HORSEPOWER
LOSS
TORQUE IN 1000 LB.-IN., AND HORSEPOWER LOSS
R. P. M.

FIG. 33.

When tests were made with the gland on the stuffing box slacked off, no difference in friction was detected during the length of the test. However, the leakage was not increased when slacked off.

When the gland was set up tight, increased friction was noticeable. The heavy black spots in Fig. 33 indicate the increased horsepower loss due to this tightening. The speed was uncertain when the reading for 100 R.P.M. was made.

Conclusions

Except at low speeds, coefficient of friction varied but little with bearing pressures from 10 to 40 lb. per sq.in. As the error in estimating the conditions on lignum-vitae bearings may be greater than this variation, one curve, Fig. 34, is drawn giving values that may be used in estimating friction loss in bearings for bearing pressures under 40 lb. per sq.in. The coefficients of friction for low speeds are quite high and very uncertain.

With stuffing box glands set up tight and with little or no water passing through the packing, the friction was fairly constant with change in speed. Curve 4, Fig. 35, is a representative curve, although the values expressed may be slightly low. With a small leak or a fairly large flow of water through the packing, the friction varied slightly with speed. Curve 1, Fig. 35, is a representative curve and the values may be used for average conditions. Friction on packing is affected by temperature, but if run for a length of time, the effect of a temperature difference is not so noticeable. Seemingly, a rate change of temperature affects the frictional resistance.

There is fairly close agreement between the calculated and actual test values for the power loss on the U.S.S. HAMILTON starboard shaft as shown in Fig. 36.

A good percentage of the calculated power loss, see Appendix IV, was due to the fluid friction on the dummy hub, curve 4, Fig. 36. A smooth surface was estimated for this calculation. The heads of two 3/4-in. tap bolts extended past the surface of the dummy hub; otherwise only the smooth steel plate surface of the dummy hub came in contact with the water. On another test when the heads of the tap bolts were removed, an observer stated that the water was less disturbed than when the heads were in place.

S.H.P. values for the starboard shaft were taken from the U.S.S. HAMILTON April, 1931, trials. The loss of power due to bearing and stuffing box friction and "windage" was taken from the U.S.S. HAMILTON April 17, 1933, shaft friction test. The per cent of power loss due to friction and "windage" is shown in Fig. 37. These values are considerably lower than those estimated in the past. It would appear that a greater error was made, in the past, by estimating the

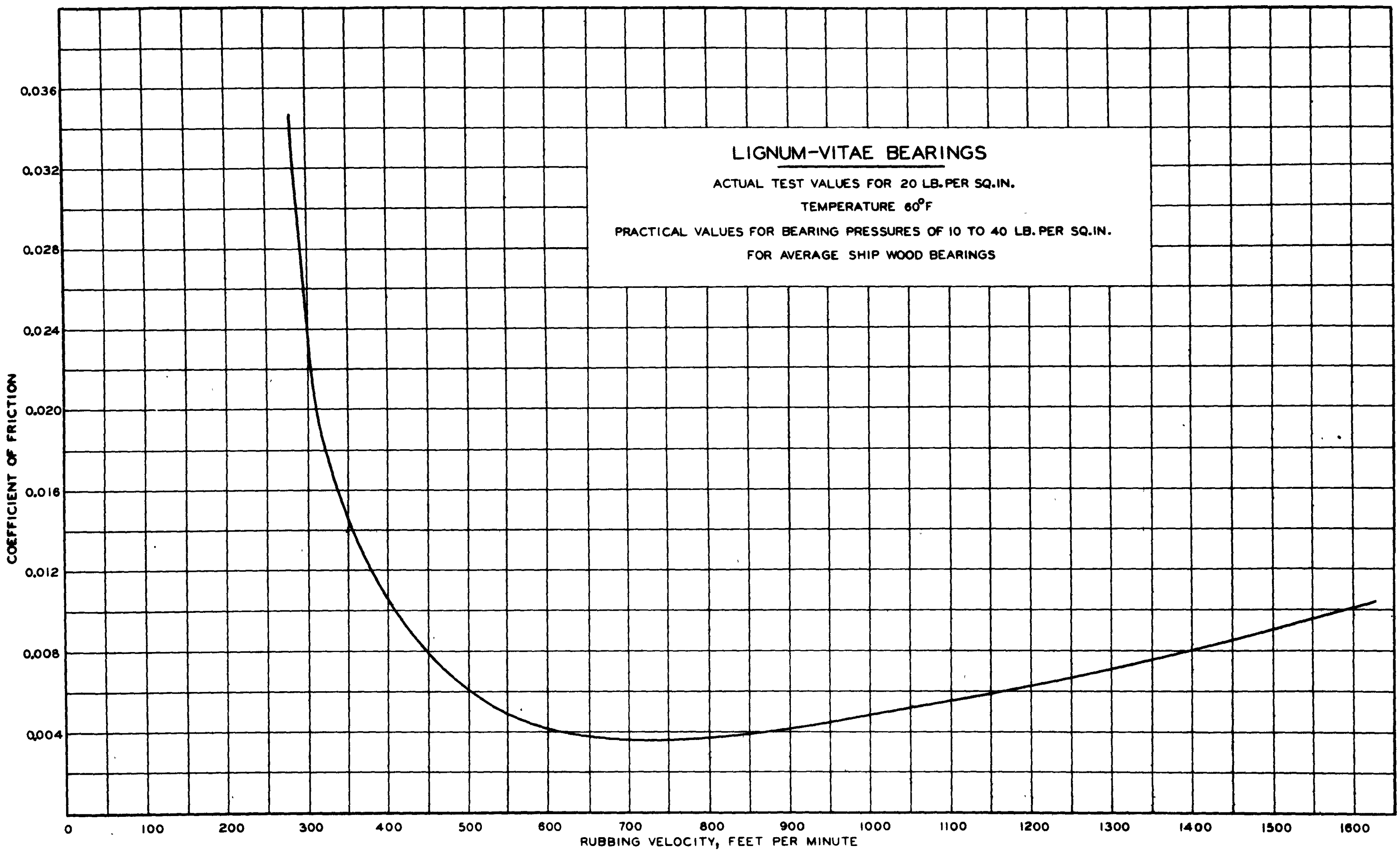
LIGNUM-VITAE BEARINGS
ACTUAL TEST VALUES FOR 20 LB. PER SQ.IN.
TEMPERATURE 60°F
PRACTICAL VALUES FOR BEARING PRESSURES OF 10 TO 40 LB. PER SQ.IN.
FOR AVERAGE SHIP WOOD BEARINGS
COEFFICIENT OF FRICTION
0.036
0.032
0.028
0.024
0.020
0.016
0.012
0.008
0.004
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
1300
1400
1500
1600
RUBBING VELOCITY, FEET PER MINUTE

FIG. 34.

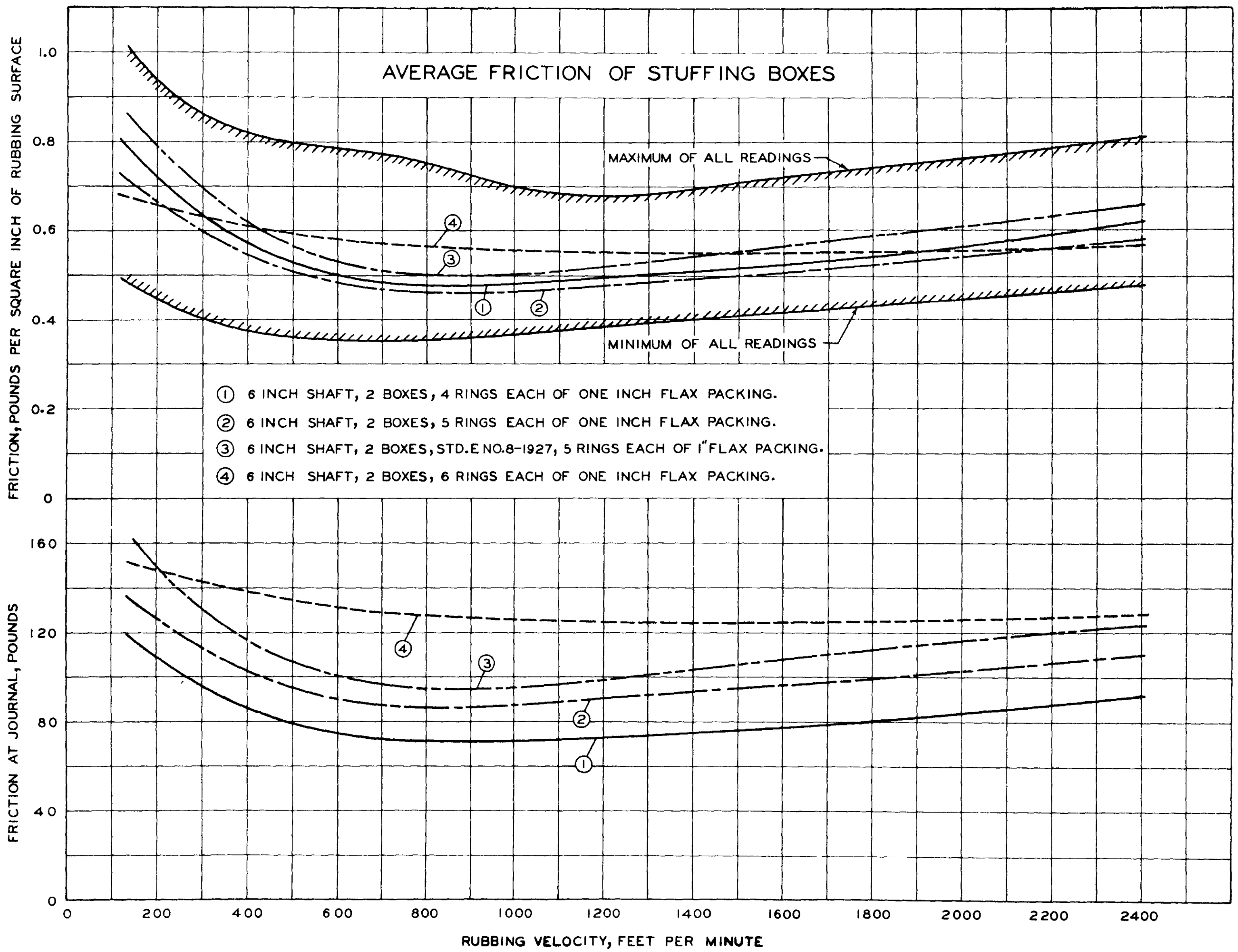
AVERAGE FRICTION OF STUFFING BOXES
MAXIMUM OF ALL READINGS
MINIMUM OF ALL READINGS
① 6 INCH SHAFT, 2 BOXES, 4 RINGS EACH OF ONE INCH FLAX PACKING.
② 6 INCH SHAFT, 2 BOXES, 5 RINGS EACH OF ONE INCH FLAX PACKING.
③ 6 INCH SHAFT, 2 BOXES, STD.E NO.8-1927, 5 RINGS EACH OF 1" FLAX PACKING.
④ 6 INCH SHAFT, 2 BOXES, 6 RINGS EACH OF ONE INCH FLAX PACKING.
FRICTION, POUNDS PER SQUARE INCH OF RUBBING SURFACE
FRICTION AT JOURNAL, POUNDS
RUBBING VELOCITY, FEET PER MINUTE
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
1.0
0.8
0.6
0.4
0.2
0
160
120
80
40
0

FIG. 35.

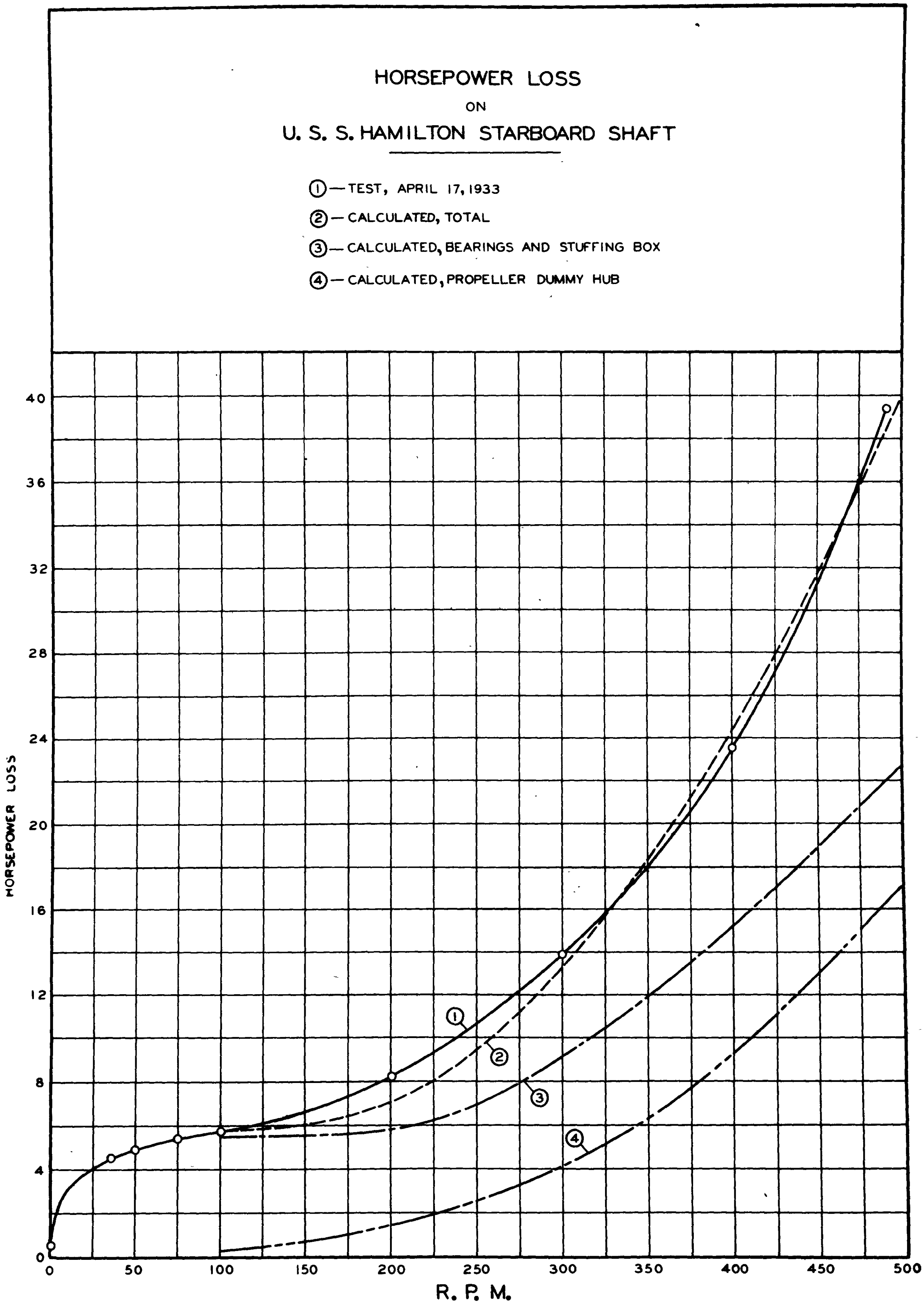
HORSEPOWER LOSS
ON
U. S. S. HAMILTON STARBOARD SHAFT
1 — TEST, APRIL 17, 1933
2 — CALCULATED, TOTAL
3 — CALCULATED, BEARINGS AND STUFFING BOX
4 — CALCULATED, PROPELLER DUMMY HUB
HORSEPOWER LOSS
0
4
8
12
16
20
24
28
32
36
40
0
50
100
150
200
250
300
350
400
450
500
R. P. M.
1
2
3
4

FIG. 36.

power loss than if no allowance had been made for shaft friction.

Excessive power can be absorbed by friction in bearings due to misalignment or lack of lubricant. Too high bearing pressures, especially at low speeds, for lignum-vitae bearings with water as lubricant may lead to high frictional values.

A set deformation, close to that for the starting torque, can be taken by a heavy shaft when at rest. Because of this a torsionmeter zero reading cannot be taken from a rest position. Likewise an error is evident by rotating a shaft slowly in both directions and averaging for a zero reading. Friction at low speeds is high, uncertain, and not equal, even at the same speed, for both directions of rotation.

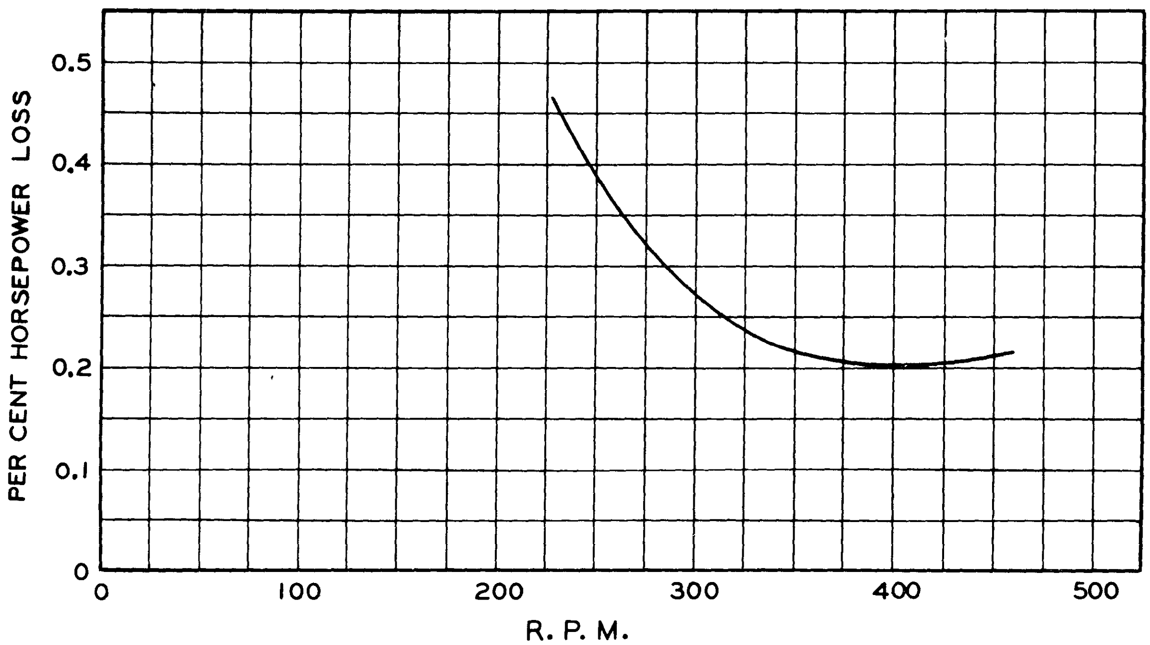

FIG. 37 POWER LOSS DUE TO FRICTION AND WINDAGE.

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APPENDIX I

Theory of Friction

Laws of Fluid Friction*

Resistance is (1) independent of the pressure between masses in contact, (2) directly proportional to the area of rubbing surface, (3) nearly proportional to the square of the relative velocity at low speeds, (4) independent of the nature of the surfaces of the solid against which the stream may flow, but dependent to some extent upon their degree of roughness, and (5) proportional to the density of the fluid, and related in some way to viscosity.

Laws of Solid Friction**

(1) The laws of sliding friction differ with the character of the bodies rubbing together. (2) The friction of fibrous material is increased by increased extent of the surface and by time of contact, and is diminished by pressure and speed. (3) With wood, metal, and stones, within the limit of abrasion, friction varies only with pressure, and is independent of the extent of surface, time of contact, and velocity. (4) Friction is greatest with soft and least with hard materials. (5) The friction of lubricated surfaces is determined by the nature of the lubricant rather than by the solids themselves.

Laws of Friction of Well-lubricated Journals***

(1) The coefficient of friction, with the surfaces efficiently lubricated, is from 1/6 to 1/10 that for dry or scantily lubricated surfaces. (2) The coefficient of friction for moderate pressures and speeds varies approximately inversely as the normal pressure; the frictional resistance varies as the area in contact, the normal pressure remaining constant. (3) At very low journal speeds the coefficient of friction is abnormally high; but as the speed of sliding increases the friction diminishes, and again rises with increased speed, varying approximately as the square root of the speed.

Laws of Unlubricated Friction#

The coefficient of unlubricated friction decreases materially with velocity, is very much greater at minute velocities, falls very rapidly with minute increases of velocities, and continues to fall much less rapidly with higher velocities up to a certain varying point, following closely the laws which obtain with lubricated friction.

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*Friction and Loss Work - Thurston

**Rennie's Experiments.

***John Goodman - Trans. Inst. C.E. 1886.

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Notes on Theory of Bearing Friction

The friction of lubricated surfaces approximates that of solid friction when a journal is run dry, and that of fluid friction when it is flooded with oil.

Increasing the viscosity of a lubricant will increase the frictional resistance.

When gradually reducing the speed (with constant bearing pressure) of any bearing, the coefficient of friction, after reaching a minimum, abruptly increases to a high value because with low speeds liquid friction is giving way to semi-dry friction.

Owing to the wedging action of the lubricant as it is dragged under the journal, the load will not be able to force the journal quite into contact with the bearing (perfect lubrication); instead, the journal will find a position of equilibrium with a certain eccentricity.

Increased bearing pressure or use of lower viscosity oil, within certain limits, reduces the friction losses.

Reduction of wear is obtainable by increasing the viscosity or the area, but either of these steps involves increased power loss when running.

The greater the minimum film thickness the greater the safety, but there is no definite "factor of safety" since danger occurs only when contact of the metals occurs.

The efficiency of a bearing is increased after running.

"It is well known that bearings under laboratory conditions may be operated with speeds and loads far exceeding those applicable in practice; the latter remain, and probably must always remain, matters of judgment and experience." Albert Kingsbury.

APPENDIX II

Reactions on a Bearing Assuming the Normal Radial to the Journal

The bearing as tested with the lever type of testing apparatus was made to float freely on its journal, as will be seen in description of apparatus, with the load applied to the bearing. When the load, W, is directly above the center line of the journal at zero velocity, the center of pressure and the normal reaction, N, are on a line of contact vertically below the load.

When the journal is rotated with no lubricant, see Fig. 38, the bearing will roll upward until the angle included between the line of application of the load and a radial line to the point of contact, A, equals the angle of repose. Then the bearing will begin to slide with contact at that point. Note that during the test, the load was applied vertically above the center line of the journal. The normal reaction, N, is taken as radial to the journal but not necessarily passing through the point of contact. The case where N is not radial is discussed in Appendix III. The reactions on the bearing are: the load, W, the normal, N, the frictional force, F, and the balancing force, P.

$$\text{Taking } \Sigma Mo = 0$$
$$Pa = Fr$$

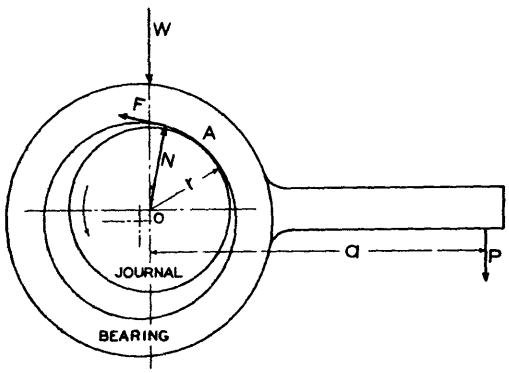

FIG. 38 REACTIONS ON A BEARING WITH NO LUBRICANT

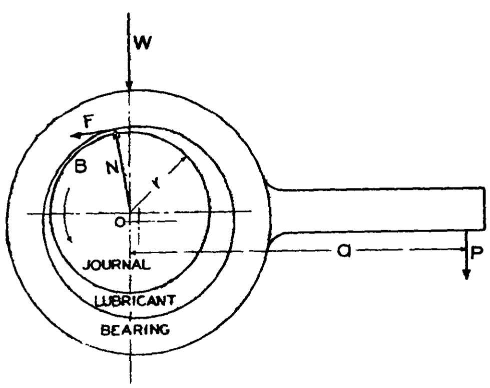

FIG. 39 REACTIONS ON A BEARING WITH LUBRICANT

If lubricant is then applied and ample speed is obtained, conditions alter, see Fig. 39. The lubricant is drawn in by the rotating journal, wedging the bearing finally completely away from the journal and moving the point of nearest approach, B, to the other side until the bearing finds a position of equilibrium with a certain eccentricity depending on speed, bearing pressure, and lubricant. The normal reaction, N, is again taken as radial to the journal but not necessarily passing through the point of nearest approach or even through the point of maximum pressure. There may be conflicting theories as to the reactions and position of the journal relative to the bearing; but if the normal, N, is radial to the journal, whether there is perfect lubrication or not, the same equation is obtained.

$$\text{Taking } \Sigma Mo = 0$$
$$Pa = Fr$$

The frictional force, F, can then be easily determined.

$$F = Pa/r \quad . \quad . \quad . \quad . \quad . \qquad (1)$$

APPENDIX III

Reactions on a Bearing When the Normal Is Not Radial to the Journal

The coefficients of friction as determined by the lever type apparatus were slightly lower than those obtained by the torsionmeter type. It was assumed that the normal reaction, N, was radial to the journal.

The higher values, for the torsionmeter method, may be accounted for in two ways: (1) The torsionmeter method measured the frictional resistance on the journal which included the bearing friction, the slight friction of the small rubber diaphragm acting as the stuffing box, and the fluid friction between the extended journal and the water in the tank; (2) The lever method measured the friction on the bearing with the assumption that the normal reaction was radial to the journal.

As the added friction due to the rubber diaphragm and the fluid friction would appear negligible, it will be assumed that the difference in results of the two methods were due to the normal reaction not being radial to the journal as assumed for the lever method of test. Then the reactions on the bearing are: the load, W, the normal, N, the frictional force, F, and the balancing force, P. See Fig. 40.

$$\text{Taking} \quad \Sigma Mo = 0$$

$$Pa = Fr \pm Nx \quad \ldots\ldots\ldots \quad (5)$$

where x is the perpendicular distance from the normal reaction to the center line of the journal.

This equation can be established for either a dry bearing or a bearing with perfect lubrication.

For small frictional values, N can be taken as equal to W. It is assumed that the normal, N, is on the same side of the center line of the journal as that of the point of contact or nearest approach.

$$\text{Then } Pa = Fr - Wx$$

$$\text{or} \quad F = (Pa + Wx)/r \quad \ldots\ldots\ldots \quad (6)$$

Then the friction as measured by the lever method would be too small by the value $W(x/r)$.

The value x is a variable depending on speed, journal diameter, lubricant, and clearance; and for all practical purposes it is indeterminate. However, x will be approximated by assuming the frictional values correct as determined by the torsionmeter method.

$$x = (Fr - Pa)/W \quad \ldots\ldots\ldots \quad (7)$$

Where P is the balancing force for the lever method and F is the frictional force for the torsionmeter method.

The curve, Fig. 40, gives the value of x as calculated for a bearing pressure of about 20 lb. per sq.in., water temperature of 60° F. but with variable speed.

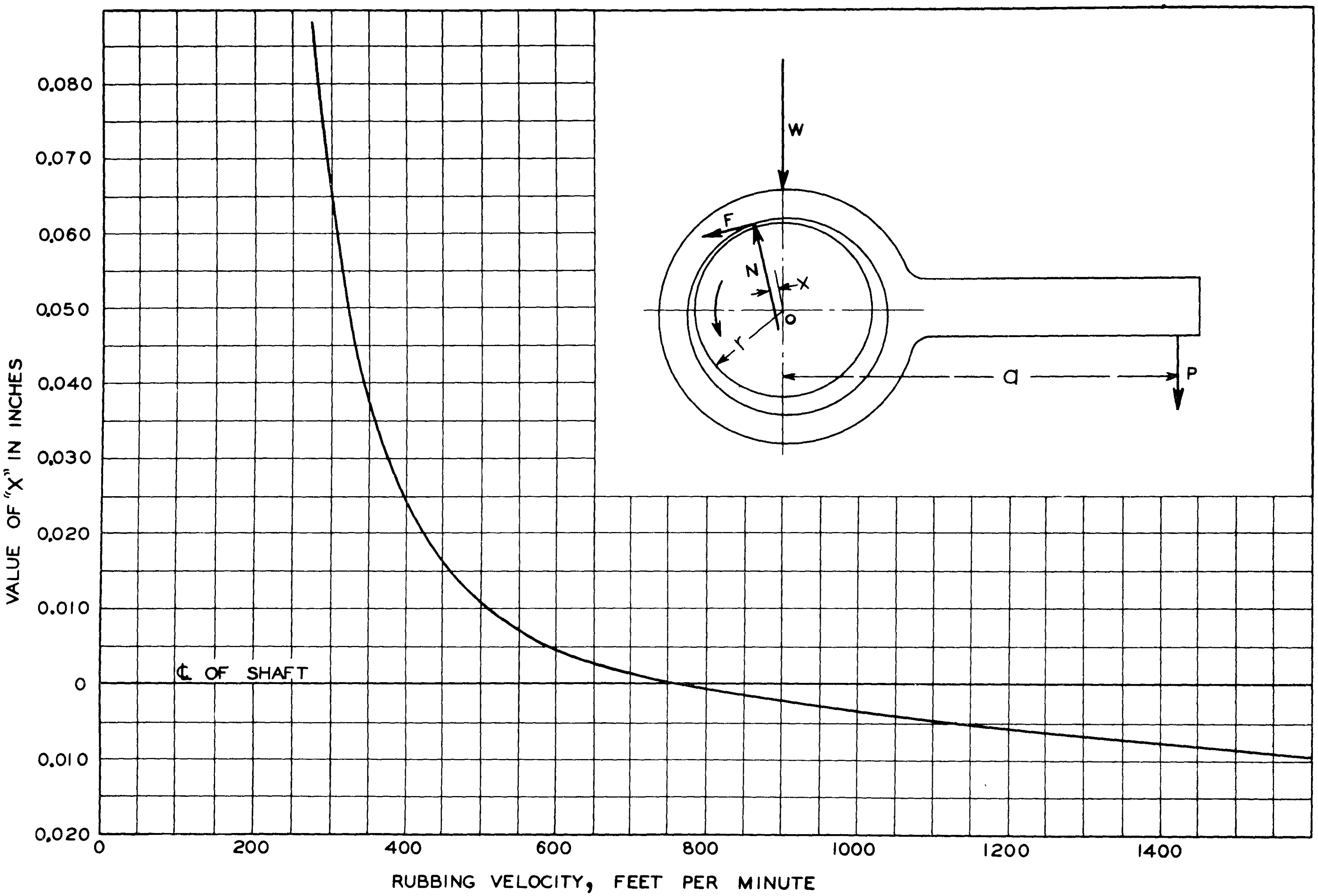

FIG. 40. Reactions on a Bearing

APPENDIX IV

Calculated Power Loss in Bearings, Stuffing Box, and Dummy Propeller Hub On the Starboard Shaft of U.S.S. HAMILTON

WEIGHTS

1. Propeller	5,425 lb.
2. Propeller shaft, nuts and sleeves	5,541
3. Coupling between propeller shaft and stern tube shaft	1,200
4. Stern tube shaft with sleeves	6,744
5. Line shaft No. 3, coupling, bolts and nuts	3,555
Total	22,465 lb.

BEARINGS AND STUFFING BOX

	Length, in.	Diameter, in.
1. Two – Lignum-Vitae	52	12-1/4
2. One – Lignum-Vitae	26-1/2	12-1/4
3. One – Metal (white)	14-1/4	11-3/8
4. One – Stuffing Box	6	12

DISTRIBUTION OF LOAD

1. Lignum-Vitae bearing No. 1	
Propeller	5,425 lb.
Estimated 2/3 propeller shaft	3,688
Total	9,113 lb.
2. Lignum-Vitae bearing No. 2	
Estimated 1/3 propeller shaft	1,844 lb.
Coupling	1,200
Estimated 1/2 stern tube shaft	3,372
Total	6,416 lb.
3. Lignum-Vitae bearing No. 3	
Estimated 1/2 stern tube shaft	3,372 lb.
Total	3,372 lb.
4. Metal bearing	
Estimated 2/3 line shaft	2,370 lb.
	2,370 lb.

BEARING PRESSURES

1. Lignum-Vitae bearing No. 1

$$\frac{9113}{52 \times 12\text{-}1/4} = 14.3 \text{ lb. per sq.in.}$$

2. Lignum-Vitae bearing No. 2

$$\frac{6416}{52 \times 12\text{-}1/4} = 10.1 \text{ lb. per sq.in}$$

3. Lignum-Vitae bearing No. 3

$$\frac{3372}{26\text{-}1/2 \times 12\text{-}1/4} = 10.4 \text{ lb. per sq.in.}$$

4. Metal bearing

$$\frac{2370}{14\text{-}1/4 \times 11\text{-}3/8} = 14.6 \text{ lb. per sq.in.}$$

HORSEPOWER LOSS

1. Lignum-Vitae bearing No. 1

R.P.M.	ft/min.	f	F	Hp. loss
100	318	0.0184	164.0	1.59
200	638	0.0037	34.6	0.67
300	956	0.0045	41.0	1.19
400	1274	0.0068	62.0	2.41
500	1590	0.0098	89.0	4.38

2. Lignum-Vitae bearing No. 2

100	318	0.0170	109.0	1.06
200	638	0.0036	23.1	0.45
300	956	0.0043	27.5	0.80
400	1274	0.0066	42.3	1.64
500	1590	0.0091	58.4	2.84

3. Lignum-Vitae bearing No. 3

100	318	0.0170	57.3	0.56
200	638	0.0036	12.1	0.24
300	956	0.0043	14.5	0.42
400	1274	0.0066	22.5	0.87
500	1590	0.0091	30.7	1.50

4. Metal bearing

100	297	0.042	99	0.89
200	595	0.050	130	2.04
300	892	0.067	159	3.73
400	1190	0.080	189	5.92
500	1487	0.090	213	8.33

5. Stuffing box

R.P.M.	ft/min.	lb./sq.in. of Rubbing Surface	F	Hp. loss
100	314	0.64	144	1.37
200	628	0.50	113	2.15
300	942	0.48	108	3.09
400	1257	0.50	113	4.30
500	1570	0.52	117	5.57

Two different calculations were made in estimating the horsepower loss on the propeller dummy hub. The first was based on tests made by Malloch in 1888 in which he made tests on rotating cylinders. His conclusions were that the resistance of a rotating cylinder agreed roughly with tests made by Froude on long thin plates. That is

$$R = KAV^n$$

where

R = resistance in lb.

K = coefficient depending upon the roughness of the surface, degree of turbulence, etc.

A = area in sq. ft.

V = velocity in ft. per sec.

n = an exponent somewhat less than 2, in this case taken as 1.85.

The value of 0.0070 chosen for the coefficient K is that obtained by Froude for a surface of calico, which was judged to have about the same roughness as the dummy hub, corresponding to the turbulence at a point 50 feet from the leading edge of the plate when towed at a speed of 10 ft. per sec.

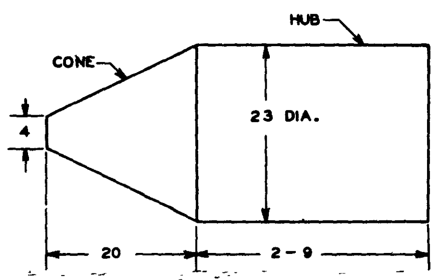

FIG. 41 PROPELLER DUMMY HUB

Calculations on this basis gave the following results:

Shaft R.P.M.	Hp. loss
100	0.17
200	1.26
300	4.01
400	9.14
500	17.20

The second calculation was based on the results of tests on a rotating disk in water (Kempf - Werft Reederei Hafen 1925). This time the following equation was used.

$$Q = \lambda A \rho r_m v_m^2$$

where

Q = torque in lb.-ft.

A = area in sq. ft.

ρ = density, taken as 1.94

r_m = mean radius in ft.

v_m = mean speed in ft. per sec.

λ = coefficient

The coefficient, λ, is a function of Reynolds' number. Reynolds' number, R, is defined as

$$\frac{v_m\, r_m}{\mu} \quad \text{or} \quad R = \frac{\omega\, r_m^2}{\mu}$$

where μ is coefficient of kinematic viscosity taken as 1.21×10^{-5} for 60° F.

The mean radius, r_m, for the hub was taken as 0.96 and for the cone as 0.56.

Calculations for the Hub

R.P.M.	v_m	R	λ	Q
100	10	0.79×10^6	0.0036	11.1
200	20	1.59	0.0028	34.4
300	30	2.38	0.0024	66.4
400	40	3.18	0.0023	113.1
500	50	3.96	0.0022	169.1

Calculations for the Cone

R.P.M.	v_m	R	λ	Q
100	5.9	0.47×10^6	0.0047	0.8
200	11.8	0.93	0.0033	2.3
300	17.7	1.40	0.0027	4.2
400	23.5	1.86	0.0026	7.1
500	29.5	2.33	0.0024	10.4

TOTAL for Propeller Dummy Hub

R.P.M.	Q_{hub}	Q_{cone}	Q_{total}	Hp. loss
100	11.1	0.8	11.9	0.23
200	34.4	2.3	36.7	1.40
300	66.4	4.2	70.6	4.02
400	113.1	7.1	120.2	9.12
500	169.1	10.4	179.5	17.07

The horsepower loss for both methods were practically the same. The results of the second method will be accepted for estimating the horsepower loss.

Total Horsepower Loss

Shaft R.P.M.	Bearing and Stuffing Box Hp. Loss	Dummy Hub Hp. Loss	Total Hp. Loss
100	5.47	0.23	5.70
200	5.55	1.40	6.95
300	9.23	4.02	13.25
400	15.14	9.12	24.26
500	22.62	17.07	39.69

APPENDIX V

General Description and Requirements for Stern-Tube Bearings

"The stern-tube bearings consist of brass shells, in halves, fitted in each end of the stern tubes. The bearing shells are grooved with long channels slightly dovetailed in cross section to receive similarly dovetailed strips of lignum-vitae. That portion of the stern shaft revolving within the stern tube is covered with a composition water-tight casing to protect it from the sea water and to furnish a good bearing material on the lignum-vitae. Strips of lignum-vitae are inserted longitudinally in the interior of the bearing shells and set so as to present the end of the grain to the shaft. The arrangement is therefore somewhat similar to that for a metal bearing, except that lignum-vitae is substituted for the bearing metal, and is placed in continuous channels running the entire length of the bearing shells with intermediate spaces between. All the lignum-vitae shall be well water-soaked and bored out to perfect alignment and to a loose fit on the shaft casing, the lignum-vitae being so fitted that the shaft will bear on the strips on the lower half of the bearing." General Specifications for Machinery for Vessels of the U.S. Navy—Edition May 1, 1919.