AN ANALYSIS OF THE
FULL FLOATING TEXTILE SPINDLE BEARING

A THESIS

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FULL FLOATING TEXTILE SPINDLE BEARING

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SUMMARY

The textile industry is one of the largest consumers of power in this country; hence, any practical means of reducing the power consumed could result in substantial savings.[4] A large portion of this power is consumed in the textile spindles. The spindle, which holds the bobbin of thread, is mounted in a bolster containing lubricating oil and two bearings--a journal bearing at the top and a pivot or conical bearing at the bottom. The most common type of spindle uses cast-iron in both the bolster and the bearings.

The purpose of this investigation is a study of some of the variables involved in an attempt to establish the type of lubrication occurring. In this way only can the proper oil be selected to fit the situation existing in any particular textile plant. Selection of such an oil can result in less consumption of power and longer life of the bearings.

To accomplish the above purpose, the friction developed by the bearings was measured for different speeds, loads and viscosity oils. From these data an experimental curve was plotted giving the relation between the coefficient of friction and the Sommerfeld number, a dimensionless ratio of the variables involved. These data were compared with a curve derived from the hydrodynamic theory of lubrication. The results indicated that the journal bearing was probably operating hydrodynamically. Further data, similar to the above but with the addition of a series of vertical loads, were taken. Evaluation of these data permitted separation of the journal friction from that of the pivot bearing and showed that the journal bearing was operating as a full 360° bearing, in the mid-range of Sommer-

feld numbers, in accordance with the hydrodynamic theory. Periodic examination of the pivot bearing showed that it was making metal to metal contact, as evidenced by continued wear, and was therefore operating in the boundary region of lubrication. Evaluation of the data in Figure 9 shows that the pivot bearing produces friction varying from approximately eleven to two hundred and forty per cent of that of the journal bearing depending upon the vertical load and Sommerfeld number.

The effects of vibration and oil whip were quite troublesome during the investigation. These effects should be investigated more fully with the possibility of correlating them with the Sommerfeld number. The effect of varying the point of application of thread load to the spindle is also a possible, important phase for future investigation.

CHAPTER I

INTRODUCTION

Since the textile industry is one of the largest consumers of power in this country, it is profitable to investigate all possible means of reducing power consumption in this industry.[4] The major portion of power used is consumed in the spinning rooms. Here, in all except the smallest plants, thousands of spindles are found. Each spindle by itself requires only a very small amount of power, but when the large number of spindles in each mill is considered the power required to overcome spindle friction becomes appreciable. One means for reducing power consumption would be selection of the proper lubricant for the spindle. Since the vertical journal bearing normally found in the textile spindle has many other applications in industry, the calculated prediction of the most suitable lubricant would be of wide benefit. The purpose of the investigation herein reported upon is the analysis of the textile spindle bearing and the determination of its lubrication characteristics so that the type of oil most suitable under a given condition could be predicted. The execution of this purpose involved specifically the following objectives: (1) to determine whether the spindle bearings operate hydrodynamically, (2) to separate journal and pivot bearing friction, and (3) to determine the partiality (bearing angle) of the journal bearing.

The most common spindle has a plain cast-iron bolster equipped with a pivot or a conical bearing at the bottom and a journal bearing at the top. This bearing was previously investigated by five graduate students at the Georgia Institute of Technology.[1,2,3,4,10] Various phases of the power consumption problem were analyzed. Some disagreement and inconsistency in the results is probably due to the many variables as well as the minuteness of the quantities involved. The apparatus developed by Cheverton[10] and modified slightly for the present investigation is capable of measuring the friction forces involved to a reasonable degree of accuracy.

CHAPTER II

INSTRUMENTATION AND EQUIPMENT

The frictional force was measured by the apparatus originally devised for measurement of power consumption of the textile spindle as modified by the student author of reference 10. The apparatus is capable of measuring the friction force developed by the journal and pivot bearings. This is accomplished by mounting the bolster in ball bearings in an aluminium base which in turn is mounted on ball bearings as shown in Figure 1. A thread wrapped around the bolster is connected to a small aluminium cantilever beam. Upon this beam are cemented a pair of SR-4 (AB-7) strain gages. This beam and the thread provide the restraining force for the bolster and bearings, and the strain gages indicate the force developed. This force is recorded on a Foxboro SR-4 strain-time recording instrument as a strain in the beam. By proper calibration, this will indicate the force developed by friction in the bearings.

The load on the journal bearing is provided by a cord attached to the aluminium base and passing over a pulley as shown in Figures 2 and 3. Weights hanging on this cord (shown in Figure 4) provide constant tension in the driving belt. In order to assure a constant radial force on the bearing, the line of action of the cord is adjusted by moving the eccentrically mounted base until the line of action passes through the center of the bearing. Since the journal bearing is mounted in a ball bearing, the resultant of other forces must also pass through its center and must be equal to the applied force.

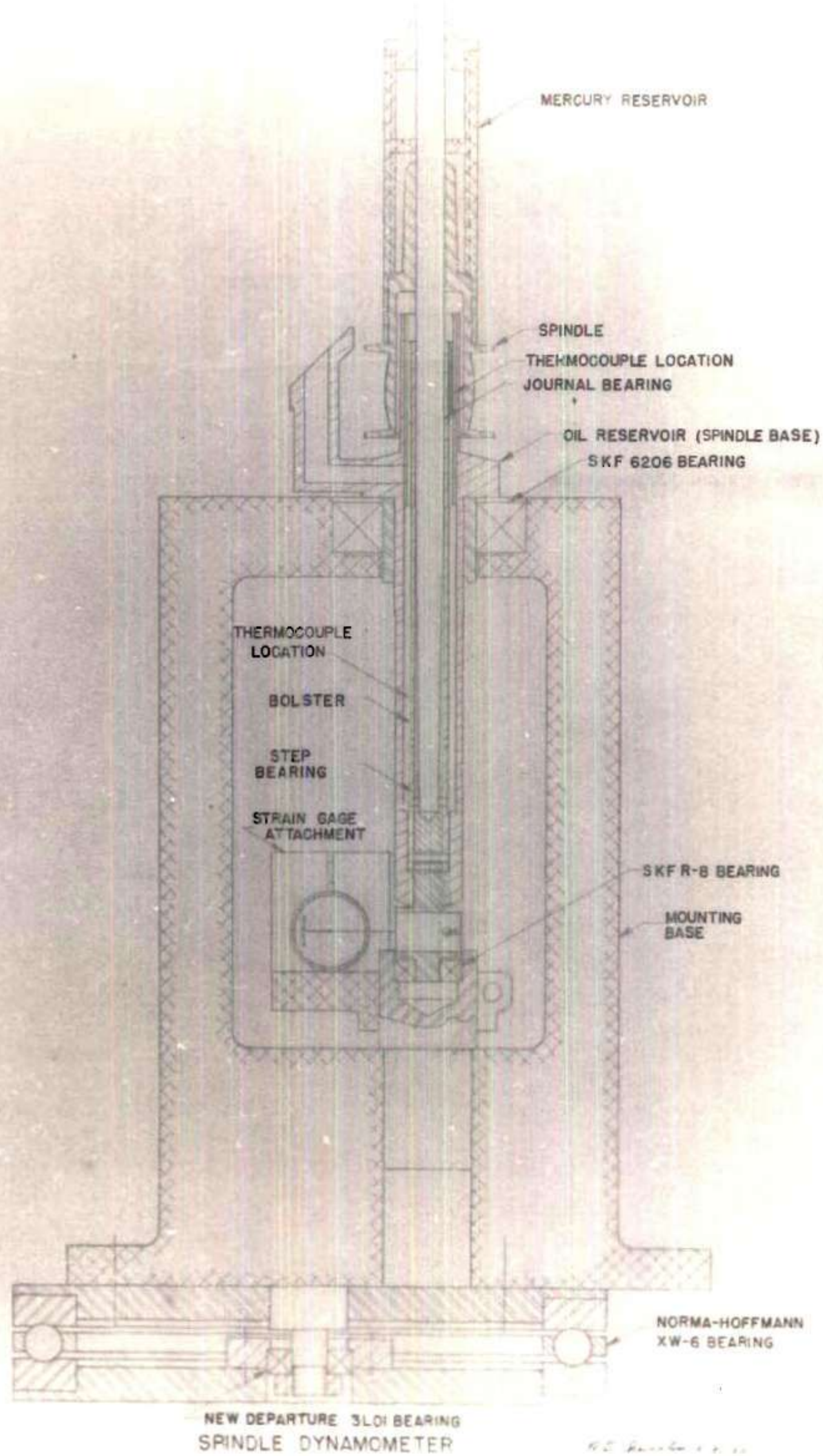

Fig. 1. Spindle Dynamometer

Figure 2

Spindle Dynamometer--Assembly

1. Dynamometer Mounting Base (Aluminium)
2. Mercury Slip Ring
3. Thermocouple Leads--Oil Reservoir
4. Thermocouple Leads--Journal Bearing
5. Cord for Maintaining Belt Tension
6. Friction Measuring Device
7. Pulley Used in Calibrating Friction Measuring Device
8. Thrust Bearing Assembly
9. Weights for Calibrating Friction Measuring Device
10. Weights for Producing Vertical Load

Fig.2. Spindle Dynamometer - Assembly

Figure 3

Spindle Dynamometer--Operating Equipment

1. Dynamometer Mounting Base (Aluminium)
2. Cord for Maintaining Belt Tension
3. Spindle Drive Belt
4. Idler Pulley
5. Stroboscope
6. Nine Speed Transmission
7. Cast-Iron Rubber Mounted Base
8. Point of Journal Load Application
9. Thrust Bearing Adjustment
10. Mercury Slip Ring

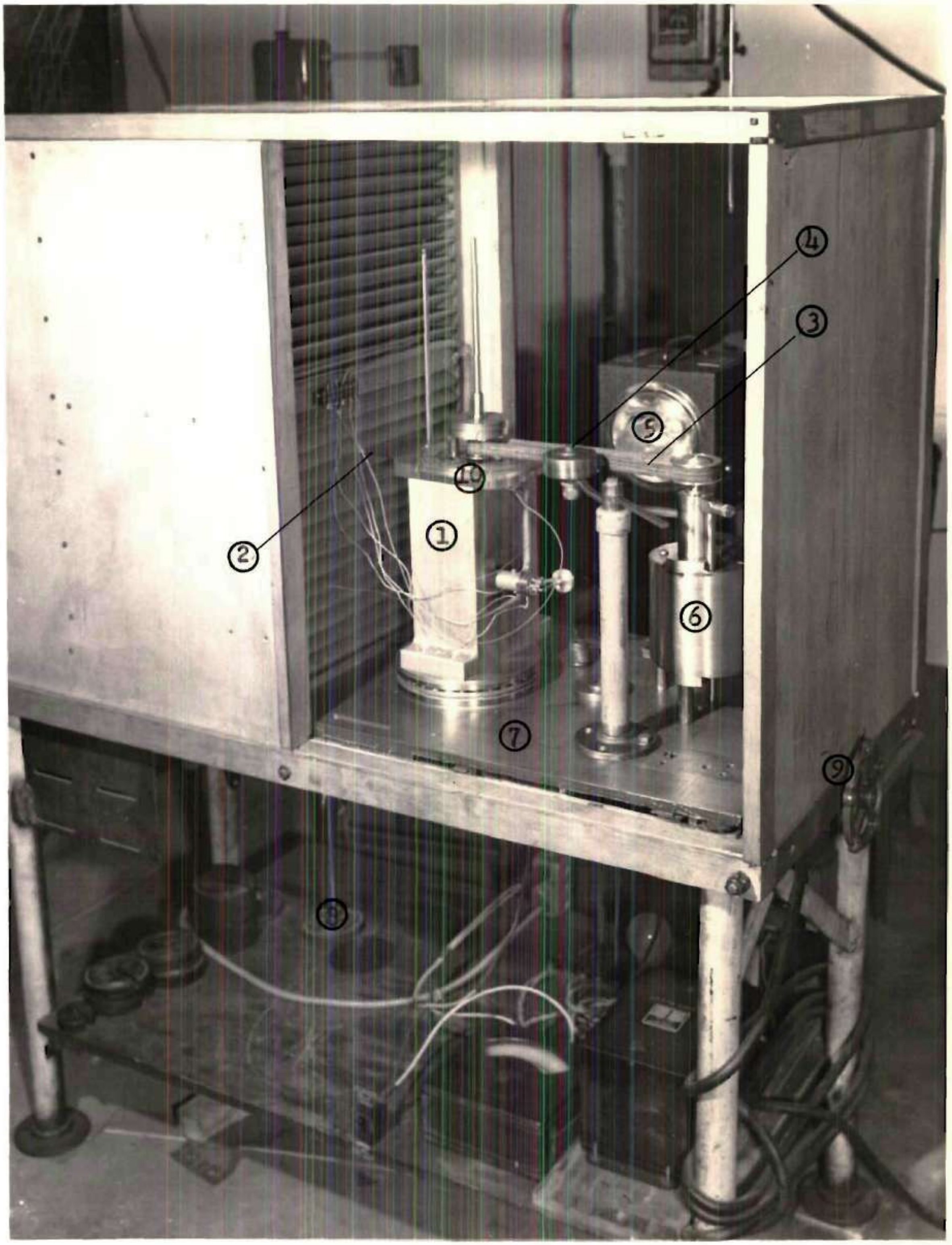

Fig.3. Spindle Dynamometer - Operating Equipment

Figure 4

Spindle Dynamometer--Operating Equipment

1. Voltage Regulator
2. Ammeter in Driving Motor Circuit
3. By-pass Switch to Ammeter
4. Point of Journal Load Application
5. Synchronous Motor
6. Mechanism for Adjusting Thrust Bearing

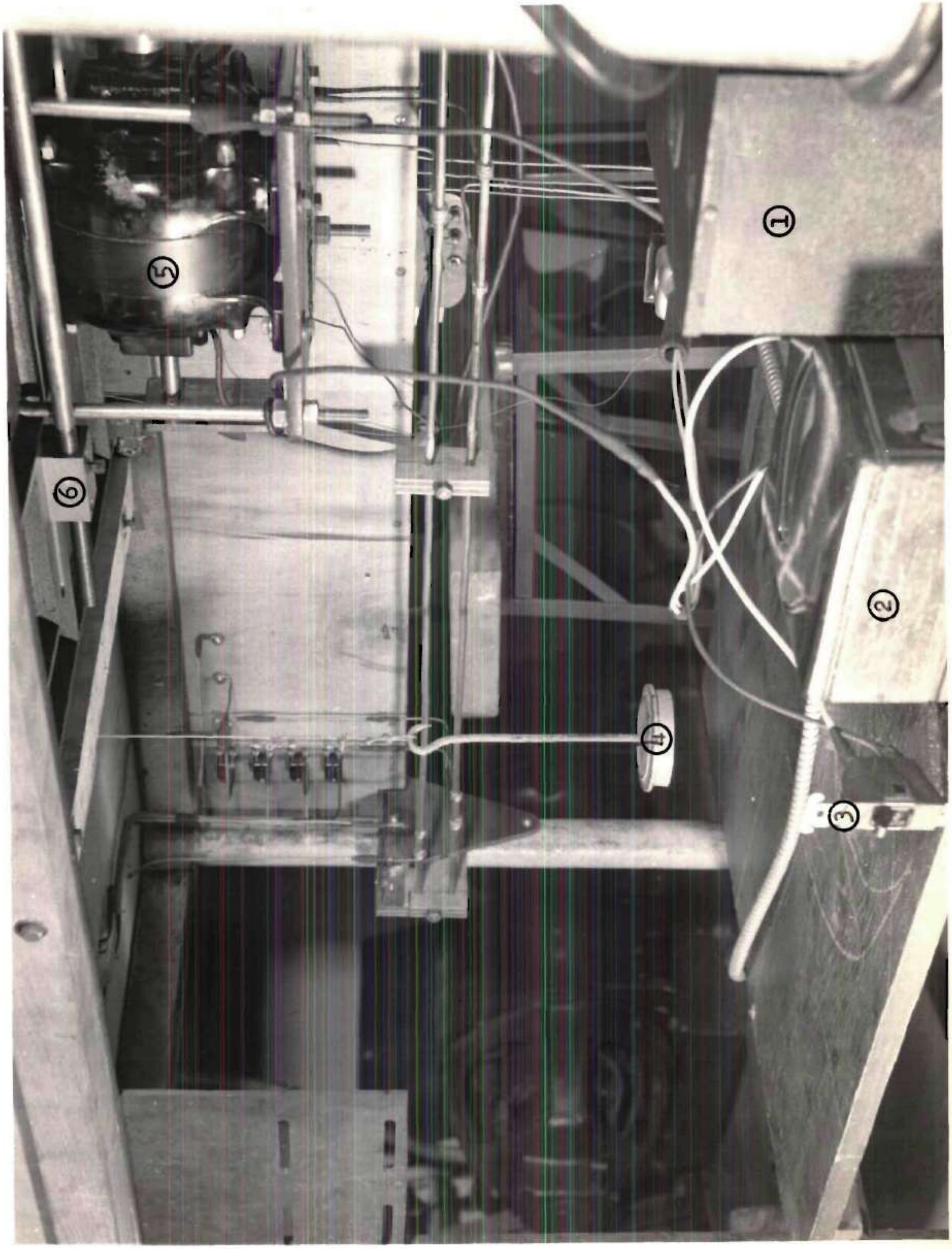

Fig. 4. Spindle Dynamometer - Operating Eqiupment

Figure 5

Spindle Dynamometer--Component Parts of Spindle and Friction Measuring Device

1. Cantilever Housing
2. Strain Measuring Element with Aluminium Cantilever Beam
3. Spindle and Mercury Reservoir
4. Cast-iron Bolster with Journal Bearing Thermocouple
5. Special Spindle Oil Reservoir

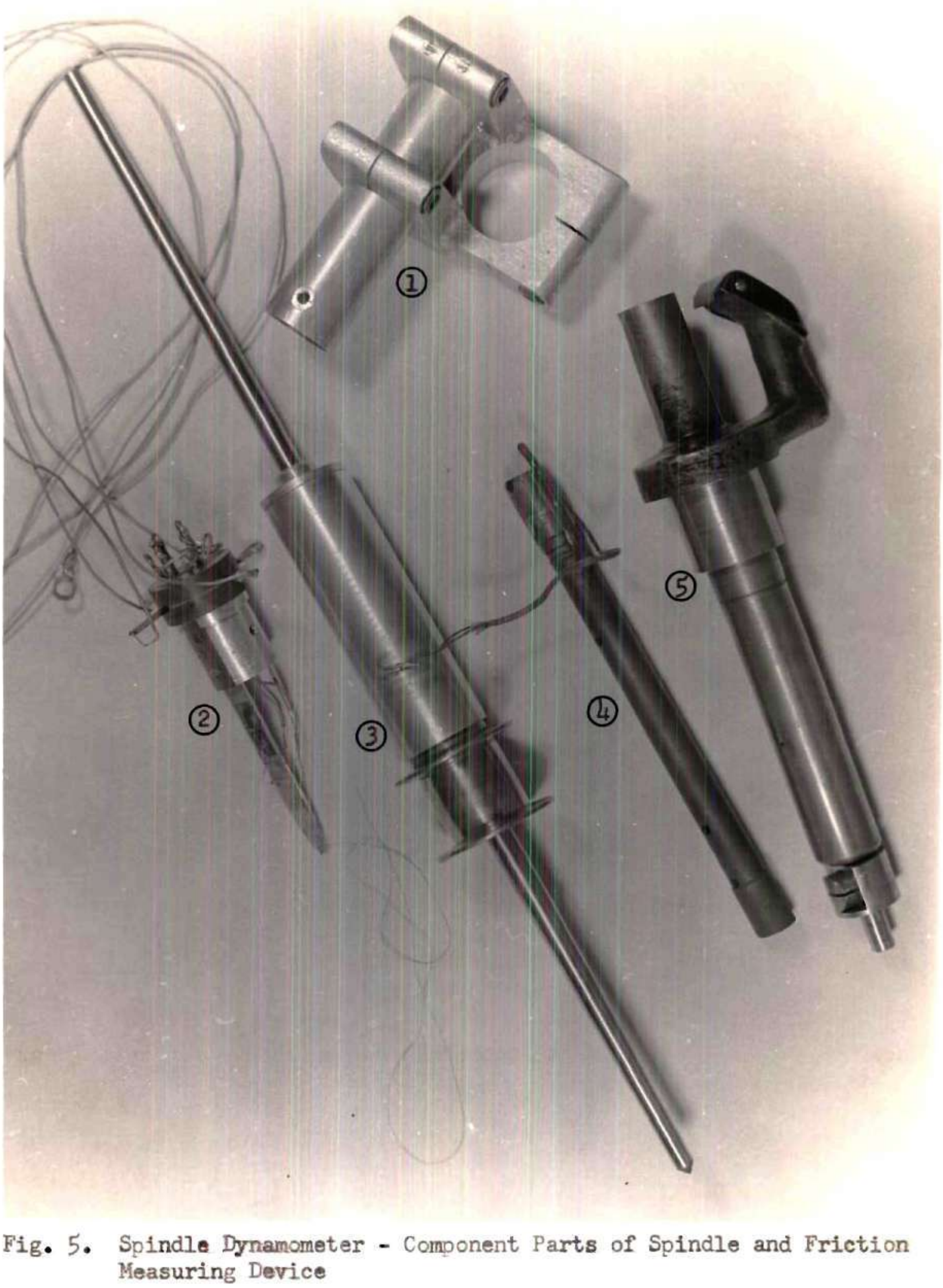

Fig. 5. Spindle Dynamometer - Component Parts of Spindle and Friction Measuring Device

Figure 6

Spindle Dynamometer—Controls and Recording Instruments

1. Foxboro SR-4 Strain Time Recorder
2. Foxboro Portable Indicator (Potentiometer used with Thermocouples)
3. Rheostat and Switches for Controlling Ambient Temperature
4. Stroboscope
5. Heating Elements and Fan Housing
6. Regulated Voltage Switch and Outlet
7. Thermocouple Selector Switch

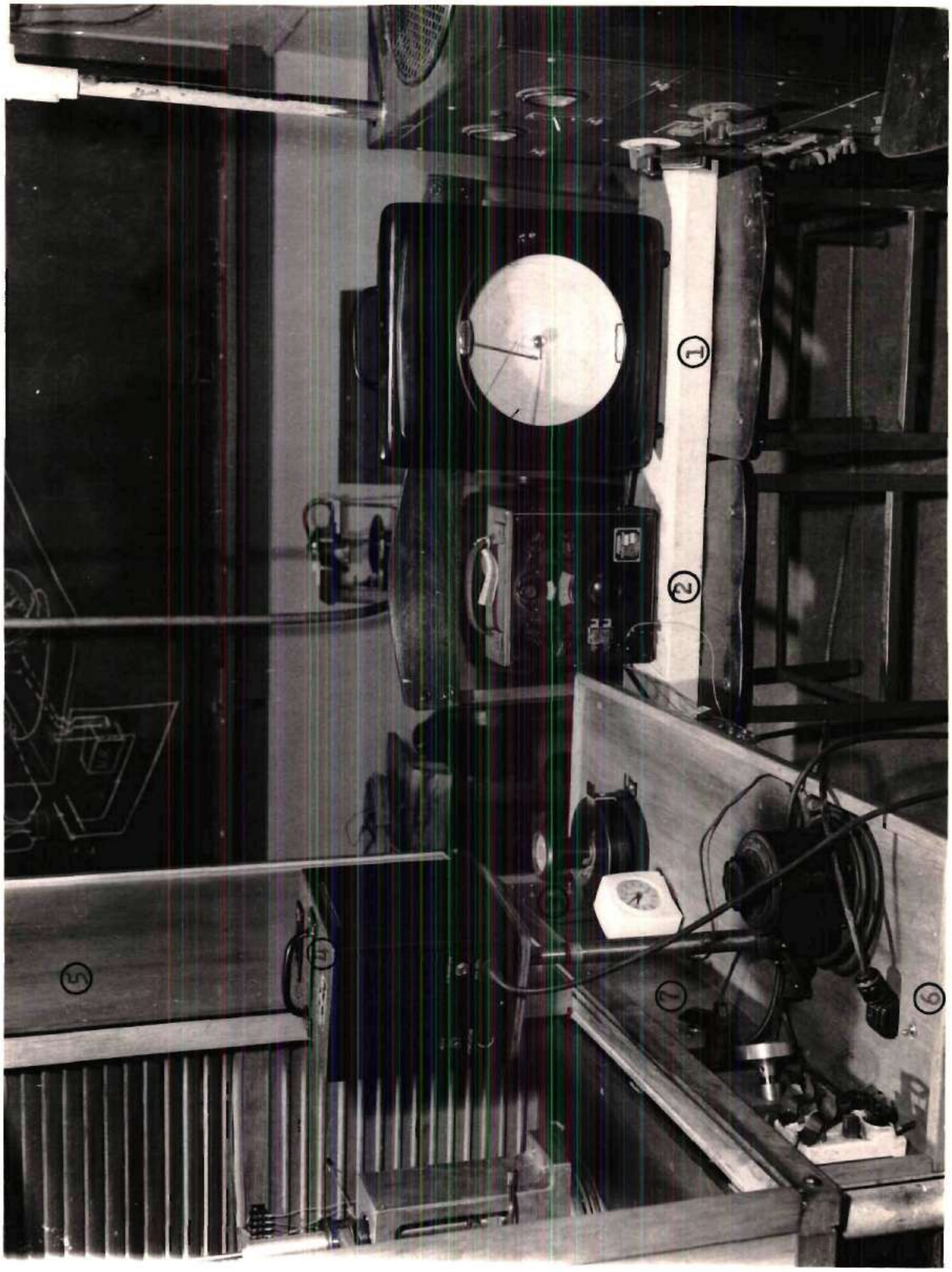

Fig. 6. Spindle Dynamometer--Controls and Recording Instruments

Temperatures in the oil reservoir were measured near the pivot bearing and in the journal bearing by means of thermocouples. A mercury slip ring was added to the apparatus so that temperature measurements could be made continuously without interference with friction measurements. A Foxboro portable indicator was used to give temperature readings to the nearest degree Fahrenheit. Accurate temperature readings are needed for determination of the viscosity of the lubricant.

The ambient temperature was controlled by a fan circulating warm air from light bulbs and was maintained at 85°F.

The spindle was driven by a synchronous motor through a transmission and a belt. Speeds of from approximately 4,000 to 12,000 revolutions per minute in increments of one thousand can be developed.

To provide for increments of vertical load, cast-iron weights were turned on a mandrel cut to the taper of the spindle. (See Figure 2)

It was found that the position of the bolster had a marked effect on the friction developed. This must have been due to a slight "out-of-round" condition of the bearing. To provide uniform results, the bolster was adjusted to the same position by changing the position of the friction measuring device for each run.

CHAPTER III

PROCEDURE

Analysis.—Accomplishment of the purpose of the investigation required first the determination of whether or not the bearing operates hydrodynamically. Thus a curve was developed following the hydrodynamic theory as a basis for comparison with the experimental data. The Westinghouse Electric Corporation supplied a series of charts[14] pertaining to the hydrodynamic lubrication of an infinite journal bearing along with a series of additional graphs presenting correction factors needed to convert the infinite bearing values to those for the actual bearing. A series of computations were thus made correcting the friction curve for end leakage. These data were then plotted on $\sinh^{-1}$ graph paper as used by Westinghouse. The friction curve thus obtained has as coordinates the coefficient of friction of the journal and the Sommerfeld number. It is a theoretical curve showing what may be expected under actual conditions according to the hydrodynamic theory. For purposes of comparison, the experimental data were evaluated and similarly plotted. The Sommerfeld number, normally dimensionless, was multiplied by sixty making its units sec./min. for comparison with the theoretical curve. The difficulty in evaluating the data arose from the fact that the friction force was measured on the bearing, while the theoretical curve was calculated for journal friction. Published investigations[13] of the theory conclude that it is impractical to convert the theoretical curve and refer it to the bearing. An analysis was therefore made of the practicability of referring the experimental data to the journal. Shaw and

Macks[13] proved using simple geometry that the difference between the friction when referred to the journal as compared to the bearing is due to the eccentricity. Since the clearance between the journal and bearing is very small and the eccentricity is, in most cases, even smaller, it is assumed that the friction force as measured on the bearing can be assumed equal to the force exerted by the journal. An evaluation of this assumption, using only the geometry of the problem, shows that, in the middle range of Sommerfeld numbers, the error was approximately one per cent or less. Hence, the assumption is valid for that range. An experimental curve with the same coordinates as the theoretical curve could therefore be obtained purely from the geometry of the apparatus and measured or known quantities. Comparison of this experimental curve with the theoretical curve should indicate whether the bearing is operating hydrodynamically.

The separation of the friction of the pivot and journal bearings of the spindle was conducted for the purpose of establishing the operating angle of the journal bearing. By placing successive weights on the spindle and securing lubrication curves as before for each weight, it was expected that a family of curves would result, each indicating a greater friction. This increase in friction should be due to the pivot bearing only as the vertical load added should affect mainly this bearing and produce little if any radial or horizontal load on the journal bearing. Hence, by properly interpreting these data, a curve for zero vertical load may be obtained. This would correspond to the condition in which the journal bearing only is operating and the resulting curve would indicate the lubrication characteristics of the journal bearing. Its relation to the theoretical curves will provide a further check on whether the bearing is operating hydrodynamically and also shows the degree of partiality (bearing angle).

Calibration of the friction measuring device.--The friction measuring device must be calibrated in order that it indicate proper values. The first step is to clean all ball bearings thoroughly, reassemble and add a few drops of light oil to the bearing races.

Then the thread around the bolster should be wrapped in the same direction as when it is connected, but instead of the end of the thread's being fastened, it is run over the end of the ball bearing pulley attached to the base. The pulley is shown in Figure 2. Next, known weights are hung on the thread to produce forces on the beam over the expected range of operation. Readings are always taken when the beam is being loaded, as values taken when it is being unloaded include friction in the ball bearings. The motor should be running during the calibration so as to jar loose any static friction. The data obtained are used to plot a calibration curve of force in grams versus SR-4 recorder readings (multiplied by the scale constant) (See Figure 7). These values, K, indicate strain in micro-inches per inch. The values in grams are converted to pounds to give a conversion curve of pounds of force versus SR-4 recorder readings (Figure 10).

Determination of friction force.--The force, read from Figure 10, is the force measured on the outside surface of the bolster. The equivalent force on the inside of the journal bearing is obtained from the fact that the torque transmitted is constant. Hence, the force at the bolster multiplied by its radius is equal to the force on the inside of the bearing multiplied by its radius. Since the ratio of the outer radius to the inner one is 2.367, the force on the bearing will be 2.367 times greater than that on the bolster. This force on the bearing is the one which we assumed equal to the force on the journal earlier in the discussion.

<u>Speed determination</u>.—The exact speed of the spindle is determined by use of a stroboscope. Frequent checks should be made during runs, and the average value of the speed used.

<u>Ambient temperature control</u>.—The temperature in the enclosure is maintained at 85°F. by electric bulbs furnishing heat and a blower to maintain uniform conditions.

<u>Oil temperature and viscosity determination</u>.—The oil temperature in the journal is read directly from the Foxboro indicator. Time must be allowed before recording friction forces to allow the apparatus to attain temperature equilibrium, as a change of one or two degrees produces a marked change in friction. The absolute viscosity in micro-Reyns (10^{-6} lb. sec./in.^{2}) is read directly from Figure 11, plotted from data furnished by The Texas Company for their oils.

<u>Test procedure</u>.—The test procedure was as follows:

1. The oil reservoir was filled with the oil to be used in the test.
2. The motor, heating elements, fan, temperature indicator and SR-4 strain recorder were turned on and allowed to attain temperature equilibrium; a process requiring about half an hour. The ambient temperature was adjusted to 85°F.
3. The temperature indicator and the stroboscope were calibrated.
4. The line of action of the horizontal load on the bearing was adjusted through its center.
5. The filling tube of the bolster was adjusted to a position opposite the driving pulley.

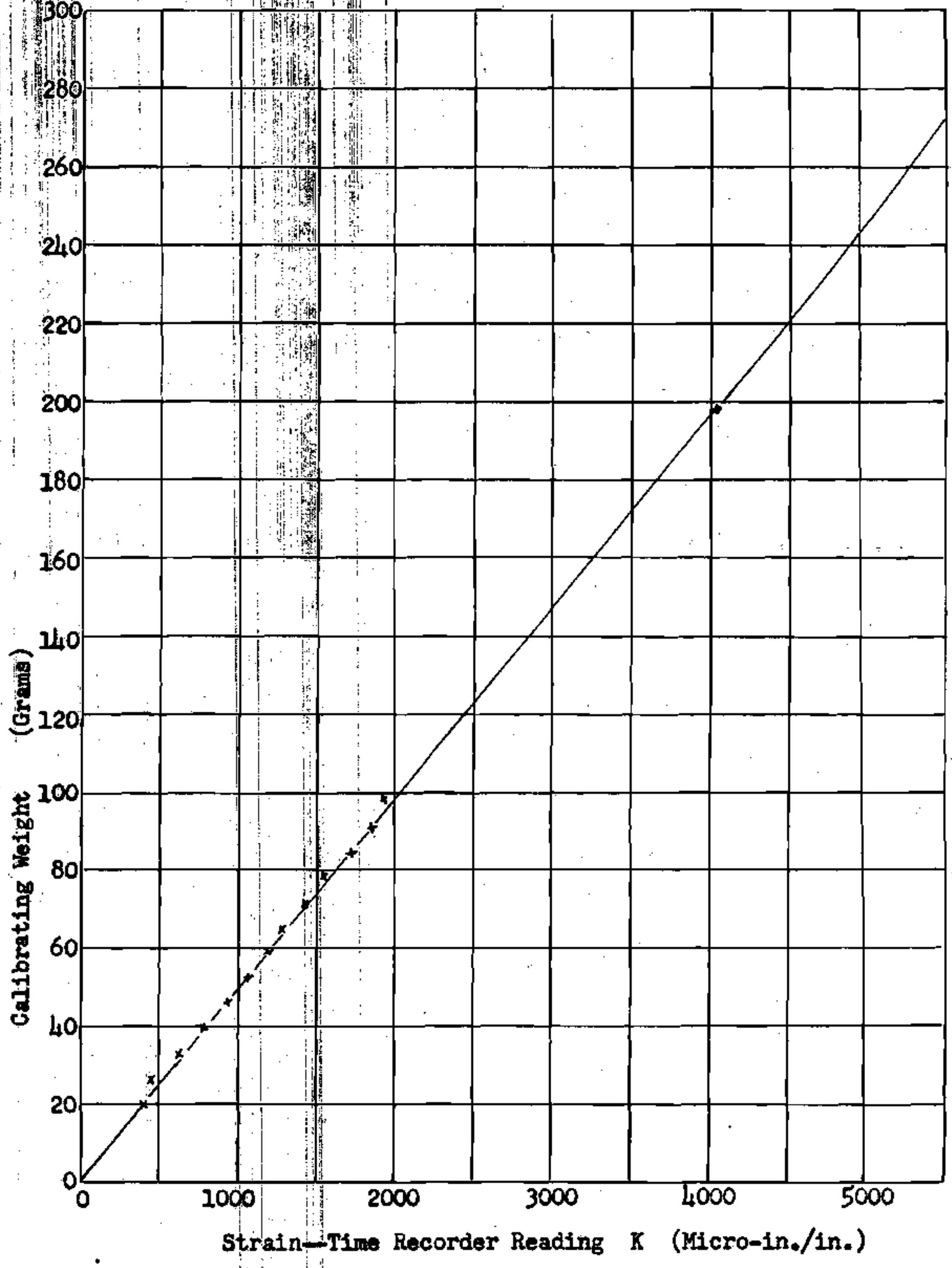

Figure 7. Calibration of Friction Measuring Device

6. When thermal equilibrium was obtained, the run was started. It was found that three minutes were usually adequate for each run. However, any indication of fluctuation of temperature or friction necessitated longer runs to obtain average conditions.
7. Runs were made in the order of lowest temperature conditions first, so that it was not necessary to wait for the reservoir to cool off before the next run.

CHAPTER IV

DISCUSSION OF RESULTS

Spindle friction curves with vertical loads constant.--The plot of all data taken with a constant vertical load on the spindle (shown in Figure 8) when compared with the solid curves obtained from theoretical data shows that up to a Sommerfeld number of 100 the bearing is operating above the 360° bearing curve with approximately the same characteristics as the theoretical curve. On the left, the curve rises, due to the assumption made that the friction force on the journal is equal to that on the bearing. This assumption produced an error which makes the coefficient of friction as computed from the data, increase rapidly as a Sommerfeld number of zero is approached. Also, this rise may be due in part to the Sommerfeld numbers being in the boundary lubrication region. On the right of the curve, the trend is below the 360° bearing curve, dropping generally to the 140° curve. This was more pronounced with the more viscous oils. Some points below the 360° line were obtained for more viscous oils with even as low a Sommerfeld number as thirty. This is probably due to one or both of the following:

1. Oil whip is a self-induced vibration which according to Hagg becomes critical when the angular velocity, or whirl, of the journal center about the bearing center approaches the critical frequency of the rotor.[11] This whirl is approximately one-half the angular velocity of the journal bearing for lightly loaded bearings and one-third for heavily loaded bearings. Oil whip could cause breaking of the film of oil around the journal and, hence,

cause the bearing to act as a partial bearing producing less friction, as indicated by the data.

2. Damping of vibrations by the heavier oils, thus causing the journal to run nearer the central position and create less friction with the bearing.

In any event, the data indicates that the journal bearing is probably operating hydrodynamically, since the experimental curve closely approximates the theoretical curve if the friction due to the pivot bearing is deducted. This will be discussed in more detail later.

Spindle friction curves with varying vertical loads.--The data for increasing vertical loads were plotted in Figure 9 for comparison with the theoretical 360° curve. The data were only plotted for the region shown since for lower Sommerfeld numbers, the data converge rapidly and for higher Sommerfeld numbers, the trend in the data is to fall off as explained above. Attempts were made to plot straight lines through each set of data, but the methods of "selected points", "averages" and even "least squares" did not provide satisfactory results. The difficulty arose from the fact that the quantities measured are so small, and there are so many variables, that the data were too eratic to produce a family of reasonable curves. However, the following method was devised for extrapolating to zero vertical load. An envelope was drawn which included all the points plotted (Figure 9). The top line was assigned the value of the greatest vertical load and the bottom line, the smallest load. From the geometry of the figure, using similar triangles, the following proportion was set up:

AB:BC equals EB:BD

(a-b):(b-c) equals (1303.45-300.55):(300.55-0)

or c equals b - (a-b):(3.34)

Using this relation, points were plotted in the range shown. These points showed a very close agreement with the 360° bearing curve. This technique of separation of journal and pivot bearing friction was not applied to the lower range of Sommerfeld numbers because the data there had a large error as previously mentioned. Nor was it applied to the higher range of Sommerfeld numbers since the trend of the data fell off in that range.

Vibration.--In some cases, particularly speeds around 10,000 revolutions per minute, excessive vibration developed and frequently the friction rose so rapidly the instrument had to be stopped to prevent damage to the friction recording apparatus. Sometimes with a journal load of around five pounds vibration developed in the heavily loaded region. Two general patterns of vibration were evident on the recording charts. One was characterized by fairly steady high and low traces with some irregularities in between. The other was merely a rapid up and down oscillation. The former was probably due to oil whip, and the latter to undamped unbalanced vibration.

Pivot bearing.--The pivot bearing was visually examined from time to time. When the spindle had run only one hundred hours there was a small portion of the tip of the pivot bearing which was very highly polished. As the spindle was run more, it was noticed that this area continued to enlarge. This indication of a polishing by "metal-to-metal" contact shows that the pivot bearing operates in the boundary region of lubrication. An examination of Figure 9 reveals that the increasing vertical loads on the pivot bearing produces a marked increase in the coefficient of friction f_j, indi-

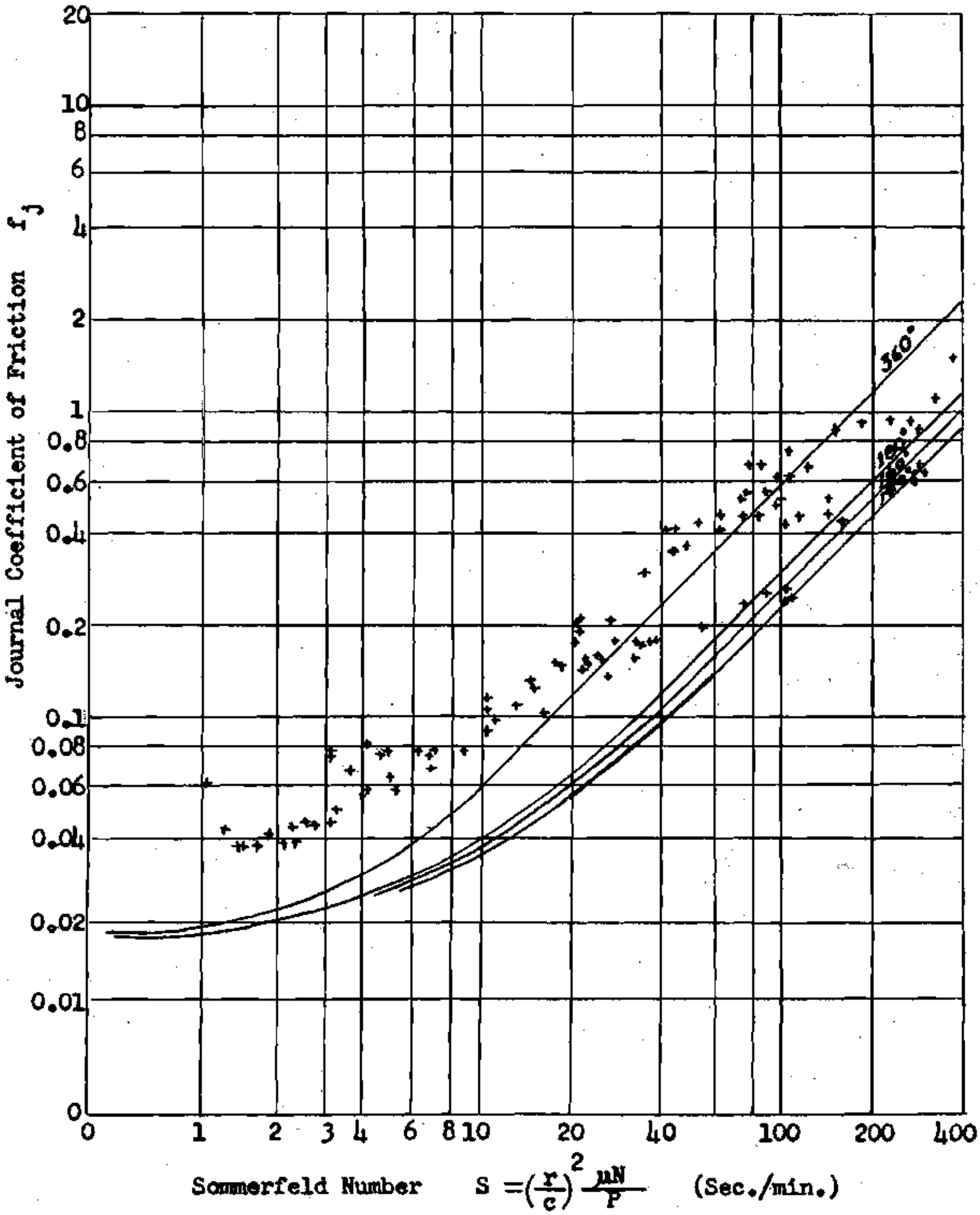

Figure 8. Spindle Friction Curves With Vertical Load Constant

cating that the pivot bearing does provide an appreciable amount of friction.

Further analysis of the data in Figure 9 showed that at a Sommerfeld number of twenty the largest vertical load produced an increase of two hundred and forty per cent in the friction. At a Sommerfeld number of one hundred and only the spindle as vertical load the combined friction was approximately eleven per cent greater than the 360° line. Other points gave per cent values between these extremes.

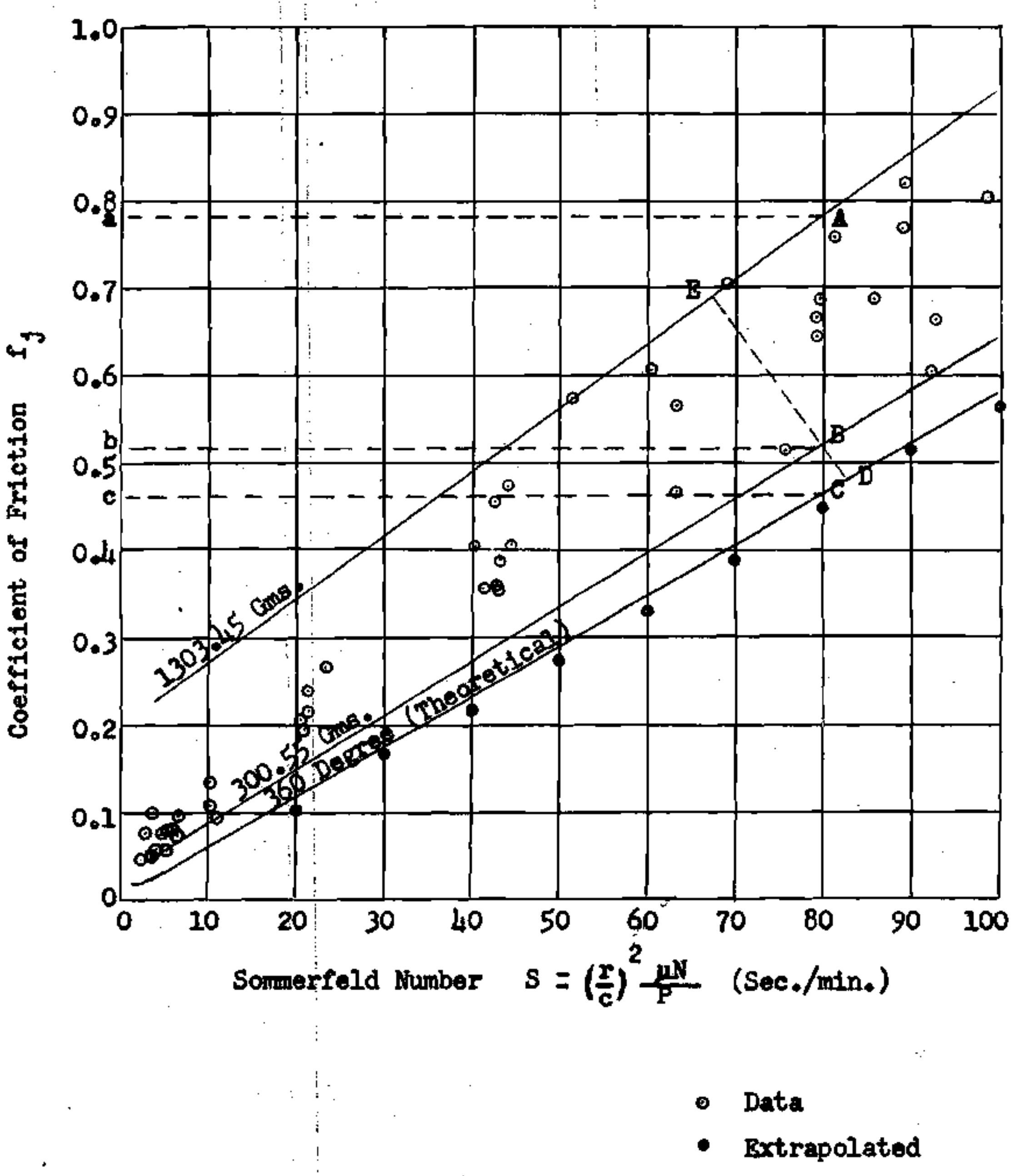

Figure 9. Spindle Friction Curves With Varying Vertical Loads

CHAPTER V

CONCLUSIONS

The limited amount of data accumulated during this investigation coupled with the non-existance of a quantitative theory for the spindle pivot bearing do not appear to permit absolute generalization. However, insofar as the conditions of spindle operation herein described are concerned the following can be concluded:

1. The journal bearing of the textile spindle investigated operates hydrodynamically.

2. The pivot bearing operates in the boundary region of lubrication.

3. The journal bearing operates as a full 360° bearing at least in the range of Sommerfeld numbers from 20 to 100.

4. The journal bearing operates with increasing partiality (decreasing bearing angle) as the Sommerfeld number increases beyond approximately one hundred, dropping to about a 140° partial bearing at a Sommerfeld number of 400.

5. The pivot bearing increases the friction in the bolster by a value varying from approximately eleven to two hundred and forty per cent depending upon the vertical load and the Sommerfeld number.

CHAPTER VI

RECOMMENDATIONS

In this investigation the effects of varying the point of application of thread load were not considered. The study of this variable is recommended as a possible future investigation.

The effects of vibration and oil whip should be investigated more fully, and the correlation between these and the Sommerfeld number, if any, should be established to guide future designers.

The pivot bearing should be redesigned, or work should be done with additives to reduce its friction.

To secure more accurate data from the present apparatus, a more positive method must be devised for insuring that the line of action of the journal load passes through the center of the bearing. The bolster must also be carefully positioned in the same location for each run so that more uniform results become possible.

A P P E N D I X

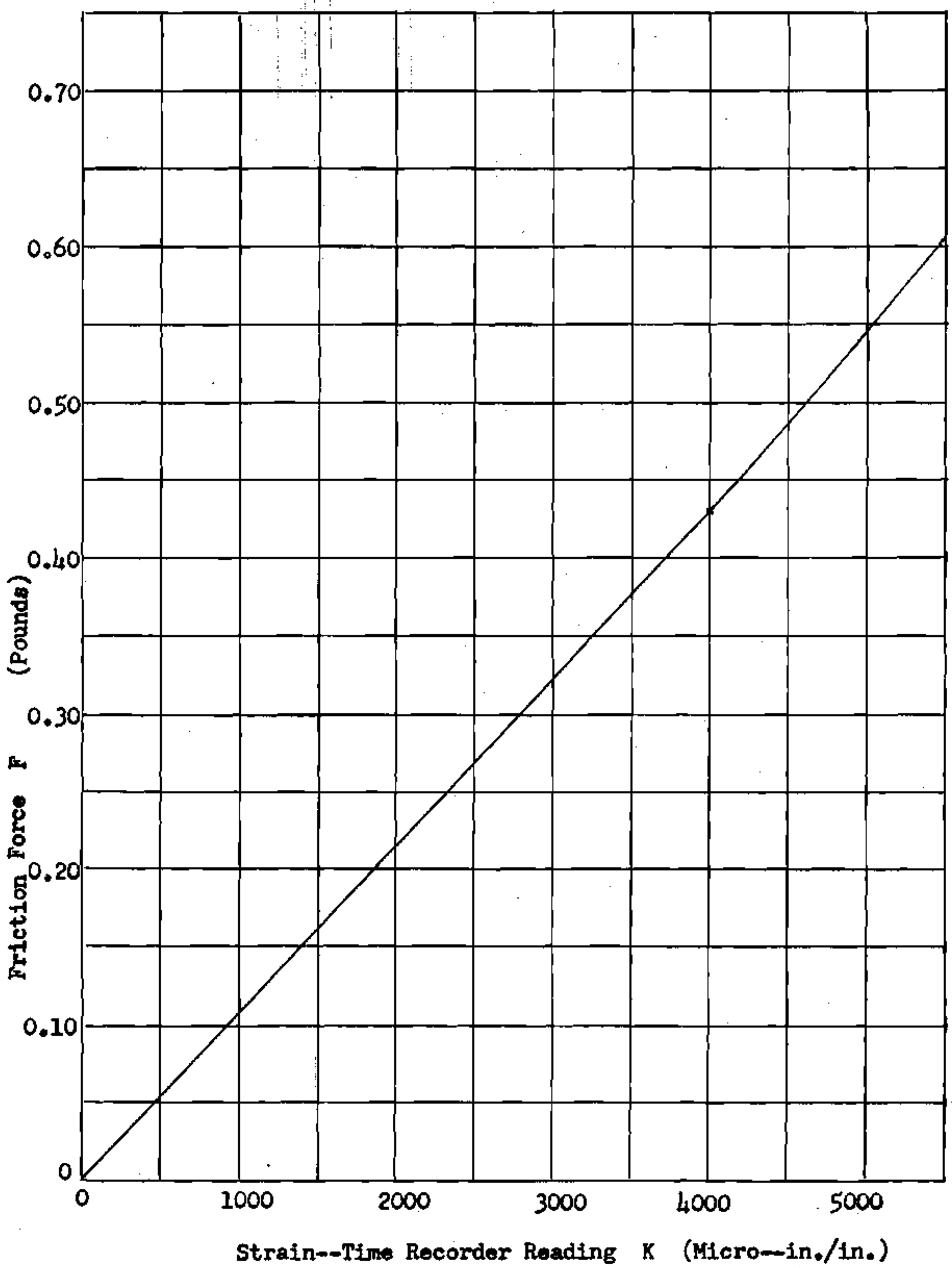

Figure 10. Conversion Curve K — F

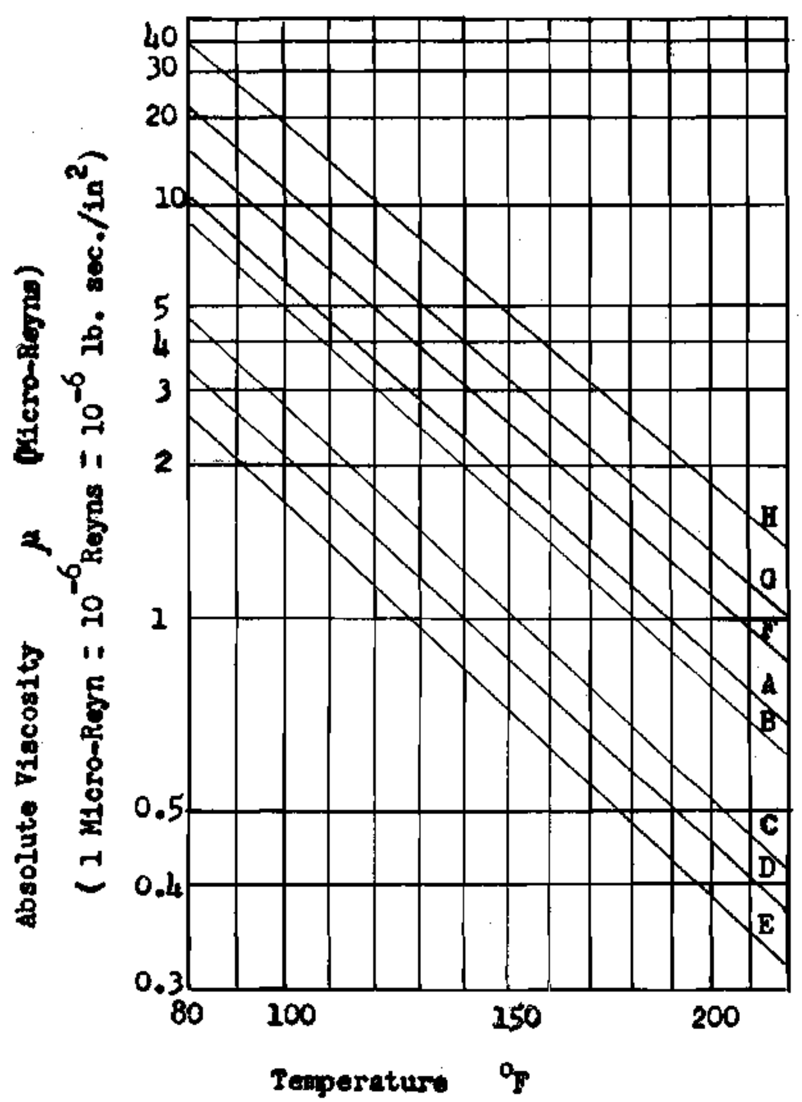

Fig. 11. Viscosity—Temperature Curves

Table 1. Calibration of Friction Measuring Device

Weights Reference Number	Grams	Cumulative Weight Grams	Strain K Micro-in/in
1	20.693	20.693	400
2	6.103	26.794	460
3	6.049	32.846	646
4	6.748	39.593	790
5	6.690	46.283	940
6	6.188	52.471	1060
7	6.648	59.119	1180
8	6.170	65.289	1290
9	6.594	71.883	1430
10	6.651	78.534	1550
11	6.501	85.035	1720
12	6.680	91.716	1860
13	6.838	98.554	1930
--	99.975	198.529	4050
--	200.085	298.639	5900

Table 2. Tabulated Data for Theoretical Curves

Sommerfeld Number S Sec/min	Coefficient of Friction of Journal, f_j, for Bearings of 360°	180°	160°	140°
400	2.280	1.175	1.018	0.894
300	1.735	0.870	-	-
200	1.168	0.586	0.508	-
100	0.581	0.296	0.256	0.224
40	0.229	0.118	0.107	-
20	0.117	0.0634	0.0593	0.0544
10	0.0594	0.0382	-	-
4	0.0291	0.0251		
2	0.0221	0.0202		
1	0.0194	-		
0.6	0.0188			
0.2	0.0183			

Table 3. Tabulated Data for Friction Curves with Vertical Load Constant at 378.55 Grams

Spindle rpm	Oil Type	Journal Load lbs	Journal Temp °F.	Strain K micro-in/in	Coefficient of Friction f_j	Sommerfeld Number S sec/min
3950	E	0.5	86	780	0.407	44.64
3950	E	1	88	810	0.212	21.24
3950	E	2	88	825	0.108	10.62
3950	E	3	89	892	0.0778	6.92
3950	E	4	90	984	0.0642	5.06
3950	E	5	90	1020	0.0568	4.05
3950	E	6	92	1146	0.0500	3.29
3950	E	7	93	1180	0.0442	2.71
3950	E	8	93	1210	0.0395	2.37
3950	E	9	93	1350	0.0392	2.12
3950	E	10	94	1580	0.0413	1.84
3950	E	11	95	1610	0.0382	1.66
3950	E	12	95	1740	0.0378	1.52
3950	E	13	95	1910	0.0383	1.40
3950	E	14	97	2400	0.0437	1.27
3950	E	15	105	3575	0.0608	1.01
3940	E	2	88	894	0.117	10.60
5750	E	2	90	1020	0.133	14.75
7650	E	2	94	1150	0.150	17.99
9540	E	2	98	1340	0.175	20.60
10450	E	2	103	1350	0.176	20.33
11390	E	2	101	1140	0.149	23.01
10450	E	0.5	104	1310	0.685	79.68
11460	E	0.5	105	1300	0.791	85.74
11460	E	1	106	1340	0.350	41.72
3950	E	2	85	750	0.098	11.27
5800	E	2	86	950	0.124	15.23
7700	E	2	90	1120	0.143	18.49
9600	E	2	97	1080	0.141	22.22
11470	E	2	101	1180	0.154	25.83
7680	E	1	92	1160	0.303	35.55
9580	E	1	99	1310	0.342	40.51
11460	E	1	101	1380	0.366	43.72

Table 3. (Continued)

Spindle rpm	Oil Type	Journal Load lbs	Journal Temp °F.	Strain K micro-in/in	Coefficient of Friction f_j	Sommerfeld Number S sec/min
3290	B	1	90	1575	0.412	64.20
5725	B	1	98	1775	0.463	74.48
7620	B	1	105	1775	0.463	82.75
9500	B	1	107	1900	0.495	97.45
11380	B	1	113	2060	0.525	100.50
3880	H	1	93	2300	0.587	238.0
5720	H	1	98	2350	0.598	287.0
7600	H	1	105	2500	0.639	306.0
9500	H	1	115	2500	0.639	285.0
11360	H	1	120	2650	0.677	298.0
3940	H	2.5	100	2290	0.235	74.8
5750	H	2.5	106	2500	0.255	90.3
7660	H	2.5	110	2350	0.240	107.2
9500	H	2.5	115	2400	0.245	114.0
11380	H	2.5	124	2600	0.266	107.2
4020	H	5	105	3500	0.178	32.7
6400	H	5	118	3500	0.171	34.5
6400	H	5	115	3500	0.178	38.4
8585	H	5	127	3500	0.178	36.9
11250	H	5	135	3500	0.178	38.8
13600	H	5	145	3500	0.178	36.8
3850	H	0.5	95	2600	1.326	437.0
5570	H	0.5	106	2900	1.528	378.0
8600	H	0.5	112	2800	1.430	580.0
10950	H	0.5	110	2700	1.382	773.0
12750	H	0.5	115	2600	1.326	771.0
3800	G	0.5	86	2200	1.122	330.0
3800	G	0.5	86	1700	0.436	165.0
3800	G	3	86	2300	0.196	54.9
3800	G	5	86	3025	0.154	32.9
4000	G	6	105	3200	0.136	27.3
4000	G	8	105	3650	0.117	20.5
4000	G	10	105	4250	0.109	16.4

Table 3. (Continued)

Spindle rpm	Oil Type	Journal Load lbs	Journal Temp ^{o}F.	Strain K micro-in/in	Coefficient of Friction f_j	Sommerfeld Number S sec/min
4000	G	5	98	3050	0.156	22.8
6360	G	5	115	3000	0.154	22.3
8780	G	5	122	3060	0.156	26.0
11120	G	5	134	3100	0.158	24.6
13450	G	5	142	3300	0.168	24.4
4000	A	1	90	1620	0.415	78.5
6320	A	1	96	1700	0.436	105.0
8720	A	1	103	1800	0.462	119.6
11080	A	1	105	1800	0.462	145.2
13380	A	1	111	2100	0.535	148.3
3970	A	0.5	91	1700	0.871	154.0
6250	A	0.5	100	1830	0.938	183.0
8650	A	0.5	104	1830	0.938	230.0
11000	A	0.5	105	1750	0.895	288.0
13250	A	0.5	115	1850	0.944	267.0
3950	D	0.5	88	840	0.437	54.7
6300	D	0.5	93	1070	0.559	78.2
8620	D	0.5	98	1130	0.592	96.5
11000	D	0.5	102	1180	0.617	111.6
13300	D	0.5	105	1260	0.660	127.2

Table 4. Tabulated Data for Friction Curves with Varying Vertical Loads

Spindle rpm	Oil Type	Journal Load lbs	Journal Temp °F.	Strain K micro-in/in	Coefficient of Friction f_j	Sommerfeld Number S sec/min
			Vertical Load 300.55 Grams			
3950	D	0.5	90	840	0.438	52.5
6240	D	0.5	95	1015	0.622	74.2
8570	D	0.5	100	1100	0.558	91.0
10870	D	0.5	105	1200	0.616	106.0
13150	D	0.5	113	1500	0.767	107.2
3960	D	5	102	1600	0.0819	4.02
3960	D	10	98	1700	0.0436	2.21
3960	D	1	89	785	0.204	26.95
3960	D	2	90	835	0.109	13.16
3960	D	3	90	875	0.0758	8.78
3960	D	4	93	1250	0.0798	6.12
3960	D	5	95	1500	0.0767	4.72
3960	D	6	107	1800	0.0760	3.03
3960	D	7	100	2100	0.0768	3.01
3960	E	0.5	88	700	0.355	42.9
6250	E	0.5	92	900	0.464	63.1
8600	E	0.5	97	1000	0.511	75.8
11000	E	0.5	100	1175	0.602	92.2
13200	E	0.5	108	1300	0.664	93.0
3970	E	0.5	88	700	0.360	43.0
3970	E	1	89	725	0.194	21.0
3970	E	2	89	700	0.090	10.5
3970	E	3	90	800	0.0686	6.8
4020	E	4	90	900	0.0568	5.2
4020	E	5	90	1000	0.0516	4.2
4000	E	6	94	1100	0.0450	3.1
4000	E	7	95	1230	0.0450	2.5

Table 4. (Continued)

Spindle rpm	Oil Type	Journal Load lbs	Journal Temp °F.	Strain K micro-in/in	Coefficient of Friction f_j	Sommerfeld Number S sec/min
			Vertical Load 663.15 Grams			
3950	F	0.5	90	780	0.402	40.8
3950	F	1	90	815	0.206	24.4
4000	F	2	90	825	0.107	10.4
4000	F	3	90	860	0.073	6.9
4000	F	4	91	875	0.057	5.05
4000	F	5	90	1100	0.057	4.1
4000	F	6	90	1175	0.049	3.4
4000	E	0.5	88	750	0.388	43.4
6300	E	0.5	91	1100	0.564	63.5
8650	E	0.5	96	1250	0.640	79.4
13100	E	0.5	110	1500	0.767	89.0
			Vertical Load 1001.10 Grams			
3950	E	0.5	87	925	0.474	44.0
6250	E	0.5	93	1175	0.602	60.5
8620	E	0.5	96	1300	0.668	79.1
10900	E	0.5	100	1375	0.701	91.3
13050	E	0.5	105	1500	0.768	98.7
3950	E	0.5	88	880	0.455	42.8
3950	E	1	88	925	0.237	21.4
3980	E	2	90	1025	0.133	10.3
3980	E	3	90	1125	0.097	6.86
3980	E	4	90	1225	0.078	5.14
3980	E	5	90	1150	0.058	4.12
3980	E	6	90	1200	0.052	3.43

Table 4. (Continued)

Spindle rpm	Oil Type	Journal Load lbs	Journal Temp °F.	Strain K micro-in/in	Coefficient of Friction f_j	Sommerfeld Number S sec/min
		Vertical Load 1303.45 Grams				
5100	E	0.5	92	1125	0.578	51.4
7400	E	0.5	95	1375	0.702	69.0
9680	E	0.5	100	1500	0.767	81.2
11850	E	0.5	105	1600	0.820	89.6
13100	E	0.5	105	1575	0.806	99.1
4000	E	6	85	1230	0.0528	3.87
4000	E	2	85	1000	0.127	11.60
4000	E	1	85	1050	0.268	23.20

Table 5. Lubricating Oils

Oil Type	Viscosity v Centistokes at 100°F	130°F	210°F	Company Designation
A	46.38	22.67	6.27	TL-1514
B	39.54	19.70	5.70	TL-1517
C	21.28	11.63	3.87	TL-1513
D	17.13	9.76	3.49	TL-1515
E	13.32	8.40	2.96	TL-1512
F	66.87	31.44	8.04	TL-2301
G	90.20	40.75	9.67	TL-2517
H	150.36	63.40	13.12	TL-2518

Sample Calculation

Determination of a point on the experimental curve.-- For the following data from Table 3.

Speed	3950 rpm
Oil	E
Journal load	0.5 lb.
Journal temp.	86° F.
Strain (K)	760 micro-in/in

From Fig. 10 for K find the force to be.................. 0.0860 lbs.
Multiply by 2.367 and divide by 0.5 to secure the coefficient of friction at the bearing (which we assume to be the same for the journal)..0.407
For 86°F. from Fig. 11 find the viscosity of Oil E to be..2.26 micro-Reyns
Calculate the Sommerfeld number from the relation:

$$S = \left(\frac{r}{c}\right)^2 \frac{\mu N}{p} = \left(\frac{0.1794}{0.00315}\right)^2 x \frac{2.26x10^{-6}x3950}{0.5/0.365x2.13} = 44.64 \text{ sec/min}$$

where: S is Sommerfeld number (sec/min)

r is radius of journal bearing

c is radial clearance between journal and bearing

μ is absolute viscosity of oil in micro-Reyns (10^{-6}lb.sec/in^2)

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