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Turbulence in journal bearings

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TURBULENCE IN JOURNAL BEARINGS

by

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A

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ABSTRACT

In this investigation, an experimental study was performed on a journal bearing under laminar and turbulent flow conditions. Using relationships for friction factor developed previously by other investigators a comparison was made with results of this present investigation. Since the flow changes from laminar to turbulent with a change in speed of rotation, curves for friction factor as a function of speed were plotted. Various attempts to make the recording and measuring systems more sensitive were also made and are discussed in this thesis.

ACKNOWLEDGEMENTS

The author wishes to thank Professor J. A. Jones and Dr. R. B. Oetting for their guidance and help during the course of this investigation. Appreciation is also extended to Messrs. Lee Anderson and Dick Smith for their help in modification of the existing equipment.

Last but not the least, the author wishes to thank Miss Janet Davidson for her help in typing of this thesis.

This presentation is dedicated to the author's fiancee, Miss Malini Desai, whose inspiration was the incentive.

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I. INTRODUCTION

In this modern age of high speed machinery, there are various factors to be considered for efficient operations. The effect of high speed of rotation on the oil film in a journal bearing is one of these factors. It is known that when a liquid flows through a pipe, the flow depends on velocity; that is, it is laminar or turbulent depending upon whether the velocity is lower or higher than the critical speed. In this same way when the speed of rotation is less than the critical speed, the lubricant in the bearing is in laminar flow and when the speed is higher than the critical speed, the flow is turbulent. The governing factor is Reynold's number.

When there is turbulence in the bearing lubricant, the bearing performance is affected considerably. In this presentation, the effects of turbulence on various factors such as friction factor, horsepower loss, temperature, oil flow, and load carrying capacity are studied.

An attempt to experimentally verify the above effects was made, but results could not be obtained. The reasons for this failure are also given in this paper.

II. LITERATURE REVIEW

The investigation of turbulence in journal bearings is relatively new and is in the primary stages of development. A journal bearing is equivalent to two rotating cylinders. The journal which rotates is the solid inner cylinder and the bearing which is stationary is the hollow outer cylinder.

The idea of instability in flow due to speed of rotation occured first to G. I. Taylor (1)*, who in the year 1923, carried out an experiment on two concentric cylinders. Taylor proved that when the outer cylinder is stationary and the inner cylinder is rotating at a high speed, concentric to that, the fluid flow between the two cylinders is unstable beyond a certain speed.

When a liquid flows through a circular pipe, the type of flow depends upon the diameter of the pipe, velocity of flow, and the viscosity of the fluid. The governing factor is the Reynold's number, given by the equation

$$R_e = \frac{V \times d}{\nu} \qquad (1)$$

When the value of the Reynold's number is less than 2,000, the flow is in the laminar region. Between the

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*Numbers in parenthesis indicate references as listed in the Bibliography.

Reynold's numbers 2,000 - 10,000,* the flow is said to be in the transition zone. During transition, the flow is neither completely laminar nor completely turbulent.

Wilcock (2) was the first to verify the effects of turbulence on the performance of journal bearings through experiment. He also applied the concept of Reynold's number to the bearing which is summarized below.

Assuming a triangular velocity profile, laminar flow condition, and negligible radial clearance (i.e. radius of the journal and the bearing are the same for the calculations of frictional torque), the shear stress is given by the equation

$$\tau = \mu \frac{u}{c} \tag{2}$$

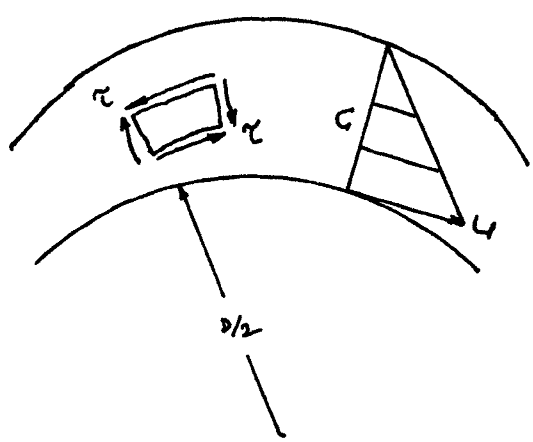

Figure 1

Representation of Velocity Profile in the Clearance

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*There are two opinions about the upper bound of the transition zone. Some authorities in fluid mechanics mention the value of Reynold's number as 3,000 for the beginning of turbulence.

The shear force is given by shear stress times the area and is equal to

$$F = \mu \frac{U}{C} \left(\pi D L \right) \tag{3}$$

The torque exerted is equal to the force times the lever arm, which in this case is the radius of the bearing. Thus, the torque is equal to

$$T = \mu \frac{U}{C} (\pi D L)(D/2) \tag{4}$$

Substituting U in terms of speed in revolutions per second and μ equal to $\rho \nu$, the torque can be expressed as

$$T = \frac{\pi^2 D^3 N L \nu \rho}{2c} \tag{5}$$

Taylor (1) changed this equation into dimensionless form in order to obtain a term which he defined as Reynold's number, given by the equation

$$R_e = \frac{\pi D N C}{\nu} \tag{6}$$

Taylor further defined the Reynold's number to be equal to an empirical relation given as

$$R_e = \pi^2 \sqrt{\frac{D}{0.0577\, C}} \tag{7}$$

The expression for critical speed (the speed at which the flow changes from laminar to turbulent) can then be obtained by equating Equations (6) and (7) and simplifying, resulting in (8)

$$N_{cr} = \frac{6.54 \times \nu}{r \times c} \left(\frac{r}{c}\right)^{1/2} \tag{8}$$

Thus, Equation (8) gives the value of critical speed in terms of radial clearance - c, radius - r, and the kinematic viscosity - ν. Once the dimensions of the bearing are fixed, the critical speed depends on the kinematic viscosity alone. Thus, it can be seen from Equation (8) that lubricants with low kinematic viscosity have low critical speeds.

Wilcock (2) conducted his experiment in the year 1950 on specially designed test equipment, which was capable of driving the bearing at very high speeds. On this unit, Wilcock tested various diameter bearings on test shafts. The test apparatus was designed to give a speed range from 250 rpm to 3,000 rpm. The bearing was loaded by a hydraulic cylinder and the maximum load which could be applied was 133,000 lbs. Wilcock selected various sizes of bearings with different clearances as listed in Table I.

TABLE I. DIFFERENT SIZES OF THE BEARINGS USED BY WILCOCK

Bearing Number	Diameter (in)	Length (in)	Radial Clearance	Re_{cr}
1	8	4	0.0064	1027
2	8	4	0.0100	818
3	8	4	0.0175	311
4	4	4	0.0075	668

Thermocouples were connected to these bearings along the circumference to measure the temperature of the oil. The inlet oil temperature, outlet oil temperature, and maximum observed temperature were recorded.

Readings of speed, torque, horsepower, and oil flow for each bearing at various speeds below and above the critical speed were obtained and the following curves were plotted.

(1) Viscosity versus Temperature

(2) $(T/\rho N^2D^4L)$ versus $(\pi DNc/\nu)$, which is called the Taylor plot.*

(3) Temperature versus Speed

(4) Power Loss versus Speed

(5) Oil Flow Rate versus Speed.

1. <u>Viscosity versus Temperature</u>. The viscosity varies with the temperature as shown in Figure 2.

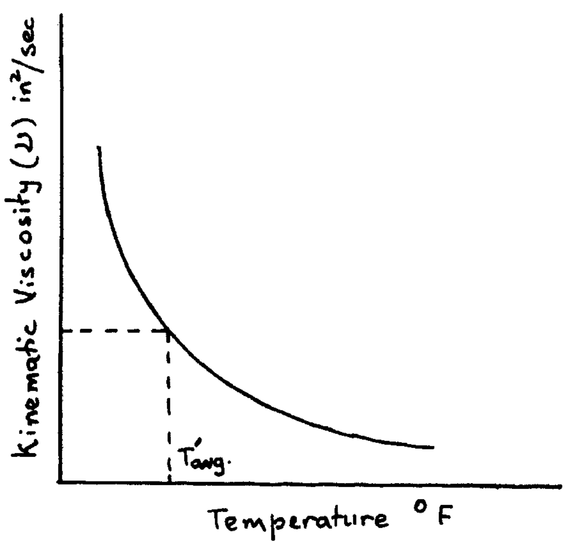

Figure 2. Viscosity as a Function of Temperature

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*The curve describes the nature of flow of the lubricant in the bearing.

The average temperature during the test can be approximately taken as

$$T'_{avg} = T'_{inlet} + \frac{T'_{outlet} - T'_{inlet}}{2} \qquad (9)$$

2. Taylor Plot, $(T/\rho N^2D^4L)$ versus $(\pi DNc/\nu)$.

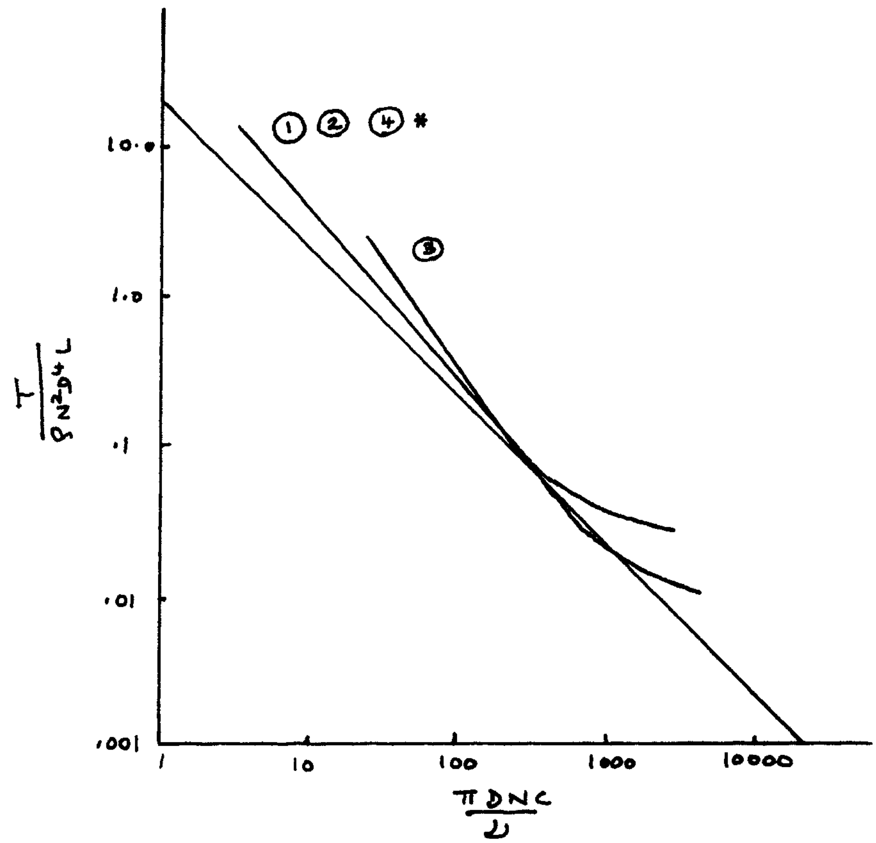

Figure 3. Taylor Plot

The theoretical curve, which is a straight line with a slope of 45°, is for the laminar region. The critical Reynold's number for each bearing is calculated by substituting the values of the corresponding critical speed in Equation 6.

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*Numbers in the circles refer to the bearings in Table I, page 5.

These critical Reynold's numbers are listed in Table I.

3. Temperature Rise Versus Speed.

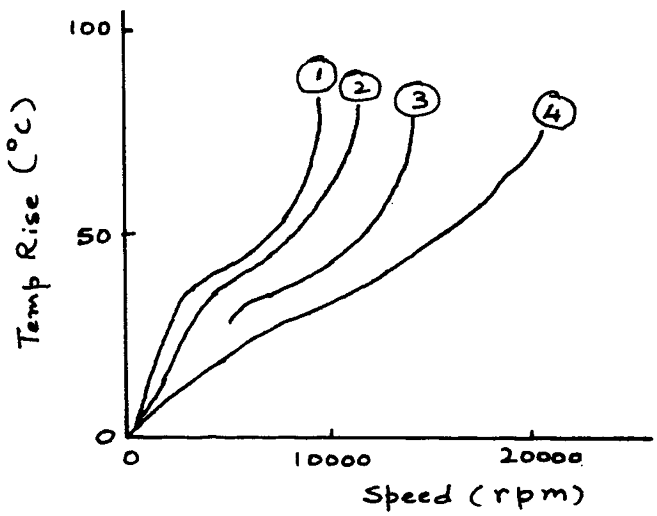

Figure 4. Temperature Rise as a Function of Speed

Figure 4 gives the value of increase in temperature above the inlet temperature as the speed increases.

4. Horsepower Loss Versus Speed.

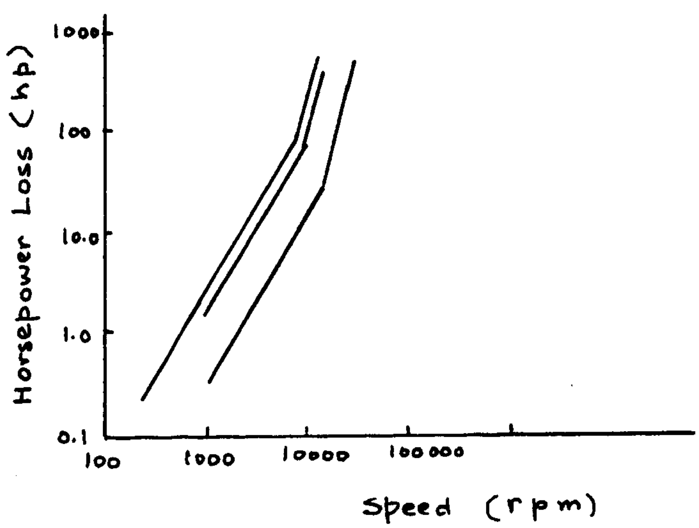

Figure 5. Horsepower Loss as a Function of Speed

When the horsepower loss is plotted against the speed, the increase in loss of horsepower due to turbulence is revealed. The horsepower loss varies as the 1.4 power of the speed below the critical speed and as the 2.7 power of the speed above the critical speed.

5. <u>Oil Flow versus Speed.</u>

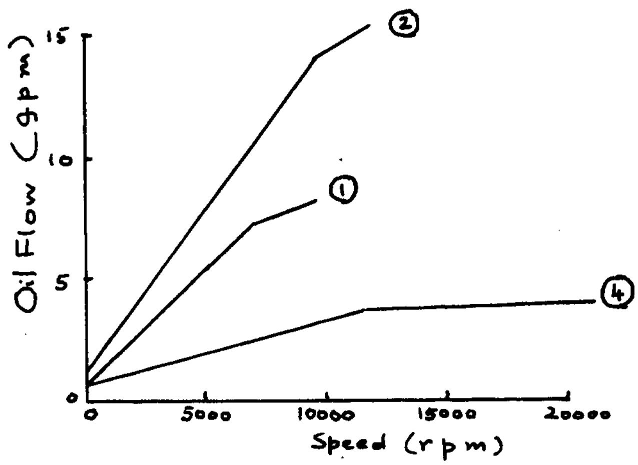

Figure 6. Oil Flow as a Function of Speed

Figure 6 shows depression in the flow at speeds higher than the critical speed. This is also an effect of turbulence because the oil flow should increase when higher inlet oil temperatures occur in high speed runs.

Smith and Fuller (3) also investigated this problem in the year 1956. They performed an experiment and studied the performance of the bearing under the condition of turbulence in the lubricant. They also developed a few expressions which would enable a calculation of the pressure distribution, friction factor, and load carrying capacity of a bearing under turbulent conditions in the lubricant.

The differential equation given below is known as Reynold's differential equation, and was derived from the equation of continuity and the Navier-Stokes equations assuming that the flow in the lubricant film is laminar, and the lubricant is a Newtonian fluid. The equation can be used to evaluate the performance characteristics of a bearing in laminar flow. The equation is given as

$$\frac{\partial}{\partial x}\left[h^3 \frac{\partial p}{\partial x}\right] + \frac{\partial}{\partial z}\left[h^3 \frac{\partial p}{\partial z}\right] = 6 \mu U \frac{dh}{dx} \tag{10}$$

where: h = film thickness

p = pressure

μ = viscosity

U = journal peripheral velocity.

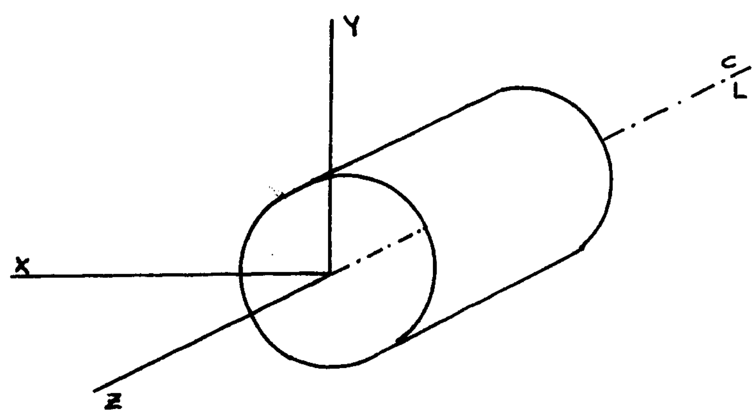

Figure 7. Co-ordinate Axis Along the Journal

Neglecting the end leakage (flow in the direction of the z-axis) the expression for pressure gradient for the laminar flow condition is given by

$$\frac{dp}{dx} = 6\mu U \frac{h-h'}{h^3} \tag{11}$$

In order to derive a similar expression for the turbulent flow condition Smith and Fuller (3) assumed the principle of superposition. The flow of lubricant is made up of two components; one flow due to turbulent shear, or the shear component of flow, and the other due to pressure, or the pressure component of flow.

The shear component of flow remained the same as that in the case of laminar flow and is given by the area of the velocity profile (the profile is assumed triangular).

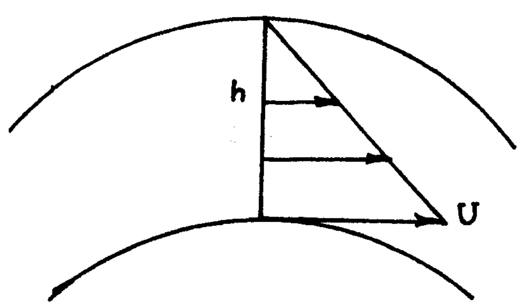

Figure 8. Velocity Profile

Thus, the shear component of flow is equal to the average velocity times the thickness and is given by the equation

$$q_s = \frac{Uh}{2} \tag{12}$$

The behavior of the pressure component of flow was studied in the laboratory by Smith and Fuller and the performance of the bearing was expected to give the nature of the pressure component of flow. It was found to be equal to

$$q_p = (U_{avg})(h) \tag{13}$$

The total quantity of flow is the sum of Equations (12) and (13), a constant, and is equal to

$$q = q_s + q_p = \frac{Uh'}{2} \tag{14}$$

The pressure gradient under turbulent flow is proportional to the square of average velocity and is given by the equation

$$\frac{dp}{dx} = -\frac{K\rho\,(U_{avg})^2}{2h} \tag{15}$$

where K is an empirical constant.

The values of U_{avg} from Equation (13) and (14) were substituted in Equation (15) and the pressure gradient was given by the equations

$$\frac{dp}{dx} = \frac{K\rho U^2 (h-h')^2}{4h^3} \qquad \text{if} \quad h > h' \tag{16}$$

$$\frac{dp}{dx} = -\frac{K\rho U^2 (h'-h)^2}{4h^3} \qquad \text{if} \quad h' > h \tag{17}$$

The pressure variation during turbulent flow of a lubricant is given by Smith and Fuller (3) as

$$p = \frac{K\rho U^2 r \varepsilon^2}{c^2\sqrt{1-\varepsilon^2}}\left[\operatorname{Arc\,Cos}\left(\frac{\cos\theta+\varepsilon}{1+\varepsilon\cos\theta}\right) + \frac{\sin\theta(\cos\theta+\varepsilon)(1-\varepsilon^2)^{1/2}}{(1+\varepsilon\cos\theta)^2}\right] \tag{18}$$

Smith and Fuller performed their experiment on a test unit which was designed to attain low critical speed, and selected water as the lubricant since the low kinematic viscosity of water would also help in suppressing the value of critical speed. The bearing was 3 in x 3 in with clearance ratio of 2.93 milli-inch per inch. The shaft was driven by a variable speed motor. Arrangements were made to record the pressure of the lubricant film at intervals of 7-1/2° over an arc of 255°.

Smith and Fuller also developed the expressions for the friction factors in the laminar and turbulent flow conditions. The friction factor in laminar flow is developed in the following paragraphs.

The shear stress is given by Equation (2) as

$$\tau = \mu \frac{U}{c} \tag{2}$$

The shear stress is also given in terms of friction factor, mass density, and velocity by an equation of the form

$$\tau = f \frac{U^2 \rho}{2} \tag{19}$$

Equating Equations (2) and (19) results in a value for friction factor of

$$f = \frac{2\mu}{\rho U c} \tag{20}$$

Substituting the values of $\mu/\rho = \nu$ and $\pi DN = U$ gives

$$f = \frac{2\nu}{\pi D N C} = \frac{2}{R_e} \qquad (21)$$

In turbulent flow the friction factor in terms of Reynold's number is given by Smith and Fuller (3) as

$$f = \frac{A}{R_e^{B}} \qquad (22)$$

where: B lies between 0 and 1, and A may be any positive value.

By multiplying Equation (19) by the bearing surface area, $2\pi rL$, and by the lever arm, r, an expression for the torque is obtained as

$$T = \pi f \rho U^2 r^2 L \qquad (23)$$

Smith and Fuller (3) evaluated A and B from their experimental results by calculating the friction factor using Equation (23), and then plotting the log of Reynold's number versus the log of friction factor as shown in Figure 9.

The values of the constants A and B were determined to be 0.078 and 0.43, respectively. This gives the equation for friction factor in the turbulent region as

$$f = \frac{0.078}{R_e^{.43}} \qquad (24)$$

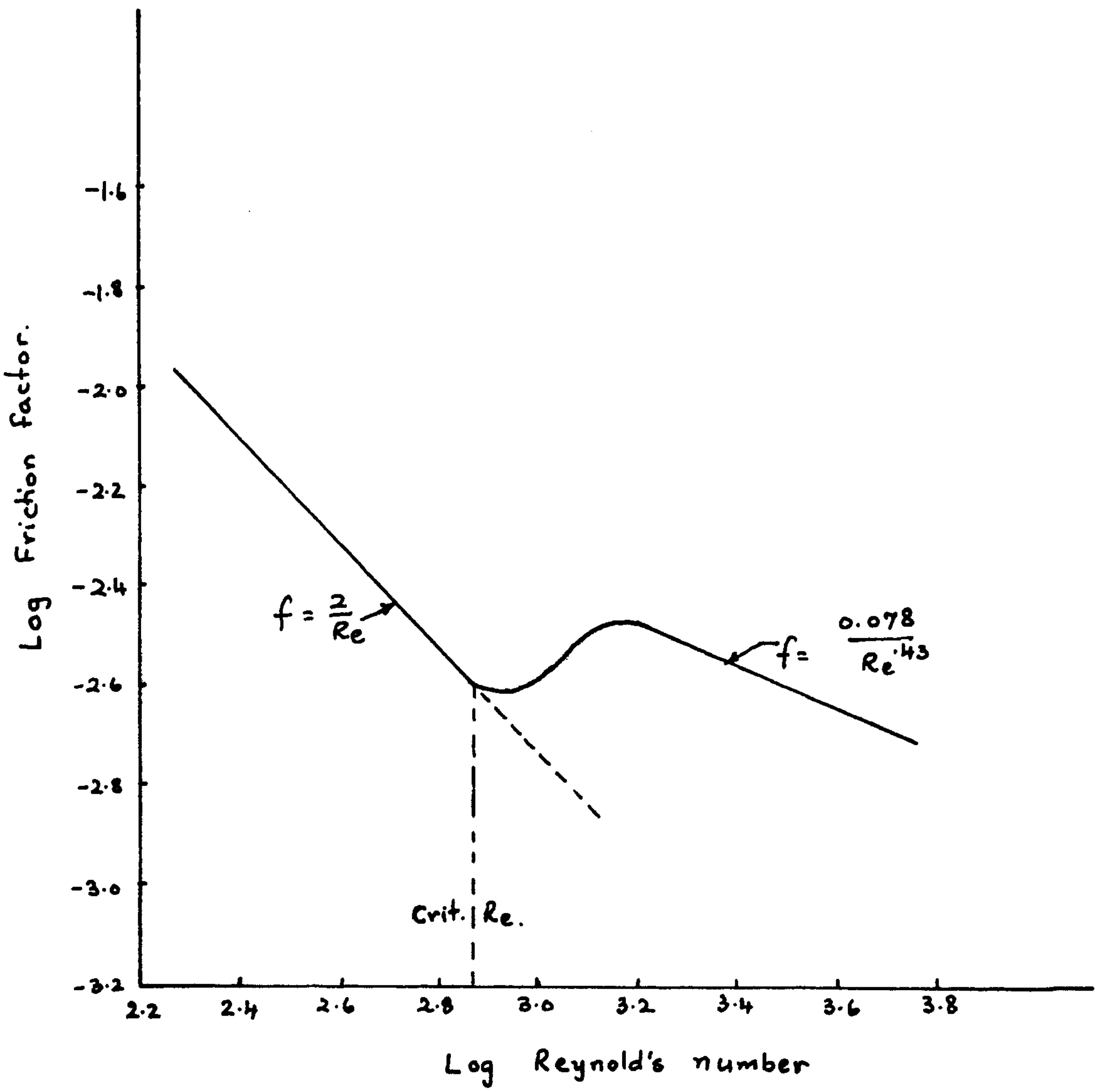

Figure 9. Variation of Friction Factor with Reynold's Number

III. EXPERIMENTAL WORK

The equipment used for conducting the experiment was designed and constructed by John Roberts (4). However, a few changes were made. The lubricant used by Roberts was oil whereas in this case water was selected in order to bring down the value of the critical speed.

A journal was made of mild steel with an outside diameter of 1.998 inches. It was mounted on two ball bearings and then connected to a small electric motor through a rubber hose. The rubber hose acted as a flexible coupling and helped to dampen vibrations to a considerable extent. The bearing was made of bronze with an inside diameter of 2.003 inches and a length of 2.0 inches. Thus, the radial clearance was 0.0025 inches. The outside diameter of the bearing was 2.375 inches so as to give 3/16 inches of wall thickness which was quite sufficient. (See Figure 10).

A bearing shell was provided which would enable introduction of lubricant to the bearing at different angles through aligned holes in the shell and bearing. The inside diameter of the shell was 2.376 inches and when fitted with the bearing, it left a diametral clearance of 0.001 inches. The purpose of such a close fit was to minimize the leakage of lubricant. The bearing was held in place by two set screws so that movement relative to the shell was prevented.

A torque arm was attached to the shell for the purpose

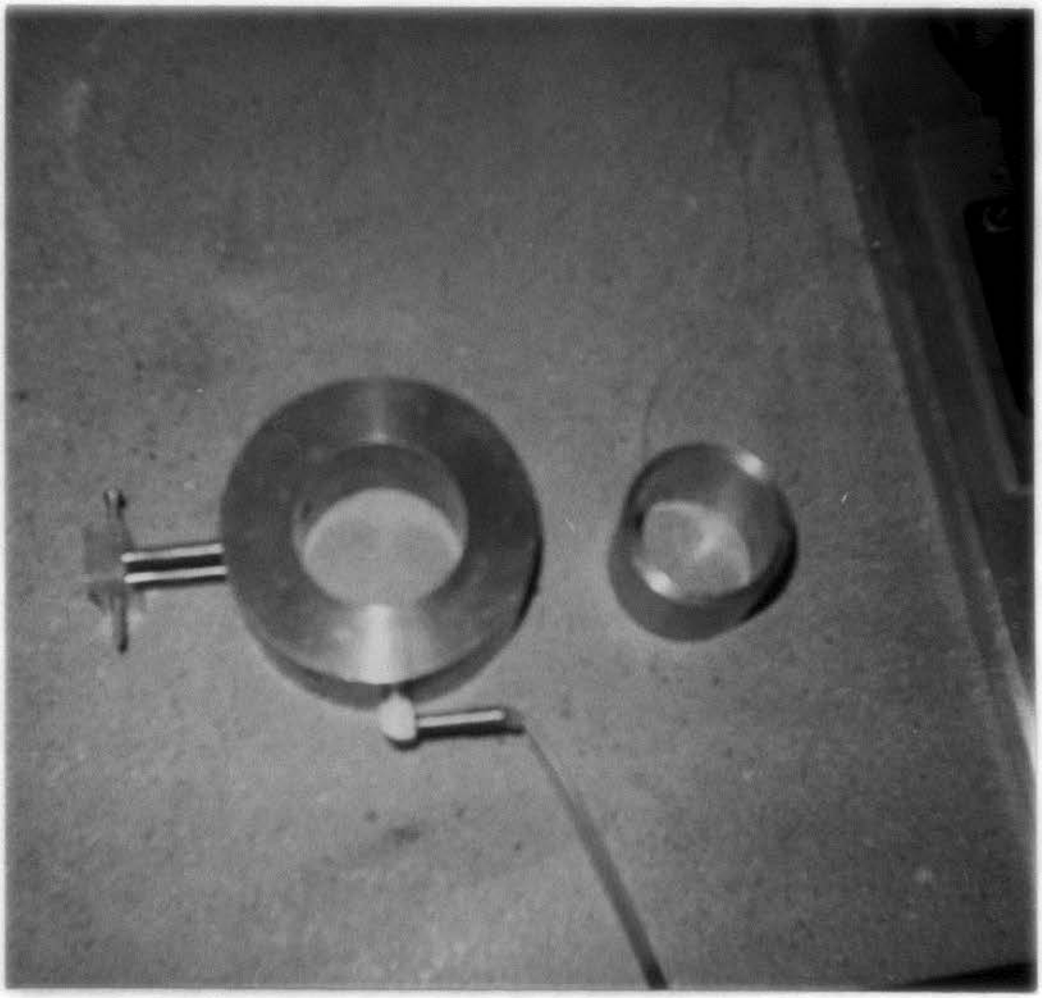

Figure 10. Bearing and the Shell

of transmitting the frictional torque to the end of a cantilever beam. The cantilever beam was mounted with an SR-4 (350 ohm) strain gage near the fixed end to measure the frictional torque of the bearing offered by the torque arm. The strain was measured with the help of a Wheatstone bridge. Dummy strain gages were used to compensate for temperature variations and to help balance the Wheatstone bridge. (See Figure 11). An industrial analyzer and brush recorder were used for the measurement of the strain. The cantilever was calibrated by suspending a known load at its free end where the torque arm made contact with the cantilever, and the curve of load versus strain was plotted. Knowing the measured strain, the load applied by the torque arm could be read from the curve.

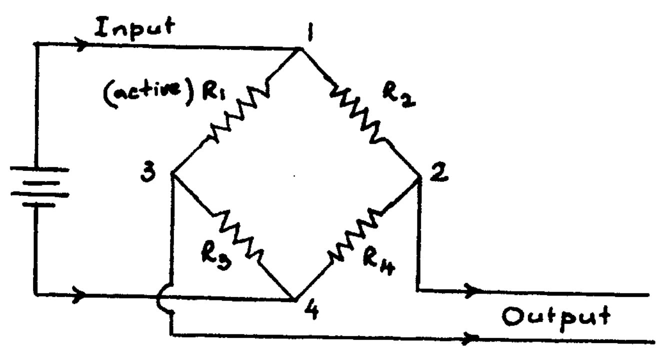

Figure 11. Wheatstone Bridge Connections with One Active Gage

Arrangements were made to supply the lubricant (water) to the bearing. For this purpose, a bottle with an outlet at the bottom was filled with lubricant and placed about 3-1/2 feet higher than the level of the bearing in order to provide sufficient head. The lubricant flowed from the bottle to the inlet hole of the shell through a plastic hose. An adjustable clip was attached to the hose at the bottle end to allow for variation in the lubricant flow to the bearing.

The lubricant coming out of the bearing was collected in a pan. The amount of water collected over a period of time was used to determine the flow rate of lubricant through the bearing.

In order to ensure that the load acted at the center of the journal, the load was applied with the help of a pulley. The pulley was connected by a small diameter wire which was wound around the shell.

The following is the summary of original dimensions.

Journal: 1.998 inches in diameter

Bearing: inside diameter: 2.003 inches
outside diameter: 2.375 inches

Radial clearance: 0.0025 inches

Cantilever: length: 10 inches
width: 1 inch
depth: 0.25 inches

The lubricating oil used in the experiment of Roberts (4) had an average viscosity of 4.3×10^{-6} ($lb_f\, sec/in^2$) with a specific gravity of 0.8. The kinematic viscosity, ν, was calculated to be equal to 0.0575 in^2/sec. By substituting the values of radius - r, kinematic viscosity - ν, and radial clearance - c into Equation (8), the critical speed of the test unit was calculated to be equal to 3010 rps or 180,600 rpm.

To reach such a high speed with the available equipment was practically impossible and measures were taken to reduce this critical speed to the range of 1000 to 1500 rpm.

The first step was to change the lubricant from oil to water because of the low viscosity (1.14×10^{-5} ft^2/sec) of water. Since the load applied to the journal is not of very high magnitude, the water film can safely carry the load and avoid metal to metal contact.

The second step was to increase the radial clearance. The journal, which was 1.998 inches in diameter, would have been easy to machine, but because it was firmly mounted in the housing and well supported in its bearing, it was not advisable to disturb the journal. The bearing, which was loose and small in size, was easy to handle and it was decided to increase the internal diameter of the bearing. The internal diameter of the bearing was machined to 2.010 inches to give a radial clearance of 0.006 inches.

Thus, water, with low kinematic viscosity, was chosen as the lubricant and the radial clearance was increased in order to reduce the critical speed. The critical speed under the new operating conditions was calculated to be equal to 22.9 rps or 1,370 rpm. To attain this speed and give a wide range of speed for operation in the turbulent zone, a Universal motor with speed capability up to 5,000 rpm was connected to the journal. The motor was firmly attached to the aluminum base plate, and electrically connected through a variac to allow for variations in speed.

The arrangements for measuring strain and supplying lubricant remained the same as discussed in the beginning of this chapter.

The adjustable screw of the torque arm was levelled against the end of the cantilever and an initial strain reading was obtained. Figure 12 shows the equipment.

The motor was started at the slowest possible speed by regulating current through the variac.

The strain reading recorded on the chart paper of the brush recorder was noted and the corresponding speed was measured with a tachometer. Then the speed was increased by a small amount and once again the strain was read out on the chart paper and the speed was measured by the tachometer.

This procedure was repeated with speed increments of 100 rpm from 600 rpm up to 3,000 rpm. This upper limit of 3,000 rpm gives about 1,700 rpm in excess of the calculated value of critical speed (1,370 rpm).

The friction factor, which increases considerably when the flow changes from laminar to turbulent, was expected to

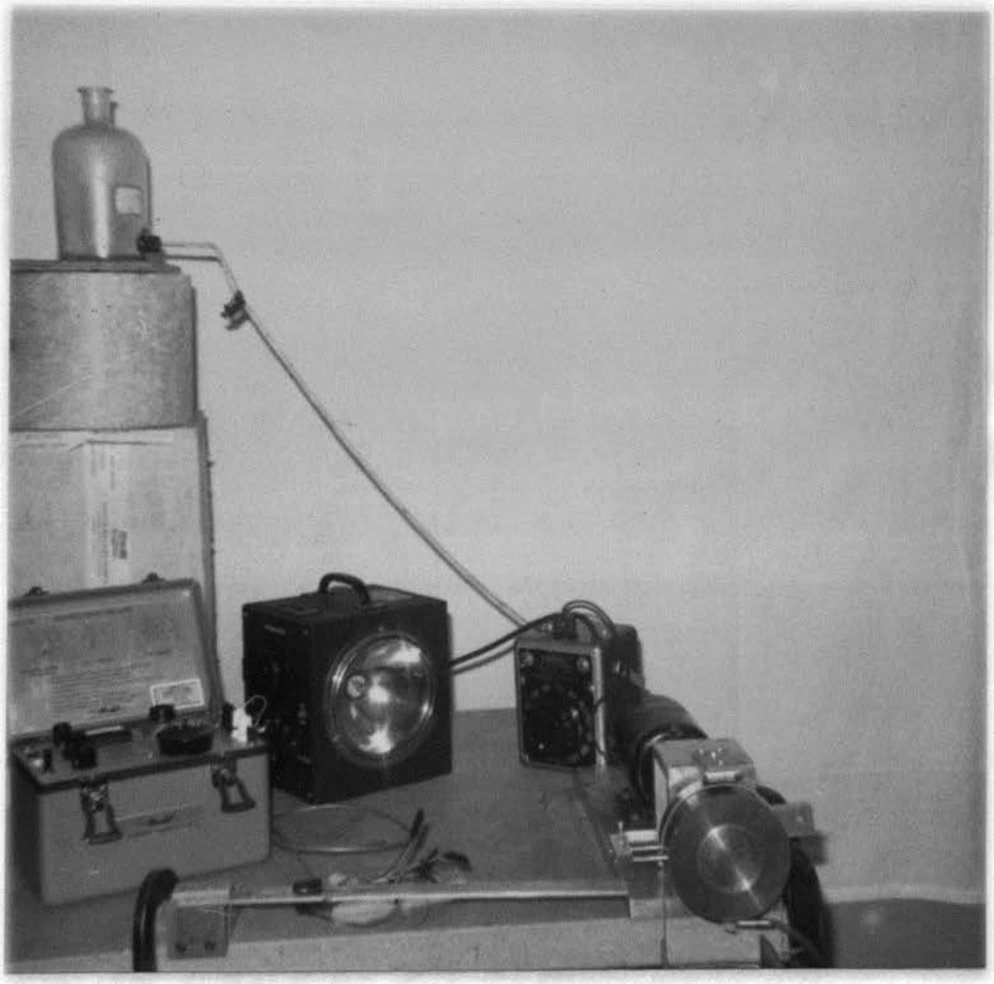

Figure 12. Set-up of the Equipment

give a higher value of frictional torque and consequently, a higher value of strain. This means that an appreciable difference in strain would have predicted the presence of turbulence in the bearing lubricant. But the strain reading did not change appreciably even when the speed was increased well above the critical speed. The pointer (pen bias) on the chart paper remained practically on the same line as it was when the motor was turned on. On the contrary, it fluctuated above and below this line due to external noise and vibrations, and it was difficult to decide whether the change was due to the increase in frictional torque or due to other external factors. Hence, it was decided to use direct strain indicators.

The use of direct strain indicators eliminated the use of the industrial analyser and the brush recorder. An active gage was mounted on the top side of the cantilever and the Wheatstone bridge circuit was balanced by three dummy gages. The strain gages were connected to the appropriate terminals of the strain indicator. The initial reading of strain without any load on the cantilever was recorded and the cantilever was once again calibrated by suspending known weights at its free end and noting the corresponding strain readings.

Once again the journal was rotated and the initial speed and the strain reading were noted. The readings of strain were also observed and noted at various speeds above the initial speed of the journal. The pointer on the dial of the strain indicator fluctuated by a small angle about the null point. This fluctuation resulted in negligible changes in the strain reading.

The cantilever of 10 inches length, one inch width, and 1/4 inch depth, was not sufficiently sensitive to the small frictional force acting at its free end. The magnitude of this force was dependent on the frictional torque. To make the cantilever more sensitive, its thickness (depth) was reduced to 1/8 inch, other dimensions remaining the same. Thus, the moment of inertia was reduced 8 times and consequently the stress and strain increased 4 times.

In order to make the cantilever even more sensitive, another strain gage was mounted on the bottom, below the first gage, and was connected in the Wheatstone bridge circuit. The connections were made as shown in the Figure 13.

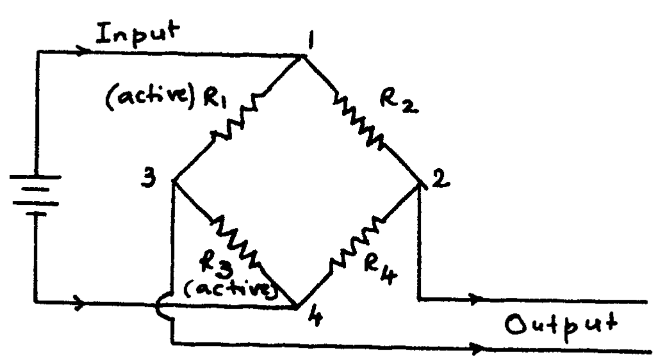

Figure 13. Wheatstone Bridge Connections for Two Active Gages

By connecting this second active gage in the adjacent branch of the Wheatstone bridge, the sensitivity was doubled and the strain read out was doubled.

The cantilever was calibrated by suspending known loads at its end and reading the corresponding strain. The readings are tabulated in Table II.

TABLE II. LOAD AND STRAIN READINGS FOR CALIBRATION

Load	Actual Strain	Theoretical Strain	Correction Factor
0	0.0	0.0	0.0
1	554	768	1.385
2	1112	1536	1.382
3	1667	2304	1.385
4	2228	3072	1.378
5	2783	3840	1.38

The corrected value of strain was calculated from Equation (25), which gives the strain in terms of load, dimensions of the beam and the Young's Modulus.

$$\lambda = \frac{(wl)(t/2)}{(E)(1/12\, bt^3)} \qquad (25)$$

The curves of the actual strain and the theoretical strain against load were plotted as shown in Figure 14.

The ratio of theoretical reading to actual reading was obtained in each case and was found to be close to 1.382;

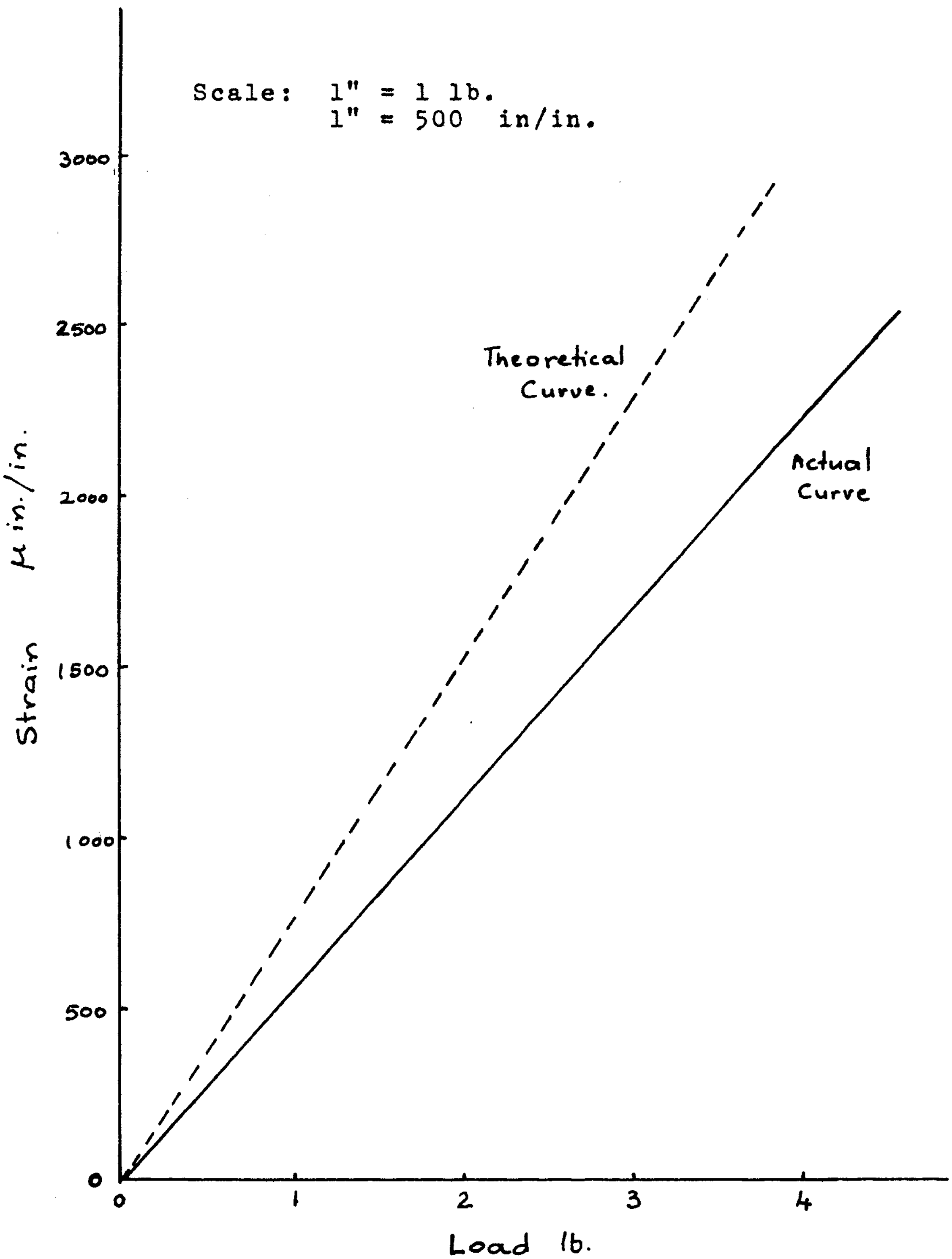

Figure 14, Calibration Curve

hence, the value of the correction factor was selected as 1.382. This factor when muliplied by the actual strain readings gives corrected values of strain.

Again the journal was run and after recording the initial reading of strain, the speed of the motor was varied and corresponding strain readings were obtained. The readings thus obtained are tabulated in Tables III and IV.

TABLE III. STRAIN READINGS WITHOUT LOAD

Sr. No.	Speed rpm	Strain μ in./in. Observed	Strain μ in./in. Corrected
1	0	0	0
2	1120	4	5.53
3	1890	2	2.77
4	2610	1	1.382
5	2720	43	59.5
6	3130	40	55.3
7	3920	22	30.4

TABLE IV. STRAIN READINGS WITH LOAD

Sr. No.	Speed rpm	Strain μ in./in. Observed	Strain μ in./in. Corrected
1	0	0	0
2	1160	8	11.08
3	1680	3	4.15
4	2500	9	12.45
5	3200	9	12.45
6	3920	9	12.45

IV. DISCUSSION

An expression for the friction factor as a function of the strain reading was obtained in the following way.

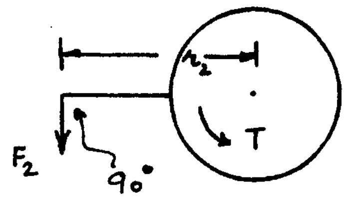

Figure 15. Force Acting at the End of the Cantilever

Torque is equal to the force times the lever arm and is given by the equation (See Figure 15).

$$T = (F_2)(r_2) \tag{26}$$

The force F_2 acting at the free end of the cantilever causes a bending moment equal to

$$M = (F_2)(\ell) \tag{27}$$

According to the flexure formula, the stress is then given by the equation

$$\sigma = \frac{(F_2 \ell)(t/2)}{1/12\, b t^3} = \frac{6 F_2 \ell}{b t^2} \tag{28}$$

The strain is equal to the stress divided by the Young's modulus and is given by the equation.

$$\lambda = \frac{\sigma}{E} = \frac{6 F_2 \ell}{b t^2 E} \tag{29}$$

Substituting for F_2 in terms of torque from Equation (26), the strain in terms of the frictional torque is given by the following equation.

$$\lambda = \frac{6\ell}{Ebt^2}\left(\frac{T}{r_2}\right) \qquad (30)$$

Equation (23), which gives the torque in terms of the friction factor and the other known quantities, is then substituted into Equation (30), giving the following expression for friction factor as a function of the strain.

$$f = \frac{\lambda E r_2 b t^2}{6\pi r^2 L u^2 \rho l} \qquad (31)$$

By using Equation (31), the friction factor for each set of readings was calculated. These values are tabulated in Table V.

TABLE V. FRICTION FACTORS FOR UNLOADED AND LOADED BEARING

Unloaded Bearing

Sr. No.	Speed rpm	Strain	Corrected Strain	Friction Factor
1	0	0	0	0
2	1120	4	5.53	0.00712
3	1890	2	2.77	0.00125
4	2610	1	1.382	0.000328
5	2720	43	59.5	0.0129
6	3130	40	55.3	0.00912
7	3920	22	30.4	0.0032

Loaded Bearing

Sr. No.	Speed rpm	Strain	Corrected Strain	Friction Factor
1	0	0	0	0
2	1160	8	11.08	.0133
3	1680	3	4.15	.00238
4	2500	9	12.45	.00321
5	3200	9	12.45	.00197
6	3920	9	12.45	.000582

Table VI gives the theoretical values of friction factor and Reynold's number for different speeds below and above the critical speed as calculated from Equations (6), (21), and (24).

A set of curves of friction factor versus speed on a log-log scale was plotted containing theoretical and experimental results. (See Figure 16).

The experimental values of friction factor showed considerable deviation from the theoretical curves. However, it did give some idea about the frictional behavior at higher speeds.

The change in friction factor did not take place at the calculated value of critical speed but at a higher speed.

For loaded bearings, the friction factor is higher and

TABLE VI. THEORETICAL FRICTION FACTORS

Laminar Flow			Turbulent Flow		
Speed rpm	Reynold's Number	Friction Factor	Speed rpm	Reynold's Number	Friction Factor
100	38.2	.0504	1100	420	.00577
200	76.4	.0252	1200	458	.00561
300	114.8	.0168	1300	497	.00542
400	153	.0126	1370	530	.00526
500	191	.01007	1400	536	.0052
600	229	.0084	1500	572	.0051
700	268	.0072	1600	611	.00494
800	306	.0063	1700	649	.00482
900	344	.0056	1800	688	.0047
1000	382	.00504	1900	726	.00458
1100	420	.00457	2000	764	.00447
1200	458	.0042	2100	804	.00438
1300	497	.0038	2200	840	.00431
1370	530	.00378	2300	878	.00423
1400	536	.00373	2400	917	.00415
1500	572	.00349	2500	955	.00406
1600	611	.00327	2600	994	.00398

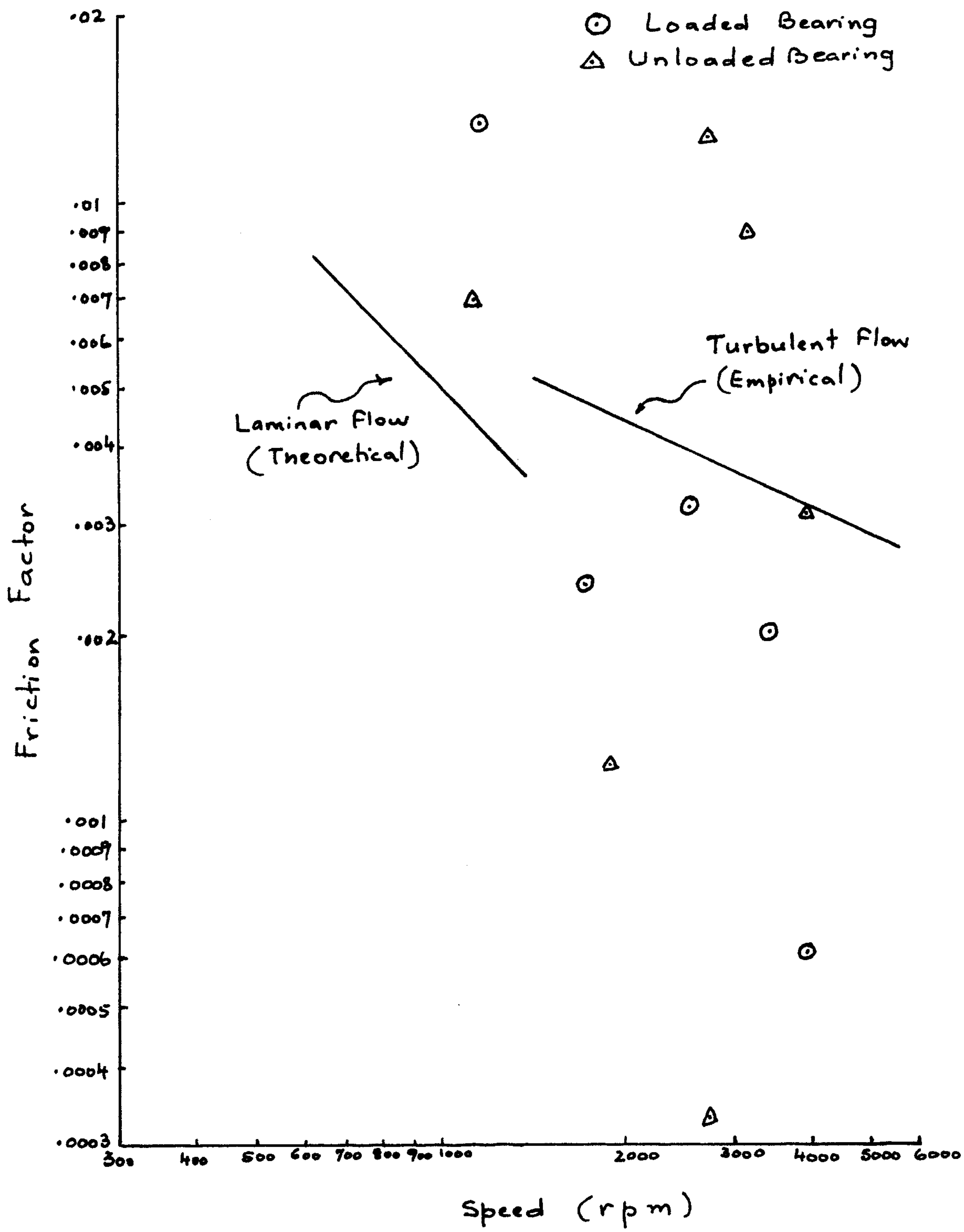

Figure 16. Variation in Friction Factor with Speed

the sudden rise in friction factor takes place at a lower speed.

Possible reasons for deviation from the theoretical friction factor curve are presented below.

The presence of turbulence is always uncertain since the range of the transition zone is uncertain. In the transition zone, the flow is neither laminar nor turbulent and consequently the frictional behavior is not predictable with the speed of rotation.

The magnitude of the frictional force is also very small. From Equations (21) and (24) developed by Smith and Fuller (3), the values of friction factor in laminar and turbulent flow at the critical speed of 1370 rpm are calculated to be*

$$f_L = \frac{2}{R_e} = 0.00378$$

$$f_T = \frac{0.078}{R_e^{.43}} = 0.00526$$

The value of Reynold's number at the critical speed is calculated by using Equation (6).

An expression for the difference in frictional torque, T_d, due to the change in the friction factor at the critical

*Suffix L and T refer to the laminar and turbulent flow conditions, respectively.

speed, is obtained by using Equation (23), resulting in

$$T_d = (f_T - f_L)\pi \rho U^2 r^2 L \qquad (32)$$

Substituting the values of each quantity in Equation (32) the difference in torque is calculated to be equal to 0.01787 in-lb.

This difference gives the value of frictional force equal to 0.004468 and the strain due to this force is 1.72 Microinch per inch. This difference is quite small and is difficult to be recorded by the instrument.

Taylor proved that the flow of lubricant between a rotating outer cylinder and a stationary inner cylinder remains more stable than between a rotating inner cylinder and a stationary outer cylinder. This implies that a bearing as shown in Figure 17 is more stable for lubricant from the consideration of turbulence.

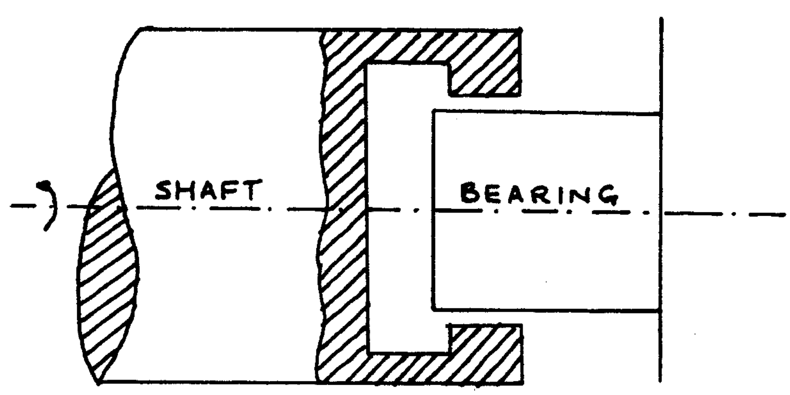

Figure 17. Proposed Journal Bearing System for Study

A study comparing the bearing of the above type with a standard type journal bearing of similar dimensions can be made and the difference in the critical speed for each system can be determined experimentally.

The following suggestions are made to make the system more sensitive and consequently to obtain better results.

The magnitudes of the frictional torque and the frictional force are small and the cantilever is not sensitive enough from the strain consideration. Equation (31), when rewritten in the following way, gives the value of strain in terms of the friction factor and other quantities.

$$\lambda = \left(\frac{6 \pi r^2 L u^2 \rho l}{E r_2 b t^2} \right) f$$

For a given value of friction factor the strain can be increased by increasing the value of the terms in the numerator and reducing the value of the terms in the denominator. Thus, the system can be made more sensitive by: reducing the width of the cantilever, shortening the distance between the torque arm and the center of the journal, using a longer bearing. An increase in the length of the cantilever will result in a higher strain reading; however, the vibrations caused by an increase in length of the cantilever could make the accurate reading of strain difficult. Further, an arrangement for amplification of the output voltage of the Wheatstone bridge could be used to improve the system sensitivity.

If the wire strain gages mounted on the cantilever beam are replaced by wire gages or film gages of higher gage factor, strain readings will be higher for a small change in frictional force and better results should be obtained.

SYMBOLS AND UNITS

D	Diameter of Bearing (in)
C	Radial Clearance (in)
f	Coefficient of Friction (dimensionless)
h	Film Thickness (in)
h'	Film Thickness at the Maximum Pressure Point (in)
L	Length of the Bearing (in)
N	Speed (rps)
p	Pressure (psi)
P	Load per Projected Area (psi)
r	Radius of Bearing (in)
R_e	Reynold's Number (dimensionless)
U	Journal Peripheral Velocity (in/sec)
μ	Lubricant Viscosity (Reyns = $lb_f sec/in^2$)
ρ	Lubricant Mass Density ($lb\ sec^2/in^4$)
τ	Shear Stress (psi)
Φ	Angle of Attitude (°)
ν	Kinematic Viscosity (in^2/sec)
K	Empirical Constant
V	Velocity of Flow (in/sec)
d	Diameter of Pipe (in)
e	Eccentricity (in) (Distance between the center of journal and center of bearing)
ε	Eccentricity Ratio = e/c (dimensionless)
q_s	Shear Component of Flow (cubic in/sec)
q_p	Pressure Component of Flow (cubic in/sec)

T_d	Difference in Torque (in-lb)
F_2	Force Acting at the End of Cantilever (lb)
r_2	Distance Between the Torque Arm and the Center of the Journal (in)
l	Length of Cantilever (in)
b	Width of Cantilever (in)
t	Thickness of Cantilever (in)
M	Bending Moment on the Cantilever (in-lb)
σ	Stress in the Cantilever (psi)
I	Moment of Inertia of the Cantilever Cross Section (in^4)
W	Load (lb)
E	Young's Modulus for the Material of Cantilever (psi)
λ	Strain (micro in/in)
T'	Temperature (°F)
θ	Angle Measured from Reference Line (°)

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