GAS LUBRICATION OF BEARINGS AT VERY HIGH TEMPERATURES

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ABSTRACT

The prime objectives of this program have been recently extended to include thrust as well as journal pneumodynamic and pneumostatic bearings and means of evaluating the lubricant-bearing systems at temperatures from 80 to 1500°F or higher at low lubricant flow rates. Specific equipment details and program requirements are given in the paper.

Flow orientation studies have been conducted at room temperature to speeds of 64,000 rpm (DN equivalent value 2.44 million). Gas bearings have been operated over the temperature range 80 to 1500°F with flow rates from 1.3 to 30 lb/hr over a range of speeds and loads.

Theoretical analyses result in generalized flow data for Type I (orifice compensated) and Type II (modified step) bearings which serve as a guide for experimental studies. Analyses of experimental data lead to an explanation of lubricant flow deviations between theoretical and experimental results at room and high temperatures. Generalized curves compare flow and load capabilities of many gases and saturated vapors from 0 to 2400°F with nitrogen as a reference lubricant.

INTRODUCTION

This paper covers work administered under the direction of the Applications Laboratory, Directorate of Advanced Systems Technology, Wright Air Development Division, with Mr. G. A. Beane acting as project engineer, and Mr. R. C. Sheard as Section Head. The overall objective of this program is to advance the state of the art of gas lubrication of high-speed bearings at very high temperatures.

We have accumulated a considerable amount of new theoretical and experimental information regarding gas lubrication of bearings at very high temperatures and high speeds. However, time available today does not allow for a discussion of all of the interesting details of our investigation. I will, therefore, highlight only those results achieved to date which may be of most interest to this general audience.

GAS LUBRICANTS

Viscosity

High-temperature, gas-lubricant-bearing systems involve both fluid-flow and heat-transfer considerations of gases. One of the most important properties of high-temperature gaseous lubricants is viscosity. Direct viscosity measurements of many potential gaseous lubricants are not available over wide temperature ranges. The kinetic theory of gases has been developed to the point where it is now possible to calculate rather accurately the transport properties of nonpolar gases and gas mixtures at low pressures. (The fundamental equations of rigorous kinetic theory were established by Maxwell and Boltzmann nearly a century ago and general solutions were independently obtained in 1916-17. However, it is only relatively recently that the necessary transport integrals (which must be evaluated numerically) have been computed for inter-molecular potentials approximating the forces between real molecules.

Calculated viscosities are now available for a great many gases at low pressure over wide temperature ranges. These gases include monatomic gases (H or N, etc.), diatomic gases (Air, H_2, N_2, etc.), simple polyatomic gases (CH_4, N_2O, etc.) hydrocarbons (C_2H_2, C_2H_4, etc.) and miscellaneous inorganic (N_2O_4, $SnCl_4$, etc.), and organic gases (C_2N_2, etc.). Limited viscosity data of selected binary mixtures (H_2-O_2, O_2-CO, etc.) and ternary mixtures (Ne-A-He) are also available over a range of temperatures. In this country the most generally used engineering form of absolute viscosity is the reyn (lb-sec/in^2). The reyn is a large unit of measurement since one reyn is equal to 68,950 poise (1 g-mass/sec-cm). However, the viscosity of a gas is very low compared to a liquid, and thus in our studies we have adopted a new term, the micro-milli-reyn (1×10^{-9} reyn), as the most convenient unit of gas viscosity to use when making engineering calculations (one micro-milli-reyn is approximately equal to cp/145).

Viscosity Values for Gases

Figure 1 shows values of absolute viscosity of many gases at atmospheric pressure and at temperatures from 0 to 2000°F. For all gases it is seen that there is an appreciable increase in viscosity as temperature increases. This is in contrast to a decrease in viscosity with temperature increase for liquids. The viscosity of air is close to the viscosity of nitrogen which is one of the prime decomposition products of the propellant hydrazine. It is also evident that the gases offering extremes in viscosity are neon on the high end and hydrogen on the low end. Hydrogen, another decomposition product of hydrazine, has approximately one-third the viscosity of neon and one-half the viscosity of nitrogen over the temperature range of interest. It is important, however, to note that hydrogen presents a safety hazard for use in high-temperature, high-speed gas-lubrication studies in most equipment due to critical air-hydrogen temperature-pressure relationships under "sparking" conditions which generally occur during high-speed gas-bearing studies.

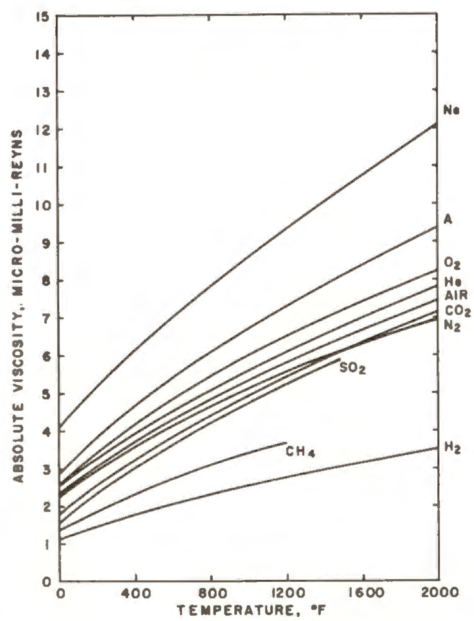

FIGURE 1. ABSOLUTE VISCOSITIES OF SEVERAL GASES FROM 0 TO 2000°F

Ambient Pressure Effects on Viscosity

Ambient pressure affects the viscosity of gases as shown in Figure 2. The viscosity increases approximately 12 percent for a pressure increase from 1 to 100 atmospheres for N_2 at room temperature. However, it is seen that this increase is only about 1 percent at 1500°F.

Oxygen dissociation of air is evident at temperatures above 3500°F whereas nitrogen dissociation occurs at approximately 4000°F. However, these temperatures are well beyond our present goal of 1800 to 2000°F. Other properties such as thermal conductivity, diffusion characteristics, critical relationships, vapor pressures, latent heats, free energies and vapor-liquid equilibra are also considerations in the design of a complete gas-bearing system for space applications.

EXPERIMENTAL GASES OF INTEREST

This study is directed toward gaseous lubricants representative of decomposition products of mono- and bi-propellants and saturated vapors of inorganic working fluids. For any reaction of rocket propellant combinations the products of decomposition may be different for every temperature. Furthermore, the percentage of each decomposition product varies with temperature. An example is given in Table 1 for hydrazine.

TABLE 1. THEORETICAL PRIMARY REACTION PRODUCTS OF HYDRAZINE AT DIFFERENT TEMPERATURES

	Percent		
Temp., °F	NH_3	N_2	H_2
1100	0	0.333	0.6667
1400	0.1248	0.3125	0.5625
1645	0.2000	0.3000	0.5000
1858	0.3040	0.2827	0.4134
2035	0.3592	0.2735	0.3673
2166	0.5000	0.2500	0.2500
2330	0.6248	0.2292	0.1460
2390	0.6856	0.2191	0.0955
2460	0.7440	0.2093	0.0467
2500	0.8000	0.2000	0

Preliminary studies of mono-propellants such as hydrazine, ethylene oxide, hydrogen peroxide, normal propyl nitrate and others indicate that the cleanest mono-propellant, from a standpoint of reaction products, is hydrazine or hydrogen peroxide. The cleanest bi-propellant is probably hydrazine plus hydrogen peroxide or hydrogen-oxygen. The decomposition products of such propellants as ethylene oxide include carbon which would result in a serious dirtiness problem in a gas lubricating system. In any gaseous lubrication system proper filtering is of utmost importance. We have found that this is particularly true in high-temperature systems.

The widest range of application for the results of this study is if two or more gases having viscosity extremes covering representative propellant decomposition product viscosities are used in gaseous lubricants. By conducting experiments with a few carefully chosen gases and correlating with theory, generalized results will be obtained which are applicable for many systems.

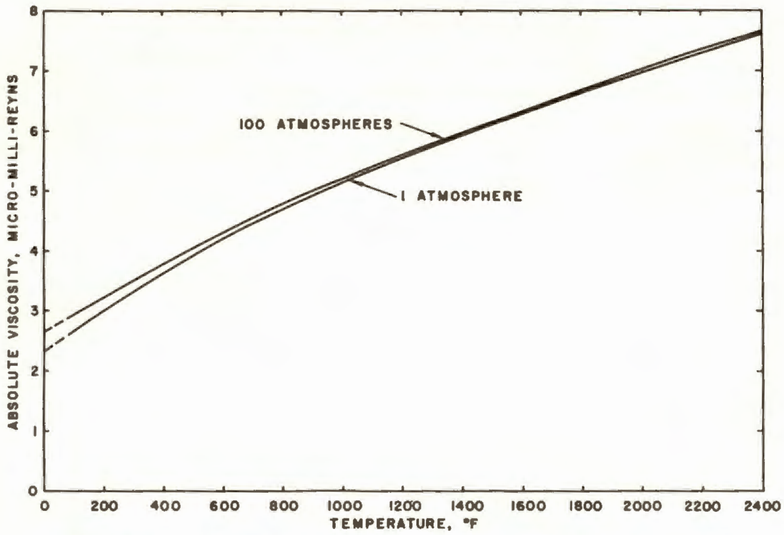

FIGURE 2. ABSOLUTE VISCOSITY OF MOLECULAR NITROGEN AS A FUNCTION OF TEMPERATURE

Viscosities of Saturated Vapors

Figure 3 shows viscosities of saturated vapors of inorganic working fluids over wide temperature ranges. The reference gases Ne, N_2 and H_2 are given for comparison. It is seen that saturated Hg vapor has a viscosity even greater than Ne while the saturated vapors of Rb, Na and K have viscosities even lower than hydrogen over the temperature range. Inasmuch as the load capacity of pneumodynamic gas bearings is proportional to viscosity, and since leakage of most pneumostatic gas bearings is inversely proportional to viscosity, H_2, Rb, Na and K are less desirable as lubricants than are the higher viscosity gases and vapors.

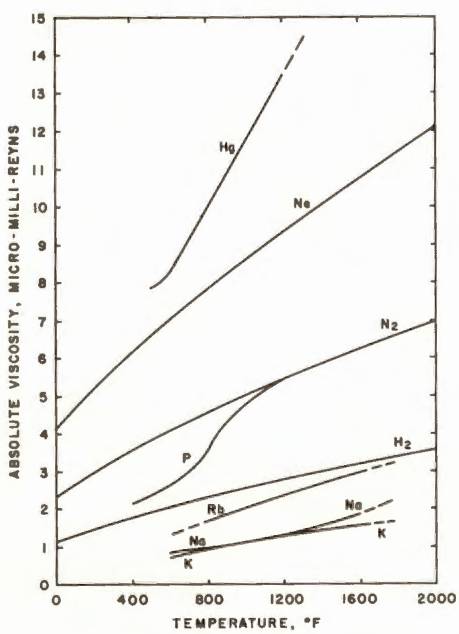

FIGURE 3. ABSOLUTE VISCOSITIES OF SEVERAL SATURATED VAPORS AND REFERENCE GASES FROM 0 TO 2000°F

Compressibility, Density and Other Gas Properties

Conventional hydrodynamic theory breaks down due to the compressibility of gases. Thus, compressibility and density are also of importance in high-temperature gas-lubrication studies.

GENERAL REQUIREMENTS

Lubrication System and Test Rigs

The general requirements of this program have recently been extended to include thrust bearings as well as journal bearings. The lubrication system and test rigs are designed with the following ultimate capabilities:

Ambient temperature, °F	80 to 1800 or higher
Lubricant inlet temperature, °F	80 to 1800 or higher
Radial load, lb	0 to 300
Thrust load, lb	0 to 300
Speed, rpm	0 to 65,000 (DN equivalent 2.5 million)

These are extremely difficult program requirements to be met by any lubricant-bearing system. In addition to operating at the very high speeds and relatively high loads, we are seeking to operate over the entire temperature and speed ranges. Our prime objective is to obtain rigorous and scientific data which we will generalize and which may be used by designers.

Gas Bearings

Pneumodynamic (self-acting) radial and thrust bearings and pneumostatic (externally pressurized) radial and thrust bearings are being investigated.

Three basic radial gas-bearing types were chosen from many considered as being the best suitable for high-temperature evaluation. These types are: (I) fully-choked, orifice-compensated pneumostatic bearing of special design, (II) modified-step pneumostatic bearing, and (III)high-ambient-pressure pneumodynamic bearing. Analyses of the bearings were made to establish rational design criteria for very-high-temperature operation. It was determined that widely different design considerations are applicable to gas bearings which must operate over a wide temperature range, and in addition under a wide range of speeds and loads, as compared to gas bearings designed for a specific operating condition at room temperature. The test bearings are fabricated from Inconel X so as to have the same coefficient of expansion as the test shaft. Various flame plated coatings have been used over the base material to inhibit seizure during contact.

Three different types of thrust gas bearings are being studied for design purposes: (a) pneumostatic step bearing, (b) pneumostatic permeable bearing, and (c) compensated stepped-convergent pneumodynamic bearing. Only those bearings offering the greatest potential for very high-temperature, high-speed operation will be manufactured and evaluated.

TEST EQUIPMENT

Two test rigs are employed in the program. Rig A is for radial load only and consists of the 1.5-inch diameter test bearing mounted in a high-temperature furnace. The test shaft is supported in cantilever fashion by two bearings outside the furnace. Pressurized air drives a turbine mounted directly on the test bearing shaft. Instrumentation to measure speed, load, flow, temperature, temperature-gradients, torque, etc., is provided. There are many special design features of this rig which permit successful operation over the wide temperature range. The test shaft of Rig A is shown in Figure 4. The test shaft after calibration at 1600°F is shown in Figure 5. Test bearings are shown in Figure 6, and the instrument panel is shown in Figure 7.

Rig B is designed to accommodate either thrust bearings to 1.5-inch diameter and/or radial bearings to 1.5-inch diameter x 2 inches long. Pneumodynamic and pneumostatic type thrust and radial bearings may be evaluated in this rig. In addition to thrust and radial loads applied

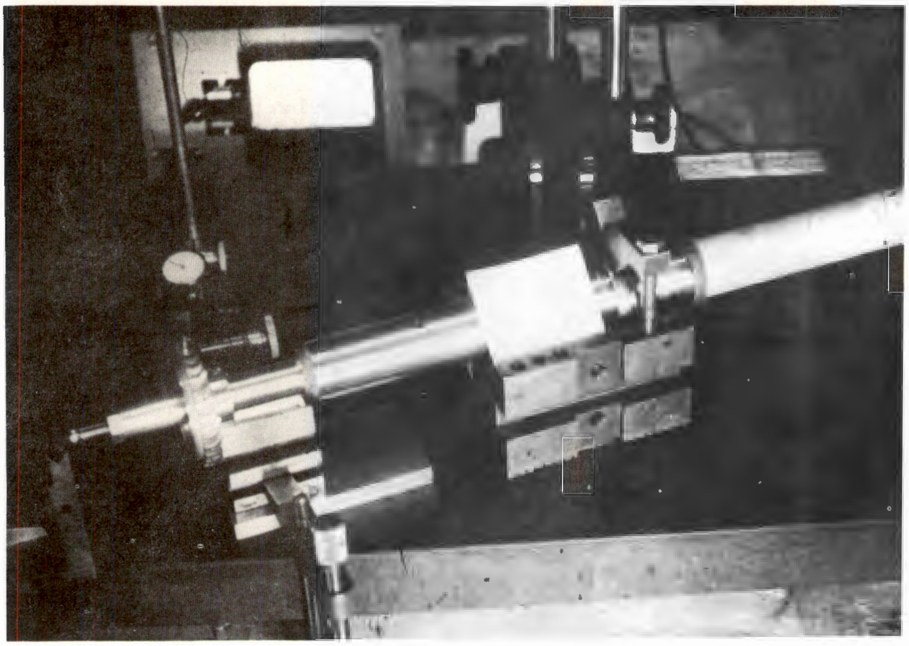

FIGURE 4. SHAFT - RIG A

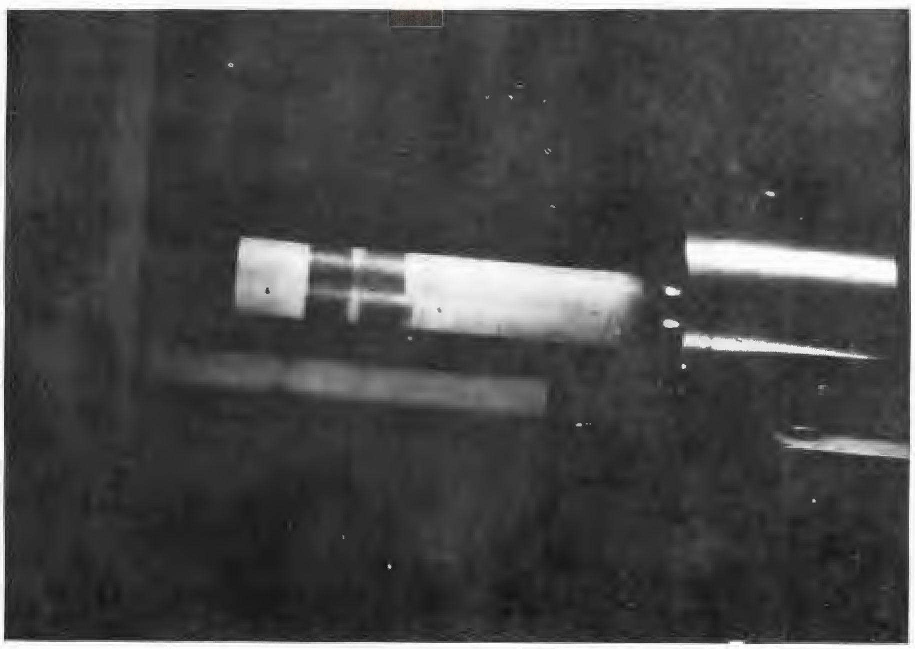

FIGURE 5. SHAFT - RIG A AFTER CALIBRATION AT 1600°F

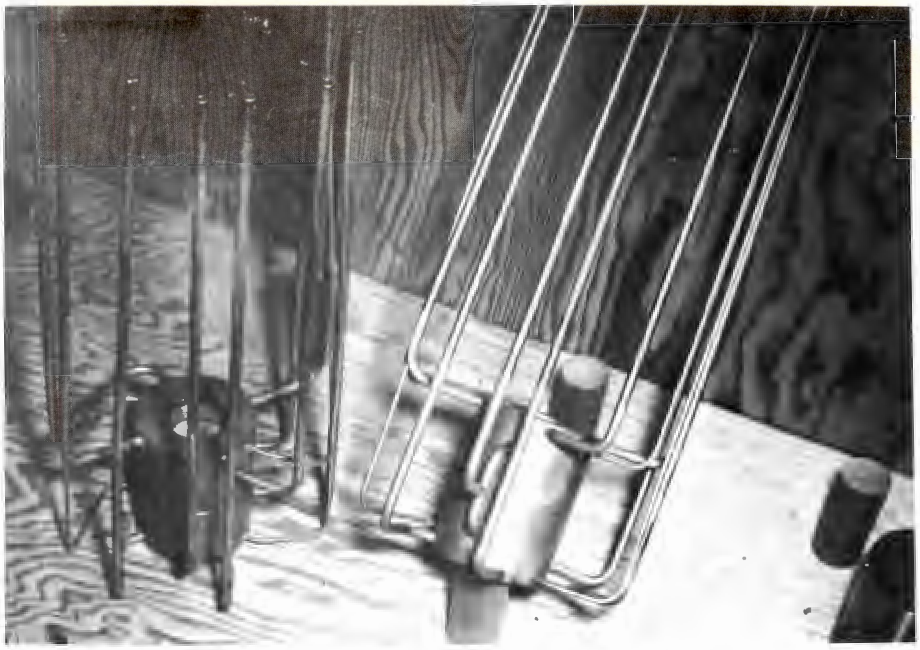

FIGURE 6. TEST BEARINGS - RIG A

FIGURE 7. INSTRUMENT PANEL - RIG A

independently, this rig is capable of utilizing combined loads in any combination simultaneously. An assembled view of Rig B made entirely of Inconel X is shown in Figure 8.

FIGURE 8. ASSEMBLY - RIG B

The lubrication systems supply clean pressurized lubricant gas over a wide range of operating temperatures to both the journal and thrust bearings. Precise flow, temperature and pressure measurements are provided to insure reliable test bearing performance. The lubricant network is constructed entirely of stainless steel tubing with special filtering means including "last chance" filters. The instrument panel for Rig B is shown in Figure 9.

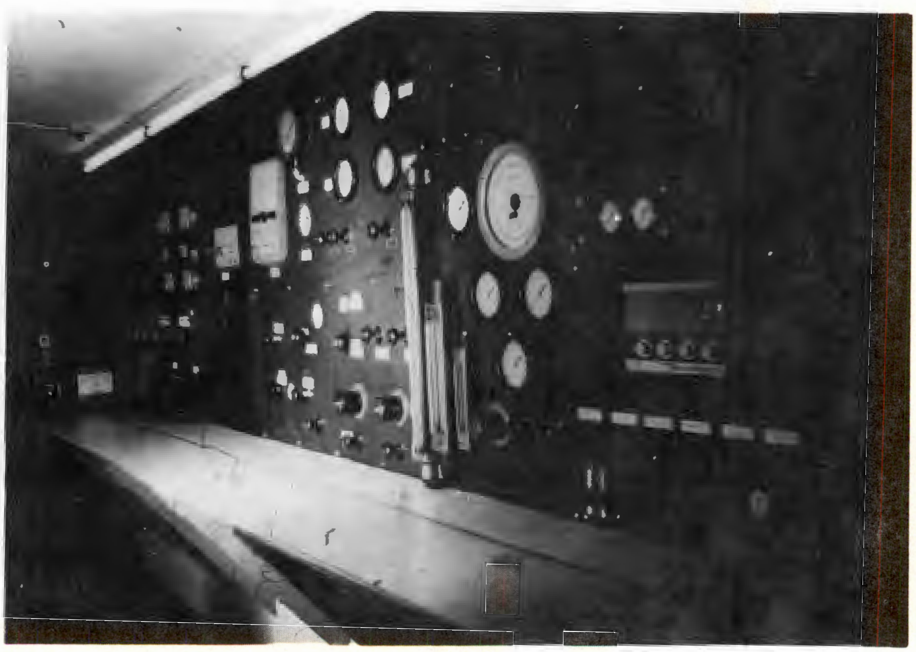

FIGURE 9. INSTRUMENT PANEL - RIG B

OPERATING CONDITIONS ANALYZED AND ACHIEVED

At the start of this very-high-temperature gas lubricant-bearing program, no bearing design or experimental data existed in the temperature range of interest. Thus, it was necessary to conduct bearing analyses over wide temperature ranges in order to design our lubricant and bearing systems, and to decide which basic bearing types to manufacture for experimental evaluation. After careful consideration of the numerous types of radial gas bearings, we chose the step bearing, the orifice-compensated bearing and the pneumodynamic bearing for more detailed studies. These analyses were conducted over the temperature range from 0 to 2000°F and higher, with gas lube flow rate as a prime dependent variable.

The majority of our theoretical and experimental studies to date have been with radial bearings inasmuch as the program has only very recently been extended to include thrust bearings In our analyses and room temperature calibration studies, we have discovered many very interesting and exciting facts about gas bearings which have been most useful in our high-temperature high-speed experimental program. Rather than bore you with long equations, which incidently are quite necessary in these studies, I have chosen a few highlights of our findings to briefly discuss.

To date we have:

(1) Conducted flow orientation studies at speeds from zero to 65,000 rpm with 1.5-inch diameter bearings (equivalent DN value of 2.5 million) over the lube-flow range from approximately 1 to 40 lb/hr.

(2) Operated gas bearings over the temperature range from 80 to 1500°F successfully at flow rates ranging from 1.3 to 30 lb/hr over a wide range of speeds and loads.

Theoretical Flow, Step Bearing (Type II)

The theoretical nitrogen lube flow as a function of temperature for Type II bearing with lubricant inlet pressure as parameter is given in Figure 10. The sharp reduction in flow rate

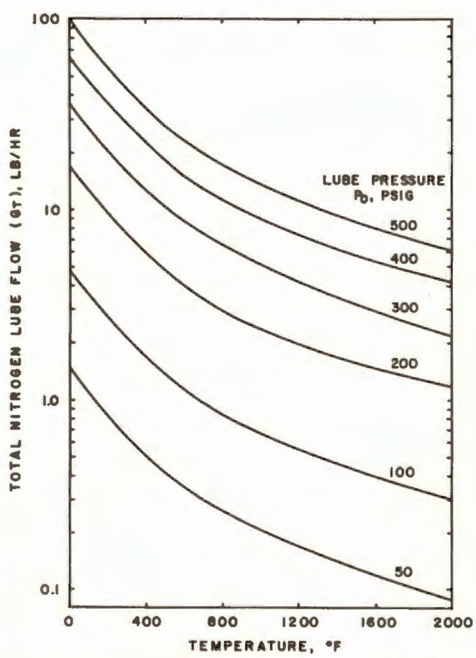

FIGURE 10. TOTAL NITROGEN LUBE FLOW AS A FUNCTION OF TEMPERATURE FOR TYPE II BEARING WITH LUBE INLET PRESSURE AS PARAMETER

caused by an increase in operating temperature is apparent. This curve may be used only as a first approximation to estimate flows required in a design application inasmuch as one-dimensional flow analysis was employed.[1] Corrections must be applied if bearing dimensions other than those indicated are employed.

Theoretical Flow, Orifice-Compensated Bearing (Type I)

The theoretical nitrogen lube flow as a function of temperature for Type I bearing with lubricant inlet pressure as parameter is given in Figure 11 for an orifice diameter of 0.015 in. and a diametral clearance (C_D) of 0.0006 in. Data for a wide range of clearances are given in our quarterly reports. The analysis shows that the flow will be laminar at all temperature and pressure conditions investigated with the exception of 1000 psi and 80°F. Calculations were based, in part, on the method of Reference 2. Flow is given for one row of 8 orifices. Type I bearing may be designed with one or more rows of orifices per bearing. For a bearing with 2 orifice rows, total N_2 lube flow below 3 lb/hr is predicted for a wide range of operating conditions. For example, a total N_2 lube flow of approximately 3 lb/hr may be expected at a lubricant supply pressure of 100 psig at a temperature of 1000°F with a bearing diametral clearance C_D equal to 0.0012 in. The flows at low pressures for C_D = 0.0006 in. are not well defined since the orifice area is really too large for this clearance-pressure condition. The bearing load-capacity under these operating conditions is theoretically very small.

Effect of Speed (Type II Bearing)

The effect of shaft speed is shown in Figure 12 for a double Type II bearing. The flow shown is for each bearing half. This bearing arrangement provides greater axial stability than does a single bearing. It is significant to note an appreciable speed effect over the speed range to 60,000 rpm (2.3×10^6 DN). In certain larger clearance tests to higher speeds, we have experienced discontinuities in these curves which may be due to a transition to turbulent flow.

Optimization of Variables for Type II Bearing

In addition to the other requirements, the entire program is directed toward operation at minimum gas-lubricant flow rates. This makes the high-temperature experimental work even more of a challenge inasmuch as low flow rates are generally associated with low bearing clearance values. Thermal distortions and differential thermal expansions make it very difficult to operate journal gas bearings over wide temperature ranges with clearance values which are normally quite satisfactory for room temperature operation. Thus, we are continually attempting to optimize our bearing designs from a minimum lube flow standpoint. An example is given in Figure 13 which shows the effect of the ratio of step height to clearance on flow with load as parameter. It is seen that an optimum value of δ/C_D exists for each load as indicated by line A-A. This is one curve of an entire series of orientation studies made to optimize the design parameters of Type II bearing from a flow standpoint.

Temperature Effects on Lube Flow

A comparison of the temperature effects on N_2 lube flow for Type I and Type II bearings is given in Figure 14. The flow for Type I bearing (fully-choked) is dependent on temperature as follows:

$$Q = f\left(\frac{1}{T^{1/2}}\right) \tag{1}$$

However, the flow for Type II bearing is dependent on temperature as follows:

$$Q = f\left(\frac{1}{\mu T}\right) \tag{2}$$

Our analyses indicate that for operation over a wide temperature range Type I bearing will operate partially choked. Thus curve C is shown in Figure 14 as representative for operation from 0 to 2400°F. It is seen that curve C lies between curves A and B over the entire temperature range.

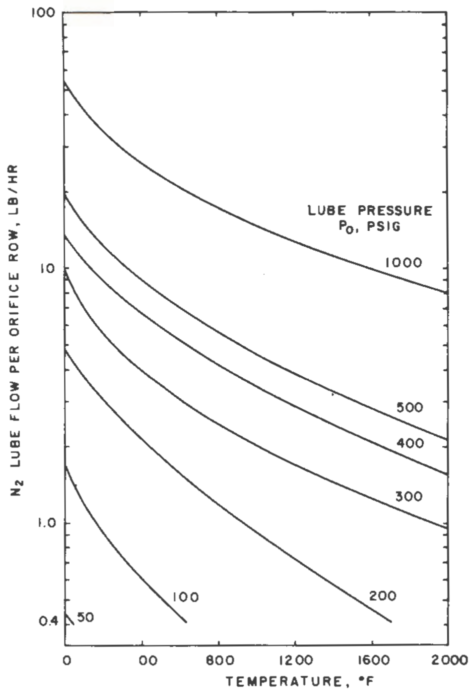

FIGURE 11. NITROGEN LUBE FLOW AS A FUNCTION OF TEMPERATURE WITH LUBE SUPPLY PRESSURE AS PARAMETER FOR TYPE I ORIFICE-COMPENSATED BEARING WITH C_D = 0.0006

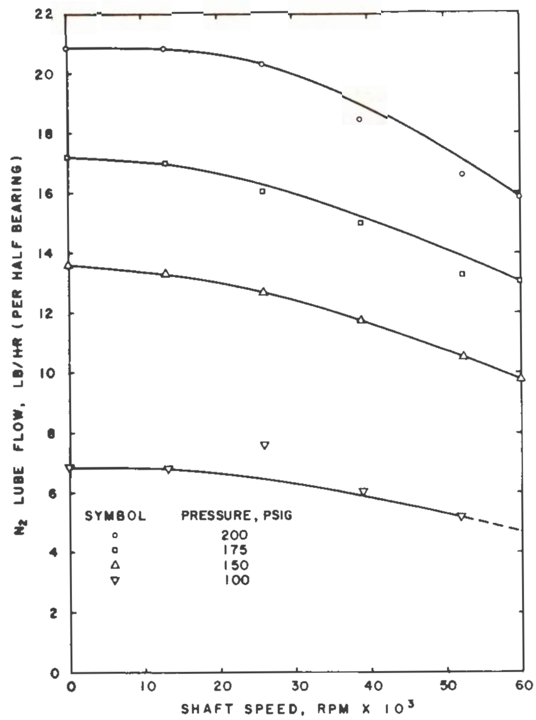

FIGURE 12. LUBRICANT FLOW AS A FUNCTION OF SPEED AT ROOM TEMPERATURE WITH LUBRICANT PRESSURE AS PARAMETER FOR DOUBLE-TYPE II PNEUMOSTATIC BEARING (TEST SERIES 101 B-1)

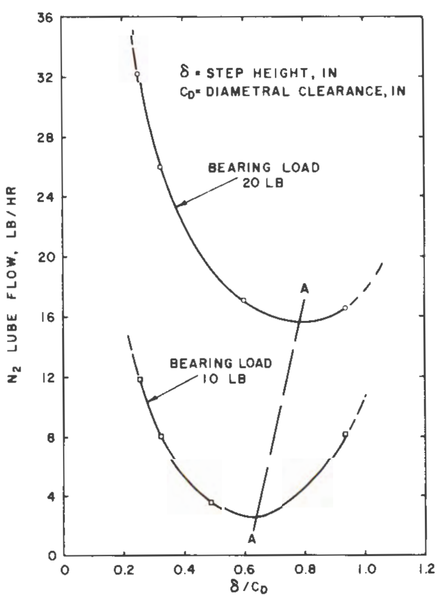

FIGURE 13. EFFECT OF δ/C_D ON NITROGEN LUBE FLOW WITH BEARING LOAD AS PARAMETER FOR PNEUMOSTATIC STEP BEARING

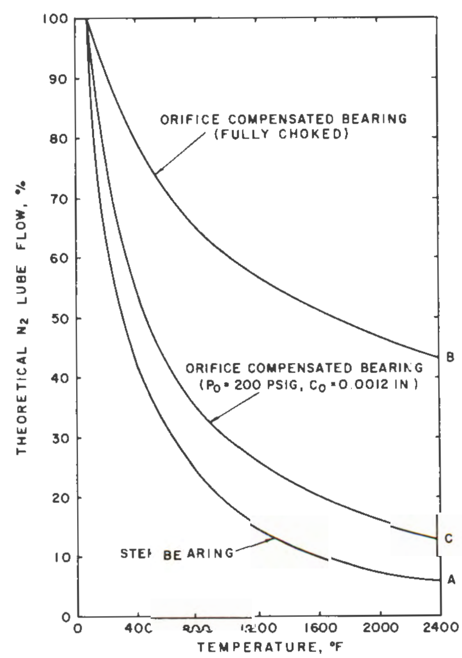

FIGURE 14. THEORETICAL NITROGEN LUBE FLOW FOR TWO BEARING TYPES FROM 0 TO 2400° F

Dynamic High-Temperature Results

Gas-lubricant flow rates as a function of operating temperature with lubricant pressure as a parameter are shown in Figure 15 for a speed of 13,100 rpm and a load of 10 lb. Lubricant pressures vary from 150 up to 225 psig, and flows are seen to range from approximately 3 up to 29 lb/hr. It may be seen that there is a significant reduction in lubricant flow for otherwise identical operating conditions as the temperature is increased over the range from 80 to 1500°F. The reduction in lubricant flow rate with an increase in operating temperature, although appreciable, is not as large as that predicted by theory.

Comparison Between Data and Theory

A N_2 flow comparison between good experimental data, bad experimental data and theoretical results is given in Figure 16 where the flows are all matched at 80°F for comparison purposes. As an example of the deviation between theoretical and experimental flow rates, the flow rates realized in many of our tests have been in the neighborhood of 300 to 500 percent greater at room temperature than at 1500°F, whereas theory predicts flow rates approximately 900 percent greater at room temperature than at 1500°F for Type II bearing (theoretical predictions are based solely on considerations of absolute viscosity and absolute temperature, neglecting speed and turbulence effects). The data of curve B may look fairly close to the theoretical curve A but at 1500°F there is a deviation of about 100 percent. Curve C may appear exaggerated in its location but it is representative of many we obtained before extreme care was taken regarding instrumentation and temperature gradients within the rig during operation.

Cause of Deviation of Experimental Flow Data from Theory

Several factors can contribute to the deviation between experimental flow data and theoretically predicted data, for example, speed and turbulence effects. In addition to the foregoing are the overall effects influencing bearing geometry. In this regard a slight difference in temperature between the bearing and shaft results in an increased or decreased clearance value which is significant regarding lubricant flow inasmuch as theory predicts that flow varies as the cube of the clearance if everything else remains constant. Also, a slight axial temperature gradient can cause a tapered clearance resulting in deviations from theoretical flow. In addition, the very high testing temperature may cause out-of-roundness of the parts which also will result in a deviation from theoretical flow.

Of the several foregoing contributing factors to the deviation between the experimental and theoretical flow, the actual temperature difference between the bearing and the shaft may be one of the most significant. Thus, an analysis has been made for Type II bearing operating at 1500°F with Inconel X as the base material for both the shaft and bearing. The results of the analysis are given in Figure 17 where the ratio of theoretical flow to experimental flow is plotted as a function of the experimental positive mean temperature differential between bearing and shaft. The results shown in Figure 17 are indeed startling in that, for a temperature differential of but 20°F, an error of 400 percent will result at a C_D value of 0.0006 in., 268 percent at a C_D value of 0.0009 in., and over 200 percent at a C_D value of 0.0012 in. The mean temperature differential includes both the actual difference in temperature plus the measurement error.

For a C_D value of 0.0009 in., a total differential from desired test temperature for both the shaft and bearing of but ± 1/2 percent at 1500°F results in a theoretical flow 216 percent greater than the experimental flow. It is difficult, indeed, to consistently maintain temperature gradients and thermocouple calibration within ± 1/2 percent at the high temperatures involved in the subject experimental program. This analysis, therefore, vividly illustrates that the deviation between experimental and theoretical flow experienced to date may well be explained merely by accumulated very small temperature gradients and very small absolute temperature measurement errors -- both of which are well within expected experimental error.

Temperature-Compensated Annular Orifice (TCAO)

Inasmuch as the conventional orifice-compensated bearing (Type I) can be designed for optimum performance at but a single set of operating variables a definite limitation exists for this bearing type for operation over a wide temperature range. However, a modified Type I

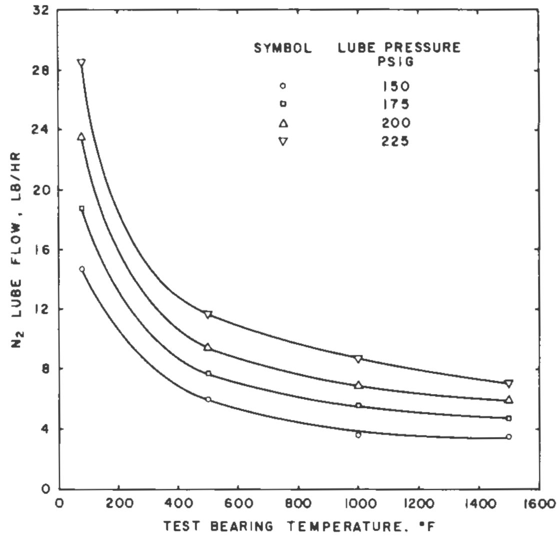

FIGURE 15. LUBRICANT FLOW OVER TEMPERATURE RANGE 80 TO 1500°F FOR TYPE II PNEUMOSTATIC BEARING AT 13,100 RPM AND 10-LB LOAD WITH LUBE PRESSURE AS PARAMETER (TEST SERIES 10 A)

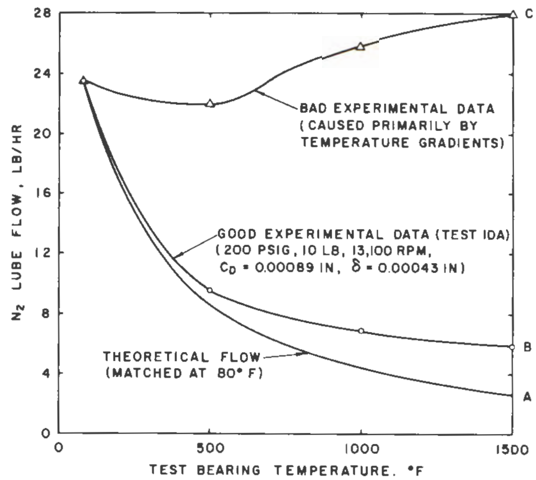

FIGURE 16. NITROGEN LUBE FLOW COMPARISON BETWEEN GOOD EXPERIMENTAL DATA, BAD EXPERIMENTAL DATA AND THEORETICAL RESULTS FOR PNEUMOSTATIC STEP BEARING

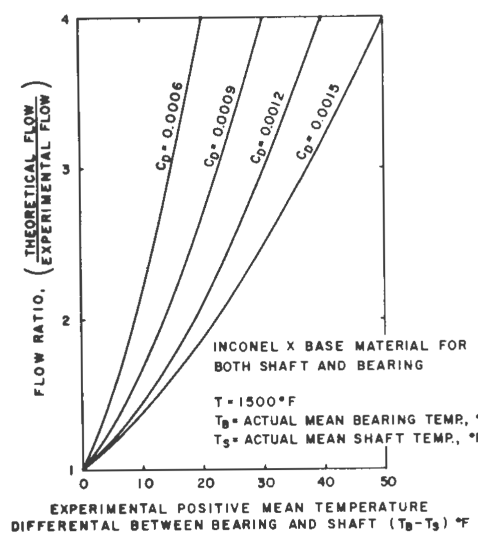

FIGURE 17. FLOW RATIO AS A FUNCTION OF POSITIVE MEAN TEMPERATURE DIFFERENTIAL BETWEEN BEARING AND SHAFT FOR TYPE II PNEUMOSTATIC BEARING WITH C_D AS PARAMETER

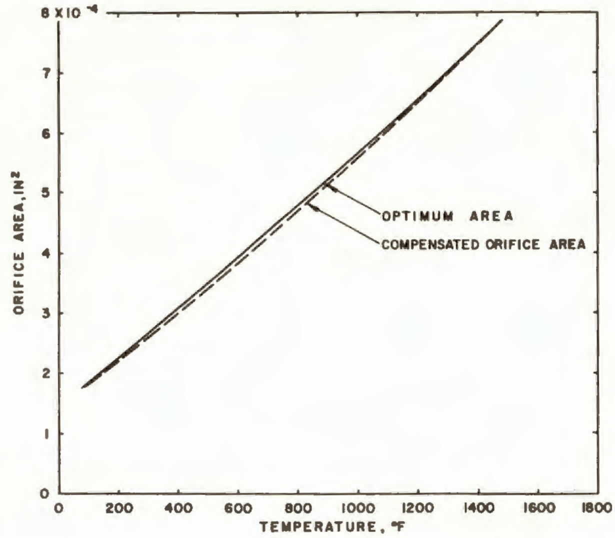

FIGURE 18. COMPARISON OF OPTIMUM-PERFORMANCE ORIFICE AREA WITH TEMPERATURE-COMPENSATED ANNULAR ORIFICE (TCAO) FOR TYPE I PNEUMOSTATIC BEARING OVER TEMPERATURE RANGE 80 TO 1500° F

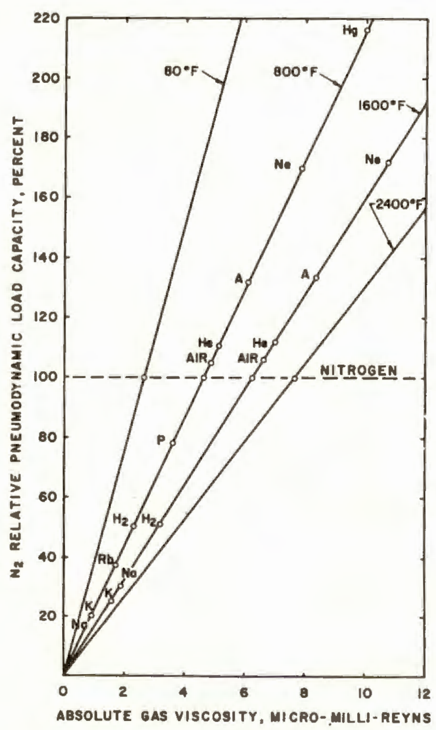

FIGURE 19. EFFECT OF GAS LUBRICANT ON NITROGEN RELATIVE PNEUMODYNAMIC LOAD CAPACITY WITH TEMPERATURE AS PARAMETER

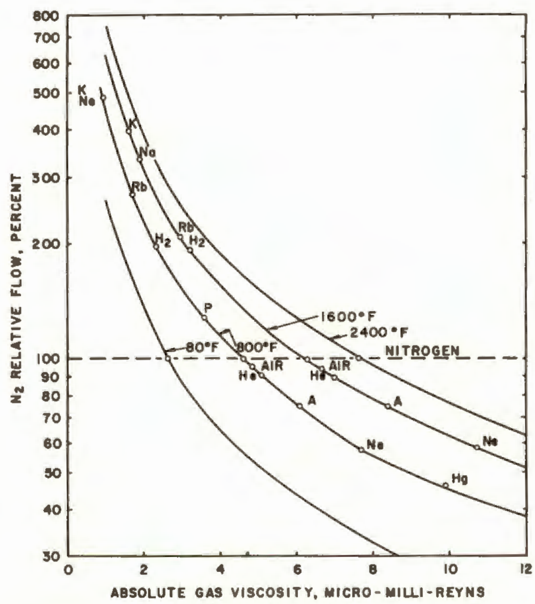

FIGURE 20. EFFECT OF GAS LUBRICANT ON NITROGEN RELATIVE PNEUMOSTATIC FLOW WITH TEMPERATURE AS PARAMETER

bearing (Type IA) with temperature-compensated annular orifices (TCAO) essentially maintains the desired pressure-flow characteristics regardless of temperature. An analysis of Type IA bearing with TCAO indicates that annular orifice materials of different coefficients of thermal expansion may be employed to produce the desired area ratios for a wide temperature operating range. Essentially this design offers a variable area orifice which will have the optimum area at two extreme temperatures. A comparison of optimum performance orifice area with TCAO is shown in Figure 18 from 80 to 1500°F. At temperatures between the two design temperatures the orifice area will not be optimum, but preliminary studies indicate very satisfactory operating characteristics over an entire practical temperature range from 80 up to 1500°F. Further generalized analyses of this bearing type are under way in an attempt to achieve a practical design for operation over the entire temperature range from 80 up to 1500°F or higher.

Effect of Gas Lubricant on Relative Load Capacity

The effect of the gas lubricant on N_2 relative load capacity for pneumodynamic bearings is given in Figure 19 with temperature as parameter from 0 to 2400°F. Inasmuch as our results are primarily with N_2 (with spot checks anticipated for 1 or 2 other selected gases) the designer may use our data and rapidly correct for his particular gas for any temperature. As an example, if his system utilizes H_2 as the working fluid and lubricant for a pneumodynamic bearing at 1600°F, he can determine by a glance that his expected load capacity is 51 percent that of N_2. Gases such as A, Ne, H_2 and saturated vapors such as Na, K, Rb, P, and Hg are included for reference purposes.

Effect of Gas Lubricant on Relative Flow

The effect of gas lubricant on N_2 relative flow for pneumostatic bearings is given in Figure 20 with temperature as parameter from 0 to 2400°F. Here again the designer may quickly determine his expected flow as a percentage of flow of the reference lubricant N_2. For example, the flow at 1600°F for a step bearing lubricated with Rb would be 210 percent the flow with N_2.

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