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The thermal contact resistance of lubricant films in vacuum

Garry A. Heizer

THE THERMAL CONTACT RESISTANCE OF LUBRICANT FILMS IN VACUUM

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H J Sauer Jr (Co-advisors) C.R. Remington

James W. Joiner

ABSTRACT

This paper presents the results of an experimental investigation on the thermal contact resistance of lubricant films and bare interfaces. The apparatus and procedure used in the determination of these resistances are described. Twelve series of tests were performed to evaluate the effects of temperature and contact pressure on the thermal resistance of lubricant films. Variations of thermal contact resistance with temperature and contact pressure for four lubricants and for bare interfaces are presented graphically. The thermal resistances of the four lubricants tested in vacuum conditions: lithium grease, graphite grease, molykote grease, and silicone lubricant, were found to lie in the range from 0.0004 to 0.0035 hr sq ft F/Btu. The thermal resistances of the four lubricants in vacuum were lower than the thermal resistance for bare interfaces in air, and one order of magnitude lower than the thermal resistance of bare interfaces in vacuum. The molykote grease was least affected by temperature.

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INTRODUCTION

For one-dimensional heat flow through a homogeneous solid, Fourier's Law of heat conduction may be used to give accurate results. However, when heat flows through composite materials, temperature gradients occur at the interfaces which cannot be predicted. The interface formed by two surfaces in contact produces an additional resistance to the flow of heat from one surface to the other. Heat is transferred across the interface by conduction through the actual microscopic areas of contact. Heat may also be transferred across the interface by radiation, by convection if a fluid is present in the interface, or by a combination of all three modes of heat transfer.

In numerous types of heat transfer apparatus, heat is conducted through composite walls, and exacting heat transfer calculations must consider the additional contact resistance through the joint at the surfaces in contact. Designs for aircraft, spacecraft, satellites, cryogenic systems, electronic equipment, and nuclear power reactors, all require knowledge of the thermal contact resistances of interfaces. High heat fluxes cause thermal contact resistance to be especially important in metal to metal contacts.

Several theoretical models for thermal contact resistance have been proposed. It has generally been assumed that the actual areas of contact are circular, of the same radius, and that they are evenly distributed in a triangular array. Using this model, Jeng (1) proposed a formula for predicting the thermal contact resistance of two right circular cylinders in direct contact. Tachibana's (2) model assumed that heat was transferred by conduction only, through metallic contacts. Barzelay, Tong, and Holloway (3) concluded that none of the three modes of heat transfer has any predominance over the other, and that all three are interdependent; Fenech and Rohsenow (4) added to the verification of this theory. Clausing and Chao (5) proposed a model which divided the heat transfer area into two regions, contact and noncontact. They neglected film resistance as had been generally done by authors in previous works, but Gale (6) and Tsao and Heimburg (7) showed that surface films can have significant effects on the metal to metal contact resistance. Yovanovich (8) separated the thermal contact resistance problem into three separate problems: thermal, mechanical, and surface description. The results of these three distinct problems were then used to predict the thermal contact resistance. However, because each model is limited in application, and because experimental and theoretical results are often difficult to correlate, one must often depend on experimental data for predicting the thermal resistance of an interface.

The thermal contact resistance of interfaces has been a subject of experimental investigation by many authors for various purposes. Effects of contact pressure, surface roughness, surface flatness, nature of material in the interface, and nature of the materials forming the interface have all been investigated in experiments. Brunot and Buckland (9) were interested in the influence on temperature rating of electrical equipment with laminated metal components. Weills and Ryder (10) were interested in the removal of heat from aircraft engine cylinders. Barzelay, Tong, and Holloway (3) were concerned with the ability of aircraft parts to compensate for localized heating by conducting heat to less adversely affected areas. Hargadon (11) developed data for use in the thermal design of thermoelectric generator hardware. Tsao and Heimburg (7) studied the effects of surface films on thermal contact resistance. Fadler, Sauer, and Remington (12) investigated the effects of various types of adhesives on thermal contact resistance. Gyorog, Smuda, and Fletcher (13) compared the insulating capabilities of various materials under compressive loads.

However, discrepancies in the results from previous works show that experimental measurements for thermal contact resistance are of little value quantitatively, unless the experimental conditions are exactly duplicated. But most investigations do agree qualitatively on the effects produced by various parameters.

Lubrication plays a vital role in our modern and complex society. To estimate the importance of lubrication one need only consider that every moving part of every machine is subject to friction and thus to wear and heat. Throughout the ages, one of man's most persistant problems has centered around the reduction and control of friction, and wear and heat. Friction consumes and wastes energy; it has been estimated that from one-third to one-half of the total energy produced in the world is consumed in overcoming friction. Wear and heat can cause changes in dimensions and eventual breakdown of the machine element and the entire machine and all that depends on it. High temperatures cause a rather rapid deterioration of the lubricant itself, evidenced by chemical breakdown and the formation of harmful acids. Temperatures in excess of 250 ^{o}F can initiate softening of bearing materials.

The sources of heat are the metal to metal contacts of the rubbing surfaces and the lubricant film. The heat generated must be removed in order for the unit to reach some steady-state operating temperature. Thus the lubricant film must be able to effectively dissipate the heat generated. Although much work has been done on lubricants in the area of stress analysis, little has been done in the area of heat transfer. This investigation studied the effects of contact pressure and temperature on the thermal contact resistance of four lubricant films.

TEST PROCEDURE AND APPARATUS

Twelve series of tests were performed to determine the effects of temperature and contact pressure on the thermal resistance of lubricant films. A schematic of the test apparatus is shown in Figure 1. The test specimens were four-inch long, one-inch diameter cylinders of Type 304 Stainless steel. After the interfaces had been turned on a lathe, surface roughness measurements were made using a Type QB Profilometer. Heat was supplied to the top of the upper test cylinder by an electrical resistance heater; input power was controlled by a Variac. Cooling coils located below the lower test cylinder were used as a heat sink. A Lauda/Brinkmann circulator maintained the heat sink temperature.

Each test cylinder contained four thermocouple holes arranged along the longitudinal axis as shown in Figure 2. All thermocouple holes were 0.500 inch deep; one-half the diameter of the cylinder. The thermocouples were dipped in an extremely high thermal conductivity grease, and the thermocouple holes were also filled with this grease. Then the thermocouples were placed in the holes and the excess grease was removed. The thermocouple leads were then wrapped around the test cylinders several times to minimize the error due to conduction along the leads. The same set of iron constantan thermocouples was used

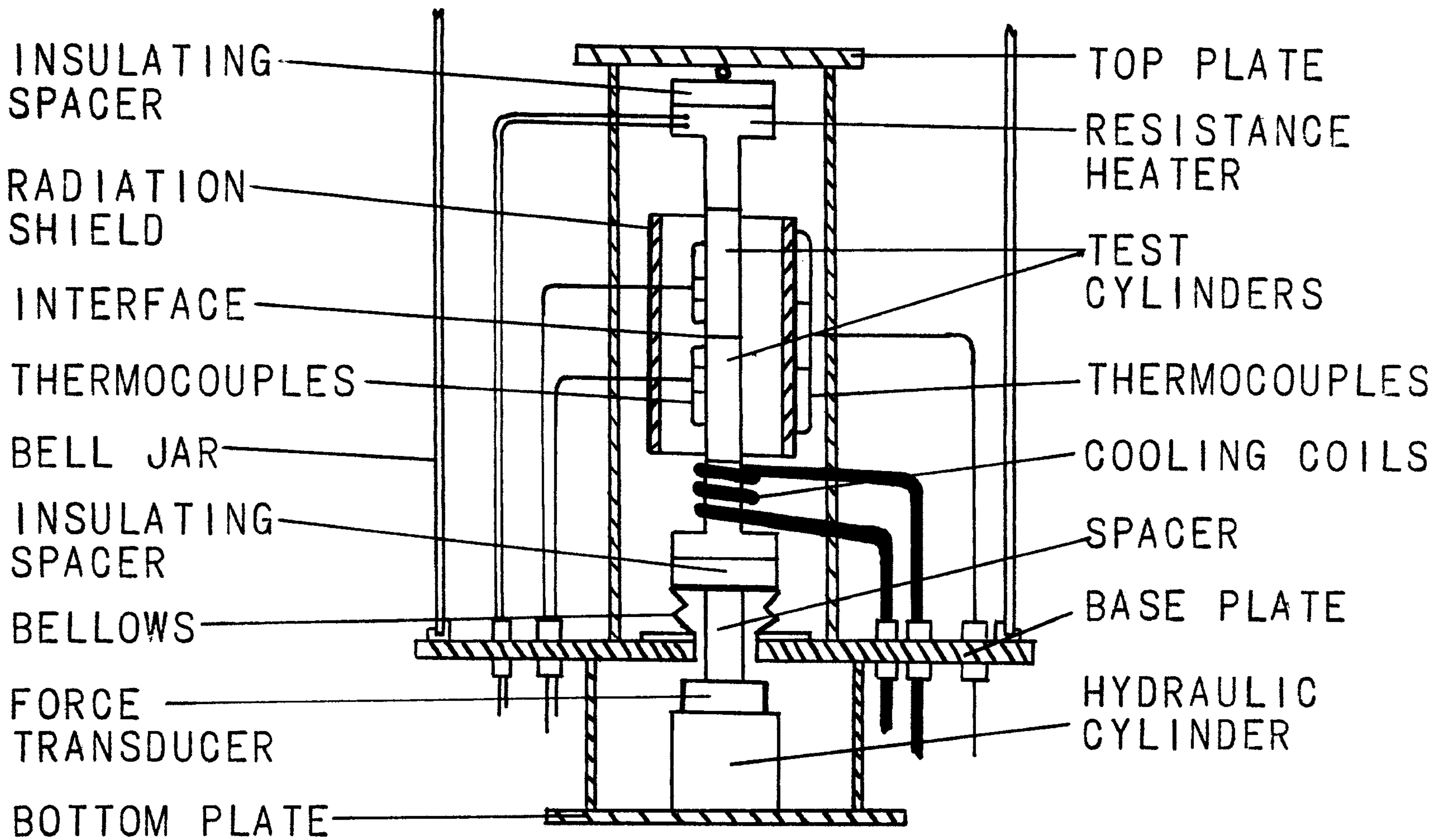

Figure 1

Schematic of Test Apparatus

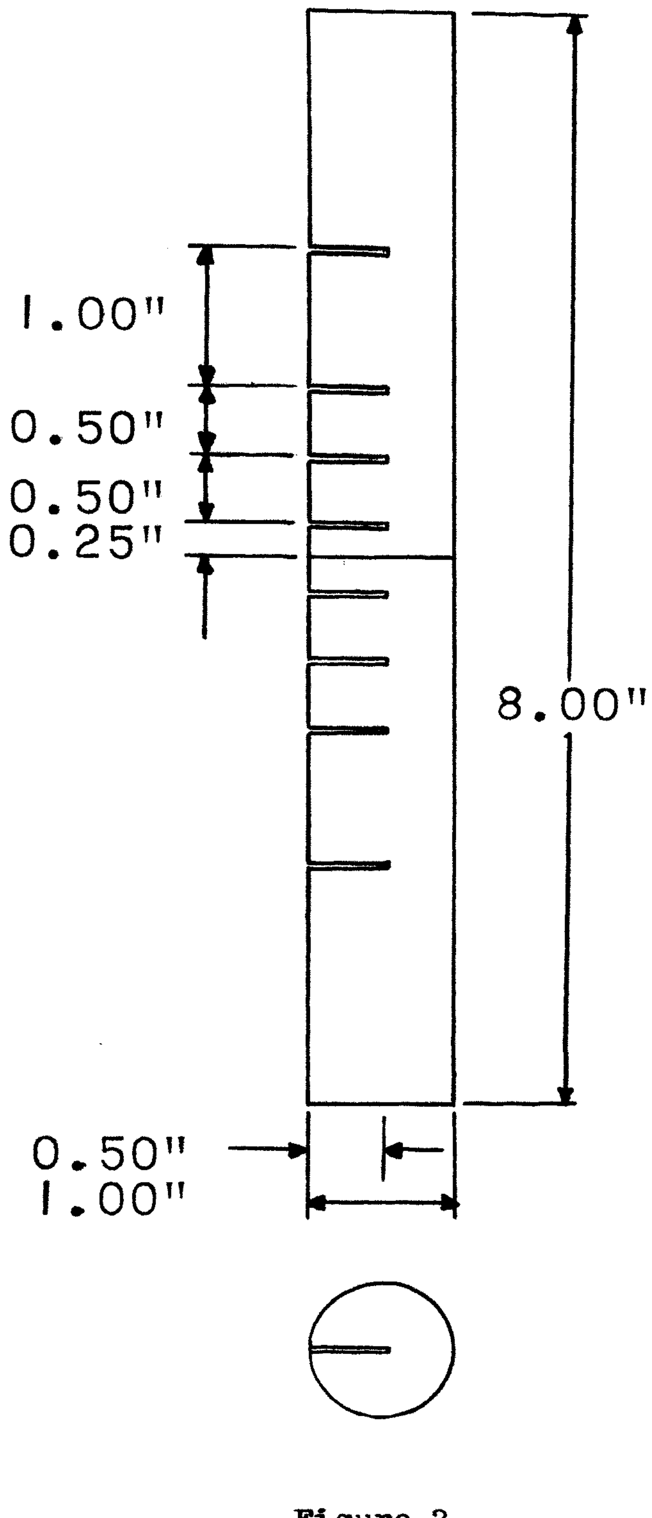

Figure 2

Thermocouple Positions in Test Cylinders throughout the investigation for measurement of temperature gradients and heat flows.

A radiation shield was used to minimize radiation losses from the test cylinders. It consisted of a cylindrical aluminum shell with the inner surface covered with 0.25 inch of insulation. The insulation was then covered with reflective aluminum foil, and the shield was placed so that it was not in contact with the test cylinders, the heat sink, or the heat source. With the vacuum condition removing the convective mode of heat transfer, and the radiation shield minimizing radiative heat losses, one-dimensional, downward, conductive heat flow was obtained.

Preliminary vacuum test runs were conducted to check out the apparatus for proper operation and to determine the time necessary to reach steady-state conditions. Four hours were required for the apparatus to stabilize thermally after initial startup, and one hour was required for stabilization after a normal interface contact pressure change of fifty psi.

In the preliminary vacuum test runs a commercial heat meter, a Hy Cal Sensimeter, was used to check the heat flows determined from the thermocouple readings. The heat flows determined from the thermocouple readings were always within five percent of the heat flows indicated by the heat meter. For a typical test run at a given interface temperature, the contact pressure was increased in increments of 50 psi from 50 psi to 400 psi. Test runs were performed at two interface temperatures, 100 ^{o}F and 200 ^{o}F; and ten of the twelve tests were conducted in a vacuum of 0.5 torr. The other two tests were performed in air at 15 psia for comparative purposes. The interface contact pressure was provided by a hydraulic pump, and was determined by using a force transducer positioned outside of the vacuum environment.

DATA REDUCTION

For steady-state conditions, the thermal contact resistance is expressed as:

$$R = \frac{\Delta T}{Q}$$

where

R = thermal contact resistance, hr sq ft F/ Btu

ΔT = temperature drop across the interface, deg F

Q = heat flow, Btu/ hr sq ft

The temperature drop across the interface was obtained by extrapolating from the temperature profiles measured by the thermocouples. Typical temperature profiles are shown in Figure 3. The heat flows were determined by using Fourier's Law for one-dimensional heat flow:

$$Q = \frac{K \Delta T}{X}$$

where

Q = heat flow, Btu/ hr sq ft

K = thermal conductivity of Type 304 Stainless steel, Btu/ hr ft F

ΔT = axial temperature gradient from the temperature profile, deg F

X = axial length corresponding to the measured T, ft

The average of the heat flows determined from the upper

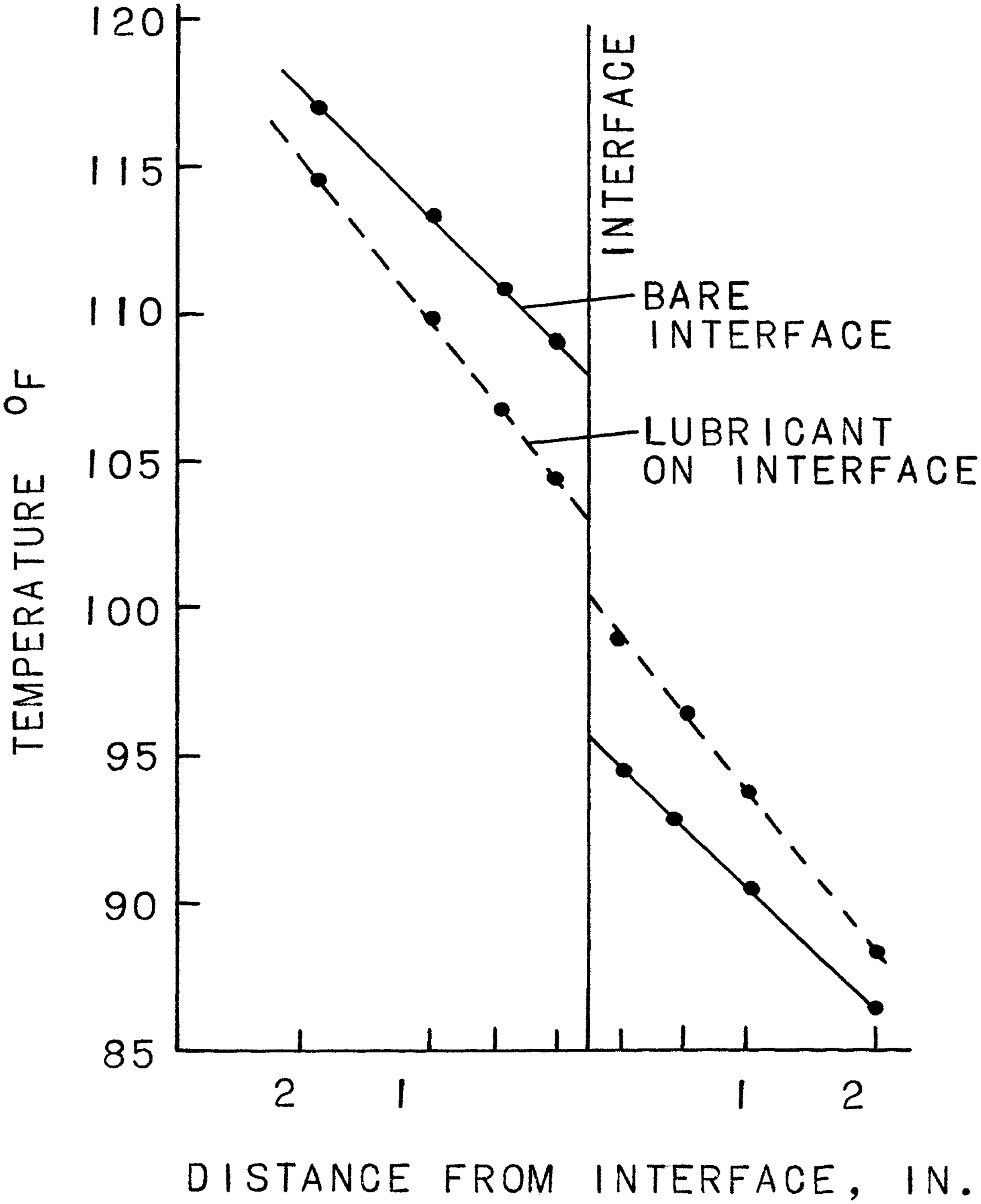

Figure 3

Typical Temperature Profiles

and lower test cylinders was used in determining the thermal resistance. Heat flow calculations for the upper and lower test cylinders compared favorably; within five percent for all test runs conducted in vacuum conditions. For the test runs performed in air, the heat flow determined from the upper test cylinder was as much as sixteen percent greater than the heat flow in the lower test cylinder. This was caused by convective heat losses.

RESULTS

Twelve series of tests were performed; ten were conducted in a vacuum of 0.5 torr. and two were conducted in air at 15 psia. Bare interface tests were conducted both in air and in vacuum. The four lubricants tested were: silicone spray lubricant, molykote grease, lithium grease, and graphite grease. Test cylinders of Type 304 Stainless steel were used throughout the investigation; the test surfaces were cleaned with alcohol and acetone between test runs. The test specimens had a surface roughness of 15-20 microinches rms.

Figures 4 and 5 plot thermal resistance against contact pressure, and the curves show the dependence of the thermal resistance of the lubricant films on contact pressure. As contact pressure is increased, the thermal resistance decreases. This decrease is due to a decrease in the thickness of the lubricant film and also to greater metal-to-metal contact. By comparing Figures 4 and 5, it is evident that the molykote grease was least affected by changes in temperature. The plots for the other three lubricants: silicone spray lubricant, lithium grease, and graphite grease indicate that they are significantly affected by temperature. When the temperature was changed from 100 ^{o}F to 200 ^{o}F, the thermal resistance of the molykote grease remained in the range from 0.0010 to 0.0017

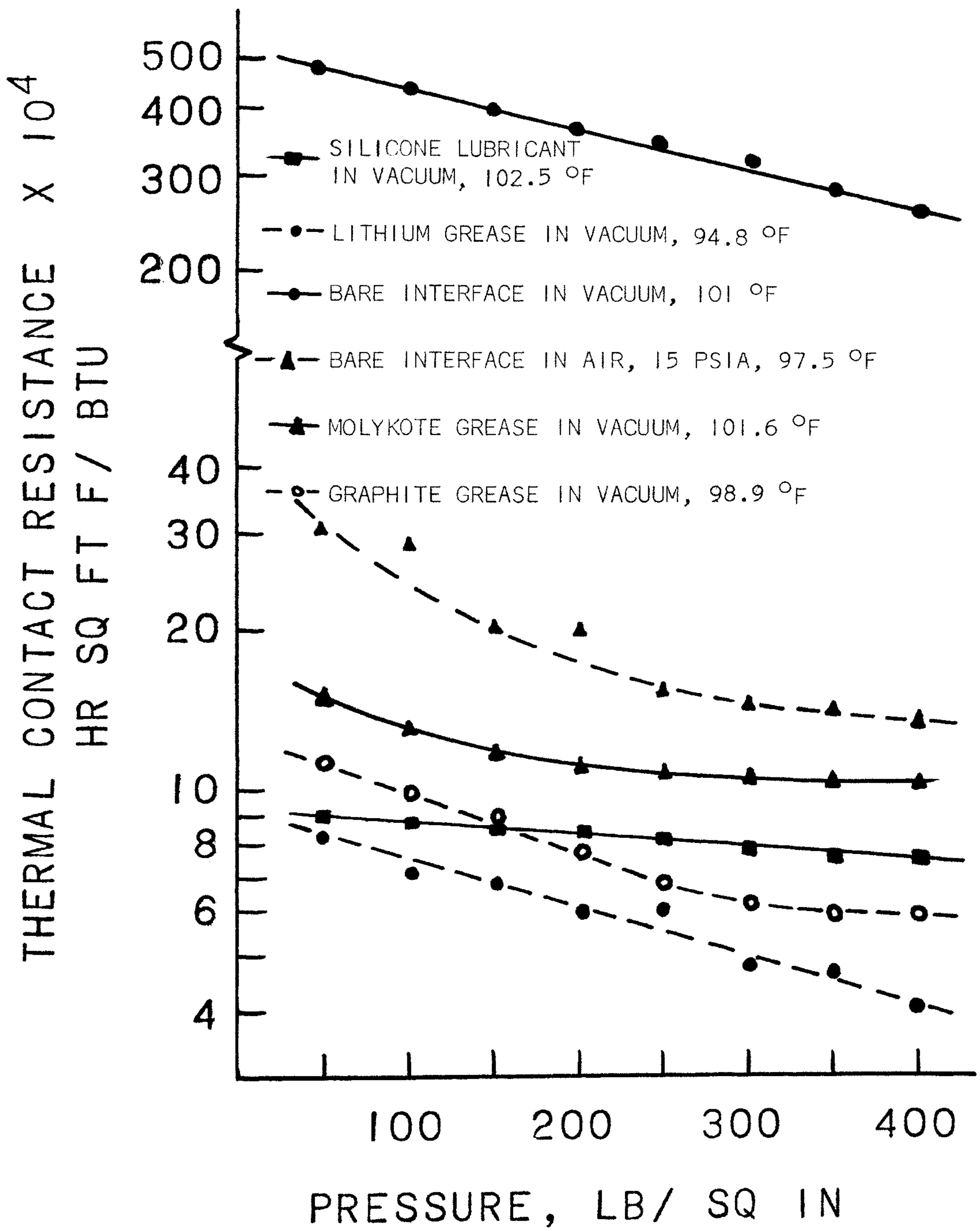

Figure 4

Thermal Resistance of four Lubricant Films at 100 °F

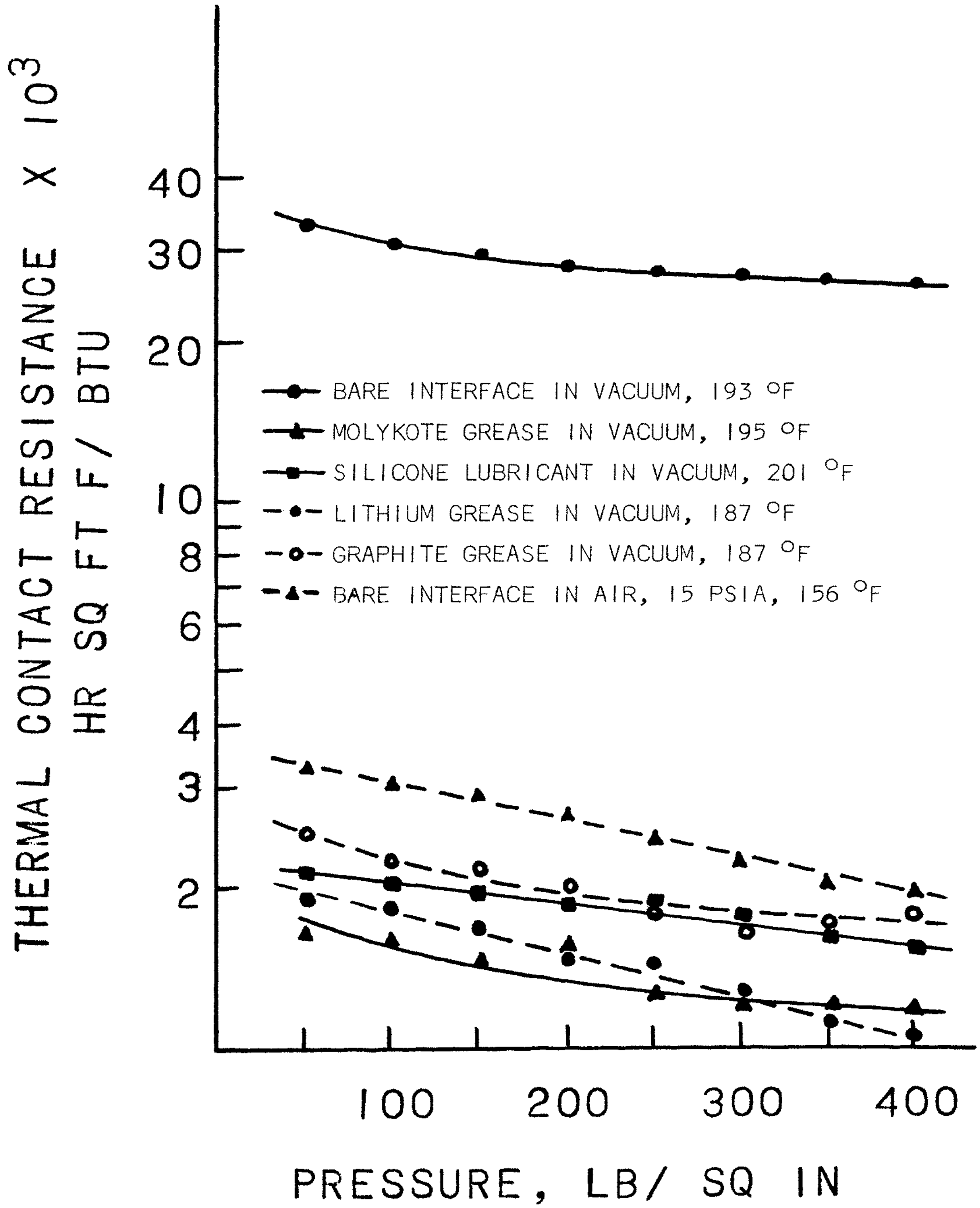

Figure 5

Thermal Resistance of four Lubricant Films at 200 °F hr sq ft F/ Btu, but the thermal resistance of the other three lubricants doubled. Of the lubricants tested, the lithium grease had the lowest thermal resistance. From Figure 6 it is evident that the thermal resistance of bare interfaces is much greater in vacuum than in air. The thermal resistance is higher in vacuum because of the removal of the convective mode of heat transfer. The results of this investigation for bare interfaces and those of Brunot and Buckland (9), Fried (14), and Hargadon (11) agree favorably. The curves in Figure 6 are:

A-this investigation, 0.5 torr., 100 ^{o}F, 20 u inch rms

B-this investigation, 0.5 torr., 193 ^{o}F, 20 u inch rms

C-Fried, (G.E. Report No. 64SD652), 0.1 atmos, 75 ^{o}F, 125 u inch rms, (14)

D-Hargadon, (ASME 66-WA/NE-2), Run 4, 10^{-4}mm Hg., 135 ^{o}F, 50-70 u inch rms, (11)

E-this investigation, air-15 psia, 156 ^{o}F, 20 u inch rms

F-Hargadon, (ASME 66-WA/NE-2), Run 3, argon-15 psia, 250 ^{o}F, 50-70 u inch rms, (11)

G-Brunot and Buckland, (ASME-April 1949), cold rolled steel, air-15 psia, 200 ^{o}F, 125 u inch rms, (9)

H-this investigation, air-15 psia, 100 ^{o}F, 20 u inch rms

The differences in surface roughness, interface temperature,

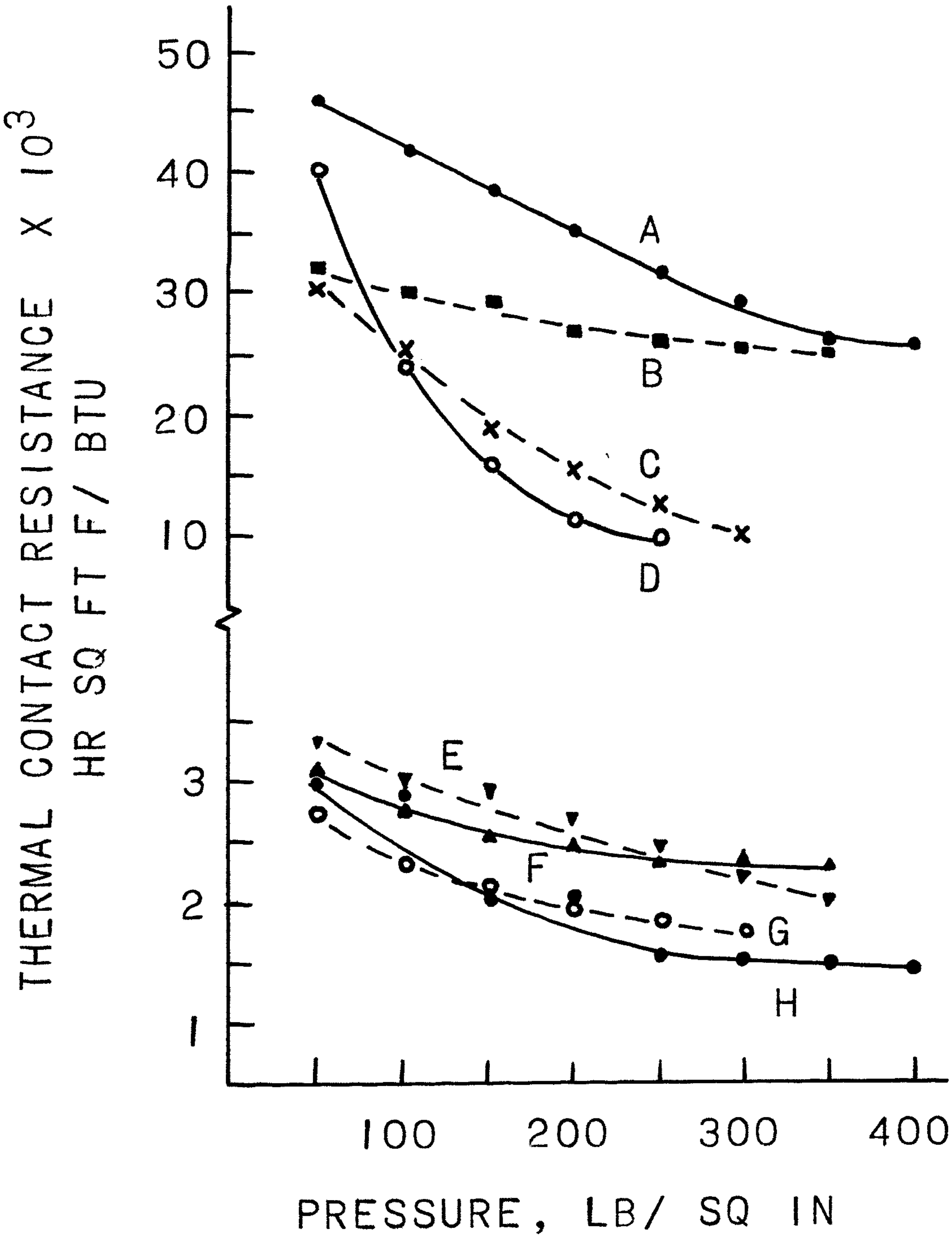

Figure 6

Thermal Resistance of Type 304 Stainless Steel with Bare Interfaces and experimental environment are responsible for the differences in the curves in Figure 6.

The thermal resistances of the four lubricants tested in vacuum: lithium grease, graphite grease, molykote grease, and silicone spray lubricant were found to lie in the range from 0.0004 to 0.0035 hr sq ft F/ Btu. The thermal resistances of the four lubricants in vacuum were lower than the thermal resistance for bare interfaces in air, and one order of magnitude lower than the thermal resistance of bare interfaces in vacuum. The molykote grease was least affected by temperature, and the lithium grease had the lowest thermal resistance.

ESTIMATED ACCURACY

The two quantities that directly contribute to the uncertainty of the interface resistance measurements are the interface temperature difference and the heat flux. The test runs at low temperatures have the largest error due to the comparatively small temperature differences and low heat fluxes. For bare interfaces a heat flux of 500 Btu/ hr sq ft and a temperature difference of 12 °F for a low temperature test run compared to 2500 Btu/ hr sq ft and 57 °F for a high temperature test run. The heat flows determined from the upper and lower test cylinders are in good agreement; for vacuum test runs they are within five percent.

The thermocouples were checked at four temperatures, 32, 80, 138, and 212 °F to determine if there were any discrepancies in the readings. At 32 °F and 212 °F all the thermocouples gave excellent results; the maximum difference in thermocouple readings at these temperatures was 0.10 °F. At 80 °F and 138 °F the maximum difference in thermocouple readings was 0.26 °F and 0.28 °F respectively. In the determination of the heat flows, these discrepancies in temperature readings created very little error. This was due to the relatively large axial temperature gradients in comparison to the discrepancies in the thermocouple readings; 4.0 °F compared to 0.28 °F. The small amount of error incurred in the determination of the heat flows resulted because temperature differences were being used and not absolute temperatures. However, the interface temperature differences for low temperature runs were on the order of 1.0 ^{o}F, and the thermocouple readings could cause error here.

Other sources of error such as conduction along the thermocouple leads and radiation losses to the radiation shield are considered insignificant in comparison to the error in the interface temperature difference for low temperature test runs. Since the error in the heat flows is 5 percent and the maximum error in the interface temperature difference is 25 percent for low temperature test runs and 5 percent for high temperature test runs, the maximum errors in the thermal resistance measurements are 30 percent for the low temperature test runs and 10 percent for the high temperature test runs. Although the quantitative results are of little value to a thermal designer unless the experimental conditions are exactly duplicated, the qualitative results are useful.

CONCLUSIONS

The conclusions reached from this investigation are:

1) The application of a lubricant film on the interface greatly reduces the thermal resistance.

2) The thermal resistance of the lubricant films increased with temperature but decreased with an increase in contact pressure.

3) Of the lubricants tested (silicone spray lubricant, lithium grease, molykote grease, and graphite grease), the lithium grease had the lowest thermal resistance.

4) The molykote grease was least affected by changes in the interface temperature.

5) The thermal resistances of the four lubricants tested in vacuum were found to lie in the range from 0.0004 to 0.0035 hr sq ft F/ Btu.

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VITA

Garry A. Heizer was born on December 8, 1948 in Red Bud, Illinois. He received his primary education in Creve Coeur, Illinois and in Red Bud, Illinois. He has received his college education from the University of Missouri-Rolla, in Rolla, Missouri. He received a Bachelor of Science degree in Mechanical Engineering from the University of Missouri-Rolla in May 1970.

He has been enrolled in the Graduate School of the University of Missouri-Rolla since June 1970, and has held a National Science Foundation Traineeship since that time.