Technical Report

Final Report
F-C1882

REVIEW OF LITERATURE ON THERMAL CONTACT CONDUCTANCE IN A VACUUM

by

S. P. Carfagno

April, 1967

Prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
George C. Marshall Space Flight Center
Huntsville, Alabama 35812

Contract NAS8-20373

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BENJAMIN FRANKLIN PARKWAY • PHILADELPHIA, PENNA. 19103

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FOREWORD

This report is concerned primarily with the subject indicated by the title, a review of literature on thermal contact conductance in a vacuum. After the review was initiated, however, it became clear that a strict interpretation of the subject would impose an unnatural limitation on the scope of the review. In selecting references, therefore, the tendency was to include even those not concerned directly or exclusively with a vacuum environment.

A attempt was made to match the coverage given a reference with its relevance and importance. Some are reported thoroughly. In some other cases, only the nature of their content is indicated. All articles cited have been included in the list of references. All articles bearing on thermal contact conductance which we decided not to review have been listed in the bibliography*, provided they had not already been listed in the extensive bibliography by Atkins [5]**.

*Appendix C.
**Numbers in brackets indicate references listed in Section 9.

ABSTRACT

The literature on thermal contact conductance in a vacuum is reviewed. Following a discussion of the fundamentals of heat transfer across an interface, the results of several theories which ignore surface waviness and one which attempts to account for it are presented. Attempts at data correlations are also described. Surface structure and the deformation of surfaces under load, as they affect the actual contact area and thermal conductance at an interface, are discussed in some detail. The use of interface fillers and the results of studies of heat transfer across bolted and riveted joints are also discussed. References are given to the dominant sources of data in the literature. In addition to a list of references reviewed, this report includes an extensive bibliography of references which did not appear in the 1965 bibliography by H. Atkins.

ACKNOWLEDGMENTS

The author wishes to acknowledge the contributions of William Hauser who conducted the literature search; Alec Peters, Director of the Franklin Institute Science Information Service, who supervised the literature search and helped in reading foreign documents; and G. Peter Wachtell, Principal Scientest, who reviewed this report.

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NOMENCLATURE*

a	Radius of contact area (m).
A	Area (m^2).
A_{ap}	Apparent contact area (m^2).
b	Radius of cylindrical heat channel (m).
B	Brinell hardness (n/m^2).
d	Separation between two parallel walls (m).
d	Flatness deviation (m).
E	Modulus of elasticity (n/m^2).
E_m	Harmonic mean of two moduli, $E_m = 2E_1E_2/(E_1+E_2)$; (ND).
F	Normal load on the interface (n).
$\mathcal{F}$	View factor (see Section 3.3).
g	Constriction alleviation factor (ND).
h	Thermal contact conductance (w/m^2 °K)
H	Hardness (n/m^2).
k	Thermal conductivity (w/m °K).
k_m	Harmonic mean value of the thermal conductivities of two materials in contact; $k_m = 2k_1 k_2/(k_1 + k_2)$; (ND).
L	Length (m).
ΔL_m	Length of contact members having same thermal resistance as the interface resistance (m).
M	Molecular weight (kg/kg-mole).
n	Number of contact points per unit area ($1/m^2$).
N	Number of contact points (ND).
p	Pressure (n/m^2).
p_{ap}	Apparent contact pressure (n/m^2).

*Units in the MKS system are given to help clarify the meaning of the symbols, but they do not necessarily agree with the units to be used in the formulas in which the symbols appear. The designation, ND, means the quantity is non-dimensional.

NOMENCLATURE (CONT.)

q	Heat flow rate per unit area, or thermal flux (w/m^2).
Q	Total heat flow rate (w).
r	Radius (m).
R	Thermal contact resistance per unit area (m^2 °K/w).
R'	Total thermal contact resistance (°K/w).
R_L	Macroscopic thermal constriction resistance (°K/w).
$\mathcal{R}$	Universal gas constant (8317.0 joules/°K kg-mole).
s	Temperature jump distance (m).
S_u	Ultimate strength (n/m^2).
T	Temperature (°K).
ΔT	Temperature drop across an interface between two solids in contact (°K). See Section 3.1.
x	Constriction ratio, $x = a/b$, (ND).
Y	Yield stress (n/m^2).
Y_o	Initial elastic limit (n/m^2).

NOMENCLATURE (CONT.)

GREEK LETTERS

α	Accommodation coefficient (ND).
γ	Ratio of specific heat at constant pressure to specific heat at constant volume (c_p/c_v).
δ_o	Film thickness (m).
δ	Depth of gap (m).
ε	Emissivity (ND).
ξ	Empirical deformation factor (see Section 4.1.1), (ND).
ζ	Elastic conformity modulus (see Section 4.2), (ND).
θ	Mean of the absolute values of the slopes of the surface texture (ND).
ρ	Density (kg/m^3).
σ	Root mean square deviation of surface height from the mean plane (m).
σ	Stefan-Boltzmann constant (5.6697×10^{-8} w/m^2 $°K^4$).

NOMENCLATURE (CONT.)

SUBSCRIPTS

1,2	Surfaces 1 and 2.
ap	Apparent.
f	Fluid.
g	Gas.
L	Macroscopic constriction or contact area.
m	Mean.
M	Metal.
o	Surface film.
r	Roughness.
ref	Reference value.
s	Microscopic constriction or contact area.
t	Total. (Roughness plus waviness.)

1. INTRODUCTION

This report is a review of literature directly or indirectly related to thermal contact conductance in a vacuum.

It is difficult to provide for the dissipation of heat from electronic components and other equipment in vehicles sent into space, where the vacuum environment eliminates convection in and around each component. In its path to heat sinks and radiating surfaces, heat must frequently cross the interface between surfaces in contact - such as bolted and riveted joints. One of the weakest links in the analysis of conduction in a complex structure is the prediction of thermal conductance across such joints.

Joint conductance can have an important influence on temperature distributions and thermal stresses. This influence is discussed in the case of aircraft structures, in Refs. 18c and 51a.

Although one is usually concerned with facilitating the flow of heat, which requires high thermal conductance, there are applications in which high thermal contact resistance is required for insulation. Ref. 99a, for example, includes papers which discuss the use of multilayer insulation for space vehicles using cryogenic fluids.

Although advances have been made in our understanding of some of the many factors which influence thermal contact conductance, the problem is so complex that it is still not possible to predict values of thermal contact resistance for real joints, except for a limited range of conditions. Inadequate awareness and control of conditions affecting contact conductance has sometimes resulted in experimental data applicable for the specific apparatus used in the studies but having little general usefulness. Experiments performed under conditions simulating idealized mathematical models have been more successful in ellucidating the basic phenomena involved. On the other hand, the extent to which ideal models have deviated from actual interfaces has limited their usefulness in predicting values of thermal contact conductance for real joints. More work is required both in studying some of the basic phenomena under ideal conditions and in integrating the results of such studies into models conforming to real joints accurately enough for predictive use. Because of the many factors affecting contact resistance and the inter-relations among them, however, it may not be feasible to eliminate the empirical approach for many situations of practical interest.

2. SUMMARY

Heat transfer across the interface between two solids in contact is not a simple phenomenon. The basic cause of complexity is that the unevenness of real surfaces causes extremely imperfect mating between contacting surfaces, introducing a resistance to heat flow. Even the smoothest surfaces are bumpy on a micro scale, and generally they are not flat. Thus, surface unevenness usually consists of small-scale *roughness* superimposed on large-scale *waviness*. When surfaces are placed in contact they touch at small areas governed by their waviness, and within these small areas real contact occurs at isolated spots governed by the roughness. The actual contact area is a small fraction of the apparent contact area.

Under most conditions the only significant modes of heat transfer across the interface are solid-to-solid conduction at the points of actual contact and conduction through the interstitial fluid. Because of the small gap size, convection is generally negligible; and the temperature level at the interface is usually small enough for radiation to be neglected also.

The fluid conductance can be approximated simply by the ratio of the thermal conductivity of the fluid to the effective gap height. Although for flat surfaces it is possible to relate gap height to the surface roughnesses, there is no general solution to the problem of estimating the effective gap height. If the fluid is a gas at such low pressure that the mean free path of gas molecules is comparable to the gap height, one must take accomodation effects into account. Under such conditions however, the fluid conductance may become small enough to be relatively unimportant by comparison with the solid-to-solid conductance. In a vacuum environment, fluid conductance is absent; heat transfer by radiation might then become more significant if the temperature level at the interface is large, but this is not usually the case.

The thermal resistance associated with conduction through the solid is caused by constriction of the heat flux through the real contact areas. While there is no temperature discontinuity at the points of contact, the temperature difference between points far removed from the interface will include a contribution due to imperfect contact. The problem of computing values of solid-to-solid conductance has not been solved. Several theories have been developed. Some assume that the surfaces are flat and that contact occurs at uniformly distributed spots of equal size. Since such theories are not applicable to non-flat surfaces, which includes most surfaces, attempts have been made to develop macroscopic constriction theories which take waviness into account. In either case, there exists no convenient, reliable method of measuring the geometrical properties of the surfaces to obtain values of surface parameters appearing in the equations.

A few attempts have been made to develop empirical approaches involving correlations of experimental conductance measurements. So far, these methods have proved to have limited usefulness.

Many investigators have attempted to take advantage of the analogy between thermal and electrical conductivities, expressed in the Wiedemann-Franz law, by substituting more easily made electrical measurements for thermal measurements. In part because of the different influences of surface contamination on the two phenomena, however, this approach has proved unsatisfactory.

Although many methods have been tried to achieve surface structure descriptions that are applicable to computations of thermal contact conductance, they have had little success. The aim is usually to predict the real contact area between mating surfaces which - especially in a vacuum environment - is intimately related to thermal contact conductance. One is also interested in a measure of the gap height. Profilometer, graphical, analog, and radiographical methods, in addition to electrical conductivity measurements, have been tried. Reasonable correlations between roughness and various finishing processes are available; but waviness depends on so many parameters that its correlation with finishing processes is generally conceded to be infeasible. The further variability introduced when surfaces are joined together makes it practically impossible to predict the actual contact area for real joints.

Deformations which occur at the interface when two surfaces are pressed together have been studied by many investigators. Most consider that the asperities deform plastically, the local stress at contact spots being equal to the hardness of the softer of the two surfaces in contact. Elastic deformation is involved in the flattening of the macroscopic irregulaties, or waviness.

It has been found experimentally that the size of individual contact areas is approximately constant (radius of equivalent circular area $\approx 30\ \mu$) for a wide variety of materials and contact conditions. As the contact pressure is increased, the real contact area increases mostly by an increase in the number of contact spots. Above approximately 1400 lb/in.^{2}, however, the size of contact spots does increase with increase in contact pressure.

All surfaces have oxide coatings or other contamination, the effect of which on heat transfer is not well understood. However, the effect on metal-to-metal thermal conductance, the dominant mode of heat transfer in a vacuum environment, appears to be unimportant.

The use of interface fillers and surface coatings is effective in increasing thermal contact conductance. Paste-like fillers have usually consisted of silicone grease, sometimes interspersed with a powder of high thermal conductivity. Foils of soft, highly conductive material (indium, for example) are effective when compressed between the surfaces of the materials joined. Similarly, surface coatings of soft materials having high thermal conductivity (gold, silver, copper) are effective. In a vacuum environment, these methods can increase thermal conductance across a joint by an order of magnitude.

In practice, interfaces of the type described occur in bolted and riveted joints. The contact area is dependent on many factors, including plate thickness, material properties, bolt and rivet tensions, and bolt spacing. Experience shows that wide fabrication variability exists even among joints which are meant to be identical. Therefore, the characterization of even carefully controlled joints typical of spacecraft design is difficult. In addition to variations in contact geometry, there may be differences between local physical properties and those of the bulk material because of surface strains resulting from machining and assembly processes. Surface contaminants may also alter the properties controlling heat transfer.

It is possible to estimate the thermal contact conductance across a bolted or riveted joint because there is appreciable contact between the plates only in the immediate vicinity of the fasteners. The procedure that has been used involves computing the contact pressure distribution in the vicinity of the fastener, dividing the area into annular zones, assigning to each zone a value of thermal conductance based on experimental conductance versus pressure data, and adding the contributions of each zone to obtain the total conductance. One difficulty with this procedure is that, because of the high contact pressures that can exist under the fastener, it may be necessary to extrapolate the experimental conductance data to pressures exceeding those at which measurements have been made. The reliability of the procedure has not been established, and some investigators think that reliable results can be obtained only by making measurements on the actual joints of interest.

Heat transfer across the interface between two materials in contact is seen to be a complex phenomenon. In addition to the conditions discussed above, other pertinent parameters include the mean interface temperature, the magnitude of the heat flux across the interface, hysteresis effects and transient effects. Much has been learned from the many theoretical and experimental studies of the problem, but there is still no general procedure for predicting the thermal conductance of real joints.

3. FUNDAMENTALS

3.1 Definition of Thermal Contact Resistance and Conductance

Sufficiently far from the interface between two solids in contact, heat flow may be uniform and unidirectional; but the flow becomes decidedly three-dimensional in the vicinity of the interface. Heat flow *lines* are redistributed so that their density is greater at points of low thermal resistance, which are usually at the contact points. Thermal contact resistance is associated with this region of influence. As a consequence of imperfect contact, the temperature difference between two points far removed from the interface will be greater by an amount, ΔT, when compared to the temperature difference for a perfect contact. In terms of ΔT the thermal contact resistance is defined as

$$R_c = \Delta T / q \quad , \tag{3.1}$$

where q is the heat flow rate per unit area. The interface conductance is

$$h = q/\Delta T = 1/R_c \ . \tag{3.2}$$

The additional temperature drop, ΔT, may be determined by extrapolation to the interface of the temperature gradients in both members in regions where the effect of the interface is negligible.

Factors which cause the temperature gradient in rod-like specimens to vary with distance from the interface and their influence on the determination of the temperature drop across the interface are considered in Ref. 25, (Ch. 3 and Sec 5.4).

3.2 Modes of Heat Transfer at an Interface

Generally, heat transfer across an interface may take place by three modes: thermal radiation, conduction at actual contact areas, and interstitial conduction. Although these three modes of heat transfer are interdependent because the resistance of each path is sensitive to the heat flux through it, the dependence is not strong, and it is usually ignored.

The mean height of the interstitial voids is so small that heat transfer by free convection is insignificant compared to that by conduction. Also, it may be assumed that the heat conducted through the gas flows entirely in the direction perpendicular to the interface because the gap heights are small relative to the distance between contact spots and the thermal conductivity of the interstitial gas is generally small compared to the solid conductivity.

3.3 Radiant Heat Transfer

The heat transferred by radiation (per unit time and area) between two parallel surfaces is given by the expression

$$q = \sigma F (T_1^4 - T_2^4) \qquad (3.3)$$

where

$$F = \frac{\varepsilon_1 \varepsilon_2}{\varepsilon_1 + \varepsilon_2 - \varepsilon_1 \varepsilon_2} = \text{view factor,}$$

σ = Stefan-Boltzmann constant,

ε = thermal emissivity,

T = absolute temperature,

and the subscripts 1 and 2 refer to the two surfaces. If $(T_1 - T_2)/(T_1 + T_2)$ is small, as it usually is, q can be approximated by

$$q = 4\sigma F T_m^3 (T_1 - T_2) \ , \qquad (3.4)$$

where

$$T_m = (T_1 + T_2)/2 \ .$$

The view factor in the above expressions must be modified for radiation between walls of other geometric arrangements (see Ref. 60).

If the coefficient of $(T_1 - T_2)$ in Eq. (3.4) is added to the corresponding coefficient in the fluid conduction equation, Eq. (3.9), and the sum is multiplied by the effective gap height, δ, the result may be regarded as an effective thermal conductivity of the gas, taking account of both conduction and radiation.

Some investigators have measured the heat transferred by radiation by performing experiments in a vacuum with the joint surfaces separated by a short distance. Sommers and Coles [97] made such measurements with stainless steel specimens and obtained a conductance of only 8.54 Btu/hr ft^2 °F. They used a heat flux of 9660 Btu/hr ft^2, which resulted in a temperature drop of 1132 °F across the joint and a mean interface temperature of 736 °F. With copper surfaces in a vacuum at a temperature of -195 °C, Jacobs and Starr [59] found that the thermal conductance remained less than 10^{-3} w/cm^2 °C (2 Btu/hr ft^2 °F) whether the surfaces were "just touching" or separated by a few millimeters. At very low contact pressures, especially in a vacuum, the heat transferred by radiation is an important factor; but it becomes a negligible factor at normal contact pressures, which usually exceed 100 lb/in^2.

3.4 Gaseous Conduction Across Small Gaps and the Effect of Low Pressure

Ignoring edge effects, the heat conducted (per unit time and area) through a gas between parallel walls separated by a distance d is:

$$q = k\,(T_1 - T_2)/d, \tag{3.5}$$

where k is the thermal conductivity of the gas and T_1 and T_2 are the wall temperatures. If the separation is relatively small, it is necessary to take into account the accommodation effect at the walls [64]. This can be done by adding the *temperature jump distances*, s_1 and s_2, to the wall separation

$$q = k(T_1 - T_2)/(d + s_1 + s_2), \tag{3.6}$$

where

$$s_j = \left(\frac{2 - \alpha_j}{\alpha_j}\right)\left(\frac{\gamma - 1}{\gamma + 1}\right)\sqrt{\frac{2\pi M T_g}{R}}\,\frac{k}{p_g}\quad, \tag{3.7}$$

α_j = accomodation coefficient at wall j, j = 1, 2,

γ = ratio of specific heats (c_p/c_v),

M = Molecular weight,

p_g = ambient gas pressure,

T_g = gas temperature,

R = universal gas constant.

The accommodation coefficient is defined as the fractional extent to which those molecules that fall on a surface and are reflected or re-emitted from it have their mean energy *accommodated* toward the value it would have if the returning molecules were in equilibrium with the wall. In terms of temperatures, the accommodation coefficient may be written

$$\alpha = (T_i - T_r)/(T_i - T_w), \tag{3.8}$$

where subscripts i, r, and w stand for the *incident* and *returning* molecules and the *wall*.

Continuum theory ceases to apply when the walls are so close together or the gas pressure so low that the mean free paths of the gas molecules exceed the wall separation and collisions between molecules are rare. One then has free molecule conduction, and d may be neglected by comparison with $s_1 + s_2$, so that

$$q = \frac{\alpha_1 \alpha_2}{(\alpha_1 + \alpha_2 - \alpha_1 \alpha_2)} \left(\frac{\gamma + 1}{\gamma - 1}\right) \sqrt{\frac{R}{8\pi M T_g}}\, p_g\, (T_1 - T_2), \tag{3.9}$$

where T_g is the mean gas temperature. By expressing the accommodation coefficient factor of the above equation in terms of other gas properties [32], it is possible to rewrite q in a form more amenable to computations [41]:

$$q = 5.53 \times 10^{-3} \left(\frac{9\gamma - 5}{\gamma - 1}\right) \frac{p_g}{\sqrt{M\, T_g}} (T_1 - T_2) \text{ (Btu/in}^2\text{ sec)}, \tag{3.10}$$

with p_g in psia and T in °R. Under these conditions the heat conduction is proportional to the gas pressure and independent of the wall separation. It does depend on the shape of the walls, however; and the above equation, which applies to parallel walls, must be modified for other configurations.

Wiedmann and Trumpler [104] reported measured accommodation coefficients for air on various metals, the values of which ranged between 0.87 and 0.97. Values of accommodation coefficients and mean free paths are also given in Refs. 32 and 64.

An example of the use of accommodation coefficients in the analysis of a thermal contact resistance problem may be found in Ref. 36. There, Sanderson's data [88a] on the thermal contact resistance between the solid fuel and its cladding in a nuclear reactor are analyzed by the theory of Fenech and Rohsenow [38] to determine values of accommodation coefficients for argon and helium.

3.5 Effect of Vacuum Environment on Thermal Contact Conductance

At low contact pressures, gaseous conduction is the primary mode of heat transfer at standard atmospheric pressure; but in the low pressure environments of outer space, this mode of heat transfer becomes negligible. It is generally agreed (see Ref. 61, for example) that inter-metallic conduction practically always dominates other modes of heat transfer, including radiation, in a vacuum environment. At the temperatures generally of interest in space vehicles the contribution of radiation is significant only at very low contact pressures. The loss of gaseous conduction causes the thermal contact resistances of joints to increase as much as ten-fold or more [99].

Experiments by Shlykov and Ganin [92] showed that the conductance of interstitial air remains almost unchanged down to ambient pressures approaching 100 mm Hg, then decreases as the pressure is lowered further, becoming negligible when the pressure is of the order of 0.1 mm Hg. Holm [56] also found, for typical joints between metals, that the contribution of conduction by air within the interstices becomes negligible when the pressure is decreased below 0.1 or 0.01 mm Hg. The variation of contact conductance with ambient pressure observed by Stubstad [98] for a copper/brass joint is illustrated in Figure 3.1.

There have been a number of investigations in which thermal contact resistance have been measured both at atmospheric pressure and

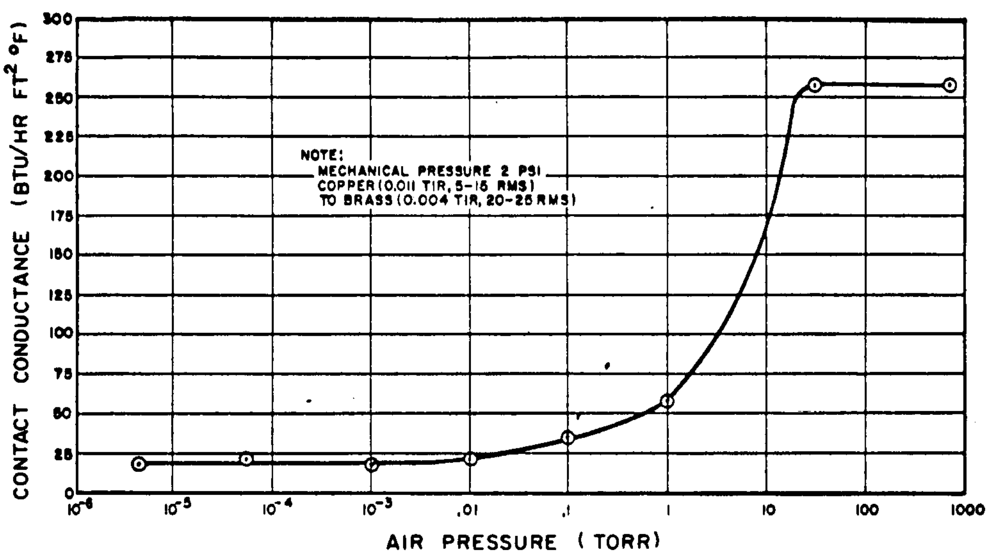

Fig. 3.1 - Dependence of Thermal Contact Conductance on Ambient Pressure. (From Ref. 98.)

(TIR stands for *total indicator reading*, which is a measure of the deviation from flatness.)

in a vacuum environment. Petri [87], for example, measured the thermal contact resistance of a joint between molybdenum and aluminum at atmospheric pressure and in a vacuum environment, at contact pressures up to 1100 lb/in^2 . At the lowest contact pressure (40 lb/in^2) he found that no more than 20 percent of the heat transfer across the joint occurred by conduction through air. At a contact pressure of 140 lb/in^2 the difference between thermal contact conductances measured in vacuum and at atmospheric pressure had decreased to a value comparable to the experimental error; and at higher contact pressures the two measurements became indistinguishable. The values of thermal conductance indicated that the depth of the interface gap was on the order of 100 times the sum of the rms values of roughness of the two surfaces. This was interpreted as due to the predominace of surface waviness over the roughness component. Petri's conclusion that the small difference between measurements at atmospheric pressure and under vacuum permits the use of data obtained at atmospheric pressure for vacuum applications is subject to question. Only one sample joint was used in obtaining the data reported in Ref. 87, and it is risky to generalize on such a basis. Futhermore, the information in Ref. 87 does not make it clear whether the test procedure eliminated the possibility of trapped gases affecting the results. Since the joint was wrapped with asbestos paper, it is possible that insufficient outgassing occurred at the interface. Stubstad [98], for example, questioned whether surfaces placed in contact at atmospheric pressure, even without any wrapping around the exposed edge, ever reaches the environmental pressure in a vacuum chamber. He stated that the pressure in the interstices may be orders of magnitude greater than the pressure measured by the vacuum gage.

Stubstad [99] measured the thermal contact resistance of interfaces between 1/8-in. thick plates of aluminum alloy, stainless steel, and copper, at contact pressures between 2 and 20 lb/in^2, both in air at standard atmospheric pressure and under vacuum. In all cases the thermal contact resistances were greater by a factor of approximately 5 in the vacuum environment.

3.6 Qualitative Influence of Various Factors on Heat Transfer Across a Contact Joint

Heat conduction through areas of actual contact increases as the thermal conductivities of the materials in contact increases. Miller [79] for example, showed this experimentally by measurements on joints between metals whose thermal conductivities ranged over a factor of about 15 between the smallest and largest values.

Experiments with reactor fuel elements [78] have shown that contact heat transfer is also dependent on the effect of radiation on the properties of the contact materials.

Increasing the thermal conductivity of the medium filling the interstices has the effect of increasing the thermal conductance across the joint. A clear demonstration of this effect was obtained by Miller [79] who investigated two steel joints at contact pressures up to 7000 lb/in^2, in a vacuum environment and also in the presence of carbon dioxide, air, and hydrogen at atmospheric pressure. Increasing the thermal conductivity of the gas not only increased the thermal conductance of the joints, but it also reduced the dependence of conductance on contact pressure. A similar effect was observed by Shlykov and Ganin [92] who conducted experiments with helium. They also concluded that conduction through an interstitial fluid is relatively more important for hard metals having low thermal conductivities. For soft metals with relatively high thermal conductivities, metal-to-metal conduction predominates. In either case, as the contact pressure is increased, the metal-to-metal component of conductance increases while the fluid conductance remains relatively unaffected.

Specimen geometry may also have an effect on the measured thermal resistance. Clausing and Chao [25], for example, showed by an analog method that the constriction resistance between cylindrical specimens is dependent upon the length-to-diameter ratio. As this ratio decreases, the constriction resistance also decreases. The effect becomes more pronounced as the fraction of interface area in actual contact is increased, but in any case it becomes negligible when the length exceeds approximately 2/3 the diameter.

The degree of conformity between the opposing surfaces at a joint is a dominant factor in determining the thermal conductance across the joint. Stubstad [98] observed that the value of contact conductance changed by a factor of more than 70 between newly machined surfaces and surfaces *broken-in* by cyclic application of a contact pressure of 10 lb/in^2 until consistent conductance values were measured in a vacuum. Evidently, the breaking-in increased the conformity of the opposing surfaces to each other. At a contact pressure of 2 lb/in^2 [99] he found that changing the orientation of 1/8-in. thick contacting plates caused the measured contact resistance to change by as much as 34 percent.

Increasing the smoothness of the contact surfaces causes an increase in the actual contact area and a decrease in thermal contact resistance. This effect, however, was not found [78] to be very significant for soft metals at high temperatures. From measurements in a vacuum environment on joints of dissimilar metals having surface finishes ranging from 5 to more than 200 μ in., CLA, at contact pressures up to 1000 lb/in^2, Kaspareck and Dailey [62] concluded that surface flatness has more influence than surface roughness on interface thermal conductance.

4. THEORIES OF THERMAL CONTACT CONDUCTANCE AND COMPARISONS WITH EXPERIMENTAL DATA

4.1 Theories Which Neglect Surface Waviness

4.1.1 Outline of Usual Theory

It is now recognized that relatively large-scale *waviness* of surfaces in contact may have greater influence than their small-scale *roughness* on the area of actual contact and, consequently, on the thermal conductance between them. Most of the earlier theories, however, did not consider the influence of waviness. In effect, the surfaces were assumed to be rough, but nominally flat.

At light to moderate contact pressures the contact spots are separated by distances which are orders of magnitude larger than the radius of the contact areas. For nominally flat surfaces under these conditions, considering that the asperity surfaces generally make angles of less than 10 deg. with the mean interface plane, each contact spot may be considered to be a small circular area concentric with a heat channel whose cross-sectional area is a proportionate part of the total interface area. As the contact pressure is increased the actual contact area also increases: first by an increase in the number of contact spots and, at sufficiently high contact pressures, by an increase in their size also. It then becomes necessary to consider the mutual influence of the contact spots [107].

Cetinkale and Fishenden [21] are among those who analyzed thermal contact conductance for the case in which contact spots are of equal size and uniformly distributed throughout the interface. Their equation for thermal conductance includes several constants which are functions of the surface roughness and must be determined experimentally. Their theory is difficult to apply because the surface parameters required are not readily obtained [41]. The practicality of Laming's theory [68], which also assumes uniformly distributed contact spots of equal size, is limited for similar reasons.

An outline of the type of analysis [18, 21, 39, 68] of metal-to-metal conduction which does not take surface waviness into account was presented by Clausing and Chao [27]. The model consists of a number of parallel heat channels as described above and shown in Figure 4.1. The usual assumptions are

a. the contact spots are circular areas of equal size ($r = a$),

b. there are N areas of actual contact, uniformly distributed over the entire apparent contact area (A_{ap}); each contact area is associated with a cylindrical heat flow channel of radius $b = xa$,

c. the asperities deform plastically,

d. the resistance due to surface films is negligible.

The above assumptions lead to the following expression for the interface conductance

$$h = \frac{2\,a\,k_m\,N}{g(x)\,A_{ap}} \quad , \tag{4.1}$$

where g(x) is a constriction alleviation factor given by the series

$$g(x) = 1 - 1.40925\,x + 0.29591\,x^3 + 0.05254\,x^5 + 0.02105\,x^7 + \ldots \tag{4.2}$$

In accordance with assumption (c), the average pressure exerted between asperities by a load, F, equals the microhardness, H. Thus, if a is the radius of the circular contact areas and N is the number of such areas, we have

$$H = \frac{F}{N\pi a^2} \tag{4.3}$$

Hohm [56a] pointed out that the average pressure across the actual contact area will be smaller than H, because part of the contact area will be merely under elastic, instead of plastic strain (see Section 5.3.2). To account for this he suggested that H be multiplied by a factor ξ, which according to experiment has values in the range $1/3 < \xi < 1$. Although ξ is often assumed to be unity, Holm reported values as low as 0.02 for polished surface. Combining the above two equations and introducing the factor ξ, we obtain the following expression for the interface conductance

$$h = \frac{2\, p_{ap} k_m}{\pi \xi H a g(x)} , \tag{4.4}$$

where

$$p_{ap} = F/A_{ap} .$$

Expressions of the form of Eq. (4.4), which assume uniform distribution of contacts over the interface, have sometimes shown good agreement with data obtained with carefully prepared specimens; but they have had little success in predicting values of interface conductance for practical surfaces.

4.1.2 Theory of Fenech and Rohsenow

One of the most thorough analyses of thermal contact conduction under the assumption that contact points are of equal size and evenly distributed has been made by Fenech and Rohsenow [37, 38, 39]. They took account of heat transfer by conduction through the contact points and through a fluid filling the voids. Their idealized model considers contact areas of radius a, distributed in a triangular array. The heat-flow channels of hexagonal cross section associated with each contact were replaced by circular cylinders of radius b. One such

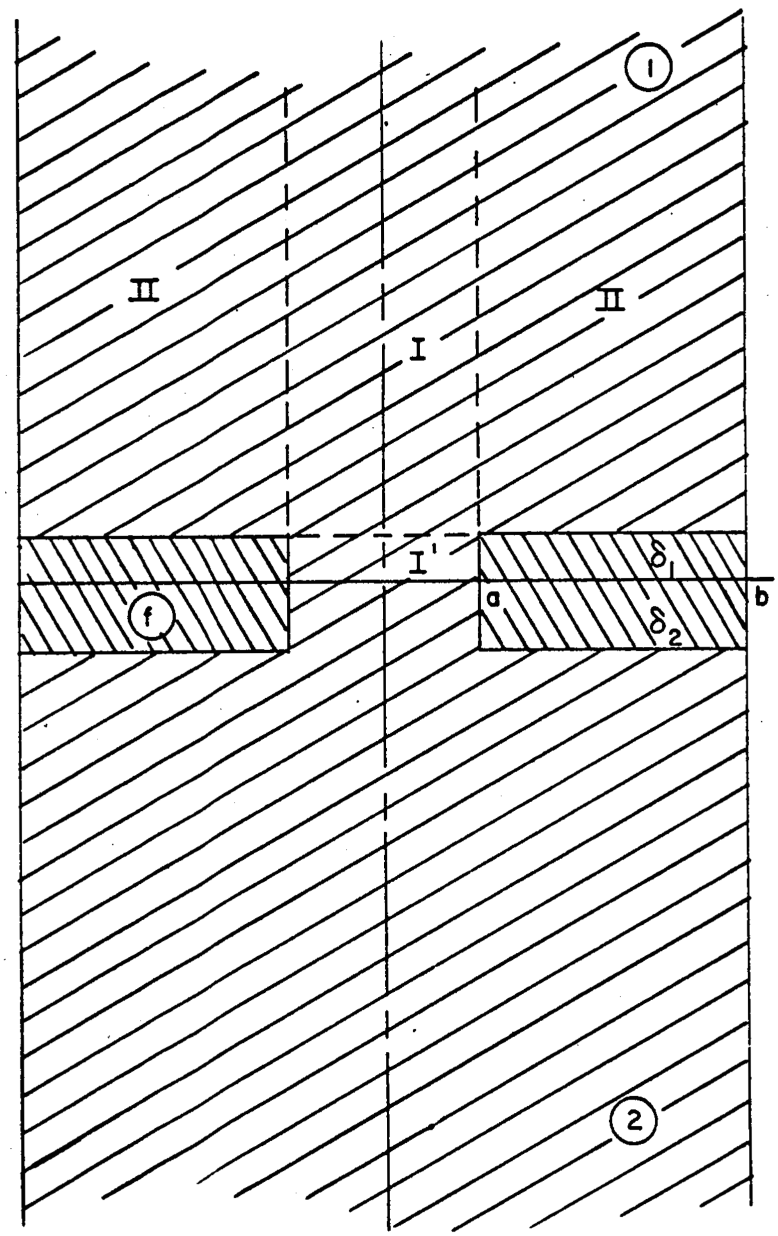

Fig. 4.1 - Model of Cylindrical Heat Channel Associated with a Contact Spot. (From Ref. 81.)

channel is shown in Figure 4.1. Since it was assumed that radial heat transfer could be neglected, no heat is transferred between channels; and the theory for one channel is applicable to the entire surface. The channel was divided into regions I, I', and II (Figure 4.1); and the steady-state heat conduction equation,

$$\nabla^2 T = 0,$$

was solved by neglecting the temperature dependence of the coefficients of thermal conductivity and imposing *average* boundary conditions between regions. The result obtained for the contact conductance is

$$h = \frac{\dfrac{k_f}{\delta_1 + \delta_2}\left[(1 - x^2)\left(\dfrac{4.26\sqrt{n}\,\dfrac{\delta_1}{x} + 1}{k_1} + \dfrac{4.26\sqrt{n}\,\dfrac{\delta_2}{x} + 1}{k_2}\right) + 1.1\left(\dfrac{1}{k_1} + \dfrac{1}{k_2}\right) x\right] + 4.26\sqrt{n}\,x}{(1 - x^2)\left[1 - \dfrac{k_f}{\delta_1 + \delta_2}\left(\dfrac{\delta_1}{k_1} + \dfrac{\delta_2}{k_2}\right)\right]\left[\dfrac{4.26\sqrt{n}\,\dfrac{\delta_1}{x} + 1}{k_2} + \dfrac{4.26\sqrt{n}\,\dfrac{\delta_2}{x} + 1}{k_2}\right]} \qquad (4.5)$$

where

k_1, k_2, k_f = thermal conductivities of the two solids and the interstitial fluid,

$n = 1/\pi a^2$ is the number of contacts per unit area

$x \equiv a/b$,

and the other quantities are defined in Figure 4.1. The second fraction in the above expression ($4.26\sqrt{n}\,x$ in the numerator) represents the heat flow across the metallic contacts; and the first fraction represents the heat flow across the voids. One of the approximations made in arriving at the above expression limits its applicability to $x < 0.1$*.

Simpler expressions for h can be written if certain criteria are satisfied. If the surface structures or the thermal conductivities of the solid materials are sufficiently similar, so that

$$\left| \left(\frac{\delta_1 + \delta_2}{\delta_1 - \delta_2}\right) \left(\frac{k_1 + k_2}{k_1 - k_2}\right) \right| > 4, \tag{4.6}$$

then the thermal conductance can be approximated by the following expression, which agrees with Eq. 4.5 within 5 percent,

$$h = \frac{k_m}{\delta} \left(\frac{1}{1 - \frac{k_f}{k_m}}\right) \left[\frac{k_f}{k_m} + \left(\frac{x^2}{1 - x^2}\right) \left(\frac{1.1\frac{k_f}{k_m} + \eta}{x + \eta}\right)\right] \tag{4.7}$$

where $\delta = \delta_1 + \delta_2$ = effective gap height,
$\eta = 2.13\ \delta\ \sqrt{n}$,

and k_m is the harmonic mean value of thermal conductivity:

$$\frac{1}{k_m} = \frac{1}{2}\left(\frac{1}{k_1} + \frac{1}{k_2}\right). \tag{4.8}$$

When the surface structures or thermal conductivities differ greatly, such that

$$\frac{\delta_1\ k_2}{\delta_2\ k_1} > 5, \tag{4.9}$$

* In the limit of $x \to o$, one expects h to be given by $h = k_f/(\delta_1 + \delta_2)$; yet neither Eq. 4.5, nor the approximate forms, Eqs. 4.7 and 4.10, reduce to this expression when x = o. Possibly, this is a consequence of the application of "average" boundary conditions (between the regions shown in Figure 4.1) in the derivation of Eq. 4.5.

Eq. 4.5 can be approximated by

$$h = \frac{k_m}{\delta}\left(\frac{1}{1 - \frac{k_f}{k_1}}\right)\left[\frac{k_f}{k_m} + \left(\frac{x^2}{1 - x^2}\right)\left(\frac{1.1\frac{k_f}{k_m} + \eta_1}{\eta_1 \frac{k_m}{k_1} + x}\right)\right] , \tag{4.10}$$

where $\eta_1 = 2.13\ \delta_1 \sqrt{n}$. (4.11)

When operating in a vacuum environment or when the contact pressure is very high, heat conduction through the voids can be neglected. Setting $k_f = 0$ in Eq. 4.5, we obtain

$$h = \frac{\frac{x^2}{1 - x^2}}{\frac{\delta_1}{k_1} + \frac{\delta_2}{k_2} + \frac{0.469\left(\frac{x^2}{n}\right)^{1/2}}{k_m}} . \tag{4.12}$$

An approximate criterion for the validity of making the above approximation is [54]

$$k_f/k_m \lesssim x^2/2 .$$

In order to evaluate Eqs. 4.7, 4.10, or 4.12 it is necessary to know the surface parameters δ_1, δ_2, n, and x. A graphical method of determining these parameters is described in Refs. 38 and 39, and an analog computer technique is described in Ref. 54. While the graphical method can be learned by a competent draftsman, it is tedious and time-consuming. The analog method, though much less time-consuming, requires the assembly of special instrumentation.

The values of the surface parameters are dependent upon the contact pressure. In Refs. 38, 39, and 54, each set of parameters was associated with an apparent contact pressure by assuming that the peaks of the softer surface deform plastically. This was based on Moore's observation [82] that asperities of the softer of two metal surfaces pressed together undergo full plastic deformation while the peaks of the harder metal become embedded in the softer surface. From this it follows that the apparent contact pressure is

$$p_{ap} = H x^2 , \tag{4.13}$$

where H is the yield pressure of the softer metal. The value of H can be measured with a microhardness test, such as the Vickers or Knoop tests. Since measured values of H depend on the size of the indentation, one should use a value corresponding to an indentation the size of which equals the average contact area x^2/n. When the value of H is not available, it can be approximated by 3 times the yield stress [39].

$$H = 3Y. \tag{4.14}$$

This deformation theory yields the relation of contact conductance to pressure only for the initial loading of a joint. In the presence of hysteresis due to cycling loading, it would be necessary to consider elastic deformation in addition to plastic deformation.

A modification of the deformation model applicable to crystalline materials (specifically, graphite) is considered in Ref. 37.

Experimental verification of the above theoretical model and the methdds of determining surface parameters was obtained [38,54] under a few conditions involving essentially flat surfaces. Figure 4.2 shows the good agreement between theory and experiment for a contact between the end faces of a rod of Armco iron and a rod of aluminum. The procedure can be extended to wavy surfaces; but, aside from scant references, no information on such applications was found in the literature.

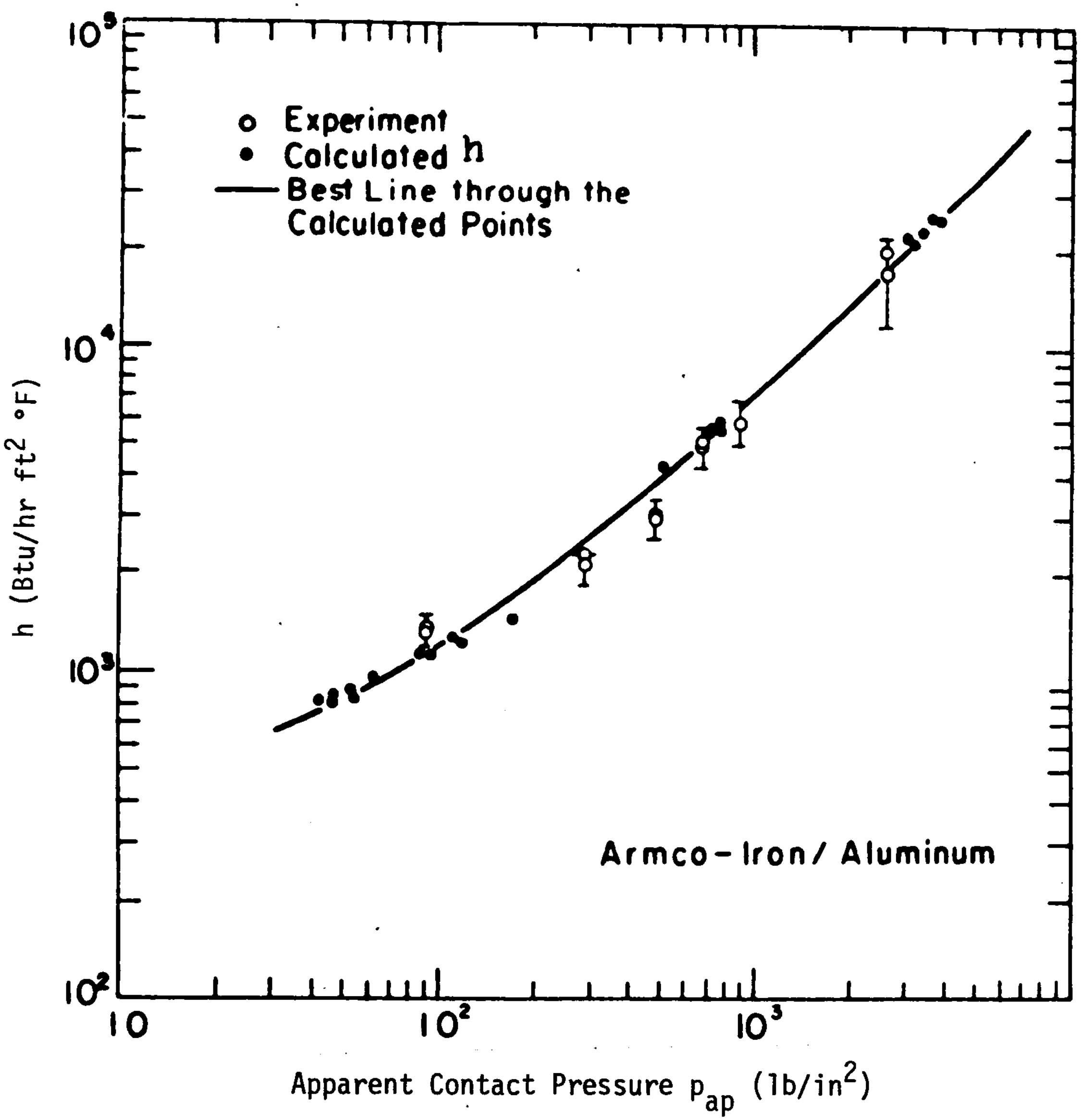

Fig. 4.2 - Comparison of Contact Conductance Theory with Experiment According to Ref. 38.

4.1.3 Theory of Mikic and Rohsenow

Based on an analysis which assumes that each contact point consists of two hemispherical asperities in symmetric contact, Mikic and Rohsenow [76] found that the actual contact area may be approximated by the ratio of load to microhardness for non-wavy metallic surfaces under apparent contact pressures between 130 lb/in^2 and 15,000 lb/in^2. The thermal contact conductance for rough, nominally flat surfaces having uniformly distributed contact spots is then given by the expression

$$\frac{\sigma h}{k_m \tan \theta} = 0.9\left(\frac{P_{ap}}{H}\right)^{16/17} \tag{4.15}$$

where σ is the root mean square deviation of surface height (measured from the mean plane) and θ is the mean of the absolute values of the slopes of the surface structure. Consideration of the symmetric model showed that approximating the actual contact area by F/H yields somewhat low values at low pressures; and, because of work hardening of the materials in contact, high values of contact area are predicted at very high pressures if the same value of hardness is used throughout. Consequently, the values of thermal contact conductance predicted by Eq. 4.15 might be expected to be somewhat low at low pressures and too high at very high pressures. Nonetheless, as shown in Fig. 4.3, experimental data for nominally flat stainless steel specimens having various roughnesses (σ = 42 μ in. to 340 μ in.) showed good agreement with Eq. 4.15.

Especially at low pressures, the contact points might not be uniformly distributed, and the influence on conductance would be the same as that of an equivalent surface waviness. An analysis of the effect of waviness showed that it is relatively significant, especially at low pressures, with a tendency to diminish in importance as the pressure is increased.

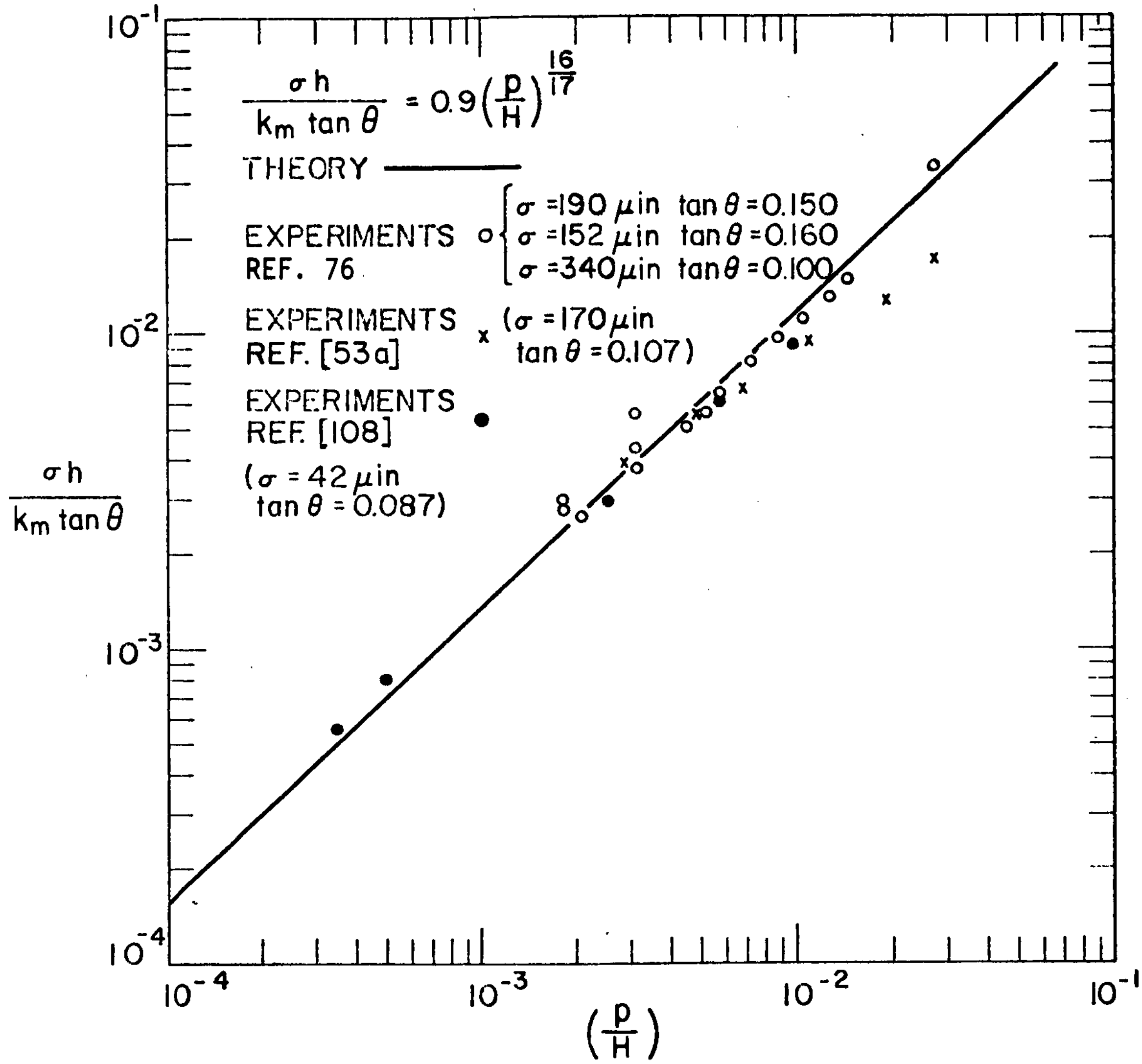

Fig. 4.3 - Comparison of Contact Conductance Theory with Experiment According to Ref. 76.

4.1.4 Theory of Shlykov and Ganin

Shykov and Ganin [92] developed a reasonably simple procedure for computing thermal contact resistance which, although it does not take surface waviness into account, had good agreement with experimental data obtained both in air and in a vacuum.* It was assumed that heat is transferred across an interface only by conduction through areas of actual contact and through the medium filling the interstices, the latter contribution being neglected for vacuum environments. The thermal resistance of the interstitial fluid was approximated by the expression

$$R_f = \delta_f / k_f \quad , \qquad \text{(See footnote **)} \tag{4.16}$$

where

$\delta_f \simeq (\delta_{r1} + \delta_{r2})/2$, the average height of the fluid layer,

δ_{r1}, δ_{r2} = average heights of microroughnesses on the two surfaces,

k_f = thermal conductivity of the fluid.

The expression derived for the thermal resistance of the metallic contacts is

$$R_M = (\pi \; r \; A_{ap}) \; / \; (2 \; k_m \; A), \tag{4.17}$$

where

A = actual contact area,

A_{ap} = apparent contact area,

r = radius of contact spots,

and

$k_m = 2k_1k_2/(k_1 + k_2)$ = reduced thermal conductivity of metals 1 and 2 joined at the interface.

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* See also Refs. 90, 91, and 93.

** Ref. 41 suggests that a good approximation of δ_f for air is

$$\delta_f = 0.64 \; (\delta_{1t} + \delta_{2t})$$

where δ_{1t}, δ_{2t} are the rms values of irregularity (roughness plus waviness) for the two surfaces. It is indicated that the factor 0.64 has different values for other gases. Note that surface waviness is taken into account by this definition of δ, but it is necessary that the waviness not exceed the capabilities of the instruments available for measuring δ_t.

In deriving the above expression it was assumed that the actual contact occurs at circular areas of equal size, uniformly distributed over the interface. Based on data in Ref. 18, the radius of the circular areas was assumed to be 3×10^{-5} m. To obtain an estimate of the actual contact area, the stress developed over the individual contact areas was assumed to be that corresponding to plastic deformation and was set equal to 3 times the ultimate strength of the softer material.* Thus

$$A = F/3S_u \tag{4.18}$$

where F = normal load on the interface,

and S_u = ultimate strength.

Since the resitances of metallic contacts and the interstitial fluid are in parallel, the total resistance, R, is related to them as follows

$$\frac{1}{R} = \frac{1}{R_M} + \frac{1}{R_f} . \tag{4.19}$$

Shlykov and Ganin [92] compared computations according to the above theory with experimental measurements on steel, stainless steel, duralumin, and copper, at contact pressures up to 7000 lb/in^2. Most of the measurements were made in air at atmospheric pressure, but some were made in a vacuum environment. On the whole, the agreement between theory and experiment was quite good. The measurements showed that increase in contact pressure causes little change in the thermal resistance contributed by the interstitial fluid and that most of the decrease in overall resistance caused by increasing contact pressure is due to an increase in the actual contact area. An interesting set of experiments on a steel joint in a helium environment showed that the fluid conductance was dominant in this case; and the total contact resistance was therefore not only lower than it is in air, but also practically independent of contact pressure.

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* For metals having a high degree of cold work, such as copper, the factor 3 should be replaced by 5 according to Ref. 92.

4.2 Macroscopic Constriction Theory

Clausing and Chao [27] point out that uniform distribution of actual contact areas would exist only if the surfaces were ideal mating pairs. Since such ideal contacts do not exist in practice, large-scale constrictions to heat flow are present and often dominate the thermal contact resistance. To account for this effect, they propose a model that divides the apparent contact area into a *contact region*, where the density of microscopic contact areas is high, and a *noncontact* region which contains few or no microcontacts. The model, as applied to cylinders having end surfaces which are sectors of spheres, is illustrated in Figure 4.4. Actual contact occurs only at *microscopic* contact areas of radius a_s, uniformly distributed within the central macroscopic contact region of radius a_L. Since the heat is first constricted to flow within the macroscopic contact area and then to the microscopic contacts and finally must flow through surface films at the interface, the total contact resistance is given by the following series combination*

$$R_t = R_L + R_s + R_o, \qquad \text{(See Footnote **)} \qquad (4.20)$$

where the subscripts t, L, s, and o refer to the total, macroscopic, microscopic, and film contributions, respectively.

Assuming clean surfaces ($R_o = 0$) and effectively perfect contact over the macroscopic contact area ($R_s << R_L$), the expression derived for the macroscopic constriction resistance is

$$R_L = \frac{g(x_L)}{2\, a_L\, k_m} = \frac{1}{\pi\, b_L^{\,2}\, h_L}, \qquad \text{(See Footnote **)} \qquad (4.21)$$

---

*The question whether it is reasonable to assume that R_L and R_S are independent resistances in series is considered in Refs. 23 and 24; and it is concluded that negligible error is encountered in doing so.

**Following Ref. 27, thermal resistances in Section 4.2 are so defined that their units are (hr °F/Btu).

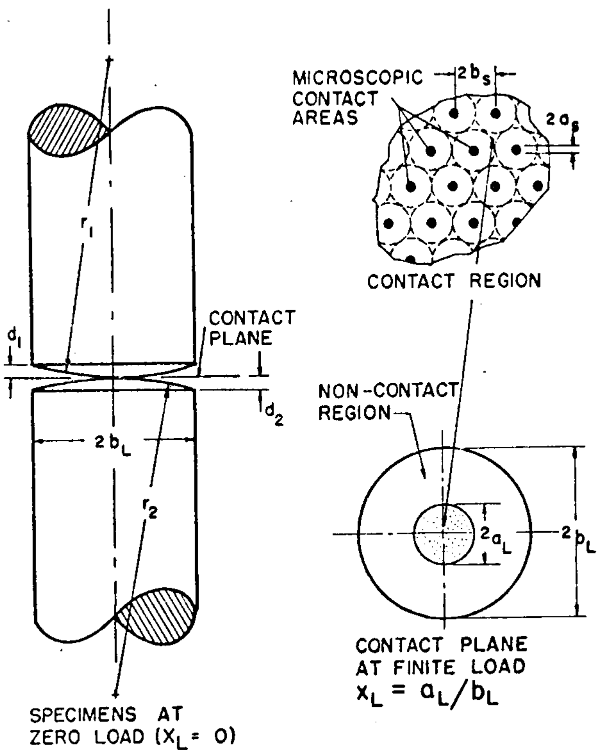

Fig. 4.4 - Model of Contact Surface for Macroscopic Constriction Theory. (From Ref. 27.)

where g(x) is the constriction alleviation factor given by Eq. 4.2 and h_L is found from the Biot modulus, which is defined as the ratio, $b_L/\Delta L_m$, of a characteristic length, b_L, to the quantity $\Delta L_m = h_L/k_m$. ΔL_m can be interpreted as the effective additional length of the contact members that would produce the same heat flow resistance as the contact. The formula found for the Biot modulus, valid for $x_L = a_L/b_L < 0.65$, is

$$\frac{h_L\, b_L}{k_m} = \phi\ (\zeta) = \frac{2}{\pi}\ \frac{y}{g(y)} \tag{4.22}$$

where

$$\begin{aligned} y &= 1.285\ \zeta^{1/3} \\ \zeta &= \text{elastic conformity modulus} = (p_a/E_m)(b_L/d_t), \\ d_t &= \text{total equivalent flatness deviation (based on} \\ &\quad \text{zero load)} = d_1 + d_2 \qquad \text{(see Footnote *)}, \\ p_{ap} &= \text{apparent contact pressure} = F/\pi b_L^2 . \end{aligned} \tag{4.23}$$

The requirement, $x_L < 0.65$, limits applicability of the above theory to moderate apparent pressures. For example [106], for aluminum surfaces having a flatness deviation of 100 μin. and a waviness wavelength equal to 1 in., the maximum apparent pressure for which the above theory is applicable is 60 lb/in^2. The range of applicability increases as the flatness deviation and modulus of elasticity increase and as the pitch of waviness decreases.

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*In Ref. 23, Clausing points out that the model of spherical contacting surfaces exaggerates the flatness deviation and that therefore d_t should probably be replaced by four times the measured value when the theory is applied to actual engineering surfaces. The reasons for choosing the factor four are not given.

The mathematical derivation involved the approximation that the Poisson ratio of both members equals $\sqrt{0.1}$, which is nearly true for most metals. Another approximation limits application to specimens having a large length-to-diameter ratio. Analog measurements [25], however, indicated that it may be reasonable to apply the results for L/b_L as low as 0.6. The mathematical derivation also involved use of Hertz's determination [100] of the contact area between two spheres in elastic contact. Limitations of Hertz's solution for the present application are given in Ref. 23, where it is mentioned that an extension of Hertz's analysis that will remove its restrictions is under way. Ref. 22 gives a numerical analysis which leads to a refinement of the above theory, relaxing some of the restrictions on x_L and L/b_L. The results of this analysis are also given in Ref. 26.

When applied to the geometry in Figure 4.4, Eq. 4.3 leads to the following expression for the component of interface conductance associated with the microscopic constriction

$$h_s = \frac{1}{R_s A_{ap}} = \frac{2\, p_{ap}\, k_m}{\pi \xi H a_s g(x_s)} \quad . \tag{4.24}$$

Some experimental confirmation of the above theory was obtained. Tests were performed under vacuum with aluminum, brass, magnesium, and stainless steel samples having ends polished and lapped to spherical shape, conforming with the model indicated in Figure 4.4. The equivalent flatness deviations, d_t, ranged from 25 to 820 μin. The mean interface temperatures in different tests ranged from 160 to 340°F, and the contact pressure was varied between 0 and 100 lb/in^2. The data were correlated in terms of the parameters $\Delta L_m/b_L$ and ζ and are plotted in Figure 4.5. Good agreement between theroy and experiment was obtained in all cases except those in which film resistance or thermal strain may have been important factors. The data clearly demonstrate the significance of the macroscopic constriction effect and the usefulness of the model in

Yovanovich [106] examined the problem of thermal contact conductance in a vacuum for surfaces having roughness and either cylindrical or spherical waviness. He neglected radiation and used deformation analyses for either purely plastic or purely elastic deformation. Based on earlier work in the field he assumed that the asperities deform plastically and elastically and that the waviness component deforms elastically for light loads and elastically and plastically for very high pressures.

A recent re-analysis of the constriction resistance was made by Greenwood [50].

5. PROPERTIES OF SURFACES

5.1 Surface Structure

Surface texture may be regarded as having several components: roughness, waviness, and error of form, as illustrated in Fig. 5.1. Profilometer traces of surface structure, in which the magnification perpendicular to the surface is usually about 50 times the magnification parallel to the surface, have sometimes created the impression that surface asperities have steep walls. Actually, the wavelengths of the irregularities of both roughness and waviness are much greater than their amplitudes. Consequently, actual contact between two surfaces occurs over a very small fraction of the interface area, often less than 1 percent.

Directional methods of mechanical surface processing give rise to a directivity, or lay, in the distribution and form of surface projections. The three main types of surface profiles in this category correspond to turning, grinding, and buffing. Nondirectional processing methods, such as electropolishing, anodizing, and lapping, result in distributions of projections having no preferred direction parallel to the surface.

One of the most common instruments used to study the topography of surfaces is the profilometer, in which a stylus traverses the surface and gives a line section of the contour. Other mechanical and optical techniques for studying the topography and structure of solid surfaces are listed and described briefly in Ref. 83a (p 705 ff).

Measures of roughness include the centerline average (CLA), also known as arithmetic average (AA), and root mean square (RMS); the latter gives more weight to the larger deviations from the centerline and is therefore somewhat larger than the former. Values of roughness can range from 2 μin. rms for very smooth surfaces to 600 μin. rms for the roughest surfaces [107]. Fig. 5.2 illustrates the ranges of roughness for different methods of surface finishing.

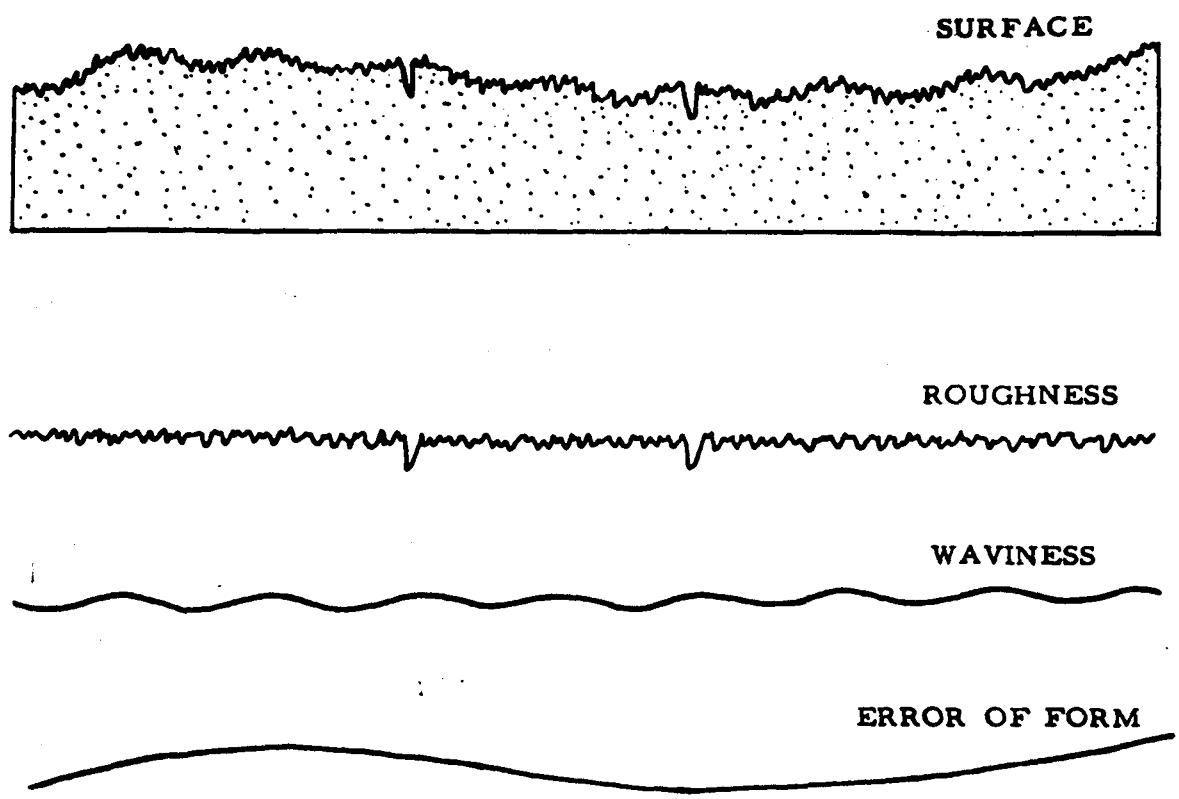
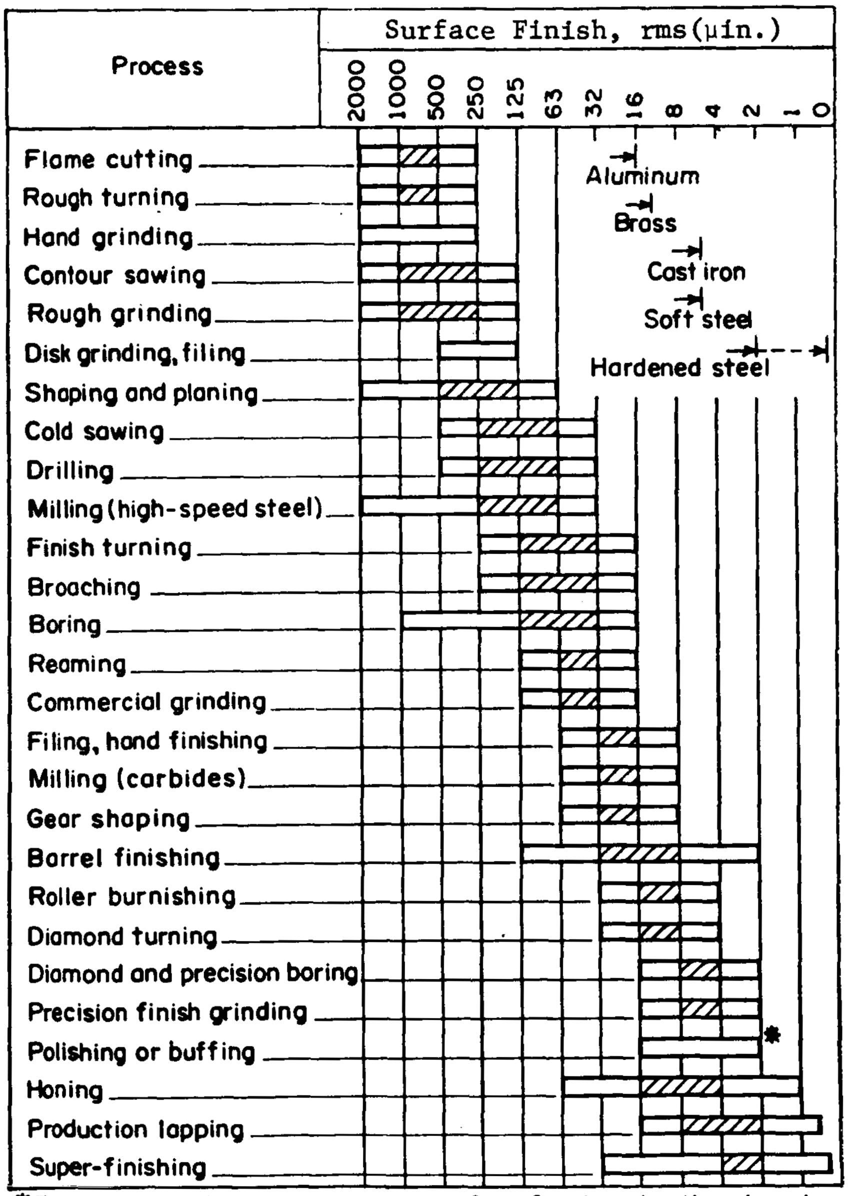


Fig. 5.1 - Relationship Between Surface Roughness, Waviness, and Error of Form. (From Ref. 88.) (Vertical magnification much greater than horizontal magnification.)



*Finish is dependent upon previous surface finish and grit and grade of abrasive.

Full range commercially used.

Usual average or economical range.

Fig. 5.2 - Relation Between Basic Finishing Processes and Ranges of Surface Roughness. Inset Shows Maximum Practical Surface Fineness for Five Basic Materials (These limits do no relate to processes listed opposite them in left column). (From Ref. 49a.)

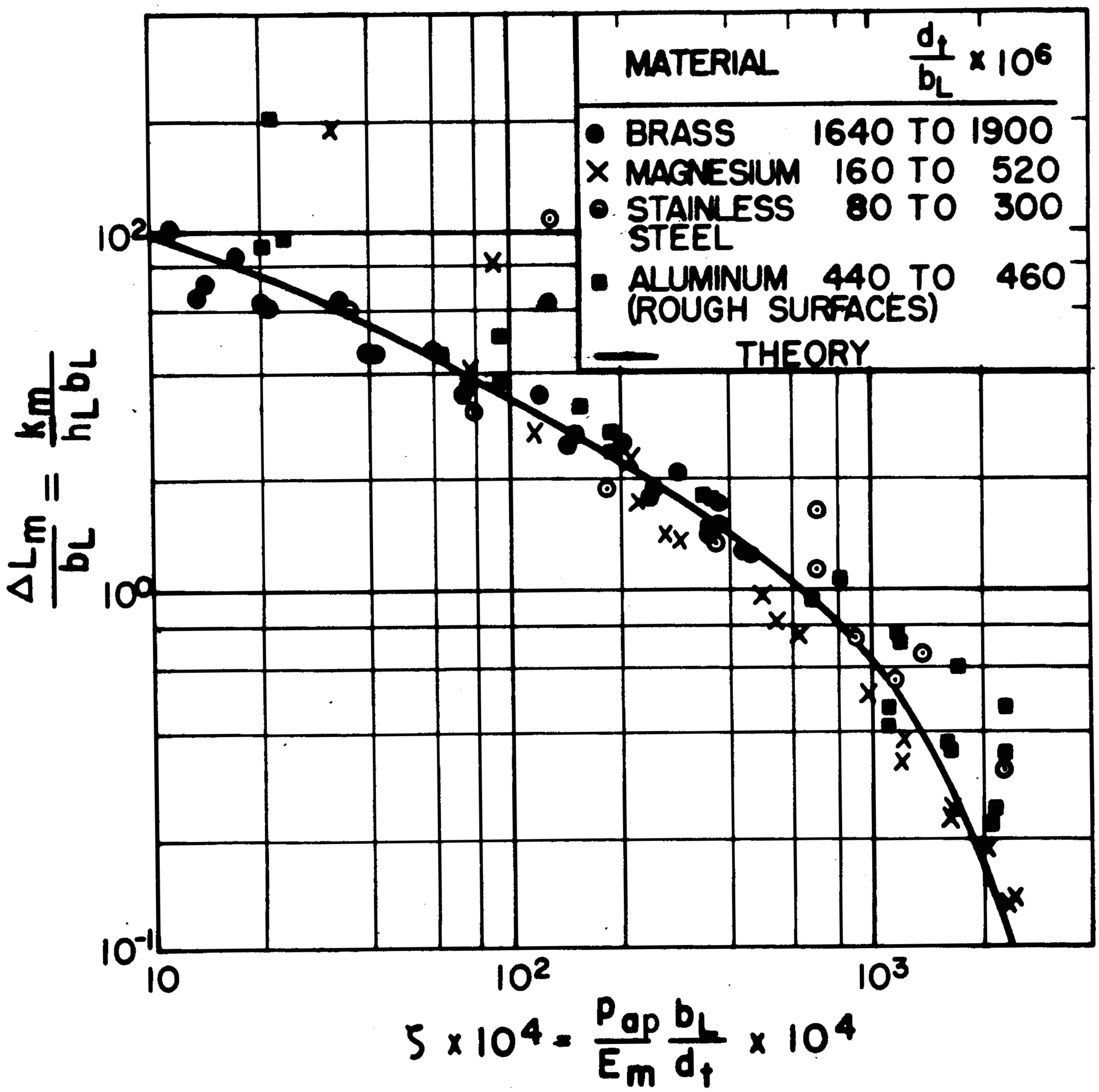


Fig. 4.5 - Comparison Between Macroscopic Constriction Theory and Experiment. (From Ref. 23.)

predicting thermal contact resistance when this effect is dominant and the geometry of spherical surfaces in contact is applicable. An obvious shortcoming of the model in its present form is the limitation to a single macroscopic contact. In Ref. 27, the authors indicated that an extension to multiregion contact was under study.

Efforts by others to apply Clausing's theory have not been very fruitful. Fried [47] had little success in attempting to correlate thermal conductance data according to the method of Clausing and Chao [25]. Bloom [13] reviewed the macroscopic constriction theory of Clausing and Chao [25] and compared it with experimental data. He found deviations and suggested several possible deficiencies of the theory. His experiments showed evidence of a strong influence due to the degree of conformity of specimens having several macroscopic contact areas, when mated in different relative orientations. A theory capable of application to such situations would be more useful than Clausing's theory. Because of the irregularity and often undeterminable nature of waviness for many types of surfaces, however, accurate prediction of conductance for such cases may be impossible

4.3 Other Analyses of Heat Transfer Across an Interface

Using an electrical analog, Padet and Cordier [86] analyzed the temperature distribution in the vicinity of a contact for the type of model shown in Figure 4.1 From its maximum value, which is constant over the area of actual contact, the temperature drops with radial distance in an almost hyperbolic manner. It was found that the temperature exceeded the value it would have for a perfect contact within a distance, from the center of a contact, that depends on the thermal properties of the materials and the height and separation of asperities. At greater distances from the center of the contact area the temperature was slightly lower than the value for perfect contact. Computations showed that the phenomenon exists in a vacuum as well as in the presence of an interstitial fluid.

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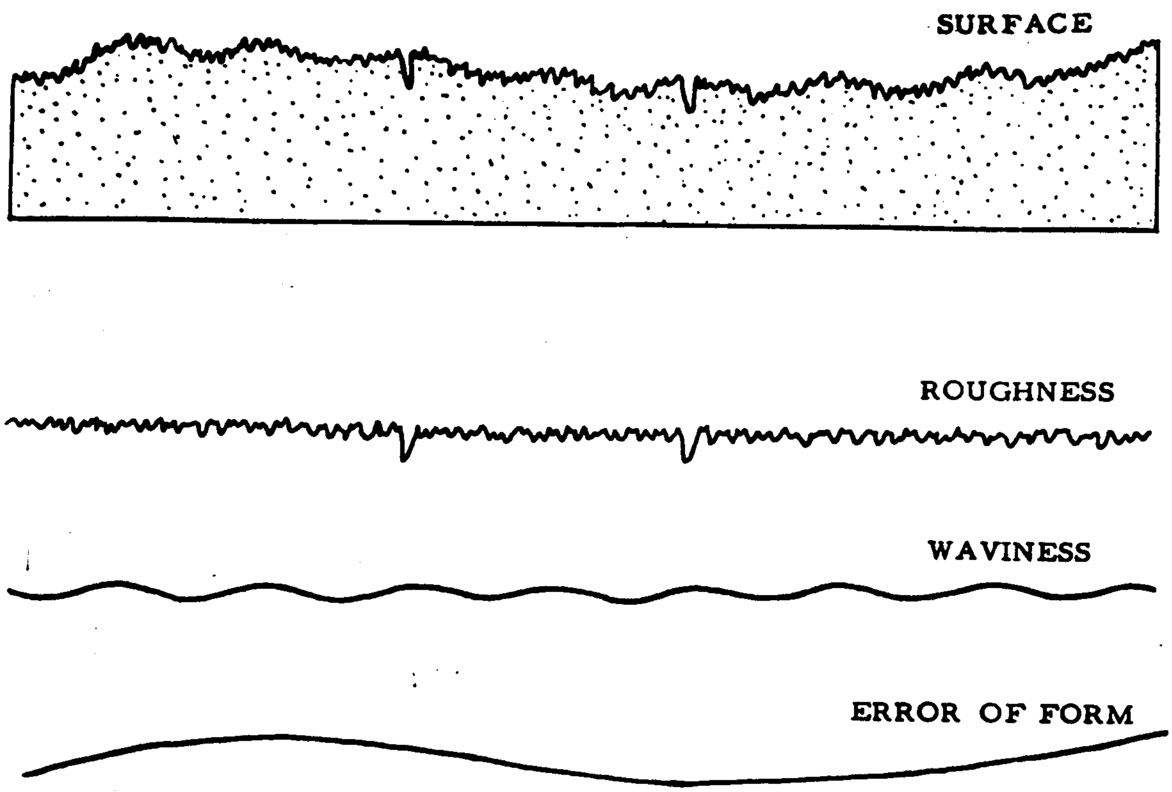

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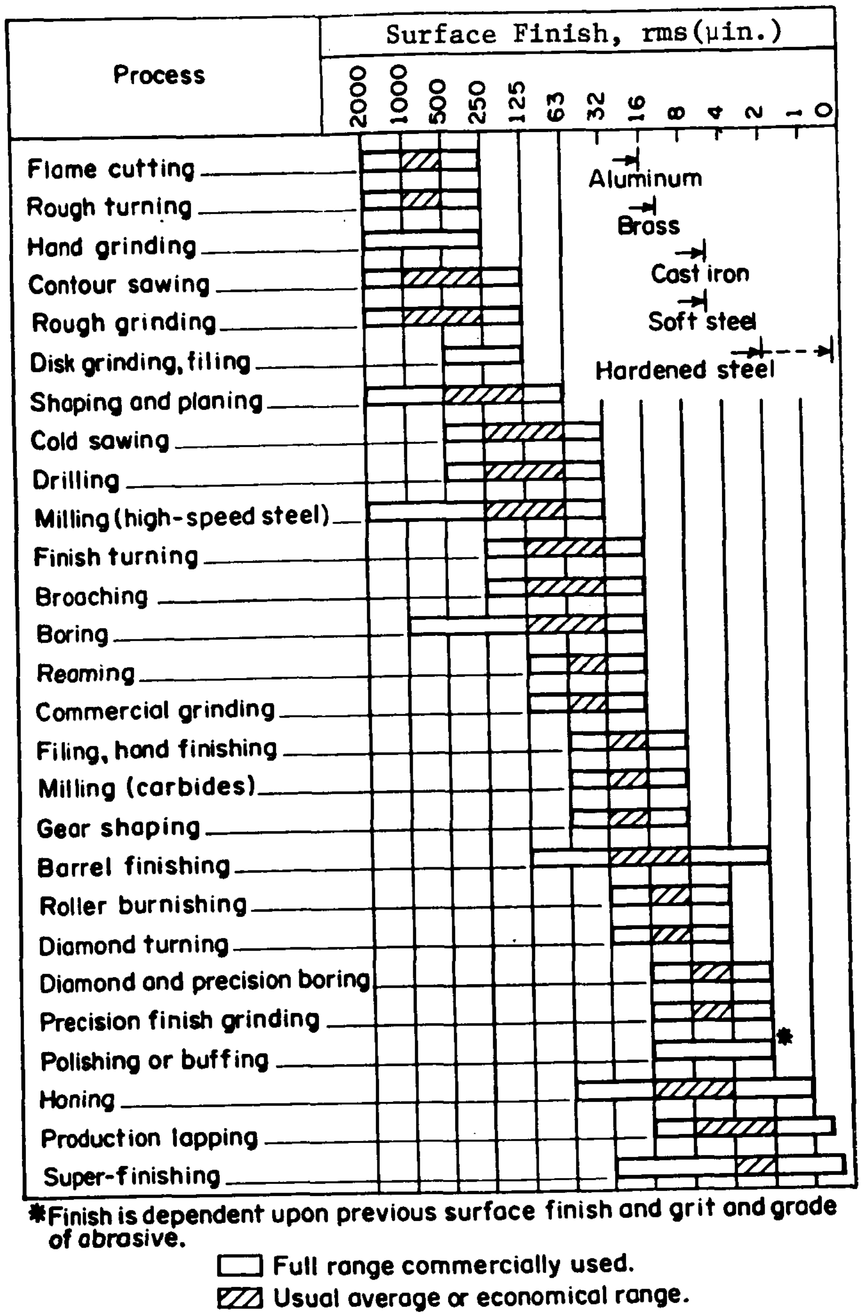

Fig. 5.2 - Relation Between Basic Finishing Processes and Ranges of Surface Roughness. Inset Shows Maximum Practical Surface Fineness for Five Basic Materials (These limits do no relate to processes listed opposite them in left column). (From Ref. 49a.)

Surface waviness can result from imperfections in the surface finishing process (such as vibration and chatter during machining), from deflections produced during assembly of fabricated products, and from heat treatment stresses. According to Ref. 107, the length of surface waves varies from 0.04 to 0.40 in. and their height varies accordingly from 80 to 1600 μin. The consensus of opinion of several experts in the field of metal finishing was reported [41] to be that the waviness of a surface is dependent on so many parameters that a correlation between waviness and roughness for a given finishing operation is impossible. The only way to obtain an estimate of waviness is to measure it.

Real surfaces contain many imperfections and impurities. The schematic diagram of a typical polished metal surface in Fig. 5.3 shows how deviations from structural and topographical uniformity increase as the free surface is approached. On top of the base metal sub-surface there is a *polish* layer which is an "amorphous fudge" of metal, metal oxide, polishing powder, and contaminants that were present when the surface was prepared. Over this there is an oxide film which may be fairly uniform or in the form of irregular needles. Finally, there is a layer of adsorbed gases, including oxygen and water vapor.

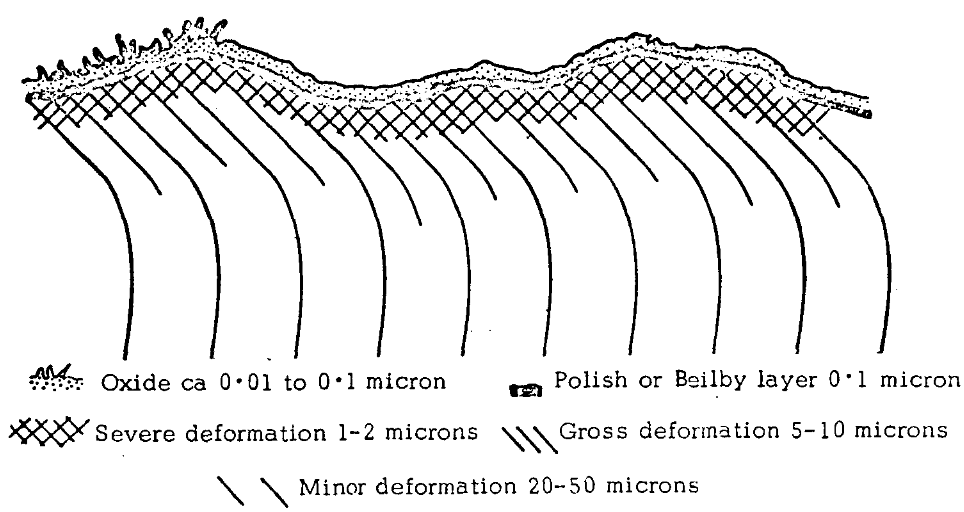

Fig. 5.3 - Schematic Diagram Showing Topography and Structure of a Typical Polished Metal Specimen. Subsurface Deformation, Heavy Surface Deformation, the Polish Layer, and the Oxide Film are Shown. (From Ref. 83a.)

5.2 Surface Hardness

The hardness of the surfaces at an interface is one of the most important properties that govern the actual contact area and, consequently, the thermal contact conductance. In fact, in the simplest contact theories it is assumed that the asperities deform plastically and that the actual contact area is related to the load on the interface by the expression

$$A = F/H \quad (5.1)$$

where H is the hardness of the softer of the two materials in contact. This use of the term hardness is a reasonable and very useful concept*, but its limitations should be recognized.

Hardness is basically a measure of the resistance to plastic deformation, originally defined in terms of a scratch test, but also measured by static and dynamic indentation tests and by rebound tests [82a, 104b]. Each hardness test has a specific definition, and comparisons are strictly valid only under identical conditions. Nonetheless, empirical correlations of different hardness measurements have been made for certain materials. Since it is here applied to the identation of a surface by asperities on another surface, the pertinent hardness tests are the indentation type tests involving small indentation areas, comparable to the areas of contact spots between the mating surfaces. The terms 'microhardness' and 'microindentation' hardness have been used to describe such tests [82a].

Values of hardness measured by indentation tests depend on the shape and size of the indenter, the load applied, and the time the load is maintained. The area used in computing hardness is, in effect, measured after the indenter has been removed. In some tests the actual area of the indentation is used, and in others the projected area is used. Clearly, microhardness is closely related to the deformation of

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* A general discussion of hardness and its application to contact theory is given in App. I of Ref. 56a.

asperities, but it is a more complex relation than is implied by Eq. 5.1. According to Ref. 21, Meyer hardness is preferable to other hardnesses in thermal contact problems because it gives the projected area of the solid spots, which is the smallest area exposed to heat flow.

Correlations of hardness and other physical properties, although largely empirical, are useful when hardness measurements are not available. Mott [82a] discusses this subject and gives references to the contributions of Holm, Tabor, and others. The relation usually used in computations of thermal contact conductance is

$$H = 3Y \quad , \tag{5.2}$$

where Y is the yield stress. Some authors replace Y by the initial elastic limit, Y_o, which is somewhat smaller than the yield stress; and a few have replaced Y by the ultimate strength, S_u, which is equal to the yield stress for an ideally plastic material. (For example, see Sec. 4.1.2 and 4.1.4.) The elastic limit of materials which work harden increases only slightly with deformation, but the yield strength may become much greater than the initial elastic limit. The yield stress of a fully worked material will equal the ultimate strength.

According to Ref. 67 (p 12 ff) the hardness of a metal surface depends on the method by which the surface was prepared and on the depth below the surface. The outermost layers of a metal have a lower microhardness than the layers immediately below them, but the microhardness decreases at still greater depths. The occurrence of maximum microhardness at an intermediate layer is the result of maximum work-hardening occurring in this region during surface preparations. The deformation which occurs when two metal surfaces are pressed together causes further work-hardening.

In Ref. 37 it is emphasized that the hardness should be determined for the average contact size in the joint under consideration.

Data for the dependence of hardness on indentation area are given in Ref. 37 for stainless steel. Although this may be an important consideration, some reports seem to have exaggerated the possible variation of hardness with the load on contact conductance. Two such reports are discussed below.

Laming [68] analyzed thermal contact conductance between surfaces having a regularly pitched ridging and conducted experiments with specimens onto which such a ridged pattern was machined. The values of solid to solid conductance indicated by his measurements on steel /brass and steel/aluminum contacts were smaller than predicted, and they increased with increasing load at a higher rate than predicted. Laming found that the experimental data could be correlated with the theory, however, if he hypothesized that the hardness which controls the area at points of actual contact is greater at smaller loads. It is doubtful, however, whether this hypothesis provides a realistic interpretation of the data. Laming admits that the maximum values of hardness deduced by applying the hypothesis are "fabulously high". Furthermore, a corollary of the above hypothesis is that the areas of the contact spots increase with load, which contradicts the observation by others that the area remains relatively constant and only the number of spots increases with load.

Williams [104a] pursued the above point in experiments performed with nickel/nickel and steel/steel joints. By conducting his experiments under vacuum, Williams eliminated the problem Laming had to separate the solid to solid conductance from his measured values of total joint conductance. In this case, it was again possible to interpret the results by an apparent increase in hardness with decrease of contact load. However, the observations that the actual number of contact spots was much smaller than the potential number of spots for the type of surface finish used and the fact that the surface contained regions of heavy spot loading and others of much lighter loading strongly suggest that surface waviness contributed to the observed variation of thermal conductance with load.

5.3 Actual Contact Area at an Interface

5.3.1 Introduction

When nominally plane surfaces are placed in contact, the actual contact area is only a small fraction of the apparent contact area. Because of surface roughness, contact occurs only at points where asperities on one or the other surface touch the opposite surface. Depending on the mechanical properties of the materials and the pressure with which the two members are held together, each microscopic contact area may be surrounded by relatively large areas where the surfaces are separated. For a time it was usually assumed that the microscopic contacts are uniformly distributed over the entire interface, but it was later recognized that large-scale surface irregularities cause the microcontacts to be confined to isolated regions of the interface.

Fig. 5.4 illustrates the several contact areas that can be distinguished:

1. The apparent contact area which is the area of a plane passing through the interface between two contacting solids.
2. The macroscopic contact areas enclosed by contours surrounding locations where contact results from deformation of the surface undulations.

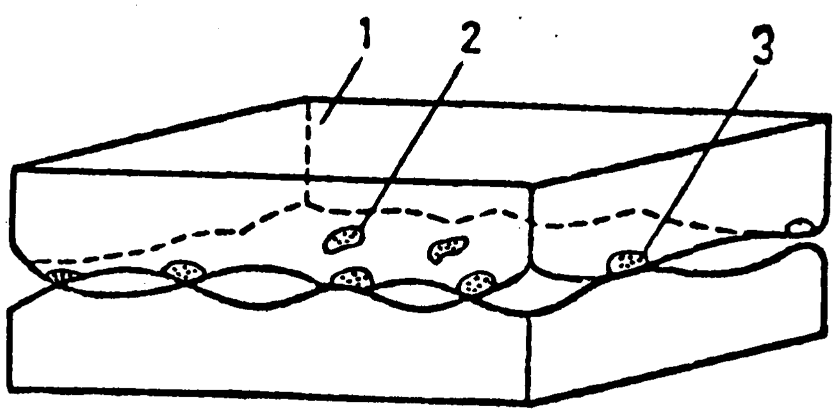

Fig. 5.4 - Schematic Diagram of Two Rough Surfaces in Contact: 1 Apparent Area; 2 Macroscopic Contact Area; 3 Actual, Microscopic Contact Area. (From Ref. 67.)

3. The real contact area, which is the sum of all the microscopic areas, where actual touching of the solids occurs.

The model shown in Figure 4.4 is an idealization of the above picture.

Many references discuss the surface deformation and real contact formed when two solids are brought together. A short resumè of the deformation of metals and non-metals and the determination of real contact area is given in Ref. 83a (pp 710-712). Hertz's solutions for the pressure between spherical bodies in contact and the size of the contact area resulting from elastic deformation are given in Ref. 100 (p 372 ff). Archard [3] reviewėd some of the literature on surface deformation and contact area and developed the theoretical relation between load and contact area for several models simulating rough and wavy surfaces deformed elastically by contact with a smooth flat surface. Refs. 33 and 67 are largely concerned with Russian work in this field. Ch. 1 of Ref. 67 includes a discussion of surface characteristics, real contact area, and the effect of deformation on surface properties; Ch. 2 gives theoretical and experimental methods for calculating contact area between rough surfaces. Ref. 33 gives theoretical and experimental methods for estimating actual contact areas and also gives the results of a number of experimental studies. A review of the process by which asperities and surface waves deform and contact areas are formed under load may be found in Ref. 108. A good review of the deformation process can also be found in Ref.39 (pp 42-54).

5.3.2 Deformation of Contacting Surfaces Under Load

When two surfaces are placed in contact they first touch at only a few places. As a load is applied the separation of the surfaces decreases and the number of discrete contact points increases. Deformation occurs at the contact points and also extends to the waviness in the surface. Most of the asperities deform plasitcally, but elastic deformation of asperities can occur at the edge of the macroscopic contact areas, where the stresses are relatively low. The stresses in the surface waves are always much smaller than those in the individual asperities, and the wave deformation is therefore elastic.

When the load is removed, the surface waves recover elastically. This is accompanied by a disruption of most of the plastically deformed contact regions so that the residual contact area after elastic recovery is considerably smaller than the contact area obtained during the initial plastic deformation.

One view of the process of surface deformation under an applied load is the following. As the applied load is increased, the onset of plastic flow occurs when the local mean pressure attains a value of approximately 1.1 times the elastic limit of the material. Because actual contact usually occurs over a very small fraction of the apparent contact area, a quite small load may suffice to cause the initiation of plastic flow. With further increase in load, fully plastic flow is attained when the local mean pressure reaches a value of about 3 times the elastic limit of the softer of the two mating materials. If the load is increased further, the area of real contact will increase; but the mean pressure will remain equal to approximately 3 times the elastic limit. Work hardening caused by the plastic deformation, however, may cause the elastic limit to increase. Experiments [18a, 18b, 56a] have shown that, except for extremely smooth surfaces, the loads which can be supported by elastic deformation are extremely small. Greenwood and Williamson [51] in fact, claim that most surfaces do not undergo a transformation from elastic to plastic deformation as the load is increased, but deform plastically even under the lightest loads. The deformation of non-metals such as polymeric solids and rubber-like materials may be much more nearly elastic than plastic in nature. Although there is some uncertainty in the behavior under very light loads, practical loads are usually large enough to cause plastic deformation.

A model for surface deformation given in Ref. 33 (pp 53 and 63) is based on the assumption, supported by experiment, that the microscopic irregularities are somewhat harder than the base material because of the cold-hardening that takes place during surface finishing. In accordance with this model, plastic deformation of the tips of surface projections and elastic sagging of the material under them begin simultaneously.

Yovanovich and Fenech [107] concluded that the deformation of asperities cannot be described as either purely plastic or purely elastic yielding. The results of experiments compared to theories assuming purely plastic and purely elastic yielding showed [107] that the deformation is plastic at low to moderate contact pressures but that there is an elastic contribution at higher pressures. Extending the work of others, Yovanovich [109] developed a formula for thermal contact conductance for surfaces which are both rough and wavy. He assumed that the waves deform elastically and that deformation of the asperities is elastoplastic. Application of his method required graphical determination of real contact area as a function of load. Test data for a sample having only roughness and another having only waviness were in reasonably good agreement with the theory. Other test data were interpreted as showing that the deformation was partly plastic and partly elastic over the entire load range (up to 10^4 lb/in^2); however, it is not clear from Ref. 109 which sample was tested. In view of the difficulty of describing such behavior analytically over a wide range of contact pressure, Yovanovich and Fenech [107] proposed an empirical approach for obtaining the geometric parameters (number of contact spots and actual contact area) for nominally flat surfaces. The method involves experimental determination of the relation between load on the joint and the relative displacement or compliance, of the two surfaces. Application of the method to experimental data, including some from Ref. 75, led to good agreement with thermal conductance theory.

When the clamping force is increased, the average interface gap height is decreased in part by plastic deformation of the peaks of the asperities and in part by elastic deformation of the material supporting the plastically deformed peaks. As the number of macroscopic contact areas increases the elastic deflection becomes less significant.*

Ling [75] analyzed the load-compliance characteristics of two rough surfaces in contact and made comparisons with the case of a rough surface in contact with a rigid, flat, smooth surface, which had been considered by others. He assumed the asperities to be right circular cones randomly distributed over the surface, so that equal numbers of cones start at every plane parallel to the surface. This leads to a geometric increase in the number of contacts between asperities on opposite surfaces as the separation of the surfaces is decreased. Assuming further that the cones are perfectly plastic, he obtained load-compliance characteristics which had steep slopes, in better agreement with experiment than the lower slopes predicted by theories for the case in which one surface is smooth and rigid. Although Ling's work is related to the thermal contact conductance problem, the computation of actual contact area and conductance was not considered in Ref. 75.

5.3.3 Surface Analysis and Estimation of Actual Contact Area

The dependence of real contact area on load depends on the shape of the asperities, their height distribution, and the metal properties - particularly the elastic modulus, yield point, and extent of work-hardening. Temperature affects the contact area obtained for a given clamping force by its effect on the physical properties of the materials. For soft metals, the size of contact area is also influenced by the time that the joint remains under load [67, p 57].

* See Ref. 74, p 33. Note that what are termed *macroscopic* contact areas in this report are called *apparent* contact areas in Ref. 74.

No simple, reliable procedures are available for computing contact areas for practical applications. The graphical method described in Ref. 39 is tedious and time consuming. The analog method described in Ref. 54 is more efficient, but requires the use of special apparatus; furthermore, the method appears to be limited to situations in which it is adequate to consider a small region of the surface or in which the radius of curvature of the waviness is sufficiently large [109]. An extensive consideration of the problem may be found in Ref 33. Theoretical methods of computing contact area are reviewed; their limitations and the difficulty of applying them to real surfaces are described.

If surface waviness had sufficient regularity the macroscopic contact area at an interface could be estimated by using the Hertz relations [100]. However, surface waviness is generally very variable, and it may be necessary to measure the macroscopic contact area experimentally. A number of models and theories have been developed for calculating the real contact area due to microscopic contacts within the macroscopic contact areas [67].

The simplest model for contact area assumes that, when two metals are in contact, the asperities of the harder metal penetrate the softer metal in a plastic manner, acting like indenters in a microhardness test. Therefore, the pressure over each contact point equals the indentation yield pressure, or hardness, of the softer metal. The actual contact area is

$$A = F/H ,$$

where F is the load between the two surfaces and H is the hardness of the softer material. Under these conditions the actual contact area is proportional to the load, is a small fraction of the apparent contact area, and is independent of the apparent contact area.

Archard [3] has shown that a similar relation between actual contact area and load may exist even when the deformations are entirely elastic. When the materials are such that elastic deformation occurs, the actual contact area and load are related according to the relation

$$A = K F^m , \tag{5.3}$$

where K is a constant. Analyses based on different mathematical models show that m lies between 2/3 (for contact between a smooth sphere and a smooth flat plate) and 1 (the value approached by more complex models* which more nearly simulate the roughness and waviness of real surfaces). As the complexity of the mathematical models increases, the number of contact areas similarly becomes more nearly directly proportional to the load, the size of each contact area becoming less dependent on load.

Henry [53] described a random process method of surface analysis which involves the determination of two statistical parameters. If the parameters are known for any two surfaces they can be combined to yield the interface geometry when the surfaces are placed in contact. It was indicated that the procedure should yield better results at high contact pressures than the graphical [39] or analog methods [54] because it takes into account conservation of material**; however, extremely little experimental verification of the procedure has been found in the literature. Special apparatus would be required for convenient determinations of the parameters.

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* The complexity is increased by covering the smooth sphere with smaller spheres and then covering them in turn with yet smaller spheres.

** When the peaks of asperities are flattened to a given level by plastic deformation, the contact area formed is greater than the bearing area obtained when the peaks are sliced off at the same level. In fact, Ref. 74 (p 31) indicates that when roughnesses are flattened to approximately 55 percent of their original height, the peaks will have flowed into and entirely filled the valleys.

Foster [42] made determinations of the density of contact spots, n, for several samples using the autoradiographic technique, Fenech's graphical method, and Henry's random process analysis. It was concluded that the latter two methods would probably yield approximately the same estimate of n, provided the surfaces were described with sufficient accuracy for the random process analysis. The autoradiographic technique gave values which were lower than those of the graphical method by a factor 2.5. This was ascribed to the possibilities that not all parts of a sample received enough tracer material during preparation and that tracer transfer did not occur at all contact spots.

Estimating the actual contact area from measurements of electrical resistance across the interface is subject to two major defects [83a]: (1) unless one can determine the number of contact spots, there is no unique relation between the total contact area and the total resistance; and (2) the presence of certain surface films may so affect the electrical resistance as to render interpretation of results difficult and uncertain. Limitations of this method are also discussed in Ref. 33 (pp 14 and 15).

Experiments have shown [18, 33, 56a, and 67 (p 51)] that the increase in contact area with increasing load is determined mainly by an increase in the number of contacting asperities, the size of the individual contact areas being almost independent of the compressive force. Boeschoten and van der Held [18] concluded that the average radius of contact spots is about 30 μ (0.0012 in.) for a wide variety of metals and a wide range of contact conditions. However, based on information in Ref. 4 to the effect that the contact conductance remains approximately constant above a certain contact pressure, they concluded that, at contact pressures exceeding about 1400 lb/in^2, the size of the contact spots increases linearly with contact pressure; there occurs a confluence of contact spots and a reduction in their number.

Similar conclusions were reached by Mustacchi and Giuliani [83], who conducted an experimental and analytical study aimed at establishing characteristic semi-empirical relations for estimating real contact areas. They obtained *bearing curves* for various joints by plotting *total contact area* vs. the *number of contact spots per unit (apparent) contact area*. Ideally, for experimental determination of bearing curves, one member of the joint should have a flat, infinitely hard, smooth surface. Generally, the bearing curves were straight lines which curved upward at high values of total contact area. This indicated that the area per contact spot remained practically constant for a range of increasing loads on the joint, but the area per spot increased with further increase in load. No differences were found among the bearing curves for aluminum (SAP), Armco iron, and carbon steel. The approach presented in Ref. 83 does not seem to be readily applicable to practical problems, but with further development it might be useful under some conditions.

A new theory of elastic contact between nominally flat surfaces was recently reported by Greenwood and Williamson [51]. They used a detailed model of elastic contact which takes account of two material preperties: the hardness and modulus of elasticity, and three topographic parameters: the mean radius of asperities, their surface density, and the spread of their heights. The theory leads to expressions for the total real contact area, the number of microcontacts, the load, and the (electrical) conductance between two contacting surfaces in terms of the separation of their mean planes. In agreement with experimental evidence, the theory indicates that the number of microcontacts and the real contact area depend only on the load, and not on the nominal contact pressure. Similarly, it also indicates that the separation of the surfaces is not very sensitive to the pressure, the separation of similar surfaces being approximately equal to the centerline average of roughness. The fact that the ratio of real contact area to load is nearly constant for elastic contacts led to the concept of an *elastic hardness* which can be used for predicting the real contact area for given loads just as the conventional hardness is used when plastic deformation is assumed. A *plasticity index*, which is essentially the ratio of elastic hardness to conventional hardness, was introduced in defining a criterion for the onset of plastic deformation. Most common surfaces have plastic indices substantially exceeding 1 and therefore deform plastically even under the lightest loads. The authors mention that a forthcoming paper will show that the above results are not limited to nominally flat surfaces but are also applicable to contacts between rough curved surfaces. The authors also describe a surface-analyzing system they have developed for computing, among others, the three topographical parameters mentioned above. It combines a profilometer, an anolog-to-digital converter, and a digital computer.

6. EXPERIMENTAL INVESTIGATIONS

6.1 Sources of Data Summaries

Since the amount of data available in the literature is too great to have been adequately summarized within the limits of this report, references to the original sources are given below.

Much of the data on thermal contact conductance available in the literature prior to 1964, including data obtained in a vacuum and with filler materials, was tabulated in Ref. 41. Data on the thermal contact conductance of stainless steel, magnesium, aluminum, copper, and iron joints in a vacuum, at contact pressures up to 100 lb/in^2 were presented in Refs. 44 and 45. The test samples were 2-in. diameter cylinders, 3 in. long. Much of the data in Refs. 44 and 45 were included by Fried in Refs. 46 and 47, which have extensive summaries of data on contact conductance measurements made in a vacuum on stainless steel, magnesium, copper, titanium, titanium alloy, and aluminum alloy joints with a variety of surface finishes.

It has been pointed out [27, 37] that thermocouples mounted too close to the interface, where heat flow *lines* are not uniformly distributed, may yield unreliable readings. This factor should be considered in evaluating data that were obtained with thin samples.

6.2 Sources of Information on Apparatus

Apparatus for measuring thermal contact resistance between cylindrical specimens in a vacuum is described in References 13, 25, 45, 46, 47, 49, 62, and 109, among others. Stubstad [98, 99] described a vacuum apparatus in which the contact surfaces could be separated for outgassing, placed in contact, and forced together with a selected pressure - all with controls external to the vacuum chamber. From its description, it appeared that the apparatus was limited to contact pressures below approximately 10 lb/in^2, but the design could probably be modified for higher contact pressures.

6.3 Use of Interface Fillers and Surface Coatings

A method of increasing thermal contact conductance is to fill the interfacial gap with a highly conductive material. Metals have been tried both as coatings on the mating surfaces and as foils placed between them. Grease and rubber have also been tried as interfacial fillers. Some of the experiments that have been performed are discussed briefly below.

Experimental measurements [49] indicate that the thermal contact conductance between metal surfaces having high microhardness can be increased substantially by plating them with a softer metal. The effect was apparent only at low contact pressures, however. Filling the interstices of the interface with a filled silicone grease also had the effect of producing a large increase in thermal contact conductance.

Bloom [13] reported large increases in thermal conductance when thin layers of vacuum grease or silicone oil were added at the interface of aluminum and stainless steel specimens tested in a vacuum. With these fillers he also found that the thermal conductance was relatively independent of contact pressure over a substantial range.

Clausing and Chao [27] found that thermal contact conductance was significantly increased when the interface between brass specimens was filled with silicone high vacuum grease. The difference between conductances with and without the grease increased as the contact pressure was increased. The effect was ascribed to an effective increase in the macroscopic contact area.

Experimental results showing the effect of silicone grease and silicone rubber fillers on the thermal conductance between aluminum plates in a vacuum were reported in Ref. 43. As the plates were only 1/8 in. thick, however, the measurements were subject to the difficulty of obtaining reliable values of interface temperature drop with thermocouples mounted close to the interface.

Jansson [61] investigated the effect of fillers on thermal interface conduction between aluminum and beryllium specimens in a vacuum. Tests were made with a flexible epoxy cement, indium and lead foils, and aluminum and gold leaf at contact pressures between 10 and 250 lb/in^2. The indium foil, an extremely soft material, gave evidence of being sufficiently compliant to completely fill the voids caused by surface irregularities; and it produced a marked increase in conductance. The epoxy cement and lead foil provided some increase in conductance, but the aluminum and gold leaf did not cause any appreciable improvement.

The use of indium foil, silicone vacuum grease, and a filled grease* as joint filler materials was investigated by Cunnington [29]. The joints were made of 6061 aluminum and AZ-31 magnesium; and they were tested under vacuum at contact pressures between 17 and 96 lb/in^2, mean interface temperatures between 60 and 250°F, and joint temperature drops of 20 to 50°F. Within these ranges the mean interface temperatures and temperature drops showed no significant effect on the thermal conductance of unfilled joints. Attempts to correlate the data for unfilled joints with the theory of Clausing and Chao did not result in good agreement, although reasonably good agreement was obtained in comparisons with the experimental data of others. The use of indium foil and vacuum grease each produced approximately ten-fold increases in the thermal conductance of the unfilled joints, and even larger increases were obtained by using the filled grease. Cunnington's indium foil data did not agree with that of Jansson [61], possibly because of differences in the foils. Data for the greases were believed to be applicable to a wider range of joint configurations because of their ability to flow and fill the interstices even at relatively low contact pressures. It is not known, however, what influence thermal cycling and vibration and long exposure to space environments might have on the filler materials.

*Dow-Corning 340 Silicone Heat Sink Compound.

In addition to indium and silicone grease, Stubstad [98] investigated a number of novel filler materials, including felt, rubber, plastic foams, wire meshes, wire brush, rubber-oil mixtures, and special fin arrangements. From tests at contact pressures up to 10 lb/in^2 he found that greased surfaces gave the highest contact conductances.

Several silicone rubbers and greases, without metal fillers, were tested under vacuum [57] with an arrangement which simulated the mounting of Gemini equipment packages on spacecraft cold plates. A mixture of silicone grease and silver dust was found to be a satisfactory filler. Tests indicated its conductance to be 230 Btu/(hr ft^2°F). A five-day, low-pressure test indicated that there was negligible evaporation or blowout of the material.

Miller [77, 80] performed experiments with steel joints demonstrating that thermal contact resistance can be substantially reduced by the use of suitable metal coatings or filler sheets. He reported that the filler or coating should have higher thermal conductivity and lower hardness than the base metal. Also, the thickness of filler sheets should not exceed twice the average height of asperities.

Mikic and Rohsenow [76] concluded that considerable reduction of contact resistance may be achieved by plating both contacting surfaces with a material of high thermal conductivity, even if it does not result in increasing the actual contact area. This follows from the fact that the thermal conductance is proportional to the harmonic mean of the thermal conductivities of the surface layers through which heat is deviated toward the contact spots.

Lindh [72] performed experiments in air with riveted thin plate lap joints having laminated fiberglass fillers of different thicknesses and expressed the results in terms of an equivalent air gap. Filler thicknesses between 0.03 and 0.1 in. produced little variation in the equivalent air gaps, the thickness of which was several thousandths of an inch; but it is possible that factors not considered in the analysis influenced the results. Other measurements made with riveted lap joints [2] demonstrated that the contact resistance could be reduced substantially by the use of aluminum sheets and paste (Prestite No. 218) as fillers.

At temperatures low enough to cause hardening of grease or paste-like fillers, their influence may be opposite to that observed at higher temperatures. Jacobs and Starr [59], for example, observed that the slightest trace of grease caused the conductance to increase at room temperature, but resulted in a seriously decreased conductance at low temperatures.

Berman [9] performed experiments at liquid helium temperatures with an 0.012 in. thick disc of Teflon inserted between the end faces of copper rods. While it is a very poor heat conductor, Teflon does not become brittle at low temperatures. With the Teflon disc in place the measured thermal resistances were greater than those measured without the disc, but when the separately measured thermal resistance of the Teflon was subtracted, the resulting values of contact resistances were appreciably smaller than the values obtained in the absence of Teflon. This suggests that Teflon might be capable of increasing the thermal conductance of joints at low temperatures, provided its thickness is small enough.

6.4 Bolted and Riveted Joints

The characterization of even carefully controlled joints typical of spacecraft design is difficult. In his investigation of practical riveted and bolted joints in air, Barzelay [6] found that the thermal conductance of production-type joints varied over a wide range for what were meant to be identical constructions. Bevans et al [12] investigated the thermal conductance of bolted joints under vacuum and reported that the "most obvious result" of their tests was the inconsistency of the results for *identical* joints. They indicated that this could be largely attributed to bowing of the bolted plates. The variation of results for *identical* joints having a filler material in the interface was even greater than that for unfilled joints.

It is difficult to match the properties of real joints with the properties of specimens (usually of the rod type) used in obtaining experimental thermal conductance data. Surface strains due to machining and assembly processes may cause local physical properties to differ from those of the bulk material. Surface contaminants may also drastically alter the properties controlling heat transfer. Since attempts [47] to correlate thermal conductance data on carefully prepared rod specimens have proved impractical, it does not seem likely that it will be feasible to arrive at general predictive schemes for bolted joints, the variability of which greatly exceeds that of the carefully prepared test specimens.

Experimental data on heat transfer across bolted and riveted joints available in the literature prior to 1964 were compiled in Ref. 41. Brief accounts of such studies, especially more recent ones, are given below.

Measurements made on a bolted joint between aluminum alloy cylinders in a vacuum are reported in Ref. 95. Although the test specimens were designed to simulate conditions in the Saturn S-IC instrument compartment, the differences between the test specimens and the real joints probably resulted in substantial differences between their contact pressure distributions and actual contact areas. Therefore, the data are useful mainly for the qualitative trends indicated. The temperature drop across the interface was found to increase as the heat flux was increased and to decrease as the torque on the bolt was increased. Varying the ambient pressure between 10^{-4} and 10^{-7} Torr had little effect on the temperature drop. This is not surprising because fluid conductance is negligible over the range of pressures investigated.

Kaspareck and Dailey [62, 63] measured the thermal contact conductance of dissimilar metal joints of a type similar to that used in mounting electronic equipment to liquid cooled panels in the Saturn IB/V vehicle. Combinations of aluminum and magnesium alloys were tested under vacuum at contact pressures up to 1000 lb/in^2. From their measurements they concluded that the temperature differential across a typical interface in a mounting bolt area would be less than 1°K, which is considered negligible for thermal design purposes. However, they had insufficient data for computing an average temperature differential over the entire component mounting surface. Since the contact pressure decreases rapidly with distance from the bolt centerline, the temperature differential could be substantial at points outside the immediate vicinity of the bolts.

Wellnitz [102] made measurements, in a vacuum, of the thermal contact conductance across bolted joints between a heat sink plate and a plate to which a resistive heating element was attached. The plates were of magnesium, and they were bolted together with titanium screws. The nominal contact areas were of the order of 1 in^2. The rate of heat generation in the resistors was varied from 2 to 10 w, the higher values being used for tests in which a silicone grease was used as a filler. Most of the conductances were in the vicinity of 1000 (Btu/hr ft^2 °F) for joints without a filler and about twice this amount for joints with the silicone grease filler. When the greased joints were cleaned and retested, the conductances were practically identical to the values obtained before the joints were greased, indicating that any possible impregnation of grease into the surface had little effect. Variation of the screw torque from 0.5 to 3 in-lb produced very little variation in conductance. This was explained by the fact that increasing the screw torque affects the contact pressure in the immediate vicinity of the bolt but has little effect on the much lower contact pressure over the span between bolts; since the span length was many times the bolt diameter, there was little effect on the average conductance. This explanation implies that the area of relatively high conductance near the bolts is such a small fraction of the total interface area that it has little influence on the overall conductance of the joint.

Coulbert and Lin [28] used an interferometer technique to obtain effective values of contact resistance for aircraft structure joints of the riveted or bonded lap-joint type. They pointed out that parameters such as sheet thickness and rivet pattern are important for such joints; they considered data obtained with machined blocks at various contact pressures to be inapplicable. Variations in manufacture of joints of identical design caused variations of as much as 100 percent in the values of thermal resistance. Most of the joints they studied had a high resistance, comparable to that of a 1-mil air gap. The number of joints studied, however, was insufficient to permit drawing conclusions concerning the general effect of various design and manufacturing parameters.

Wide manufacturing variability was also evident in contact resistance measurements made under vacuum on riveted joints which had skin-stringer configurations typical of space vehicle structural joints [31]. The joints consisted of 2024 T-3 clad aluminum sheet (0.040 and 0.060 in. thick) riveted to bare 2024 T-3 aluminum angle extrusions (0.063 and 0.125 in. thick). The measured values of interface conductance ranged from 71 to 1295 (Btu/hr ft^2 °F). Although there were some variations in experimental conditions, most of the variation in thermal conductance was attributed to manufacturing variability and to the unpredictable bowing and warping caused by thermal stresses. The use of thicker plates or stringers tended to yield higher thermal conductances, probably because their greater stiffness reduced the degree of inter-rivet warping.

Measurements of the thermal contact conductance of riveted and spot-welded skin stiffener joints and riveted and bolted lap joints were reported in Ref. 19. For the skin stiffener joints, the thermal conductance varied between 150 and 220 (Btu/hr ft^2 °F) but the values appeared to be independent of the joining method (riveted or spot welded) and the sheet thickness (1, 1.5 and 2 mm). In earlier work [103] with riveted and bolted lap joints, the same investigators found that increasing the plate thickness from 4 mm to 8 mm increased the conductance considerably. Increasing the number of rivets or bolts per unit area also had the effect of considerably increasing the thermal contact conductance across the lap joints.

10 in-lb. In Ref. 35, it is claimed that the use of a soft shim material of low thermal resistance between a flange and mounting plate has little practical value in increasing thermal conductance across the joint, but it is doubtful that this conclusion was justified on the basis of the brief investigation that was made of this factor. The results given in Ref. 35 could be used to predict the thermal performance of joints similar to the ones studied.

Measurements of interface gaps for some riveted lap joints indicate that the average gap size depends much more on plate thickness than on the rivet pitch [3b]. Macroscopic contact areas were found only very close to the rivets. The region over which force was transmitted from one plate to the other was limited to an annulus with an outside diameter approximately equal to the diameter of the rivet head. It was also found that variations in workmanship in preparing joints could result in considerable differences in the magnitude and distribution of gap heights.

The difficulty of obtaining reliable estimates of plate deflections in bolted and riveted joints by analytical approaches was shown in Ref. 41, where the work of several authors is discussed. This presents a problem for joints in air because approximately 90 percent of the total conductance for an average joint occurs across the region, outside the immediate vicinity of the fastener, where the plates are separated by gaps much larger than the surface irregularities. In fact, for this situation Fontenot [41] concluded that one must resort to the use of experimental data. The problem is less serious for joints in a vacuum environment, where most of the heat is conducted through small areas surrounding the fasteners.

Bevans *et al* [12] developed an analytical study of the problem when one has a uniform heat flux normal to the external surfaces of plates joined by bolts. Based on the observation that the apparent contact area was confined to a small region surrounding the bolts, they used a model which considered radial heat flow in a circular plate to a circular sink at the center. The total resistance was assumed to be dominated by the resistance to heat flow within the thin plates, to and from the bolt area. The resistance of the area of apparent contact near the bolt was determined by computing the pressure distribution under the bolt, dividing the area into annular zones and assigning to each zone a conductance obtained from the conductance versus pressure data given in Ref. 13 for a comparable joint.* The total resistance was the sum of the resistances through the two plates and the resistance of the apparent contact area near the bolt. Resistance computations for entire bolted plates were made by sectioning the plate into rectangular and triangular areas that could be associated with individual bolts and then representing each of these areas by a circular sector in order to fit the analytical model. The agreement between experimental and computed values of joint conductance for several bolt configurations was considered by the authors to be adequate.

*Unfortunately, in Section 7.2.3 of Ref. 12, where a sample computation is given, it is not clear how the bearing area was obtained from the measured values given for bolt and nut bearing diameters. Furthermore, the computed stresses were so high that over most of the contact area it was evidently necessary to use extrapolated values of conductance versus pressure, measured values of which were available only for pressures up to approximately 5000 lb/in^2.

7. DATA CORRELATIONS*

7.1 Holm

Holm [55 and 56] examined measurements of thermal contact resistance for metal joints in a vacuum environment and found that they can be correlated with the load acting on the joint. Under conditions that make metal-to-metal conduction the dominant mode of heat transfer, it is understandable that there should be a correlation with load instead of pressure (based on apparent contact area). The thermal resistance is in this case dependent on the real contact area, which is a function of the load. The apparent contact area becomes important when conduction through a fluid filling the interstices contributes to the heat flow.

Holm expressed the thermal resistance in the form

$$R' = \phi(M)\ f(F) \tag{7.1}$$

where $\phi(M)$ is a dimensionless function dependent on the metals in contact and $f(F)$ is an empirical relation. He found that a suitable expression for $\phi(M)$ is

$$\phi(M) = (k_{ref}/k)\ (H/H_{ref})^{1/2} \tag{7.2}$$

where k is the thermal conductivity, H is the microhardness, and the subscript ref applies to a reference metal. Taking silver as the reference metal, Holm obtained

$$\phi(M) = 0.019\ \sqrt{H}/k, \tag{7.3}$$

where H is in n/m^2 and k in (watt/m °K). The empirical function f(F) is the line drawn in Figure 7.1.

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* Theoretical data correlations are also given in Sections 4.1.2, 4.1.3, and 4.2 of this report.

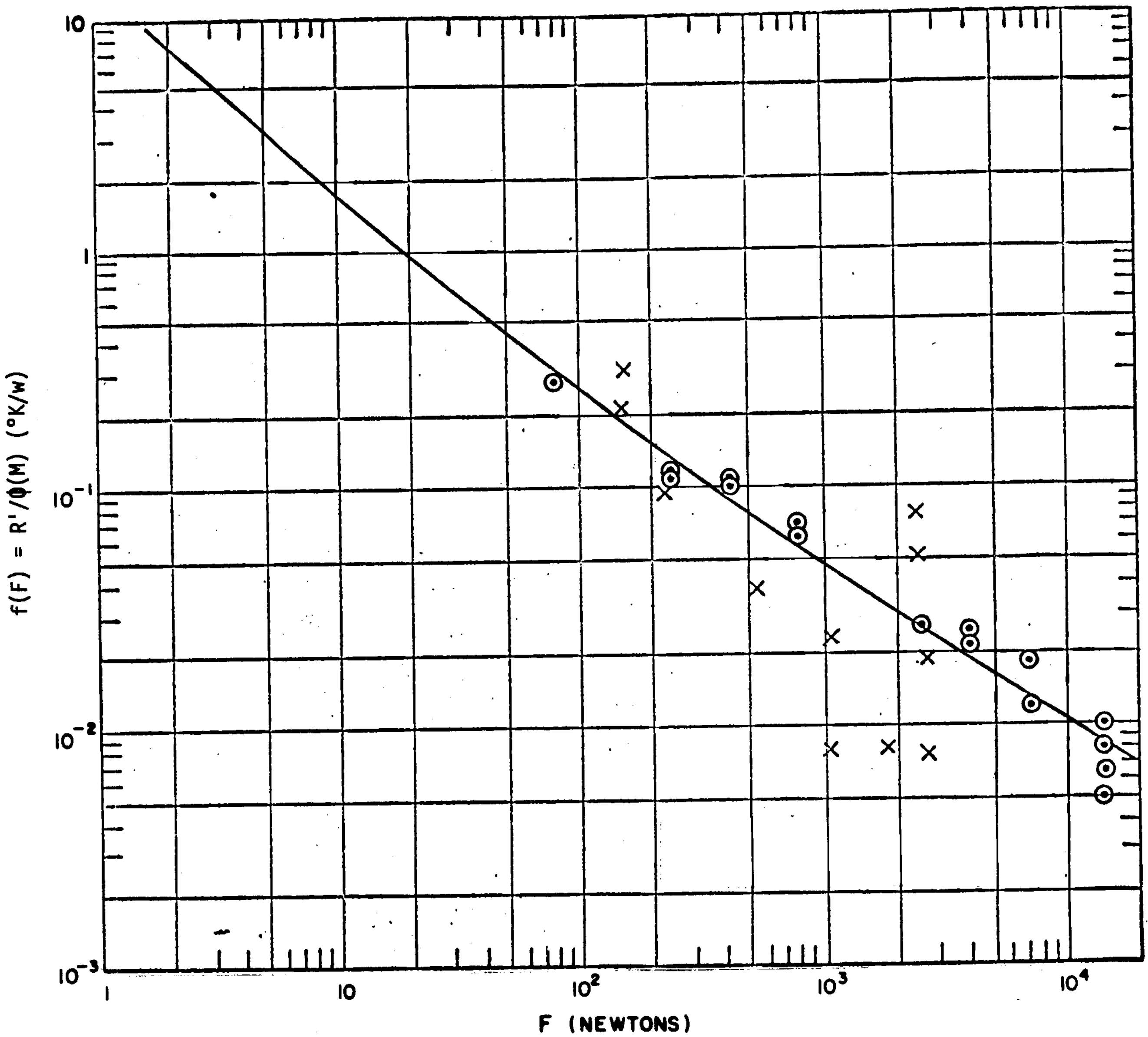

Fig. 7.1 - Empirical Correlation of Thermal Contact Resistance Data According to Holm [55].

F = total force on interface
$f(F) = R/\phi(M)$, see Eq. 7.1.

Holm considered Eq.(7.1) capable of yielding values of thermal contact resistance suitable for normal design applications in a vacuum environment. He indicated that experimental data that deviated substantially from his empirical curve, Figure 7.1, did not correspond to situations likely to be met in practice. In view of the limited amount of data represented in Figure 7.1*, however, the general usefulness of Eq. (7.1), even in practical cases, does not seem to have been established. An attempt by Fried [47] to apply Holm's suggested correlation, with slight modifications, did not prove practical for predicting values of thermal contact conductance.

7.2 Graff

Graff [49a] suggested a design procedure based on a correlation of thermal conductance data in the literature in terms of two non-dimensional groups:

Pressure, p/B

Conductance, $hp/k\rho$

where

B = Brinell hardness number (resistance of material to surface penetration),

h = thermal conductance of joint interface,

k = thermal conductivity of material,

p = apparent contact pressure,

ρ = density of material.

Data from the literature were used to plot graphs of p/B vs. $hp/k\rho$ for ferrous and non-ferrous metals. The graphs enable one to obtain the value of $hp/k\rho$ and to compute h if the value of p/B is known. Other graphs in Ref. 49a enable one to take into account the mean interface temperature and the temperature drop across the interface. Unfortunately,

* See Table 1, Ref. 55.

for purposes of this report, the only data applicable to vacuum conditions in Ref. 49a are for optically flat surfaces of copper, silver, and gold.

Hsieh [58] prepared a well-documented correlation of data available up to 1963 according to Graff's method. Data on magnesium and aluminum alloy joints in vacuum are included in Ref. 58.

7.3 Bloom

Bloom [13] reported measurements of thermal contact conductance for 2-in. diameter specimens of aluminum alloy (7075-T6) and stainless steel (17-4 PH) tested in a vacuum environment at contact pressures up to 1000 lb/in^2. The surface roughnesses ranged between 3 and 135 μin., rms. Using graphs of log h versus log p, he correlated his data and other data available in the literature with equations of the form

$$h \sim p^c \quad ,$$

where h is the thermal contact conductance, p is the apparent contact pressure, and c is a constant. For most of the conditions covered by his plots, the value of c ranged between 0.8 and 0.9.

8. MISCELLANEOUS TOPICS

8.1 Analogy Between Thermal and Electrical Conductivities

Since thermal and electrical conductivities are related by the Wiedemann-Franz law*, many attempts have been made to deduce values of thermal contact conductance from electrical measurements. The analogy between electrical and thermal conduction is discussed in Ref. 56. Although electrical measurements may provide a convenient means of studying the thermal problem under certain conditions [56a], this has usually been found to be an unsuitable approach [9, 10, 11, 18, 21, 34, 39, 48, 52, 66, 105].

Fried [47] attempted to correlate electrical and thermal contact resistances by making both types of measurements on the same specimens in a vacuum environment, but the data did not seem to be amenable to development of a reliable scheme for predicting thermal contact conductance from measurements of electrical resistance. Berman [9] who also made simultaneous measurements of thermal and electrical conductance under vacuum, found wide deviations from the Wiedemann-Franz law. He also discovered that electrical conductance is much more dependent than thermal conductance on the prior history of a joint.

Electrical contact conductance is much more dependent than thermal conductance on oxide films which are usually present. The conduction of electricity through metallic oxides is not due to their intrinsic conductivity [39]. Thin oxide films act as potential barriers which electrons can traverse by a tunnel effect. When the oxide film thickness exceeds approximately 100 A, the film resistance is governed by the intrinsic conductivity, but it is strongly dependent on lattice

* The ratio of thermal to electrical conductivity is proportional to the absolute temperature.

imperfections in the crystals. Although approximate computations of the electrical resistance can be made, accurate values can be obtained only be experimental measurement.

8.2 Trapped Gases

Data reported in Ref. 49 indicate that air might be trapped in joints assembled in a normal atmosphere and subsequently exposed to a vacuum environment. This may have been responsible for the observation of larger thermal conductances for two such samples at low contact pressures, by comparison with conductances observed when the samples were assembled under vacuum.

When air was trapped in the interface, Bloom [13] found that thermal contact conductances were as much as 50 percent greater than the values observed when the procedure prevented such air trapping. Specimens have been outgassed for as long as 60 hours [61] to eliminate the effect of trapped gases from measurements of thermal contact resistance in a vacuum environment. The effects of trapped gases are also discussed briefly in Ref. 46.

The relation between the rate of gas leakage through an interface between metals in contact and surface roughness is examined in Ref. 34. Both theoretical and experimental results are presented showing that the leakage rate is proportional to the square of the mean roughness of the surfaces.

8.3 Mean Interface Temperature

Many investigators [7, 8, 13, 34, 59, 65, and 94] have observed that the thermal conductance increased as the mean temperature of the interface is increased. The increase is partly due to increased radiative heat transfer at higher temperature levels and is usually small.

Berman [9] found that thermal conductance was proportional to T^2 at temperatures of a few degrees Kelvin, regardless of the type of joint (copper/copper, steel/steel, copper/teflon). He found the temperature dependence to be small, however, at temperatures near 70°K.

8.4 Direction of Heat Flow

In some experiments the thermal contact resistance has been found to be dependent upon the direction of heat flow through the interface. Clausing [26], who reviewed the literature on this subject, pointed out that the effect has been a confusing phenomenon: some workers have observed it, others have not, and a number of explanations have been offered.

Clausing states that the directional effect cannot be explained with the *microscopic* model, which assumes a uniform distribution of contact points throughout the interface. His analysis and corroborating experimental work are based on the *macroscopic* model. It was reasoned that local mechanical stresses associated with deformation of asperities are usually high, while the mechanical stresses associated with formation of macroscopic contact areas are usually low. Therefore, it is expected that thermal strain will have little effect on the microscopic contact areas but might appreciably influence the size of macroscopic contact areas and the associated macroscopic constriction resistance. Clausing used a model in which the thermal strain effects result only from temperature gradients parallel to the contact plane, such gradients resulting from the presence of macroscopic constrictions or from heat transfer with the environment through the lateral boundaries of the members in contact. The analysis of the influence of macroscopic constrictions showed that, when dissimilar metals are in contact, the size of the macroscopic contact area tends to increase when heat flows from the metal of larger thermal conductivity to the one of lower thermal conductivity and *vice-versa*. This tendency, which is independent of the surface geometries, caused the thermal contact resistance to depends on the direction of heat flow. The directional effect is small for low heat flux and high thermal conductivity.

Thermal strains resulting from heat transfer with the environment through the lateral boundaries of the members in contact may cause either an increase or decrease in thermal contact resistance, depending on the original surface geometry and the direction of heat flow.

Experiments were conducted with cylindrical specimens having spherical surfaces on the ends placed in contact. The flatness deviations varied from a few microinches to several hundred microinches, and the surface roughness was about 4 μin. Figure 8.1 shows the influence of heat flux, heat flow direction, and contact pressure for a stainless steel/aluminum interface. Similar results were obtained for a stainless steel/magnesium interface. Conditions were such that the macroscopic constriction effect was dominant. At the higher rates of heat flow, reversal of the heat flow direction caused the contact resistance to change by approximately 300 percent.

These results indicate that, even in the absence of hysteresis effects, the relation between thermal contact resistance and contact pressure for a joint between dissimilar metals cannot be represented by a single universal curve, but requires, instead, a family of curves with heat flux as parameter.

The fact that the corresponding dashed and dotted curves in Figure 8.1 have common intercepts on the horizontal axis was interpreted as indicating that the directional effect disappears when there is no heat flow [22]. Actually, if the direction of heat flow from aluminum to stainless steel is considered positive and the opposite direction considered negative, one would expect the data for a given value of the parameter ζ to fall on a single smooth curve. Therefore, we don't really have *two* curves. If the dashed curves in Figure 8.1 are

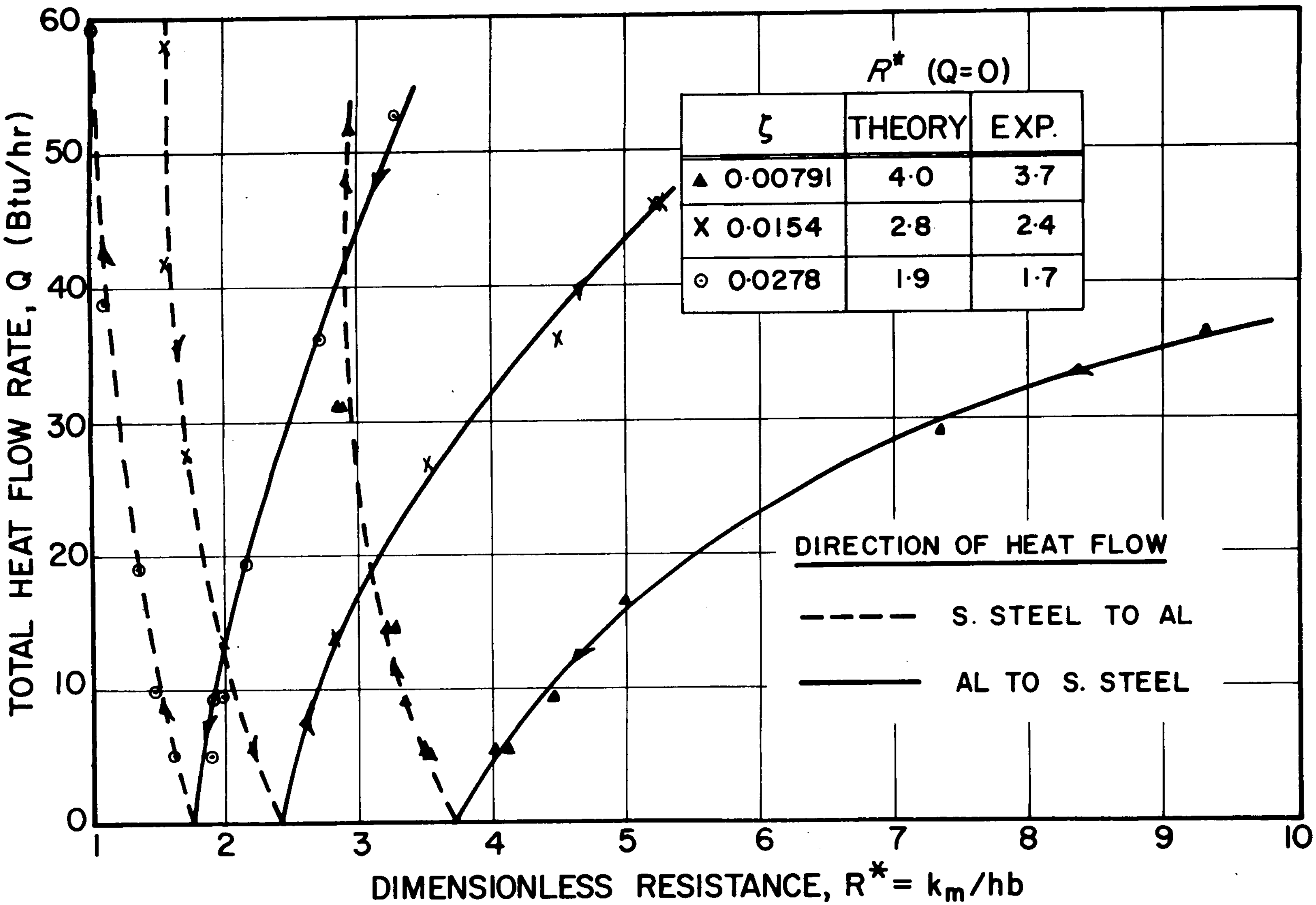

Fig. 8.1 - Influences of Heat Flow Rate, Direction of Heat Flow, and Contact Pressure on Contact Resistance. (From Ref. 23.)

Stainless Steel/Aluminum Interface (d_t = 180 μin.)

$$R^* = k_m/hb = 2\ [10^{f(x)}]$$

$$f(x) = 1.39839 - 7.44698\ x + 19.9303\ x^2 - 38.5897\ x^3 + 38.6553\ x^4 - 16.6247\ x^5.$$

reflected about the horizontal axis, each pair of curves yields a single smooth curve; but if the same is done for the magnesium/stainless steel data in Ref. 26* a change in slope occurs at Q = 0, which is difficult to explain.

The directional trend observed by Clausing is opposite to that observed by other workers for the same materials. This may be a consequence of differences in experimental arrangements. The lack of agreement among observations of directional effects emphasizes the extreme care required in designing apparatus, making measurements, and reducing data in order to obtain meaningful results.

8.5 Hysteresis

It has been observed by many investigators [21, 34, 39, 52, 101] that the values of thermal conductance during unloading of a joint are higher than those during loading. One of the explanations of this phenomenon is that the recovery of deformation during unloading occurs elastically, with the contact area proportional to $F^{2/3}$; whereas the contact area is proportional to the load, F, during the plastic deformation which may dominate the loading condition. The contact area at a given load during unloading will therefore be greater than the contact area at the same load during loading. Another contributing factor may be the formation of cold welds during loading and the adherence at such points during the unloading. The hysteresis effect is also discussed in Refs. 8, 25, 43, and 45.

8.6 Effect of Time

Clausing and Chao [25] reported observing changes of contact resistance with time. In the case of magnesium, a continuous decrease of contact resistance with time was observed, and the effect was attributed

* See Figure 7 of Ref. 26.

to creep. For stainless steel specimens, a cylic variation in thermal resistance observed at very light loads appeared to be caused by thermal strain. They also observed that property changes resulting from aging or annealing of heat treated alloys can cause appreciable change in thermal contact resistance when joints between such materials are operated at elevated temperature. In addition to a hysteresis effect, Stubstad [99] encountered a creep effect when a load was applied for many hours. He pointed out, therefore, that information pertinent to these effects, such as duration of contact and load cycling, should be considered in using experimental contact resistance data for design purposes.

8.7 Transient Effects

Most of the experimental data on thermal contact resistance have been obtained under steady-state conditions. However, some transient effects have been studied [15, 16, 17, 89]: among them, the effect of the rate at which the interstices of an interface are emptied of fluid, the effect of a sudden change in temperature environment, and the effect of a sudden change in thermal contact conductance. Such transients may occur in practice, such as when a heat source is turned on or off and when a space vehicle has its environment changed from that on the earth to that in space. Ref. 16 also discusses the problem of thermal interference among several pieces of equipment attached to a common heat sink and the problem of developing a passive thermal control device involving a variable thermal contact. Transient behavior in the presence of thermal contact conductances are also discussed briefly in Reg. 41.

Schauer and Giedt [89] developed a method for measuring the thermal contact conductance between two plates during the transeint following sudden heating of one plate. They applied the method to several aluminum/stainless steel and stainless steel/alumina contacts. For the metallic contact, the contact conductance increased by about 200 percent in approximately 70 msec following the sudden heating; then it leveled off at values that depend on the initial temperature of the plates and agreed with predicted values within ± 20 percent. The metal/ceramic contacts, however, exhibited a steady decrease in thermal conductance following a rise during the first 10 or 20 msec after the sudden heating. The results established the feasibility of using a capacitive discharge for sudden heating of one plate in contact with another and of using surface mounted thermocouples for obtaining temperature data from which the transient behavior of thermal contact conductance can be computed.

8.8 Surface Films

Oxidation and other types of contamination produce surface films, the influence of which on contact resistance is not well understood. If the films are thin, their effect may be negligible - especially in the presence of a conducting fluid filling the voids of the interface. In a vacuum environment, however, with relatively high heat fluxes through the microscopic contact areas, the effect of surface films may be important. For example, it was found that the interface conductance increased by more than an order of magnitude when magnesium samples on which a visible film had formed were repolished [27]. With a gaseous medium filling the voids, but at comparatively high thermal fluxes, Miller [78] observed a marked increase in the thermal resistance of contacts between fuel element components when an oxide film was present.

Oxide films form very rapidly on metal surfaces, a film 10-15 A thick forming almost immediately [67, p 18]. The films are usually very brittle, particularly after reaching a critical thickness. It has been shown that plastic deformation of a metal considerably increases the rate of oxidation. The rate of film growth decreases with time, ultimately becoming zero. On freshly cut surfaces the ratio of film thickness, δ_o, to thermal conductivity of the oxide, k_o, is small compared to δ/k for the base materials. Also, being much more brittle than the base materials, the oxide film will crumble under a small load at the points of contact. For these reasons, oxide films probably have negligible influence on the thermal conductance between clean surfaces. Thick tenacious films, however, may influence thermal conductance substantially, especially in a vacuum environment. Clausing and Chao [27] mention that sufficiently thick films will ". . . prevent conduction by the quantum mechanical tunnel effect, which otherwise provides a significant contribution to heat flow across gaps of less than about 0.20 A ..." in height. There are also indications that the importance of films increases as the flatness of surface is increased [27].

Fenech and Rohsenow [39] developed a recurrence relation for taking into account the additional thermal resistance due to second-order surface roughness, such as waviness. The same relation can be used to take into account the effect of oxide films [53], but the procedure seems impractical because of the difficulty of measuring the film thickness, δ_o. Furthermore, the likelihood of cracking and crumbling of the oxide film when it is subjected to a load and when the surfaces deform raises questions concerning the definition of film thickness.

Although thin oxide films may not significantly affect thermal contact conductance, they do have a serious influence on electrical conductivity. In fact, measurements of electrical conductivity [38] have been used to indicate the growth of film thickness with time. This is one of the differences between thermal and electrical effects which make it difficult to deduce thermal data from electrical measurements.

S P Carfagno
S. P. Carfagno
Sr. Staff Physicist

for G. Novicki
Z. Zudans
Technical Director
Mechanical & Nuclear
Engineering Department

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APPENDIXES

APPENDIX A

CONVERSION FACTORS FOR PRESSURE AND THERMAL UNITS

Pressure

$$1\ (\text{lb/in}^2) = 6.895 \times 10^4\ (\text{dyne/cm}^2)$$
$$= 6.895\ (\text{kn/m}^2)$$

Thermal Conductance

$$1\ (\text{Btu/hr ft}^2\ {}^\circ\text{F}) = 5.677\ (\text{w/m}^2\ {}^\circ\text{C})$$
$$= 0.3663\ (\text{w/in}^2\ {}^\circ\text{C})$$
$$= 1.356 \times 10^{-4}\ (\text{cal/cm}^2\text{sec}\ {}^\circ\text{C})$$

$$1\ (\text{cal/cm}^2\text{sec}\ {}^\circ\text{C}) = 7372\ (\text{Btu/ft}^2\ \text{hr}\ {}^\circ\text{F})$$

Thermal Resistance

$$1\ (\text{hr ft}^2\ {}^\circ\text{F/Btu}) = 2.730\ (\text{in}^2\ {}^\circ\text{C/w})$$

Thermal Flux

$$1\ (\text{Btu/hr ft}^2) = 1.929 \times 10^{-6}\ (\text{Btu/in}^2\text{sec})$$
$$= 3.154\ (\text{w/m}^2)$$

Appendix B

OTHER LITERATURE REVIEWS

A number of reviews of literature on conductance are available. Experimental measurements in a vacuum, obtained prior to 1963, were reviewed by Fried in Ref. 43. Fried later prepared a more general review [45]. An extensive review was also prepared by Clausing and Chao [25]. A review of computational methods and a compilation of data reported up to 1964 was written by Fontenot [41]. Excellent general reviews were written by Wong [105], Hsieh [58], and more recently by Minges [81]. Ref. 84 is a report of proceedings at a thermal joint conductance conference which included a brief review [100a] of work done or monitored by the Jet Propulsion Laboratory. Another brief review is given in Ref. 14.

APPENDIX C

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