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Report No. 76361-43
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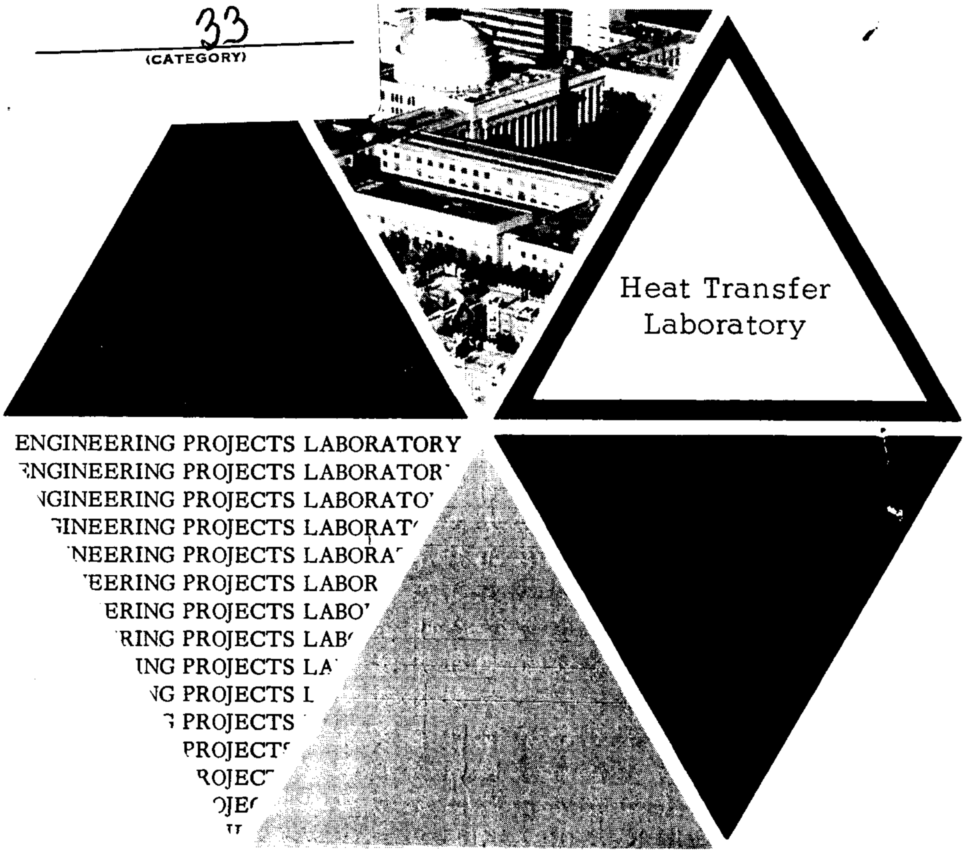
Department of Mechanical Engineering

Massachusetts Institute of Technology



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and

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A B S T R A C T

A thermal contact conductance equation was developed which considers both the effect of surface roughness and waviness. It was shown that the overall thermal contact conductance is determined by the roughness at large contact pressures or rough surfaces. It is also shown that surface roughness increases the contour radius over that predicted by the theory of Hertz. The surface roughness influences the magnitude of the waviness resistances by spreading the load at the contact over a larger region. The theory was seen to be in very good agreement with experimental data.

ACKNOWLEDGMENT

The authors wish to express their sincere appreciation for the financial support of the National Aeronautics and Space Administration under Research Grant NGR-22-009-065.

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NOMENCLATURE

a	microcontact radius
A	area
b	radius of elemental heat channel
d	out of flatness
D	diameter of macroscopic heat channel
E	modulus of elasticity
h	thermal conductance
H	material hardness
k	thermal conductivity
L	distance (pitch) between waves
n	contact spot density
N	number of contacts
Q	heat flow rate
R	thermal resistance
T	temperature
Y_o	yield stress
r,z	coordinates

<u>Greek symbols</u>

ε	$\equiv \sqrt{A_r/A_a}$
γ	factor
ρ	radius of curvature
λ	$\equiv D/L$
σ	rms value

Subscripts

1	metal 1
2	metal 2
a	apparent
c	contour
eff	effective
H	Hertzian
r	real
s	harmonic mean
t	total

INTRODUCTION

This report describes the analytic and experimental work conducted at the Heat Transfer Laboratory of the Mechanical Engineering Department at M.I.T., in evaluation of the thermal contact conductance (reciprocal of resistance) between wavy, rough surfaces placed in a vacuum. This work was a portion of a comprehensive program[1] to make possible a theory for the prediction of thermal contact resistance across interfaces formed between metal surfaces. Thermal contact conductance becomes a major consideration whenever heat transfer between touching surfaces must be accomplished in the absence of a conducting fluid. The most prominent areas of the application of the theory include space vehicles with their environmental control subsystems, space vehicle energy conversion devices, as well as space environmental-simulation chambers.

STATEMENT OF THE PROBLEM

All "worked" surfaces exhibit waviness and roughness. These surface characteristics are determined by means of profilometers[2,3] and Fig. 1 shows a typical linear profile of a wavy, rough surface. These surface irregularities are the result of the inherent action of production processes, machine or work deflections, vibrations and warping strains. The surface irregularities with the large wavelength are termed waviness. In addition to these, most surfaces exhibit finely spaced roughness that is superimposed on the waviness and is responsible for the finish of the piece. In general, the longer waves cannot be seen by either eye or microscopic examination. They may, however, play a controlling part in the behavior of the interface formed by two such surfaces.

When two clean metallic surfaces are placed in contact with each other, the heat transfer between them can only be accomplished by the presence of a temperature drop across the interface. This temperature drop is due to the additional resistance to heat flow across the contact. In the absence of a conducting fluid (vacuum conditions), the heat flow is confined to the real contact area, i.e. the heat is conducted across the interface through the contacting asperities. The thermal resistance can be thought of as the convergence of the heat flow lines by the contour area and then a pinching effect due to the contacting asperities. The contour area is determined by both the waviness and roughness of the surface. It will be shown that the Hertzian contour area must be modified due to the presence of the surface roughness. This effect is significant when the roughness is large or when the applied load is small.

In order to solve analytically the heat conduction problem between contacting metallic surfaces, the following model has been adopted. It is assumed that all microcontacts are uniformly distributed inside the contour area. Furthermore, all contact spots have the same average area of contact, circular in shape with an average radius a, Fig. 2. From the above it readily follows that inside the contour area there exist a number of identical heat channels. The density of contact spots will depend upon the surface roughness, the material properties and the applied load. In addition, for the contact in a vacuum, the contacting surface for each heat channel is considered to be flat. The last assumption is justified by the fact that surface irregularities usually have a very gentle slope[2,3]. One half of the elemental heat channel is shown in Fig. 2.

The shape of the contour area, specified by the type of surface waviness, is assumed to be circular for spherical waviness.

Finally, it is assumed that the surfaces in contact are free from any kind of film and consequently, the whole problem of thermal contact resistance is treated as the constriction phenomenon only, i.e. as the effect of constriction of the heat flow lines due to the influence of waviness and roughness.

ANALYTIC SOLUTION FOR AN ELEMENTAL HEAT CHANNEL IN A VACUUM

For the proposed thermal model, the temperature distribution and implicitly the thermal contact resistance is specified by the Laplace differential equation (for steady state conditions and thermal conductivity independent of temperature)

$$\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\,\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} = 0 \tag{1}$$

and the following boundary conditions:

$$\left.\begin{array}{llll} T = \text{constant} & \text{at} & z = 0 & 0 < r < a \\ -k\,\dfrac{\partial T}{\partial z} = 0 & \text{at} & z = 0 & a < r < b \end{array}\right\} \tag{2}$$

$$-k\,\frac{\partial T}{\partial z} = \frac{Q}{\pi b^2} \qquad z \to \infty \tag{3}$$

$$-k\,\frac{\partial T}{\partial r} = 0 \qquad \text{at} \quad r = b \tag{4}$$

$$-k\,\frac{\partial T}{\partial r} = 0 \qquad \text{at} \quad r = 0 \tag{5}$$

where Q is the quantity of heat flowing through the model per unit time and k is the thermal conductivity of the material of the heat channel.

The thermal resistance following the electrical analog is

$$R = \frac{\Delta T}{Q} \tag{6}$$

where ΔT is the extrapolated temperature difference at the interface and Q is the heat flow per unit time across the interface.

The term thermal contact conductance, when used, will represent the reciprocal of the contact resistance.

The solution to the above thermal problem is obtained and discussed in detail in reference (1).

The thermal contact resistance per elemental heat channel is found to be

$$R = \frac{4}{\pi k a} \phi \left(\frac{a}{b}\right) \tag{7}$$

where k is the thermal conductivity, a is the radius of contact for the heat channel and $\phi(\frac{a}{b})$ is a geometric parameter which depends upon the ratio of the contact radius to the heat channel radius.

Figure 3 shows values of the contact resistance factor $\phi(\frac{a}{b})$ based on several different boundary conditions:

(1) $\phi_1(\frac{a}{b})$ is based upon a parabolic heat flux over the contact area;

(2) $\phi_2(\frac{a}{b})$ is the result of considering the temperature field obtained by superposition of an infinite number of sources equally spaced on the surface z=0;

(3) $\phi_3(\frac{a}{b})$ is a linearized form of $\phi_2(\frac{a}{b})$ and is a good approximation for values of $0 < \frac{a}{b} \leq 0.6$;

(4) $\phi_4(\frac{a}{b})$ is based upon a constant heat flux over the contact area.

The case when the condition of constant heat flux prevails over the contact area has been considered for two reasons: (i) since the constant heat flux imposes a higher constriction of heat flow than the constant temperature condition over the contact area, the former should always yield the higher thermal contact resistance and could serve as an upper bound for the previous solutions; and (ii) in certain cases, for example macroscopic constriction due to the waviness effect, the condition over the contour area depends upon the contact spot distribution inside the contour area and therefore the actual situation over the contour area may approach that of the constant heat flux.

The expression for the thermal contact resistance of the form $R = (4/\pi ka)\ \phi(\frac{a}{b})$ represents the constriction resistance for one half of the elemental heat channel. The total resistance for N heat channels acting as parallel thermal resistors is

$$R_t = \frac{8}{N\pi k_s a}\ \phi(\frac{a}{b}) \tag{8}$$

where

$$k_s \equiv \frac{2k_1 k_2}{k_1+k_2}$$

The total thermal contact resistance per unit apparent area is

$$R_t A_a = \frac{1}{h} = \frac{8\phi(\varepsilon)}{\sqrt{\pi}\ k_s\ \sqrt{n}\ \varepsilon} \tag{9}$$

where n is the contact spot density and is determined by the definition of the thermal model $n = 1/\pi b^2$ and $\varepsilon \equiv a/b = (A_r/A_c)^{1/2}$

For values of $\varepsilon \equiv a/b \leq 0.60$, an approximation for $\phi(\varepsilon)$ is given by

$$\phi(\varepsilon) \simeq \frac{\pi}{16} - \frac{\varepsilon}{4} \tag{10}$$

otherwise values of $\phi(\varepsilon)$ may be obtained from Figure 3.

ANALYTIC SOLUTION FOR SPHERICAL WAVINESS

The thermal model for macroscopic heat channels will be geometrically similar to the elemental heat channel, and all expressions obtained for the latter are applicable. The parameter $\varepsilon = {}^{a}/b$ will be replaced by the parameter ${}^{D}/L$, where D is the diameter of the contour area and L is the wave length of the spherical waves.

It follows directly that the expression for the thermal contact resistance per unit apparent area due to spherical waviness is given by

$$R_w = \frac{4L \ \phi({}^{D}/L)}{k_s \ ({}^{D}/L)} \tag{11}$$

The values for $\phi({}^{D}/L)$ for different L/D can be found from Fig. 3. (formally taking ${}^{D}/L = {}^{a}/b$).

SURFACE DEFORMATION ANALYSIS

The thermal contact resistance has been expressed in terms of ε, the contour area and the wavelength through some surface characteristics and material properties. Next it will be necessary to relate ε to the pressure over the contour area and to determine the contour area as a function of the apparent pressure for the case of rough spherically wavy surfaces.

Since ε depends upon the contact spot density and radius, and these parameters are dependent upon the apparent pressure at the interface, it should be expected that local ε will depend upon the local apparent pressure. The apparent pressure over the contour area is a maximum at the center of the contour decreasing with the radius, finally vanishing at the edge of the contour. One would therefore expect ε to be a maximum at the center of the contour area, vanishing at the edge of the contour.

It would be expected that where ε is a maximum, plastic deformation prevails, while where ε is a minimum, elastic deformation prevails.

ACTUAL CONTACT AREA

A complete analysis for estimating the real contact area when two rough non-wavy surfaces are brought into contact appears in reference (1). The analysis is essentially based on a model which assumes that each contact spot consists of two hemispherical asperities in symmetric contact, Fig. 4a.

The result of the analysis can be expressed by the following relationship

$$\varepsilon^2 \equiv \frac{A_c}{A_a} = \gamma \frac{P_a}{3Y_o} = \gamma \frac{P_a}{H} \tag{12}$$

where γ is a function of the material properties of the contacting bodies, the applied load and the geometry of the surfaces in contact.

Since the slope of the asperities is less than 10°, and the applied load on the interface always exceeded 130 psi, therefore the value of γ is very close to unity, and it is permissible to use the relation

$$\varepsilon^2 = P_a/H \tag{13}$$

CONTOUR AREA FOR SPHERICALLY WAVY SURFACES IN CONTACT

The model for spherical waviness, where only the mean line of the surface is presented, is shown in Fig. 4b. It is assumed that the waviness is not too pronounced, i.e. $^d/L \ll 1$. As a consequence of the above, the radius of curvature is expressed as

$$\rho \approx L^2/8d \tag{14}$$

The height above the mean plane d will be called the flatness deviation and L the wavelength between spherical waves. For two such surfaces in contact, one can determine, by applying the Hertz theory, how the contour area (for smooth surfaces) varies with the applied load. The final result can be written in the form

$$\lambda_H \equiv \frac{D}{L} = 1.285 \left[\left(\frac{P_a}{E_s}\right)\left(\frac{L}{2d_t}\right) \right]^{1/3} \tag{15}$$

where D is the diameter of the contour area, $d_t = d_1 + d_2$, $E_s \equiv (2E_1E_2)/(E_1 + E_2)$ and E_1 and E_2 are the respective moduli of elasticity for the surfaces in contact.

If the surfaces in contact are in addition rough, one can anticipate that the actual contour area will extend beyond the contour area predicted by the Hertz theory. Since the pressure over the contour area is a maximum at the center and decreases with increasing radius, it is expected that the contact spot density will also decrease with increasing radius, being maximum in the region around the center of contact.

In order to make the relations, based on the model which assumes uniform distribution of contacts within the contour area, useful, we define here the effective contour area to be that area which would contain all the contact spots if they had been uniformly distributed inside this area.

Using the definition given above, and the assumption that the mean plane is deformed elastically according to the Hertz theory, the effective contour area was related to the Hertz contour area. The complete analysis appears in reference (1) and only the final result is given here

$$\lambda^2_{eff} = \lambda^2_H + 2 \int_{\lambda_H}^{1} \exp \left\{ - \frac{d_t}{\sigma} \lambda^2_H \, g(\frac{\lambda}{\lambda_H}) \left[2 \frac{Y}{\sigma} + \frac{d_t}{\sigma} \lambda^2_H \, g(\frac{\lambda}{\lambda_H}) \right] \right\} \lambda d\lambda \quad (16)$$

where $\lambda_{eff} \equiv \frac{D_{eff}}{L}$ and $\lambda \equiv \frac{2r}{L}$ and

$$g(\frac{\lambda}{\lambda_H}) \equiv (\frac{\lambda}{\lambda_H})^2 - 2 \left\{ 1 - \frac{1}{\pi} \left[(2 - \frac{\lambda^2}{\lambda^2_H}) \sin^{-1}(\frac{\lambda_H}{\lambda}) + (\frac{\lambda^2}{\lambda^2_H} - 1)^{1/2} \right] \right\} \quad (17)$$

is the waviness factor which is presented graphically in Fig. 5.

Since $(^Y/\sigma)$ in Eq. (16) is a function of ε where ε is given by

$$\varepsilon = \frac{1}{\lambda_{eff}} (\frac{P_a}{H})^{1/2} \quad (18)$$

it is obvious that the process of calculating λ_{eff} is an iterative process. However, from the known value λ_H and some experience, one can make a good estimation of $^Y/\sigma$ in the first step, so that only one calculation of λ_{eff} is necessary.

TOTAL THERMAL CONTACT RESISTANCE EQUATION

The analytic section will be concluded by outlining the procedure for the prediction of the total thermal contact resistance across the interface formed by two rough and spherically wavy surfaces in a vacuum environment.

Since the roughness and waviness resistances are in series (i.e. the roughness has negligible effect on the temperature distribution) from Eqs. (9) and (11) it follows that

$$(RA_a) = \frac{1}{h} = \frac{8\phi(\varepsilon)}{\sqrt{\pi}\, k_s\, \sqrt{n}\, \varepsilon} + \frac{4\, L\, \phi(\lambda_{eff})}{k_s\, \lambda_{eff}} \tag{19}$$

where the first term is the resistance due to the surface roughness and the second term represents the contribution due to the spherical waviness. The second term has been corrected for the effect of roughness upon the size of the contour area.

EXPERIMENTAL JUSTIFICATION OF THE THEORY

The analysis developed earlier in this work on thermal contact resistance made use of some approximations which we summarize.

1. The total contact resistance is the result of surface roughness (pinching of heat flow lines) and surface waviness (restrict the heat flow to the contour area in the absence of conducting fluid).
2. The contact spots are assumed to be uniformly distributed over the contour area and they have an average radius, a.
3. The elemental heat channel (in the absence of a conducting fluid) consists of a circular flat on the end of a circular cylinder.
4. The effect of the roughness resistance does not extend into the waviness region.
5. The contour area is assumed to be the result of two spherical waves interacting elastically under an applied load.
6. The thermal model for the waviness is assumed to be similar in shape to the elemental model.
7. It is assumed that roughness has the effect of increasing the size of the contour area over that predicted by the Hertz theory.

A complete description of the experimental apparatus and the test procedure is given in Ref.(1). The pertinent test results are shown in Figs. 6, 7 and 8 where the overall thermal contact conductance is shown plotted against the load on the interface. The present theory on the overall thermal contact resistance was used to predict the total resistance between three pairs of stainless steel 303 specimens having rough and wavy surfaces. The surface description of the specimens is shown in Table 1.

TABLE I

Pair 1	Specimen 1	σ_1 = 190 μin, d_1 = 95 μin, $\tan\theta_1$ = 0.150
	Specimen 2	σ_2 = negligible, d_2 = 55 μin, $\tan\theta_2$ = 0
Pair 2	Specimen 1	σ_1 = 132 μin, d_1 = 80 μin, $\tan\theta_1$ = 0.163
	Specimen 2	σ_2 = 76 μin, d_2 = 0 μin, $\tan\theta_2$ = 0.137
Pair 3	Specimen 1	σ_1 = 292 μin, d_1 = 80 μin, $\tan\theta_1$ = 0.100
	Specimen 2	σ_2 = 174 μin, d_2 = 35 μin, $\tan\theta_2$ = 0.100

The theoretical curves shown plotted on Figs. 6, 7 and 8 are based upon Eq. (19). The value of $\tan\theta$ appearing in the figures is taken to be the larger of the two values for each pair of specimens. Since $\nu \equiv \lambda_{eff}/\lambda_H$ depends upon the applied pressure, an average value was used for the pressure range from 131 psi to the pressure at which $\nu = 1$.

For comparison, the wavy conductance curve based upon $\nu = 1$, i.e. when the contour area is assumed to be the same as that predicted by the Hertz theory for smooth surfaces, is presented for each pair of specimens.

The material properties in all cases were the same, i.e., Hardness H = 370,000 psi, Young's modulus of elasticity $E = 26 \times 10^6$ psi and thermal conductivity k = 10 BTU/hr ft°F.

DISCUSSION OF TEST RESULTS

The theoretical conductance, Eq. (19) considers the effect of surface roughness and waviness. It is seen (Figs. 6, 7 and 8) that this equation agrees quite well with the vacuum test data over a wide applied load range. The effect of waviness upon the total conductance is quite significant at the low interface pressures where the convergence of the heat flow lines is greater than the pinching effect of the contacting asperities.

As the load on the interface increases, the waviness effect becomes less important and the roughness effect dominates the overall conductance. The pressure for a given pair of surfaces, at which the waviness effect is negligible, depends upon the elastic properties of the surfaces, the magnitude of the flatness deviation and the roughness of the surfaces.

When the effect of surface roughness is completely ignored, (Clausing and Chao[4])* the total conductance is dependent solely on the effect of the contour area. This theory is seen to agree with test data for lightly loaded interfaces and predicts a thermal contact conductance order of magnitude larger than test data for pressures exceeding contact pressures of 1000 psi, Figs. 6, 7 and 8.

When the effect of surface waviness is completely ignored, the total conductance is dependent solely on the pinching effect of the contacting asperities. The discrepancy between theory and test data

*Clausing had assumed that "the average size of microcontacts is the same order of magnitude as the surface roughness", pp 39 & 58 Ref.4. It has been observed by many investigators that the microcontact diameter ranges between 1 micron for pressures of 1psi to about 40 microns for pressures of 5000psi, Ref. 8. These diameters have been observed for materials such as copper, aluminum and stainless steel.

just appears at contact pressures below 1000 psi, and becomes significant for contact pressures below 100 psi, Figs. 6, 7 and 8.

The spreading effect (i.e. increasing the contour area) of the surface roughness is also seen in the comparison between the two curves labelled $\nu = 1.0$ and $\nu = 1.6$. The effect is not as dramatic as it would be if the surface roughness were smaller and/or the waviness larger.

roughness is important for the higher contact pressures. The pressure at which the transition occurs (or pressure range) is dependent upon the surface roughness, waviness, material properties and the applied load.

It is recommended that further work be done to determine the effect of surface roughness and waviness upon the overall thermal contact conductance. It is important that the effect of non-uniform distribution of contact spots over the contour area be examined more closely. Along this line it is necessary to determine whether the roughness resistance alters the temperature field significantly to alter the thermal model used for calculating the waviness resistance. It is desirable to have one equation which can predict the contour radius taking into effect the surface roughness, waviness, material properties and the contact pressure.

CONCLUSIONS AND RECOMMENDATIONS

A thermal contact conductance equation was developed which considers both the effect of surface roughness and waviness. The theory is in good agreement with limited experiment results. Neglecting the effect of surface roughness (Clausing and Chao[4]) results in a conductance equation which yields extremely large values for all contact pressures greater than 1000 psi, i.e. predicted values of conduction orders magnitude larger than measured values. Bloom[7], in a very extensive report showed very clearly that a conductance theory based solely on surface waviness is quite inadequate in predicting conductance for large pressures. He found that the macroscopic theory "tended to predict much larger values of h than data when more than 42% of the total apparent area was in macroscopic contact". This would correspond to a value of $\lambda_{eff} = 0.65$.

Neglecting the effect of surface waviness results in a conductance equation which gives values of h lower than observed for low pressures. Fried, in two reports[5,6], obtained experimental data for many contacting surfaces. All surfaces tested by Fried exhibited both roughness and waviness characteristics. He found that "when h vs. P_a was plotted on log-log paper, a definite two-regime behavior with a pronounced point of change in slope was observed for most of the test results". The slope of the best curve through his data was $\sim 2/3$ for pressures between 5 and 150 psi and then changed to ~ 1.0 for pressures exceeding 150 psi. The change in slope that Fried observed is not unlike the two regimes shown in Figs. 6, 7 and 8. The two regimes may be thought of as the waviness dominated regime and the roughness dominated regime. Waviness being important for the light contact pressures, while the

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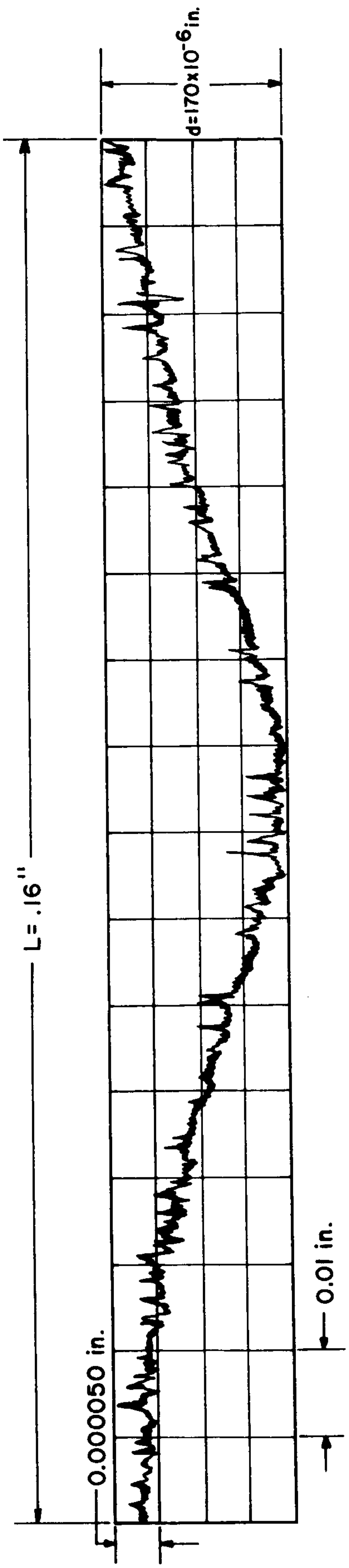


FIG. Ia TYPICAL TRACE SHOWING WAVINESS

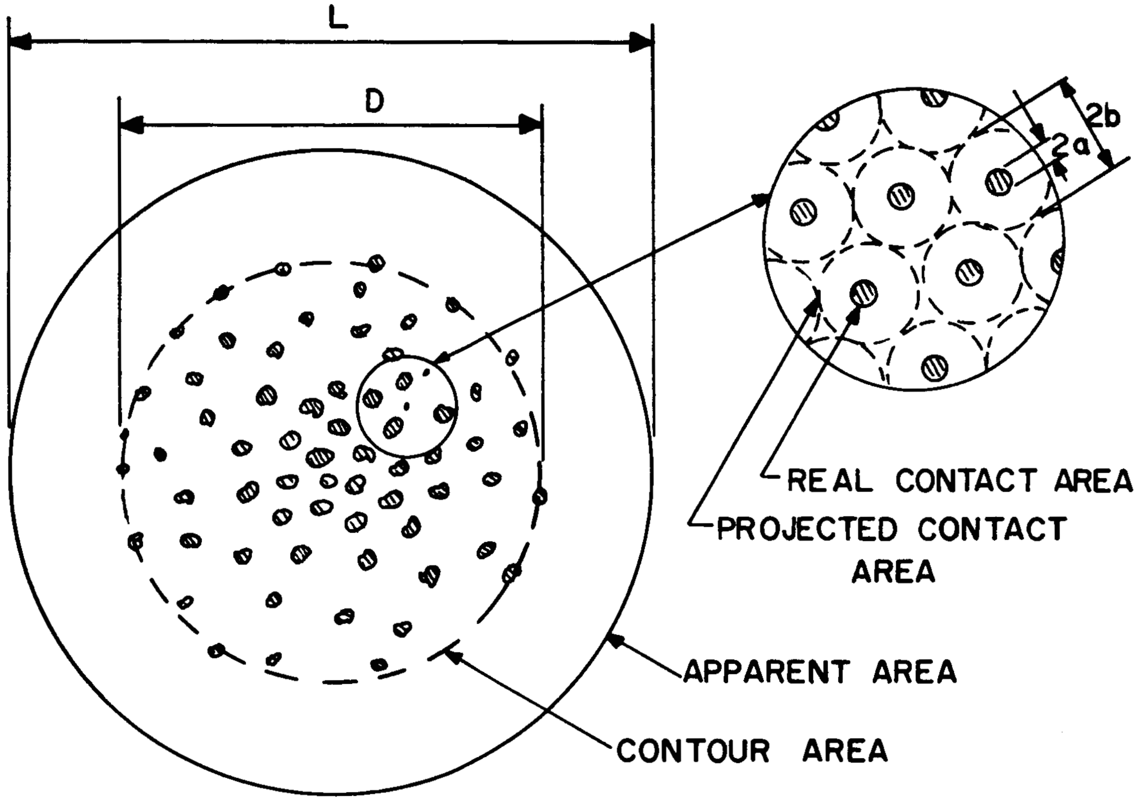


FIG. 1b SPHERICAL CONTACT MODEL

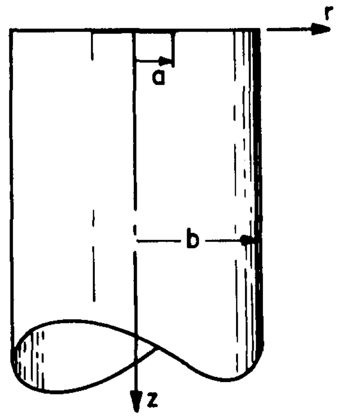


FIG. 2 MODEL OF AN ELEMENTAL HEAT CHANNEL

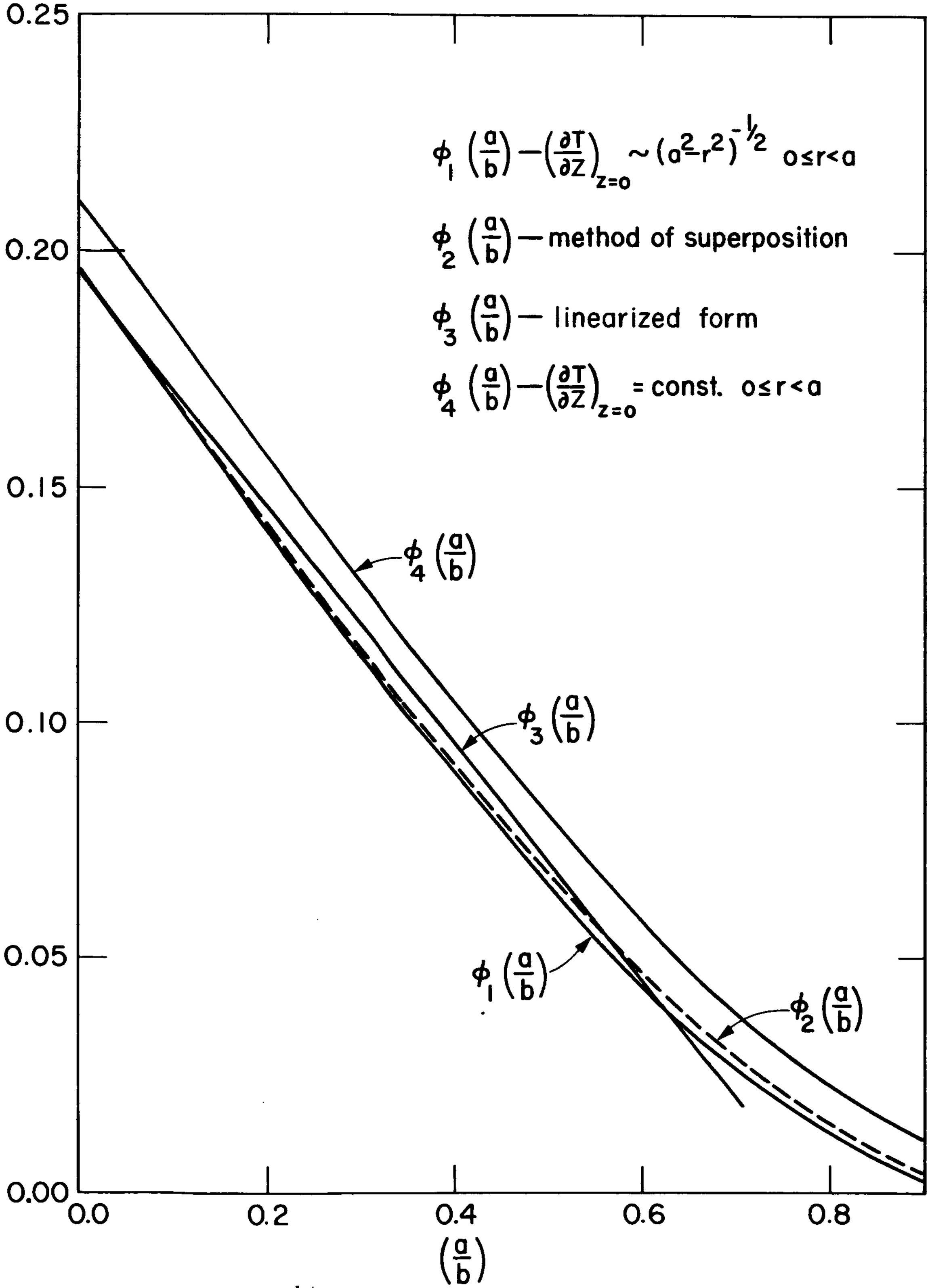
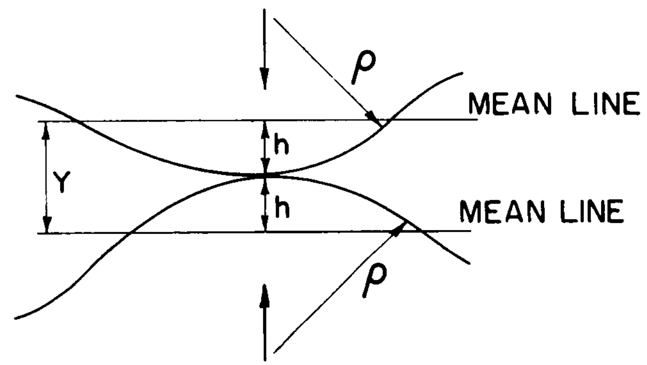
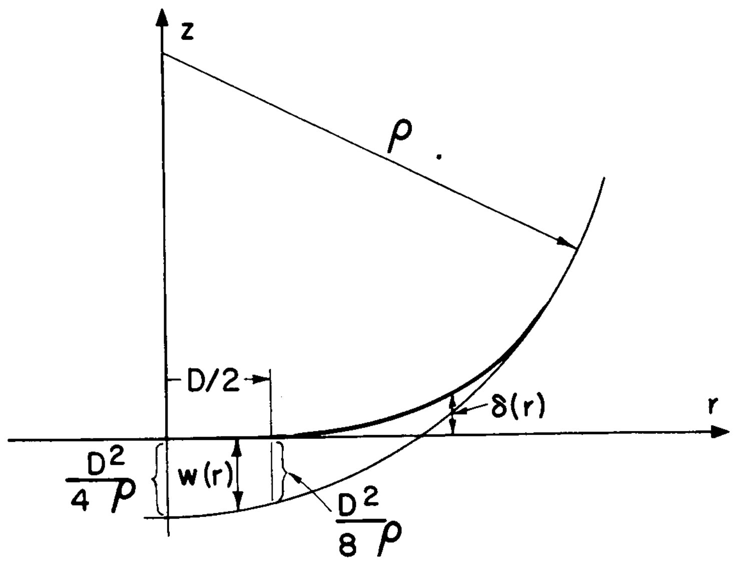


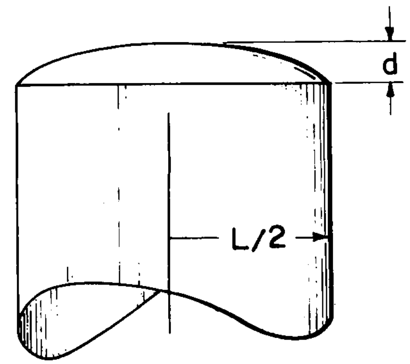
FIG. 3 CONTACT RESISTANCE FACTOR



a) MODEL FOR TWO ASPERITIES IN CONTACT



b) LOCAL DEFORMATION OF A SPHERE PRESSED AGAINST A RIGID PLANE



c) SPECIMEN FLATNESS DEVIATION

FIG. 4 MODELS USED IN DEFORMATION ANALYSIS

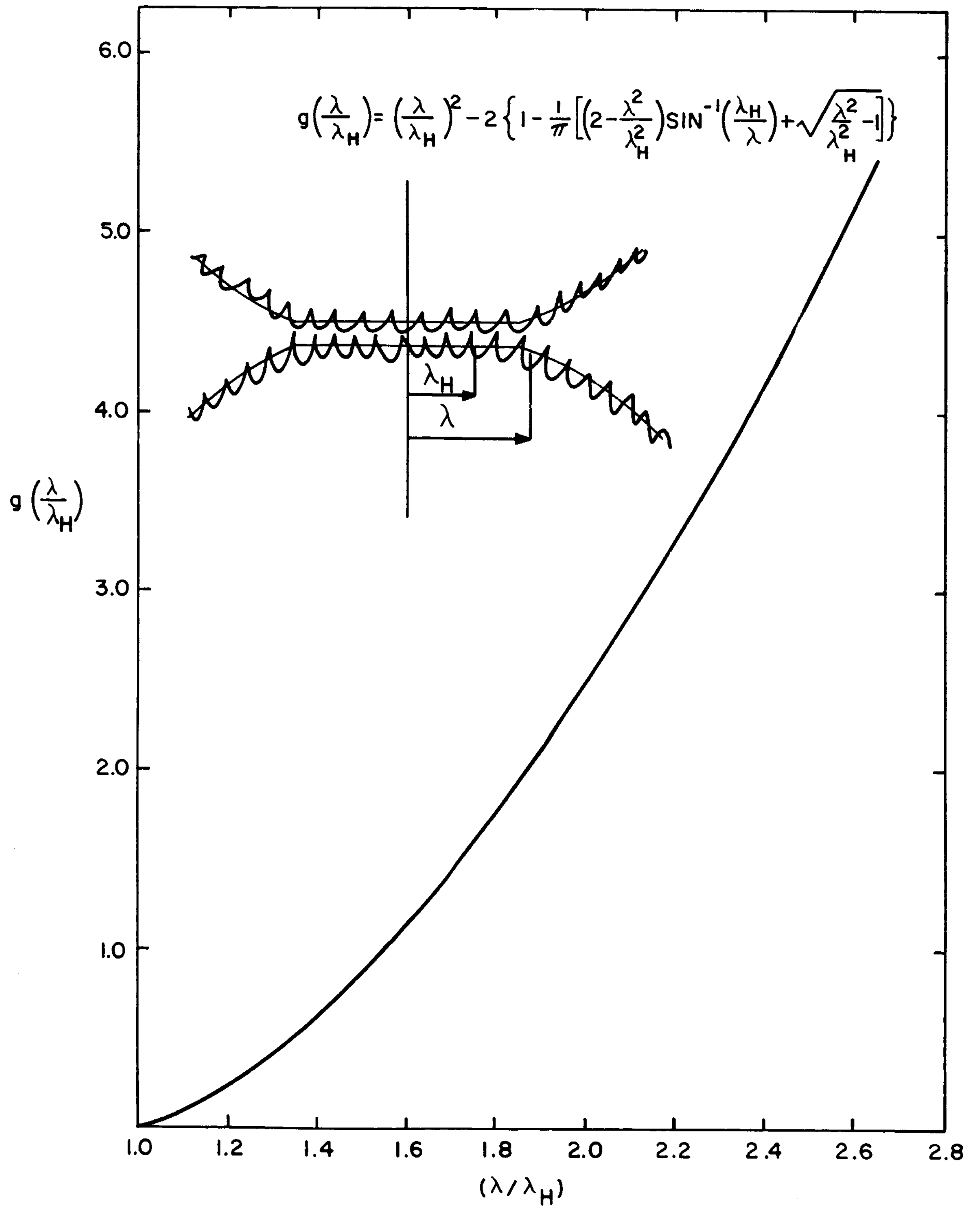


FIG. 5 WAVINESS FACTOR

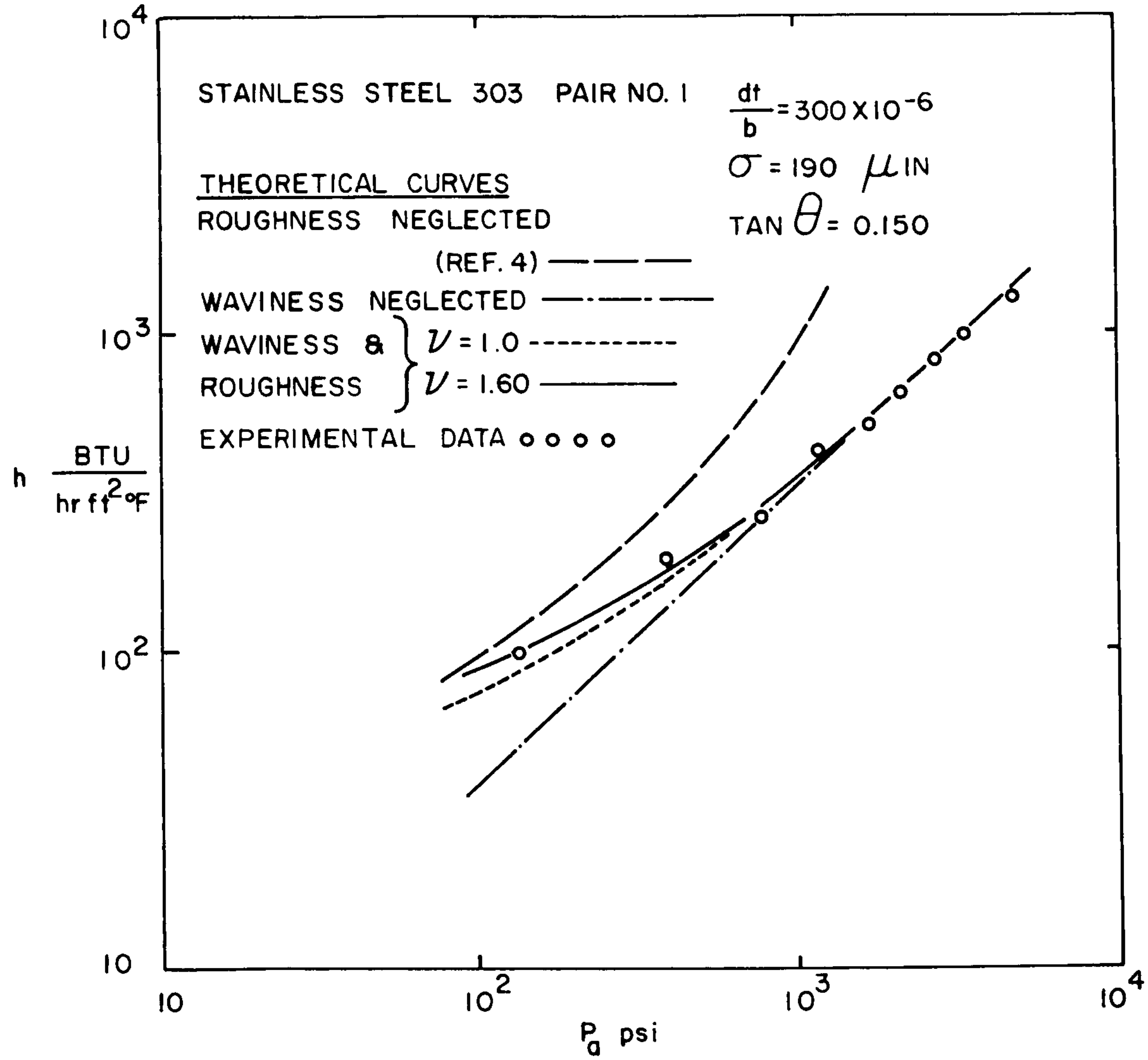


FIG. 6 CONTACT CONDUCTANCE VS APPARENT PRESSURE

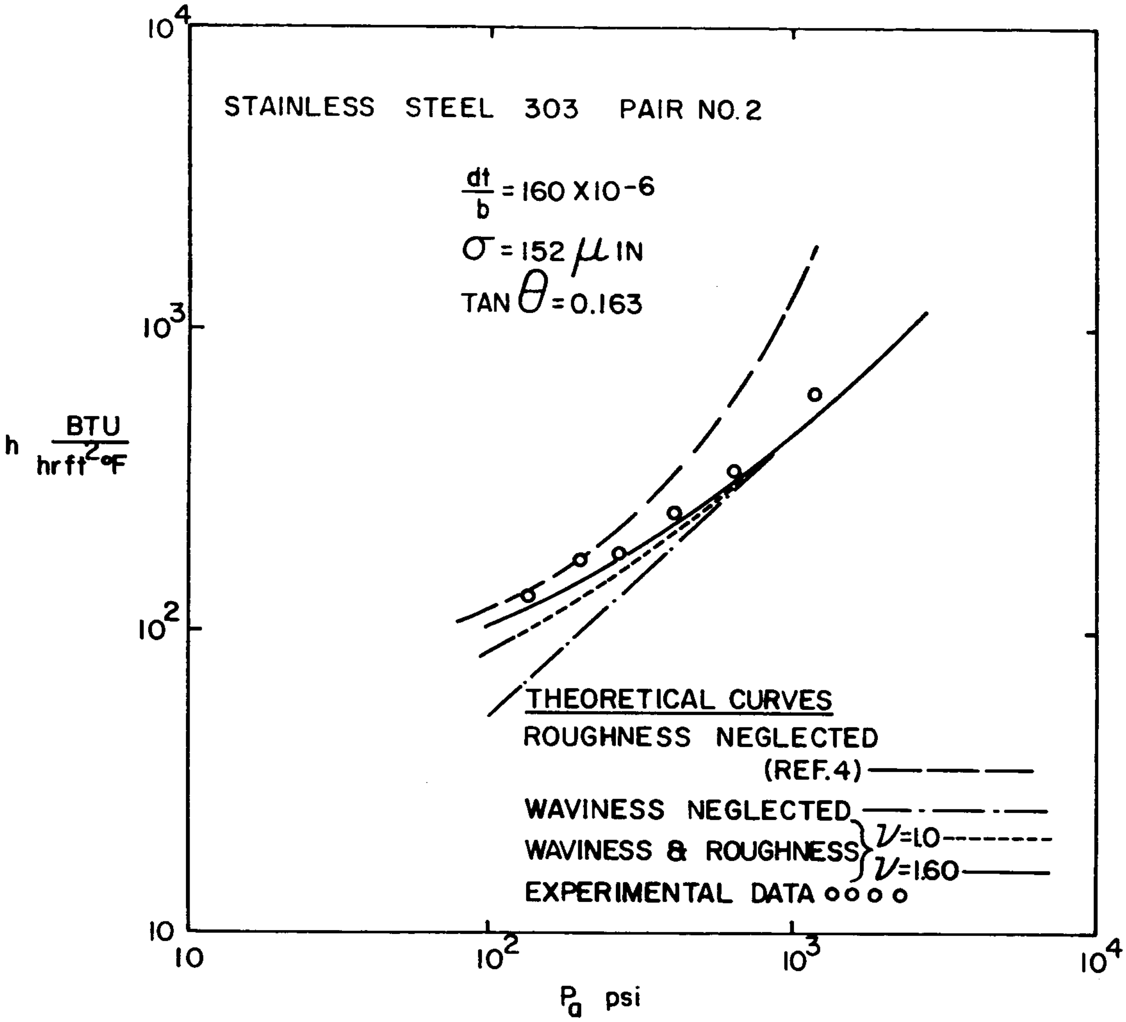


FIG. 7 CONTACT CONDUCTANCE VS. APPARENT PRESSURE

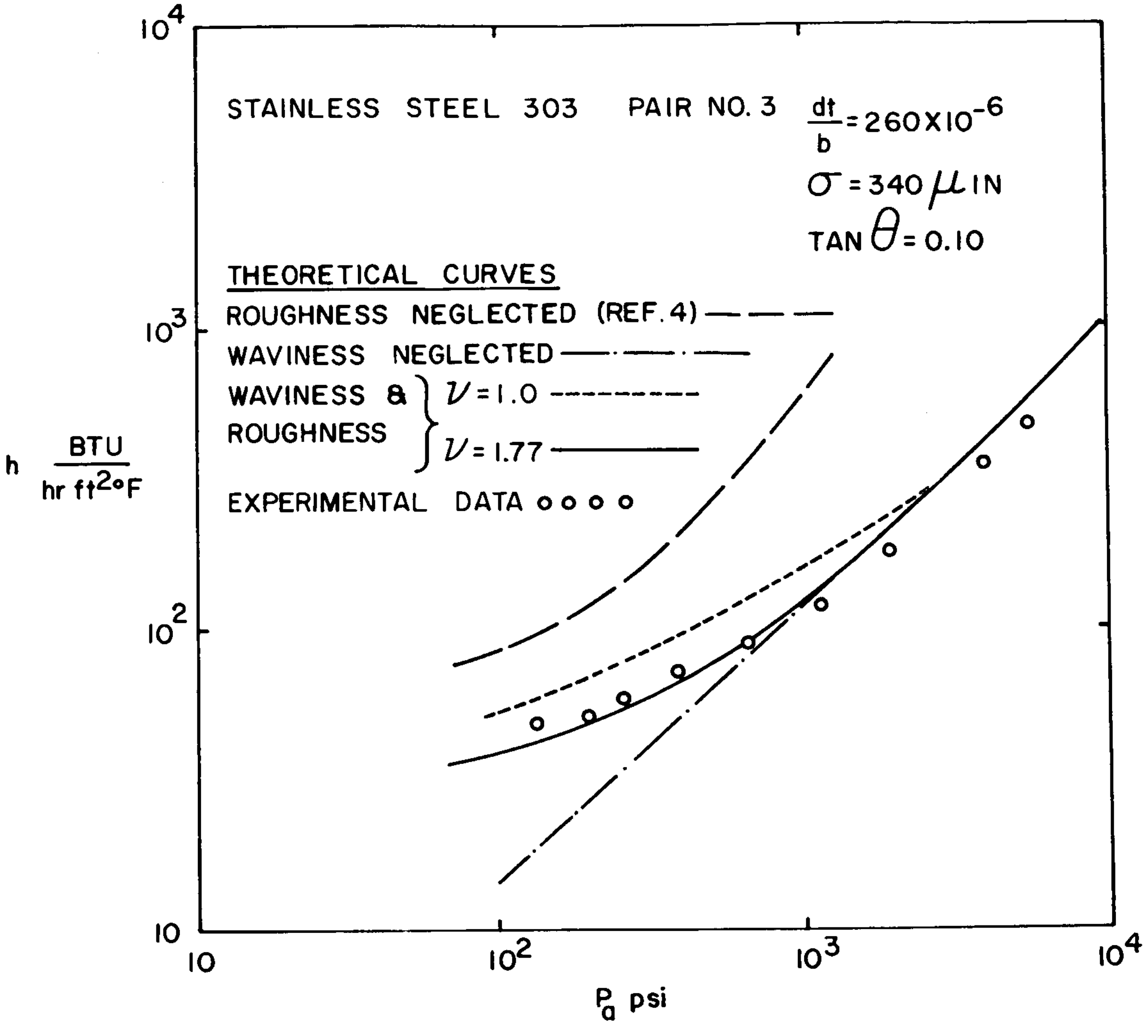


FIG. 8 CONTACT CONDUCTANCE VS. APPARENT PRESSURE