STATISTICAL STUDY OF THERMAL CONTACT CONDUCTANCE

by

T. Nejat Veziroglu
Professor of Mechanical Engineering

June 1972

NASA GRANT NGR 10-007-010
REPORT

MECHANICAL ENGINEERING DEPARTMENT
UNIVERSITY OF MIAMI
CORAL GABLES, FLORIDA

STATISTICAL STUDY OF THERMAL CONTACT CONDUCTANCE

by

T. Nejat Veziroglu
Professor of Mechanical Engineering

June 1972

NASA GRANT NGR 10-007-010
REPORT

MECHANICAL ENGINEERING DEPARTMENT
UNIVERSITY OF MIAMI
CORAL GABLES, FLORIDA

"STATISTICAL STUDY OF THERMAL CONTACT CONDUCTANCE"

by

T. Nejat Veziroglu
Professor of Mechanical Engineering
University of Miami, Coral Gables, Florida

I. INTRODUCTION

When two metal surfaces are brought into contact, they actually touch only at a limited number of spots, the aggregate area of which is usually only a small fraction of the apparent contact area. The remainder of the space between the surfaces may be filled with air or other fluid, or may be in vacuum. When heat flows from one metal to the other, flow lines converge toward the actual contact spots since the thermal conductivities of metals are so much greater than those of fluids. This converging of flow lines causes the contact resistance, which is usually high compared to the resistances offered to heat flow by metals. It is therefore becoming increasingly necessary, especially as the need for more reliable equipment grows in aerospace, nuclear, and other industries, for designers to be able to predict thermal resistance or conductance of contacts with greater accuracy.

The importance of the problem of interfacial thermal conductance has attracted the attention of many researchers, resulting in a large number of publications {1-5}*. Almost all the experimental researchers noticed that the thermal conductances of similar contacts (produced by surfaces of the same roughnesses and materials and having the same interfacial fluid and contact pressure) would not be the same in general but would vary quite widely. This is caused by the fact that no two contacts of similar surfaces are exactly the same since the surface roughnesses are not "regular" but statistical in nature; i.e., having asperities of varying heights and shapes with varying distances between them which can produce an infinite number of contact geometries. In

* Numbers in Square brackets refer to references at the end of the report

In correlating the experimental results of several researchers, Veziroglu {6} found that there would be large deviations (a mean deviation of $\pm$ 35%) in predicting interfacial thermal conductances, the primary reason being the uncertainties in estimating the contact geometries. Henry and French {7} developed a profilometer-analog computer system for estimating the actual contact areas. D'yachenko et al {8} presented experimental and analytical methods for determining the actual contact areas. Assuming that the distribution curve obtained from the surface profile has a normal distribution, Tsukizoe and Hisakado {9,10} deduced relationships for the distance between the surfaces and the size and area of the contact spots, and obtained good agreement with experiments. Assuming exponential and Gaussian distributions for the asperity heights, Greenwood and Williamson {11} and Greenwood {12} showed that the average size of the contact spots were almost constant and independent of the contact pressure for both the elastic and plastic deformation at the contact spots.

Sexl et al. {13-14} and Hsich et al. {15} applied the statistical studies of the actual contact areas to thermal contact conductances, and derived relationships for interfacial thermal conductances of contacts in vacuum. The latter group assumed the asperities to be conical with a normal height distribution, the deformations to be plastic and the constrictions to be infinite. They obtained good agreement with experimental results in the contact pressure range of 10^3 p.s.i. to 10^4 p.s.i. Sexl et al. assumed the asperities to be spherical caps and the constrictions to be infinite, but did not make any restrictive assumptions on the asperity height distribution and considered both the elastic and plastic deformations at the contacts.

The investigation reported in this paper was undertaken with two goals in mind: (1) To experimentally verify the statistical nature of interfacial thermal conductance, and (2) to develop a theory in agreement with the experimental results. The experimental results conclusively show that the conductances of apparently similar contacts may vary widely. A theoretical model has been developed by considering the statistical nature of the surface roughnesses. The agreement between the theory and the experiments, both with regard to the mean values and the standard deviations, is good.

II. EXPERIMENTAL APPARATUS

Fig. 1 shows the apparatus used for measuring interfacial thermal conductance. It consisted of two "holders" (an upper holder and a lower holder) for aligning test pieces, a vacuum chamber, a radiation guard, a spherical sliding joint and a frame. Each holder contained an electric heater, a water cooler and a 1 inch diameter X 2.22 inch long stem with a recessed head for positioning the test piece. There was an automatic voltage regulator in the electric heater circuit in order to keep the heat input constant. The cooling water system had a constant head water tank for maintaining a constant water flow rate and consequently a constant heat flux. The lower holder was attached to the lower plate of the frame of the apparatus. The upper holder was not attached to the frame. When properly installed, as shown in the figure, the upper holder fitted through an opening in the upper plate of the frame which had a clearance of 0.25 inch to permit small lateral motion. The vacuum chamber consisted of two flanges, each attached to one of the holders, and a rubber tube placed between the flanges. The flexibility of the rubber tube insured that all the axial load would be applied through the test pieces. A vacuum pump was attached to the vacuum chamber for pumping air and gases out of the chamber. The radiation guard consisted of three layers of aluminum foil arranged concentrically and was used to reduce the heat losses by radiation. The spherical joint consisted of two parts, a convex part and a concave part. The convex part was attached to the top of the upper holder and the concave part was free to float over it. During the experiments the apparatus was placed between the lower and upper plates of a compression testing machine for applying contact loads. The spherical head insured that the load was applied uniformly over the interface even if the contact surfaces of the test pieces were not exactly perpendicular to their own axes.

Iron-constantan 30 gauge diameter thermocouples were placed along the holder stems (and along the test pieces) to measure temperatures. The experimental apparatus was also instrumented for vacuum, electric power, water flow rate and water temperature measurements.

III. TEST PIECES AND EXPERIMENTAL PROCEDURE

The test pieces were made of stainless steel 303. They were cylindrical in shape with 1 inch diameter and 1.5 inch length. Their bases had a 3/32" x 3/32" peripheral recess so that they could fit into the corresponding parts in the holders for correct alignment (See Fig. 1). To make the test surfaces as nearly identical as possible, they were prepared in a shaper using the same tool and the same feed and under identical conditions. The shaper tool was prepared so that it made V-shaped parallel cuts which produced an essentially two dimensional surface roughness pattern as shown in Fig. 2. Among the 30 test surfaces prepared, the surface roughness CLA readings, measured with a Talysurf profilometer, ranged from 1828 micro-inches to 2249 micro-inches (See Table I) because of the wear in the shaper tool during the production of the surfaces. However the roughness wavelength, which was dependent on the tool feed, was constant (9800 micro-inches) for all the test surfaces. The 30 test pieces were matched in pairs to obtain 15 test interfaces. They were matched so that the deviations in the mean gaps of the interfaces were as small as possible. This was achieved by matching the surface having the greatest roughness with that having the smallest roughness, the surface having the second greatest roughness with that having the second smallest roughness, and so on. As a result of this matching, the range of the CLA sums for the 15 interfaces were narrowed to 4037 - 4269 micro-inches, as seen from Table I. Each test piece had four iron-constantan thermocouples soldered on its cylindrical surface at distances of 1/4", 1/2", 3/4" and 1" from its test surface. All the thermocouples were calibrated in place.

To start an experiment, a matched pair of test pieces were selected; their test surfaces were cleaned, and they were placed in the experimental apparatus with the lay directions of their surface roughnesses perpendicular to each other to form the test interface. After this, the experimental apparatus was placed in a compressive testing machine and an initial load of about 1000 lbs. was applied. Then the vacuum pumps, the cooling water flow in the upper holder and the electric heater of the lower holder were started, in that order. The vacuum chamber was evacuated to a pressure of about 2×10^{-4} Torr., the water flow rate was maintained at 2 g p m and the heater power at 300 watts. About 12 to 15 hours (overnight) were required for the system to reach steady-state. The load was increased by increments of about 1500 lbs. and all the readings were taken when the system reached steady-state which took about 2 hours after each load increase. The experiment was continued till a maximum load of about 10,000 lbs. was reached. A more detailed description of the experimental procedure is given by Bhandari {16}.

The above described experimental procedure was repeated fifteen times, once for each of the fifteen fresh interfaces. After the completion of the experiments, the roughnesses of the surfaces forming the interfaces were re-examined using a Talysurf profilometer. It was found that the CLA readings of the surfaces roughnesses decreased by about 5%.

IV. EXPERIMENTAL RESULTS

Using the experimental readings, the thermal contact conductance and the apparent contact pressure (the load divided by the apparent contact area or the test piece cross-sectional area) were calculated for each of the experimental points. The thermal contact conductance as a function of apparent contact pressure for each of the interfaces is shown in Fig. 3. The general trend of the curves confirms the findings of the previous researchers that for gradually increased loading (a) the thermal contact conductance increases with increase in the apparent contact pressure, and (b) the rate of increase of the thermal contact conductance with respect to the apparent contact pressure increases with increase in the apparent contact pressure. However, the results presented in Fig. 3 also show that there are wide differences between the results for different interfaces. The maximum deviations of the thermal contact conductances from the mean values range from ± 60% at 2500 p.s.i. to ± 35% at 12000 p.s.i. The sum of the roughness CLA readings of the surfaces forming the interfaces (which is a measure of the interface gap) had a maximum deviation of ± 3% from the mean. These small deviations in the interface gap cannot explain the large differences measured in thermal contact conductances. Also, for experiments carried out in vacuum the interface gap would have almost no effect on thermal contact conductance. A close inspection of Fig. 3 indicates no tendency which can be attributed to the interface gap.

The asperities of a given test surface were not all of the same height. There were large differences between the asperity heights. Because of such a distribution of asperity heights, when different interfaces are formed using similar surfaces, each interface may have a completely different number of actual contact spots for a given contact load. It is well known that the number of actual contact spots strongly influences the thermal contact conductance; increasing the number of contact spots increases the thermal contact conductance and vice versa. Consequently, the asperity height distribution could explain the large differences in the thermal contact conductances obtained under apparently similar conditions.

V. THEORY

The profiles of the test surfaces used in the experiments can be idealized as a series of wedges of equal base, which is the roughness wavelength λ, and varying height h_i as shown in Fig. 4. When two such surfaces are brought together - with their lay directions perpendicular to each other - to form an interface, each possible contact is contained in a square of area λ^2. Fig. 5 shows two views of an actual contact when the opposing wedges of heights h_1 and h_2 cut into each other by a length s. The length s is given by,

$$s = h_1 + h_2 - L \quad \ldots \quad (1)$$

where L is the distance between the planes passing through the wedge bases of the two interface surfaces. As seen from the figure, the projected contact area, S, becomes,

$$S = 1/2\, d_1\, d_2 \quad \ldots \quad (2)$$

where the diagonal d_i of the projected area is given by,

$$d_i = \frac{\lambda s}{h_i} \quad \ldots \quad (i = 1,\ 2) \quad \ldots \quad (3)$$

Substitution of equations (3) into (2) gives

$$S = \frac{\lambda^2 s^2}{2h_1 h_2} \quad \ldots \quad (4)$$

Hence the equivalent radius of the projected contact area becomes,

$$a = \sqrt{\frac{S}{\pi}} = \frac{\lambda s}{\{2\pi h_1 h_2\}^{0.5}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

The interface can be assumed to be made up of several "contact elements" such as shown in Fig. 6. Each contact element consists of two cylindrical metals (of radius b) having an actual circular contact (of radius a) in the middle of their bases facing each other with an interstitial fluid or vacuum between the remainder of the contact surfaces. Considering such a contact element, thermal contact conductance per unit apparent contact area for a contact in vacuum becomes {17}

$$u = \frac{ak}{b^2 \tan^{-1}\{\frac{b}{a} - 1\}} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

where k is the thermal conductivity of the contact materials. In the case of dissimilar contact materials, k becomes the harmonic mean of the two thermal conductivities.

The interface formed by the test surfaces can be assumed to be made of several contact elements having different dimensions, $a_1, a_2, \ldots, a_n$ and $b_1, b_2, \ldots, b_n$. When heat flows from one test piece to the other, far from the interface the heat flow lines are parallel to each other and the temperature distributions are linear. The temperature difference measured at the interface between extrapolations of the linear temperature distributions is the contact temperature drop, i.e., the temperature drop caused by the interface. The contact temperature drop is the same for all the contact elements which make up the interface. In other words, the thermal contact conductance per unit area, u, is the same for all of the contact elements, viz,

$$u = \frac{a_n k}{b_n^2 \tan^{-1}\left\{\frac{b_n}{a_n} - 1\right\}} \quad \ldots \; (n = 1, 2, \ldots, n) \; \ldots \quad (7)$$

The sum of the cross-sectional areas of the contact elements must be equal to the apparent contact area, A; i.e.,

$$A = \pi \sum_{n=1}^{n} b^{2} \quad (8)$$

Eliminating the b_n^2 between equations (7) and (8), one obtains,

$$u = \frac{\pi k}{A} \sum_{n=1}^{n} \frac{a_n}{\tan^{-1}\left\{\frac{b_n}{a_n} - 1\right\}} \quad (9)$$

The summation in equation (9) can be expressed as follows, with a good degree of approximations:

$$\sum_{n=1}^{n} \frac{a_n}{\tan^{-1}\left\{\frac{b_n}{a_n} - 1\right\}} = \frac{n\bar{a}}{\tan^{-1}\left\{\frac{\bar{b}}{\bar{a}} - 1\right\}} \quad (10)$$

where $\bar{a}$ and $\bar{b}$ are the average values of a_n and b_n respectively. If N is the maximum possible value of n (the number of contacts), then the apparent contact area A is given by,

$$A = N\lambda^2 \quad (11)$$

since each area of λ^2 can contain at most one contact. Substitution of equations (10) and (11) into (9) yields

$$u = \frac{\pi k}{\lambda^2} \cdot \frac{n}{N} \cdot \frac{\bar{a}}{\tan^{-1}\left\{\frac{\bar{b}}{\bar{a}} - 1\right\}} \quad (12)$$

If $f_1(h_1)$ and $f_2(h_2)$ are the normalized distribution functions for the wedge heights of the two surfaces forming the interface, then the ratio n/N gives the fraction of possible contacts actually occurring and can be calculated as follows:

$$\frac{n}{N} = \iint_R f_1(h_1) f_2(h_2) dh_1 dh_2 \quad (13)$$

where R indicates the region of integration defined by

$h_1 + h_2 \geq L$

Combining equations (8) and (11) and solving the resulting equation for $\bar{b}$ gives,

$$\bar{b} = \lambda \frac{N}{\pi n} \quad (14)$$

Using the definition of "a" given by (5), its average value becomes,

$$\bar{a} = \frac{\lambda}{2\pi} \iint_R \frac{s}{h_1 h_2} \cdot f_1(h_1) f_2(h_2)\, dh_1 dh_2 \quad (15)$$

Substitution of equations (13), (14) and (15) into (12) yields the following expression for the average thermal contact conductance per unit area:

$$u = \frac{\frac{\pi}{2} \cdot \frac{k}{\lambda} \cdot \left\{\iint_R f_1 f_2 dh_1 dh_2\right\} \cdot \left\{\iint_R \frac{s f_1 f_2}{(h_1 h_2)^{0.5}} \cdot dh_1 dh_2\right\}}{\tan^{-1} \sqrt{2} \left\{ \left(\iint_R f_1 f_2 dh_1 dh_2\right)^{0.5} \cdot \left(\iint_R \frac{s f_1 f_2}{(h_1 h_2)^{0.5}} \cdot dh_1\, dh_2\right)^{-1} - 1 \right\}} \quad (16)$$

Because of the model of the surface profiles (i.e., wedges with sharp edges), the deformation at the interface will be plastic. Therefore, the force F at a single contact, is given by,

$$F = MS \quad (17)$$

where M is the Meyer Hardness of the softer contact material. The average apparent contact pressure p can be calculated from a force balance at the interface, as follows,

$$Ap = n\pi \bar{a}^2 M \quad \text{or}$$

$$N\lambda^2 p = n\pi \bar{a}^2 M$$

$$p = \frac{n\pi \bar{a}^2 M}{N\lambda^2} \quad (18)$$

Substituting equations (5) and (13) into (18), the average contact pressure becomes,

$$p = \frac{M}{2} \{\iint_R f_1 f_2 dh_1 dh_2\} \cdot \{\iint_R \frac{s^2}{h_1 h_2} f_1 f_2 dh_1 dh_2\} \quad . \; . \; . \; . \qquad (19)$$

VI. COMPARISON OF THEORY AND EXPERIMENTS

In order to calculate the theoretical values of the mean thermal contact conductance and the mean contact pressure from equations (16) and (19), the thermal conductivity k, the surface roughness wavelength λ, the surface roughness wedge height distribution functions f_1 and f_2, and the Meyer hardness M of the softer contact material are needed. The thermal conductivity of the test pieces at the mean test temperature of 305°F. was calculated by comparison with a specimen of known thermal conductivity. The result was,

$$k = 8.073 \text{ Btu/hr.}^{\circ}\text{F.ft.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (20)$$

The surface roughness wavelength was determined from the surface profile recordings to be,

$$\lambda = 9800 \text{ micro-inches} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (21)$$

Since both the surfaces forming the interfaces were produced under identical conditions, the distribution functions f_1 and f_2 were the same, i.e., $f_1=f_2=f$. This function was obtained from the surface profile recordings and is given in Table II. The Meyer hardness of the test pieces (stainless steel) was measured at room temperature (68°F.) using a hardness tester and was,

$$M_o = 216{,}620 \text{ p.s.i.} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \; . \qquad (22)$$

The Meyer hardness M, taking into account the effects of the contact temperature and the duration of loading, was calculated using the expression given in reference 17.

Using the above data and equations (16) and (19), the theoretical values of the mean thermal contact conductance, the mean contact pressure and the standard deviation in contact conductance ($\sqrt{\overline{\mu^2}-\bar{\mu}^2)/n}$) have been calculated; also the experimental values of the same have been calculated using the experimental data. The results are presented in Table III. As can be seen from Table III, (1) both the experimental and theoretical conductances increase with increase in contact pressure, (2) the rate of increase of the conductances increase with increase in contact pressure,

(3) the experimental and theoretical conductances and standard deviations are of the same order, and (4) the experimental conductances are within the standard deviations of the theoretical conductances.

Fig. 7 shows the experimental points and the theoretical relationship for mean values in a contact conductance vs. contact pressure plane. As can be seen from Fig. 7, the agreement between the experiments and the theory is good. The theoretical contact conductance asymptotically approaches infinity as the contact pressure approaches the hardness value (i.e., as the contact surfaces touch each other all over the interface) as would be expected.

The thermal contact conductances, in addition to the parameters considered above, are affected by factors such as surface films, warping, waviness and actual roughness shape which were neglected in the theory. Considering all of these, the degree of agreement between the experiments and the theory is rather good.

VII. CONCLUSION

The investigation conclusively shows the statistical nature of thermal conductance between surfaces in contact and presents a theory which can successfully predict the mean conductance and standard deviation.

VIII. ACKNOWLEDGEMENTS

The work described herein was supported by the United States National Aeronautics and Space Administration under Grant NGR 10-007-010 in the University of Miami, Coral Gables, Florida, U.S.A. and by the Turkish Scientific and Technical Research Organization under Project MAG-196 in the Middle East Technical University, Ankara, Turkey. The author wishes to express his appreciation to Drs. S. Kakaç and O. Yeşin of the Middle East Technical University for their interest and encouragement, and to N.K. Bhandari, P. Nayak, S. Worthem and P. Singer of the University of Miami for their assistance.

REFERENCES

1. Gex, E. C., "Thermal Resistance of Metal-to-Metal Contacts - An Annotated Bibliography," Armed Services Technical Information Agency (Defense Documentation Center) Doc. 263181 (July 1961).

2. Atkins, H.L., "Bibliography on Thermal Metallic Contact Conductance," NASA Marshall Space Flight Center, NASA-TM-X-532227, 26 pp. (April 1965).

3. Vidoni, C.M., "Thermal Resistance of Contacting Surfaces: Heat Transfer Bibliography," University of California, Lawrence Radiation Lab., UCRL-14264, AEC Contract W-7405-Eng.-48 (June 1965).

4. Wong, H.Y., "A Survey of the Thermal Conductance of Metallic Contacts," Aeronautical Research Council, CP 973, 1968.

5. Moore, C.J., Atkins, H., and Blum, H.A., "Subject Classification for Thermal Contact Resistance Studies," Amer. Soc. Mech. Engineers, Winter Annual Meeting, Paper No. 68-WA/HT-18, Nov. 1968.

6. Veziroglu, T.N., "Correlation of Thermal Contact Conductance Experimental Results," Progress in Astronautics and Aeronautics, Vol. 20, pp. 879-907, 1967.

7. Henry, J.J., and Fenech, H., "The Use of Analog Computers for Determining Surface Parameters for Prediction of Thermal Contact Conductance," Transactions of Amer. Soc. Mech. Engineers, Journal of Heat Transfer, Series C, Vol. 86, pp. 543-551, Nov. 1964.

8. D'yachenko, P.E., Tolkacheva, N.N., Andreer, G.A., and Karpover, T.M., "The Actual Contact Area Between Touching Surfaces," English Translation from Russian by Consultants Bureau,New York 68 pp. 1964.

9. Tsukizoe, T., and Hisakado, T., "On the Mechanism of Contact Between Metal Surfaces - The Penetrating Depth and the Average Clearance," Transactions of Amer. Soc. Mech. Engineers, Journal of Basic Engineering, pp. 666-674, September 1965.

10. Tsukizoe, T., and Hisakado, T., "On the Mechanism of Contact Between Metal Surfaces: Part 2 - The Real Area and the Number of the Contact Points," Transactions of Amer. Soc. Mech. Engineers, Journal of Lubrication Technology, pp. 81-88, Jan. 1968.

11. Greenwood, J.A., and Williamson, J.P.B., "Contact of Nominally Flat Surfaces," Proc. Royal Soc., Series A, Vol. 265, pp. 300-319, 1966.

12. Greenwood, J.A., "The Area of Contact Between Rough Surfaces and Flats," Transactions of Amer. Soc. Mech. Engineers, Journal of Lubrication Technology, pp. 81-91, Jan. 1967.

13. Sexl, H., and Sexl, R.U., "A Statistical Approach to the Theory of Interfacial Conductivity I," Center for Theoretical Studies Report, University of Miami, Coral Gables, Florida, 24 pp., July 1967.

14. Sexl, H., Sexl, R.U., Burkhard, D.G., and Schocken, K., "A Statistical Theory of Interfacial Thermal Conductivity," Thermal Conductivity, Plenum Press, pp. 467-476, 1969.

15. Hsieh, C.K., Yeddanapudi, K.M., and Touloukian, Y.S., "An Analytical Study of Thermal Contact Conductance for Two Rough and Wavy Surfaces Under a Pressure Contact," 9th Conference on Thermal Conductivity, U.S. Atomic Energy Commission, pp. 554-570, March 1970.

16. Bhandari, N.K., "Statistical Study of Thermal Contact Conductance", M.S. Thesis, University of Miami, Florida, 96 pp., January 1969.

17. Cetinkale (Veziroglu), T.N., and Fishenden, M., "Thermal Conductance of Metallic Surfaces in Contact", General Discussion of Heat Transfer, Proceedings of the Institution of Mechanical Engineers and American Society of Mechanical Engineers, pp. 271-275, 1951.

NOMENCLATURE

Symbols

a	Contact element radius.
A	Apparent contact area.
b	Radius of actual contact at contact element.
d	Roughness wedge width at actual contact.
f	Normalized distribution function for wedge heights of a surface.
F	Force.
h	Roughness wedge height.
k	Thermal conductivity.
L	Distance between roughness wedge bases of contact surfaces.
M	Meyer hardness.
n	Number of contacts.
N	Maximum possible number of contacts.
P	Apparent contact pressure.
R	Region of integration.
s	Distance roughness wedges cut into each other.
S	Projected contact area.
u	Thermal contact conductance per unit area.
λ	Roughness wedge width at roughness base.

Subscripts

i	Surface 1 or surface 2.
n	Contact element number.
o	Meyer hardness measured at room temperature.
1	Surface 1.
2	Surface 2.

TABLES

TABLE I

PRE-EXPERIMENT MEAN SURFACE ROUGHNESS READINGS

Interface	Surface 1 CLA micro-inches	Surface 2 CLA micro-inches	Total CLA micro-inches
1	2086	1951	4037
2	2112	1976	4088
3	2181	1962	4143
4	2053	2100	4153
5	2133	2000	4133
6	2001	2104	4105
7	2032	2131	4163
8	2152	2031	4183
9	2081	2048	4129
10	2022	2033	4055
11	2320	1870	4190
12	2324	1898	4222
13	2215	1860	4075
14	2249	1828	4077
15	2134	2135	4269

TABLE II

WEDGE HEIGHT VS DISTRIBUTION FUNCTION

Wedge Height h (μin)	Distribution Function f (μin.$^{-1}$)x10^6	Wedge Height h (μin)	Distribution Function f (μin.^{1})x10^6
1800	0	2800	92
1810	508	2960	82
1820	1010	3120	74
1840	2100	3280	67
1920	2108	3440	60
2000	2077	3600	54
2080	1873	3760	48
2160	1356	3920	42
2240	592	4080	36
2320	370	4240	30
2400	247	4400	24
2480	173	4560	18
2560	123	4720	12
2640	102	4880	6
2720	97	5040	0

TABLE III

EXPERIMENTAL AND THEORETICAL RESULTS

Pressure (p.s.i.)	Conductance (BTU/hr.°F.ft.^{2}) Experimental	Theoretical	Std.deviation (BTU/hr.°F.ft.^{2}) Experimental	Theoretical
3000	450	600	160	250
4000	555	820	170	315
5000	710	1050	180	370
6000	900	1300	200	420
7000	1125	1550	220	450
8000	1407	1790	255	480
9000	1735	2050	285	510
10000	2130	2290	325	535
11000	2650	2600	415	555
12000	3240	2800	555	570

FIGURES

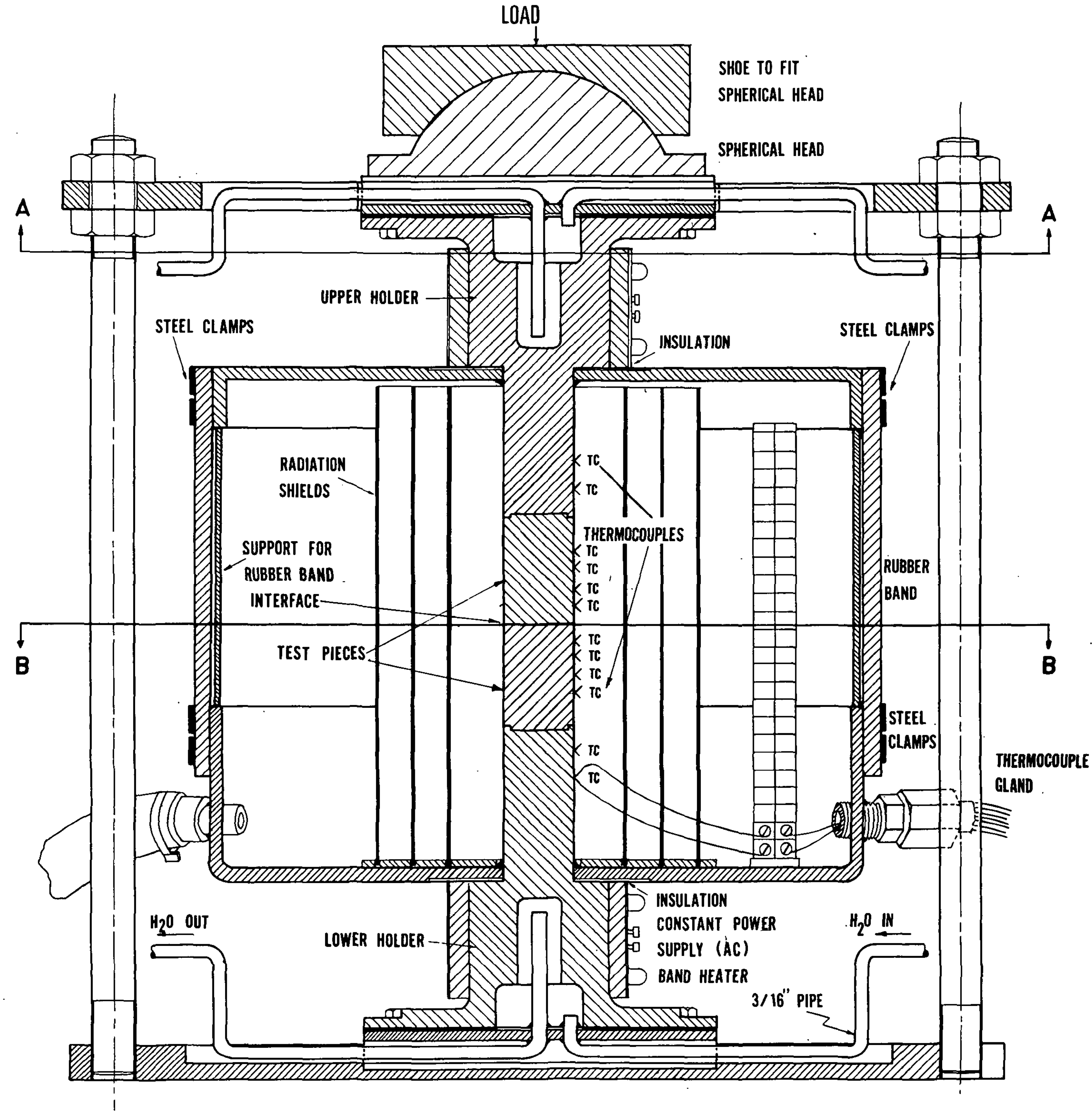


FIG.1.- APPARATUS FOR MEASURING INTERFACIAL THERMAL CONDUCTANCE

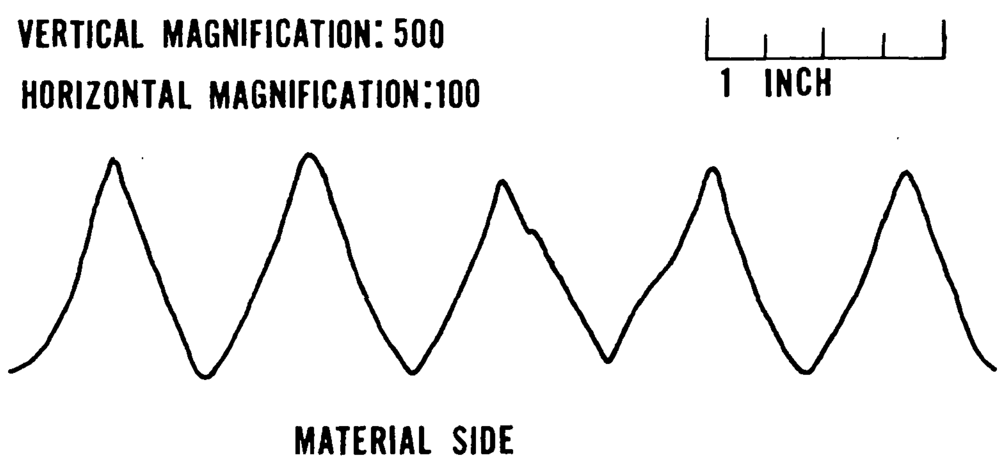


FIG.2.–TYPICAL TEST SURFACE PROFILE

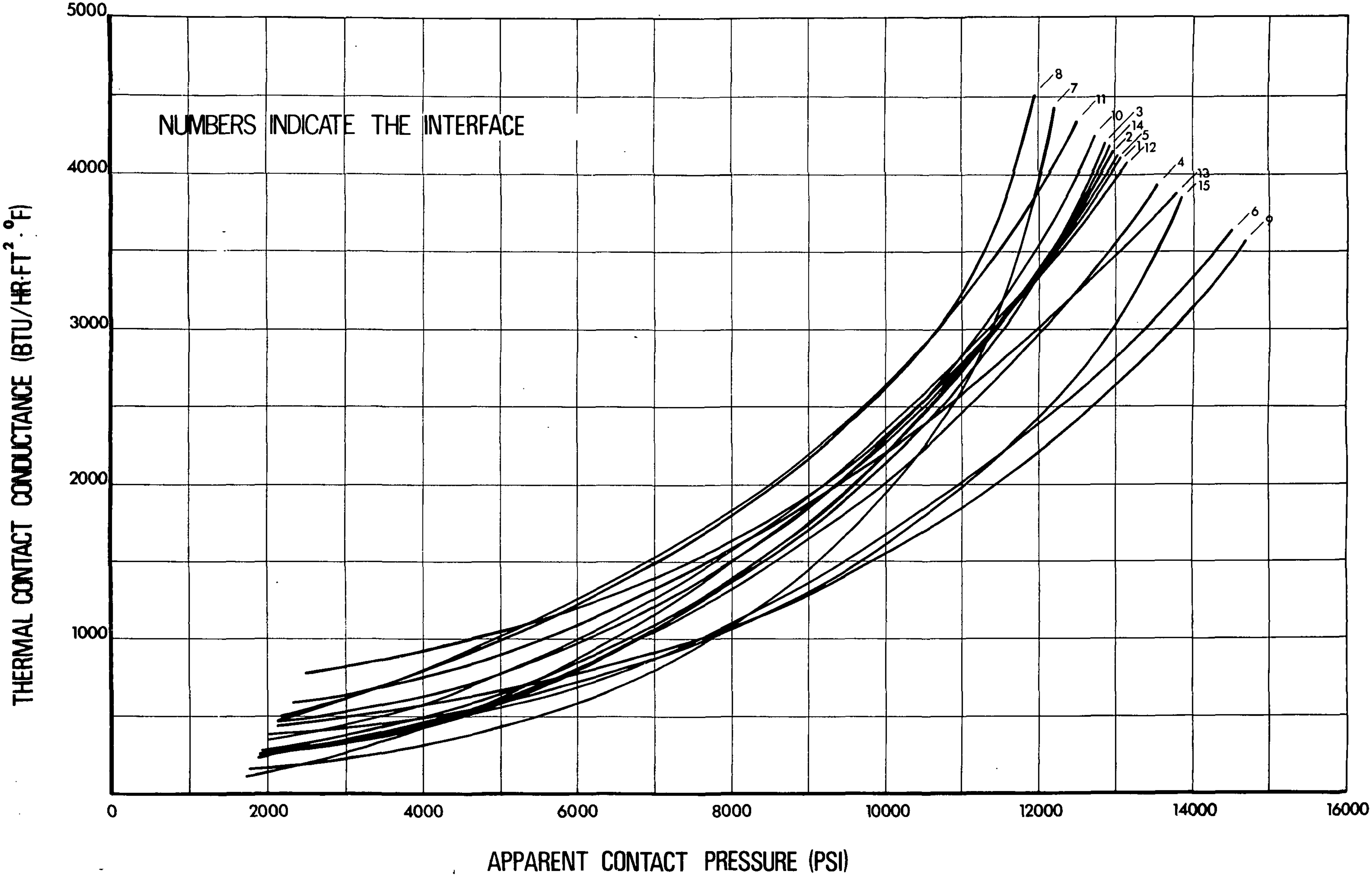


FIG.3.-EXPERIMENAL THERMAL CONTACT CONDUCTANCE VS APPARENT CONTACT PRESSURE RELATIONSHIPS

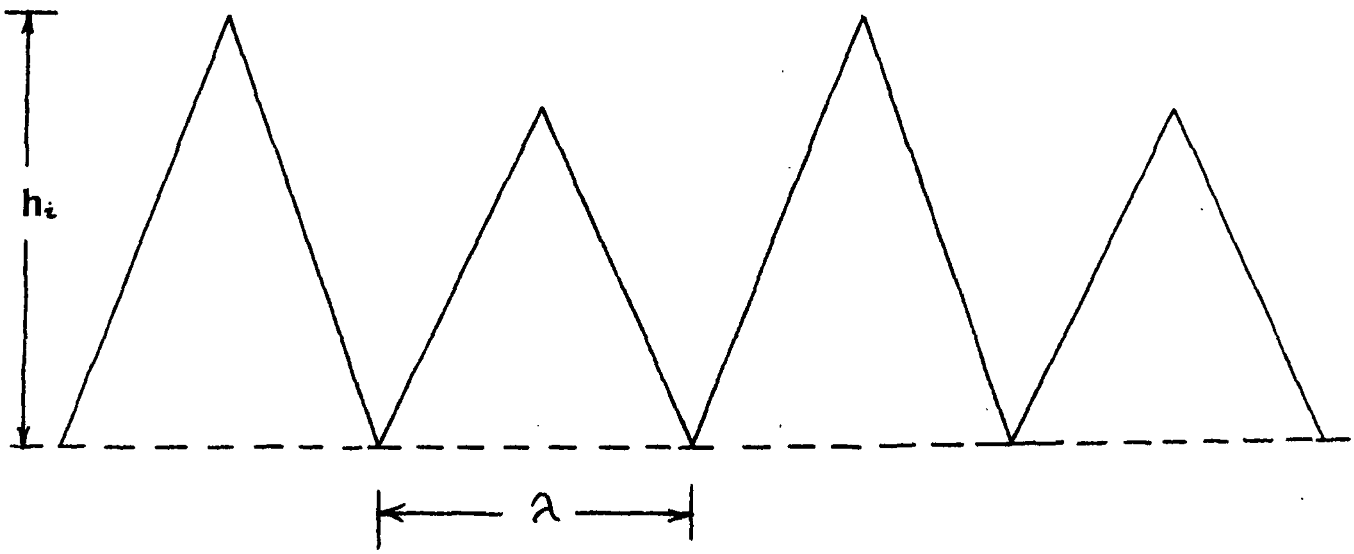


FIG. 4. ASSUMED SURFACE PROFILE

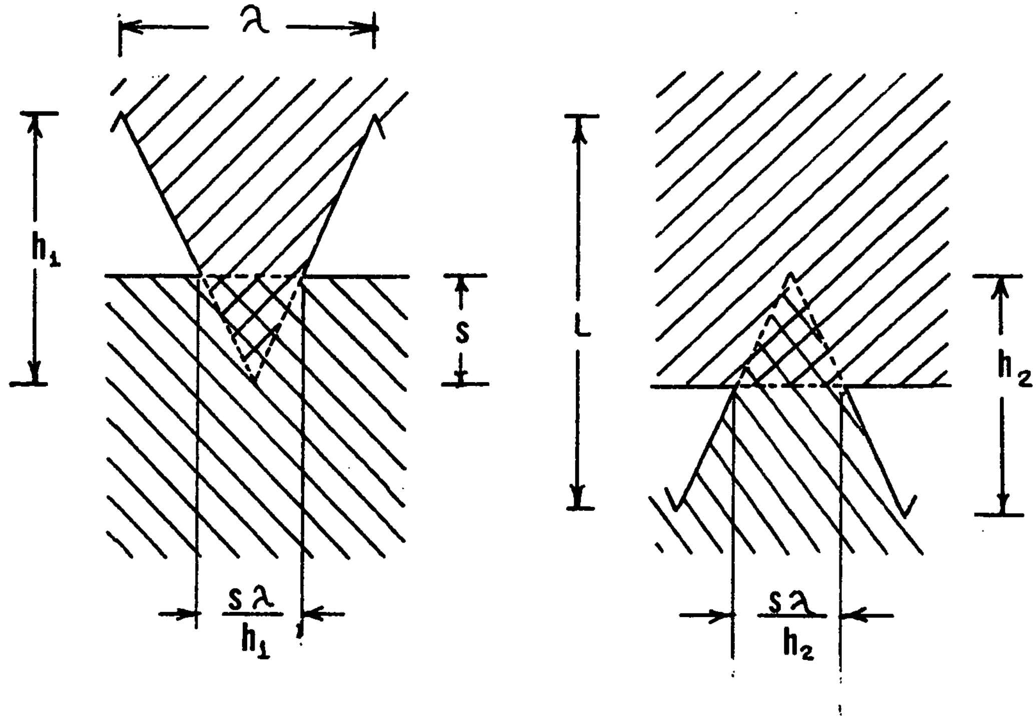
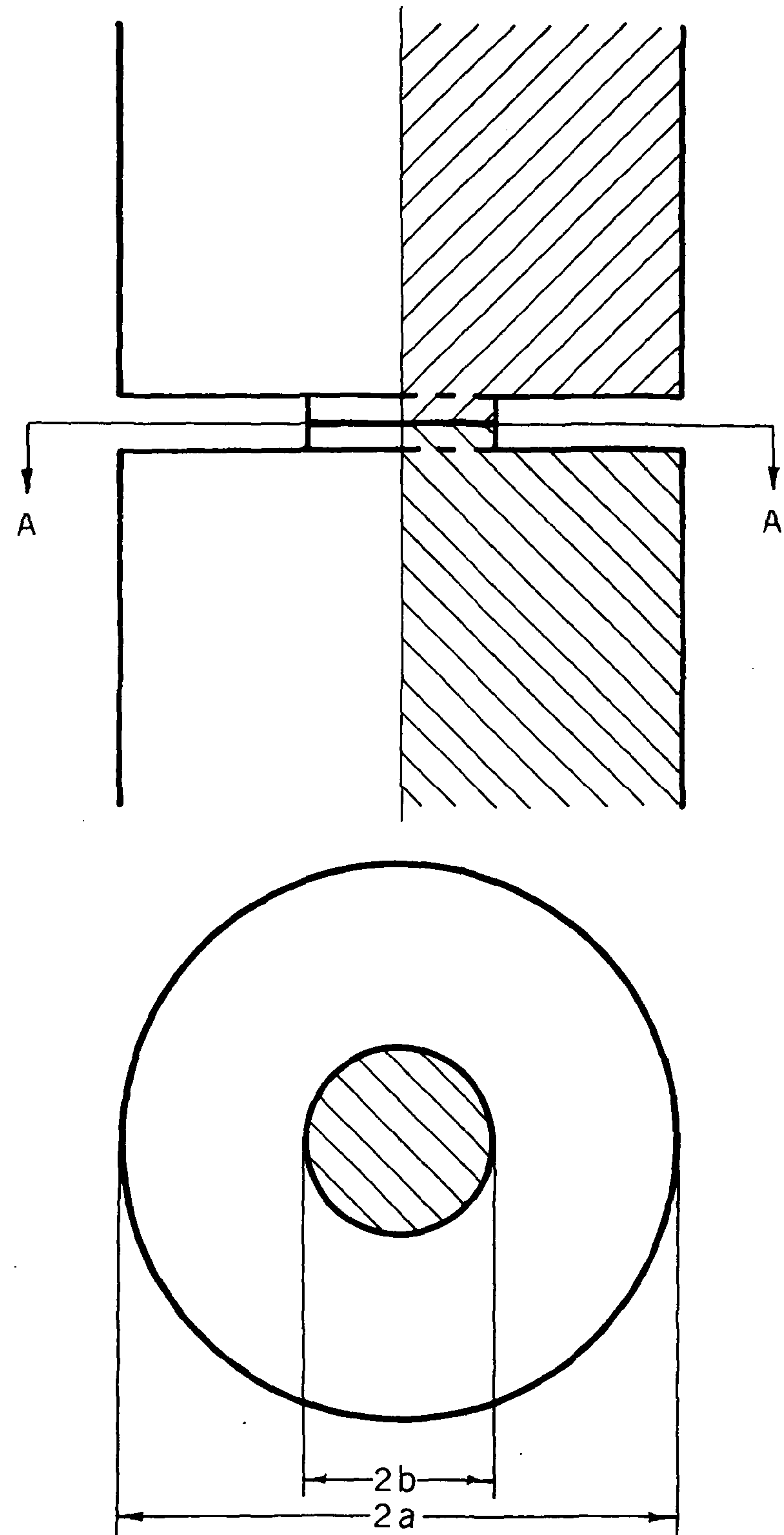


FIG.5._GEOMETRY OF A CONTACT



SECTION AT AA

FIG. 6.— IDEALIZED CONTACT ELEMENT

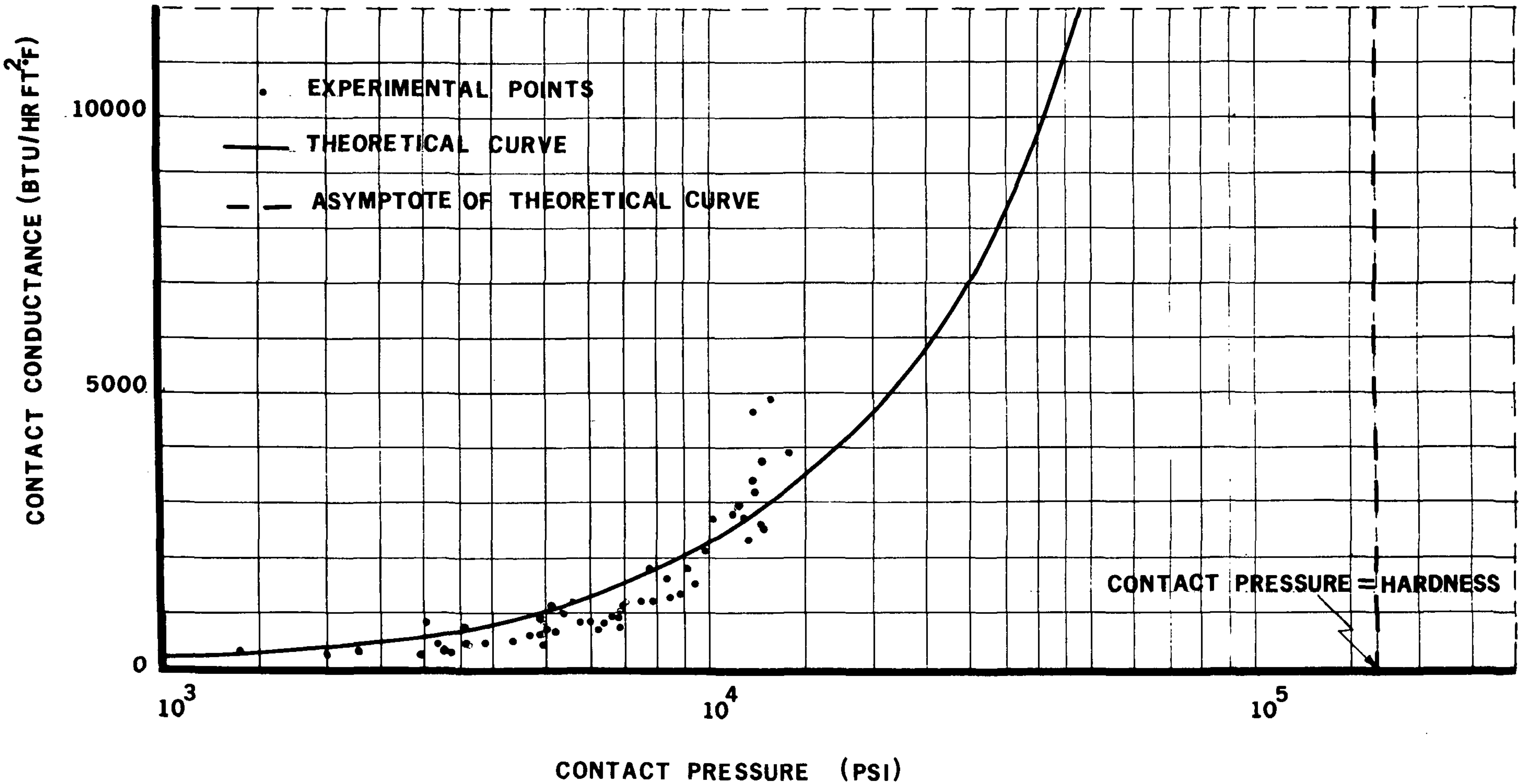


FIG.7.-CONTACT CONDUCTANCE VS CONTACT PRESSURE EXPERIMENTAL AND THEORETICAL RESULTS