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AN ANALYSIS OF THERMAL RESISTANCE AT A SOLID-STATIONARY LIQUID METAL INTERFACE DUE TO THE PRESENCE OF GAS CAVITIES

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Abstract—A mathematical analysis of the thermal interface resistance is carried out on an idealized elemental heat flux tube for the case of a gas cavity interposed between a solid and a stationary liquid metal. The thermal resistance is expressed in terms of the thermal conductivities of the solid and liquid metal, the number of gas cavities per unit apparent area and a wettability number. The last part of the paper is devoted to a qualitative comparison between the theory and the test data of Bleuven *et al.* Good agreement was observed within the assumptions made.

NOMENCLATURE

A, projected interface area;
a, radius of gas activity;
b, radius of elemental heat flux tube;
d, distances from interface;
B, coefficient of proportionality for gas cavity volume;
K, thermal conductivity;
n, gas cavity density;
P, system or gas pressure;
P^*, normalized gas pressure for data reduction, $P^* = P/0{\cdot}6$;
Q, heat flow rate;
R_c, thermal contact resistance defined by equation (29);
R_c^*, normalized thermal contact resistance for data reduction, $R_c^* = R_c/R_c(P = 0{\cdot}6)$;
r, radial distance from centre of the gas cavity in the interface plane;
T, temperature;
u, auxiliary coordinate, $u = \sinh\zeta$;
v, auxiliary coordinate, $v = \sin\delta$;
W, wettability number, $W = (Aa - Ag)/Aa$;
x, y, z, cartesian coordinates;
α, angle subtended by gas cavity;
δ, thickness of gas cavity;
ζ, oblate spheroidal coordinate;
η, oblate spheroidal coordinate.

Subscripts
a, apparent or average;
c, contact;
e, extrapolated;
g, gas;
i, interface;
1,2, components of heat flux tube, solid and liquid, respectively.

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INTRODUCTION

MANY modern power plants, whether large nuclear reactors or small spacecraft systems, utilize liquid metals as single-phase coolants. Since heat is conducted across solid/liquid metal interfaces, it is important that due account be made of the additional thermal resistance at these interfaces, especially when the heat flux is very large. At present this additional thermal resistance is determined experimentally or completely ignored.

Several experimental studies [1–9, 15, 16] dating back to the early 1950's were directed towards finding the laws which governed heat transfer between a metal wall and a flowing liquid metal. It is evident that if there is contact resistance between the metal wall and the liquid metal, the overall thermal conductance will be greatly reduced. It was found that the experimental values of Nusselt number were well below the predicted values. This discrepancy was attributed to various factors such as scale or oxide on the walls of the system, to gas entrainment and to the nonwetting of the tube walls by the liquid metal.

Kirillov *et al.* [8] were able to determine experimentally the thermal resistance between NaK liquid metal flowing in a copper tube. They observed a significant thermal resistance which persisted for many hundreds of hours of operation of the system. After about 500 h this interface resistance decreased and became so small that it could not be measured. It was concluded that the continuous flow and purification of the liquid metal removed all traces of scale and oxide; thus removing the cause of the interface resistance.

Subbotin *et al.* [9] reported that tests with mercury flowing in a steel tube with uniform heat flux showed that the interface resistance was dependent upon the Reynolds number. It was concluded that this resistance could not be due to a permanent scale or oxide layer on the tube wall.

Several, more recent, studies [10–12] examined in particular the effects of gas entrainment, nonwettability of liquid metal and the presence of oxides upon the heat transfer between a solid and a liquid metal. The other studies [13, 14] used the analogy between electrical and thermal resistance in an attempt to clarify the important parameters which influence the flow of heat between a solid and a liquid metal.

The most recent experimental study [17]

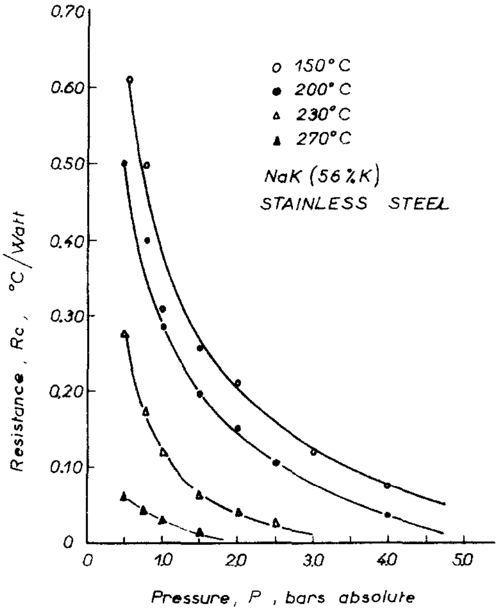

FIG. 1. Thermal resistance data from reference [17]. Wall treated with cold argon.

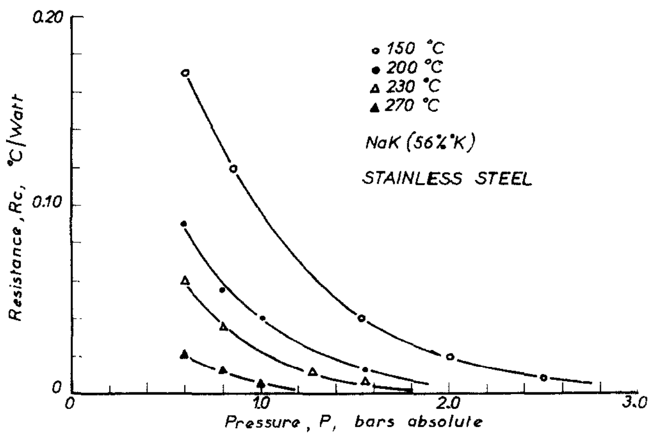

FIG. 2. Thermal resistance data from reference [17]. Wall treated with hot argon.

determined the effect of the system pressure, and the initial treatment of the wall on the thermal contact resistance between stainless steel and a stationary NaK liquid metal. They found that the system pressure had a strong influence upon the interface resistance, Figs. 1 and 2. The resistance (for example, at a system temperature of 150°C and an initial treatment of the wall with cold argon) decreased from about 0·60°C/W at a pressure of 0·6 bars absolute to about 0·15°C/W at about 3 bars absolute. Upon returning to the initial pressure of 0·6 bars without changing the system temperature, they obtained about the same (within 5 per cent) initial thermal resistance. As the system temperature was increased thereby assuring that the liquid would wet the metal surface even more, the thermal resistance decreased; but always they observed the same characteristic curve of resistance versus system pressure. When the initial treatment of the wall was accomplished with hot argon rather than cold argon the thermal resistance was markedly decreased. Again they observed the characteristic curve of resistance versus system pressure. The simultaneous return of the system to the initial system pressure resulted in the same initial thermal resistance.

Since their system was clean (no scale or oxide on the walls), and the NaK liquid metal was pure, and the tests were conducted with a static fluid, the thermal resistance was due exclusively to the presence of gas cavities formed because of the nonwettability of the liquid metal.

The aim of this work is to provide a theory which will give the important parameters and their influence upon the thermal contact resistance between a solid and a stationary liquid metal when gas cavities are present. This theory excludes the effects of scales and oxides which may be present on walls and the effects of gas entrainment by the liquid.

MATHEMATICAL MODEL

When a liquid metal is in contact with a metallic surface, intimate contact will not occur over the entire common interface, Fig. 3(a). Because the liquid metal does not perfectly wet the solid

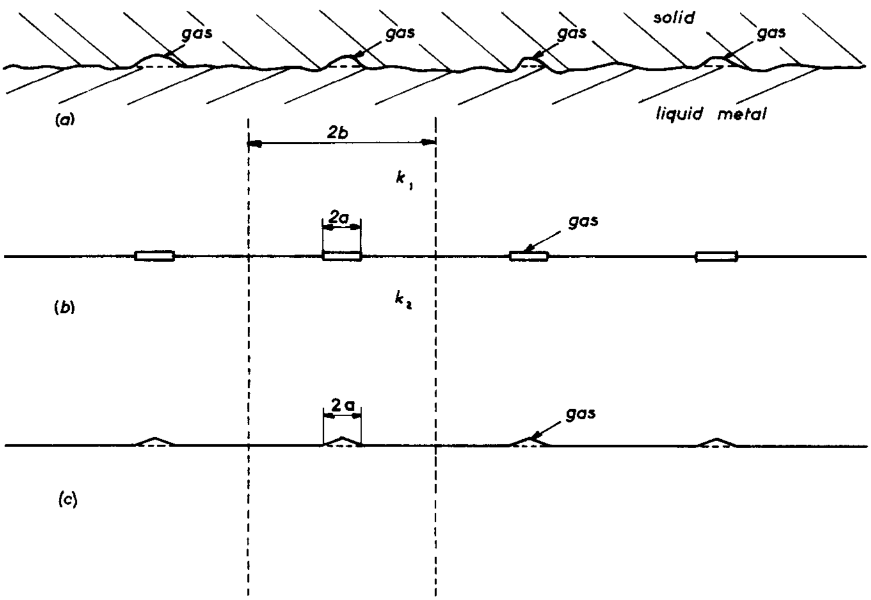

FIG. 3. Schematics of solid–liquid metal interfaces (a) Real interface between solid and liquid metal. (b) Ideal interface between solid and liquid metal. (c) Alternate ideal interface.

surface, gas, such as air or argon, will be trapped in the deepest parts of the surface. These gas cavities will in general vary in number, size, shape, and distribution over the apparent interface. It is evident that these gas cavity characteristics will depend upon a number of physical and geometric characteristics: the type of solid, of liquid metal and of gas which form the interface, the surface roughness, the surface cleanliness, and the method of initial treatment. For the following thermal analysis it will be assumed that the number of cavities is known, that all the cavities are thin circular discs, having the same size. Furthermore, it will be assumed that all the cavities are uniformly distributed over the apparent interface.

Since the thermal conductivities of the metal and liquid metal are much greater than the thermal conductivity of the gas, heat will flow around the gas cavity, i.e. the region of the interface occupied by the cavity is impervious to heat flow. As a consequence of these assumptions, there exists a number of identical heat flux tubes in the form of hexagonal cylinders having adiabatic walls and in the center of each tube there is a gas cavity. The hexagonal cylinders will be approximated by circular cylinders having the same cross-sectional area. The interface between the solid and liquid metal is assumed to be flat because for most machined solids the surface irregularities have very small slope [18].

Finally it is assumed that there is intimate contact between the solid and liquid metal beyond the cavities and, therefore, there is no thermal resistance at this part of the interface.

TEMPERATURE DISTRIBUTION

For the model of an elemental heat flux tube described above, Fig. 4, the temperature distribution, and implicitly the thermal resistance, must satisfy the Laplace differential equation in cylindrical coordinates:

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

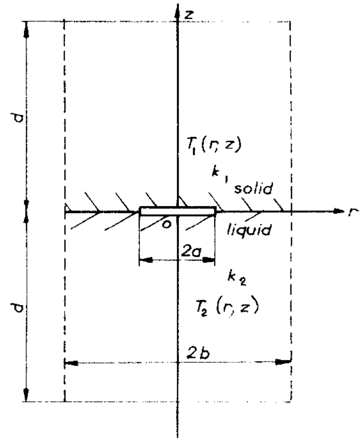

FIG. 4. Elemental heat flux tube.

and is subject to the following boundary contions:

$$K_1 \frac{\partial T_1(r,0)}{\partial z} = K_2 \frac{\partial T_2(r,0)}{\partial z} = 0, \quad 0 < r \leqslant a \quad (2)$$

$$T_1(r,0) = T_2(r,0) = T_i, \quad a < r \leqslant b \quad (3)$$

$$K_1 \frac{\partial T_1(r,0)}{\partial z} = K_2 \frac{\partial T_2(r,0)}{\partial z}, \quad a < r \leqslant b \quad (4)$$

$$\frac{\partial T_1(b,z)}{\partial r} = \frac{\partial T_2(b,z)}{\partial r} = 0, \quad (5)$$

$$\frac{\partial T_1(r,z)}{\partial z} = \frac{Q}{K_1 \pi b^2} \quad z \geqslant b \quad (6)$$

$$-\frac{\partial T_2(r,z)}{\partial z} = \frac{Q}{K_2 \pi b^2} \quad z \leqslant b \quad (7)$$

Q is the rate of heat transfer through the elemental heat flux tube per unit time, K_1 and K_2 are the thermal conductivities of the solid and liquid metal respectively, and b, the radius of the heat flux tube, depends upon the density of gas cavities per unit apparent area, i.e. $n\pi b^2 = 1$.

The gas cavity is represented by a thin circular disc of radius a and thickness $\delta(\delta/a \ll 1)$. The boundary condition (2) states that the disc is impervious to heat transfer. By using symmetry of heat flow arguments it is assumed that the common interface temperature is uniform, boundary conditions (3) and (4). Since the disc is at the origin of the chosen coordinate system, the temperature distribution is axisymmetric and the heat flow lines enter and leave the interface at right angles and tend to become parallel to the z-axis at large values of z and r. At large distances from the interface the temperature field must be nearly uniform so that isothermal planes $T_1(d, r)$ and $T_2(-d, r)$ can be placed at symmetrical distances from the interface.

The mathematical difficulties presented by the mixed boundary conditions (2) and (3) can be circumvented by the use of oblate spheroidal coordinates (ζ, η) with the origin in the center of the disc [19]. Introducing the auxiliary coordinates (u, v) we have:

$$u = \sinh \zeta, \qquad v = \sin \eta \tag{8}$$

and the cylindrical coordinates (r, z) can be expressed as:

$$r = a \cosh \zeta \cos \eta = a \sqrt{(u^2 + 1)} \sqrt{(1 - v^2)}, \tag{9}$$

$$z = a \sinh \zeta \sin \eta = a\,u\,v,$$

where a is the radius of the non-conducting disc.

Constant values of ζ and η can be chosen to represent the heat flow lines and the isothermal surfaces respectively [19]. In particular $v = 0$ describes the isothermal interface T_i and $u = 0$ describes the adiabatic disc itself. Introducing (9) into (1) we obtain a new expression for the Laplacian:

$$\frac{\partial}{\partial u}\left[(1 + u^2)\frac{\partial T}{\partial u}\right] + \frac{\partial}{\partial v}\left[(1 - v^2)\frac{\partial T}{\partial v}\right] = 0. \tag{10}$$

Solutions to (10) are found by separation of variables; by assuming $T = U(u) . V(v)$ and introducing into (10) we get:

$$\frac{1}{U}\frac{\mathrm{d}}{\mathrm{d}u}\left[(1 + u^2)\frac{\mathrm{d}U}{\mathrm{d}u}\right] = -\frac{1}{V}\frac{\mathrm{d}}{\mathrm{d}v}\left[(1 - v^2)\frac{\mathrm{d}V}{\mathrm{d}v}\right] = m^2. \tag{11}$$

Several solutions are possible for specific values of the separation constant, m^2, and they are of the type of particular integrals, since the boundary conditions have not been utilized. Thus, for $m^2 = 2$, we find as suggested by:

$$\frac{\mathrm{d}}{\mathrm{d}u}\tan^{-1} u = (1 + u^2)^{-1},$$

and

$$\frac{\mathrm{d}}{\mathrm{d}v}\tanh^{-1} v = (1 - v^2)^{-1},$$

that

$$T(u, v) = U . V = \{Au + B[u \tan^{-1} u + 1]\} . \times \{Cv + D[v \tanh^{-1} v - 1]\} \tag{12}$$

satisfies (10).

Since $T(u, 0) = T_i$, we can select $D = 0$ in (12), and add the constant T_i, because the temperature must be finite along $v = 1$, i.e. along the z-axis. The temperature in the auxiliary coordinates now becomes:

$$T(u, v) = V\{A'u + B'[u \tan^{-1} u + 1]\} + T_i. \tag{13}$$

In order for the solution to be symmetric about the common interface we must set $A' = 0$. The remaining coefficient can be determined from the condition at distances far from the interface. Since $\tan^{-1} u \to \pi/2$ as $u \to \infty$, the temperature gradient becomes, for large values of u,

$$\lim_{z \to \infty} \frac{\partial T}{\partial z}(u, v) \to \frac{\partial}{\partial z}\left[B'vu\frac{\pi}{2}\right] = B'\frac{\pi}{2}\frac{1}{a}.$$

Upon introducing condition (6) gives

$$B' = \frac{2a}{\pi}\frac{Q}{K_1 \pi b^2}.$$

The temperature distribution in the upper half of the heat flux tube can now be expressed as:

$$T_1(u, v) = T_i + \frac{2}{\pi} a \frac{Q}{K_1 \pi b^2}[u\,v \tan^{-1} u + v] \tag{14}$$

or as

$$T_1(\zeta, \eta) = T_i + \frac{2}{\pi} a \frac{Q}{K_1 \pi b^2}[\sinh \zeta \sin \eta \tan^{-1} (\sinh \zeta) + \sin \eta]. \tag{15}$$

This equation satisfies (1) and all the boundary conditions (2)–(6). A similar expression can be obtained for the lower half of the heat flux tube, but here K_2 is to be used and the sign before the second term will be negative.

An approximate expression for the temperature distribution in polar coordinates can be obtained if $r/a \geqslant 2{\cdot}5$:

$$T_1(r, z) = T_i + \frac{2\,a\,Q}{\pi K_1 \pi b^2} \times \left\{ \frac{z}{a} \tan^{-1}\left[\frac{\sqrt{(z^2 + r^2)}}{a}\right] + \frac{z}{\sqrt{(z^2 + r^2)}} \right\}. \quad (16)$$

THERMAL CONTACT RESISTANCE

In Fig. 5 we have shown the isothermal surfaces in the region near the interface. It can be shown that the maximum interface temperature

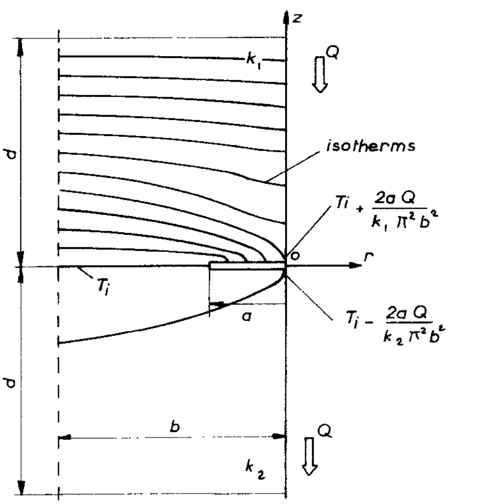

FIG. 5. Temperature distribution near the interface.

on the upstream side (solid when the heat flows from solid to liquid metal) occurs at the center of the disc and has the value $T_i + (2/\pi)(aQ/K_1\pi b^2)$. The temperature distribution over the disc from (15) with (9) is:

$$T_1(r, 0) = T_i + \frac{2}{\pi}\frac{aQ}{K_1 \pi b^2}\sqrt{\left[1 - \left(\frac{r}{a}\right)^2\right]}, \quad 0 \leqslant r \leqslant a. \quad (17)$$

From symmetry arguments it can be shown that the minimum interface temperature on the downstream side (liquid metal) occurs at the center of the disc and the temperature distribution over the disc is:

$$T_2(r, 0) = T_i - \frac{2}{\pi}\frac{aQ}{K_2 \pi b^2}\sqrt{\left[1 - \left(\frac{r}{a}\right)^2\right]}, \quad 0 \leqslant r \leqslant a. \quad (18)$$

We can define an average interface temperature for both sides of the interface as:

$$T_{a_1}(z = 0) = \frac{1}{\pi b^2}\int_0^b T_1(r, 0)\, 2\pi r\, \mathrm{d}r = T_i + \frac{4}{3K_1\pi^2}\frac{a^3 Q}{b^4}, \quad (19)$$

and

$$T_{a2}(z = 0) = \frac{1}{\pi b^2}\int_0^b T_2(r, 0)\, 2\pi r\, \mathrm{d}r = T_i - \frac{4}{3K_2\pi^2}\frac{a^3 Q}{b^4}. \quad (20)$$

Figure 6 shows the complete temperature distribution for an elemental heat flux tube, and it is seen that the temperature along the wall of the flux tube is continuous through the interface $T_1(r = b) = T_2(r = b)$ when $z = 0$. The average temperature as defined by (19) and (20) will be discontinuous across the interface. We could have shown the temperature distribution along the z-axis in both the solid and liquid metal, and this also would be discontinuous across the interface. All three of these temperature distributions $T(r = b)$, $T(r = 0)$, and T_a will approach asymptotically the same equation as $z \to \infty$, i.e.

$$T_a = T(r = b) = T(r = 0) = T_i + \frac{2}{\pi}\frac{aQ}{K_1 \pi b^2}\left[\frac{\pi}{2}\frac{z}{a} + 1\right]$$

for the solid. A similar argument holds for the liquid metal side of the elemental heat flux.

The temperature drop across the interface

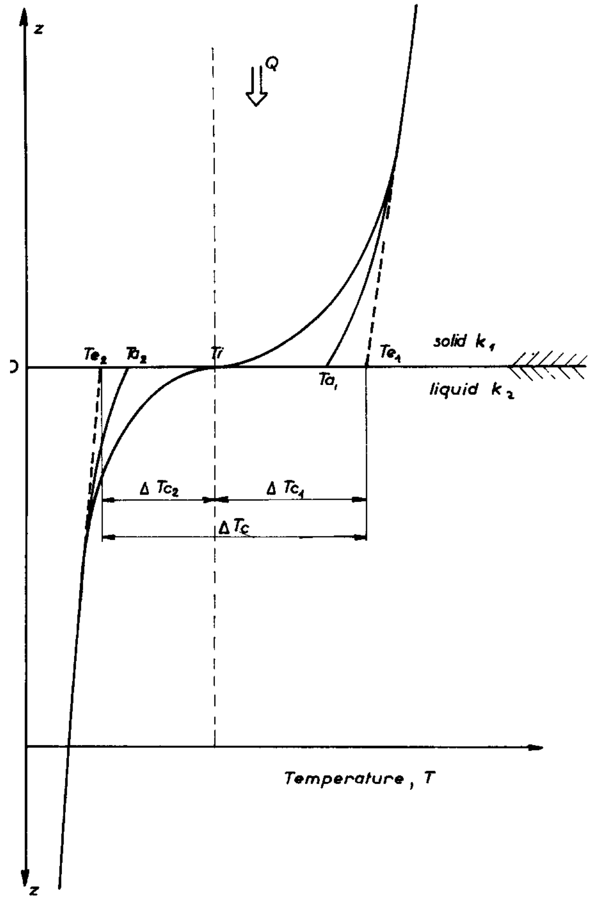

FIG. 6. Temperature distribution in heat flux tube.

due to the presence of the gas cavity is defined as the difference between the interface temperatures which are obtained by extrapolating from large z (both sides) to the interface, Fig. 6, so that:

$$\Delta T_c = T_{e_1} - T_{e_2} = \Delta T_{c_1} + \Delta T_{c_2}. \quad (21)$$

Consider one side only (solid, for example) where

$$\Delta T_{c_1} = \lim_{z \to \infty} \left[T_{a_1} - z \frac{\partial T_{a_1}}{\partial z} - T_i \right]; \quad (22)$$

but it can also be shown that

$$T_{a_1}(z \gg 0) = T_{a_1}(z = 0) + \frac{2}{\pi} \frac{aQ}{K_1 \pi b^2} \left[\frac{z}{a} \frac{\pi}{2} \right],$$

and also that

$$\frac{\partial T_{a_1}}{\partial z} = \frac{2}{\pi} \frac{aQ}{K_1 \pi b^2} \left[\frac{1}{a} \frac{\pi}{2} \right],$$

and so that (22) reduces to

$$\Delta T_{c_1} = T_{a_1}(z = 0) - T_1, \quad (23)$$

where $T_{a_1}(z = 0)$ is defined by (19).

Similarly for the liquid metal side we have:

$$\Delta T_{c_2} = T_i - T_{a_2}(z = 0). \quad (24)$$

Adding (23) and (24) gives us the total temperature drop across the interface due to the presence of the gas cavity, thus:

$$\Delta T_c = \frac{8a^3 Q}{3\pi^2 K b^4} \quad (25)$$

where $K = 2\,K_1 K_2 / K_1 + K_2$.

The thermal resistance across the interface is defined as:

$$R_c = \frac{T_{el} - T_{e2}}{Q} = \frac{\Delta T_c}{Q} = \frac{8a^3}{3\pi^2 K b^4}. \quad (26)$$

where K represents the physical characteristics and a, b represent the geometric characteristics of the interface. Equation (26) can be written in terms of the gas cavity density, and the wettability number W which is defined as the ratio of the wetted area to the total apparent area, i.e. $W = A_w/A_a = (A_a - A_g)/A_a$.

Thus we have:

$$\frac{a}{b} = [1 - W]^{\frac{1}{2}}, \quad (27)$$

and

$$b = 1/\sqrt{n\pi}. \quad (28)$$

Thus, by substitution of (27) and (28) into (26), we obtain the thermal interface or contact resistance:

$$R_c = 0{\cdot}480 \frac{\sqrt{n}}{K} [1 - W]^{\frac{3}{2}}. \quad (29)$$

This expression shows that the thermal resistance depends upon the thermal properties of the solid and the liquid metal, the density of cavities and the wettability number. The resistance is independent of heat flux density. Once the number of cavities is fixed, the resistance can only be changed by means of the wettability number W, i.e. by the size of the cavities.

EFFECT OF SYSTEM PRESSURE

To determine the effect of system pressure let us use the model shown in Fig. 3(c). We also assume that the number of cavities and the total weight of gas is fixed, i.e. the system has been filled under a certain gas pressure, say argon at ambient conditions, and, therefore, the quantity of gas at the interface will remain constant as the system pressure is changed.

The cavity is modelled as a cone subtending a very large angle $\alpha \geqslant 170°$ (α depends upon the surface roughness, surface tension, etc.). The height δ of the cavity is very small relative to the cross section.

The volume of the cavity can be written as

$$V = B a^3 \tag{30}$$

where B is the proportionality constant and a is the radius of the cavity. Assuming perfect gas laws we can for a constant system temperature write that a_2 and a_1 are related to the pressure change by

$$\left(\frac{a_2}{a_1}\right)^3 = \frac{P_1}{P_2}. \tag{31}$$

Suppose that n is fixed, then b is also constant and we can write

$$\frac{a_2}{a_1} = \left[\frac{1 - W_2}{1 - W_1}\right]^{\frac{1}{2}}, \tag{32}$$

where W_2 and W_1 correspond to P_2 and P_1, respectively.

Utilizing (31) and letting $P = P_1/P_2$ we obtain the following relationship between W_1, W_2 and P:

$$\frac{1 - W_2}{1 - W_1} = P^{\frac{2}{3}}. \tag{33}$$

When (33) is substituted in the ratio R_{c_2}/R_{c_1} we obtain

$$R_{c_1}P_1 = R_{c_2}P_2 = \text{constant}, \tag{34}$$

which shows that for a fixed system (constant gas weight and constant gas cavity density) and constant operating temperature, the thermal contact resistance will decrease hyperbolically with increasing system pressure.

COMPARISON OF THE THEORY WITH THE DATA OF BLEUNVEN ET AL. [17]

The results of the thermal analysis developed in this paper on thermal contact resistance made use of some assumptions which are here summarized:

1. Heat transfer occurs across a flat interface between a solid and a stationary liquid metal.
2. The gas cavity is impervious to heat flow.
3. The interface is free of any oxide and the contact beyond the gas cavity is assumed to be perfect.
4. The shapes of each gas cavity and associated heat flux tube were idealized as a circular disc and a coaxial tube, respectively.
5. All gas cavities were assumed to be uniformly distributed and of the same size.
6. The quantity of gas in the cavities was assumed to be constant independent of temperature and pressure changes.

The test data of Bleunven *et al.* [17] satisfy the first three approximation listed above, and it is assumed that the system used by them satisfies the last three assumptions to a first-order approximation.

To facilitate the comparison between this theory and the experimental results, the thermal resistance data has been normalized to the thermal resistance observed when the system pressure was 0·6 bars absolute. It is clear that there is good qualitative agreement between the theory, shown as a solid line in Fig. 7, and the test data of Bleunven *et al.* The tendency for the test data to be lower than the prediction

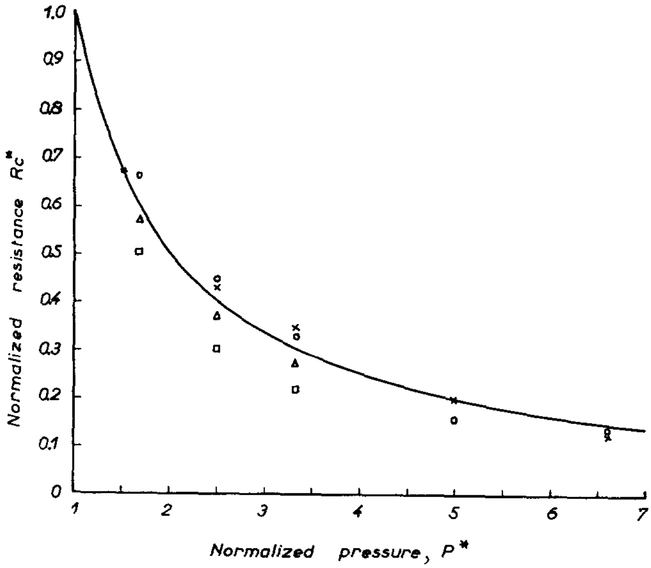

FIG. 7. Resistance R_c^* vs. pressure P^*.

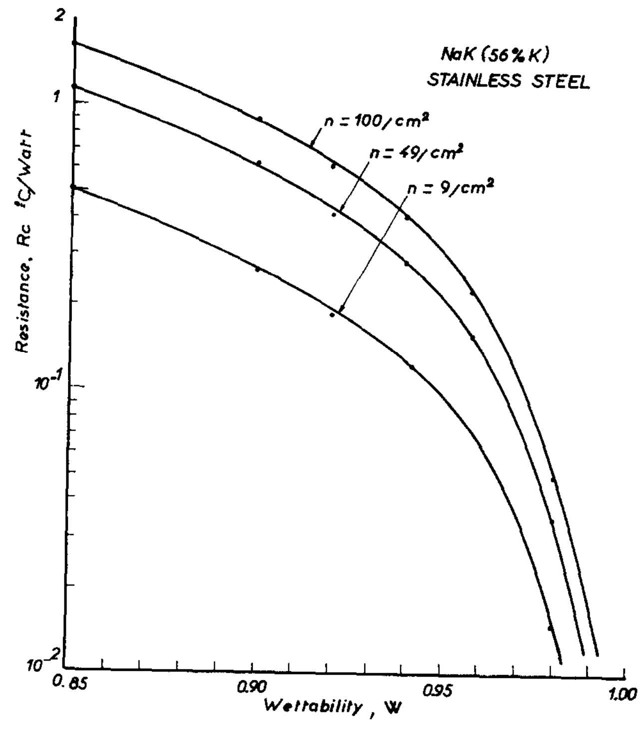

FIG. 8. Resistance R_c vs. wettability number W.

may be due to the fact that assumption 6 is not strictly true; i.e. the quantity of gas in the cavities may be a mixture of the cover and liquid metal vapor and would not remain constant with increasing system pressure.

Calculated values of the thermal contact resistance for a stainless-steel/NaK (56% K) interface are presented in Fig. 8 for several values of gas cavity density and the wettability number. It was assumed that a realistic range for the wettability number was 0·85–1·0 and most probably the values encountered in practice would lie in the range 0·94–0·995 when the liquid metal readily wets the solid. To show the effect of the gas cavity density, values of 9, 49 and 100 cavities per cm^2 have been arbitrarily selected. The lowest value will correspond to the case where the solid surface is smooth and the liquid metal easily wets the solid; on the other hand, the largest value will correspond to the case where the surface is quite rough and there is poor wetting of the solid.

It is not possible, at this time, to predict the gas cavity density or the wettability number from the description of the apparatus employed. The two parameters are not independent, but if n is fixed, W can be considered as a measure of the size of the cavities.

Note that the predicted values of the thermal contact resistance for $W > 0{\cdot}92$ qualitatively agree with the test data, Figs. 1 and 2. It appears that $n = 100$ cavities per cm^2 corresponds to the case where the system was placed under a vacuum of 10^{-3} mmHg before and after the filling with NaK, and, once filled, argon was used as the cover gas. On the other hand, $n = 9$ appears to correspond to the case where the system was filled and emptied twice with NaK under atmospheric pressure, and before the third filling, the system was placed under a vacuum of 10^{-3} mmHg, which lasted 1 h. The system was then filled with NaK at 300°C under a cover of argon. Obviously this method will reduce the number of gas cavities and also increase the percentage of the interface wetted by the liquid metal.

CONCLUSIONS

The thermal contact resistance at a solid stationary liquid metal interface based on certain conditions is shown to depend upon the thermal conductivities of the solid and liquid metal, the number of gas cavities per unit apparent interface area and upon wettability number. The theory was compared with test results obtained under conditions which completely satisfied three of the six assumptions, and to a first approximation, satisfied the other three assumptions made.

Although equations (29) and (34) give a highly simplified picture of this complex heat-transfer phenomenon, the theory provides information on the pertinent parameters which determine the magnitude of thermal contact resistance at a solid–liquid metal interface. This resistance can be made small be decreasing n and increasing W, which can be accomplished by using smooth surfaces, suitably treating the system to remove trapped gases and operating at high system pressures.

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ANALYSE DE LA RÉSISTANCE THERMIQUE D'UNE INTERFACE ENTRE UN SOLIDE ET UN MÉTAL LIQUIDE AU REPOS DUE À LA PRÉSENCE DE CAVITÉS GAZEUSES

Résumé—Une analyse mathématique de la résistance thermique à l'interface est exécutée sur un tube de flux de chaleur élémentaire idéalisé dans le cas d'une cavité gazeuse interposée entre un solide et un métal liquide au repos. La résistance thermique est exprimée en fonction des conductivités thermiques du solide et du métal liquide, du nombre des cavités gazeuses par unité de surface apparente et d'un nombre de mouillabilité. La dernière partie de l'article est consacrée à une comparaison qualitative entre la théorie et les résultats d'essais de Blewen *et al.* On a observé un bon accord compte-tenu des hypothèses faites.

EINE ANALYSE DES THERMISCHEN WIDERSTANDS AN EINER UNBEWEGTEN FLÜSSIGMETALLSCHICHT INFOLGE VON GASEINSCHLÜSSEN

Zusammenfassung—Der thermische Kontaktwiderstand wurde an einer idealisierten, elementaren Wärmestromröhre mathematisch untersucht für den Fall eines gasgefüllten Hohlraums zwischen einer festen und einer ruhenden flüssigen Metallschicht. Der thermische Widerstand wird angegeben in Abhängigkeit der Wärmeleitfähigkeit des festen und des flüssigen Metalls, der Zahl der Gashohlräume pro Flächeneinheit und einem Benetzbarkeitskoeffizienten. Der letzte Teil der Arbeit ist dem qualitativen Vergleich zwischen dieser Theorie und den Testergebnissen von Bleuven et al gewidmet. Die Übereinstimmung innerhalb der gemachten Annahmen war gut.

АНАЛИЗ ТЕПЛОВОГО СОПРОТИВЛЕНИЯ НА ПОВЕРХНОСТИ РАЗДЕЛА ««ТВЕРДЫЙ-НЕПОДВИЖНЫЙ ЖИДКИЙ МЕТАЛЛ»», ВЫЗВАННОГО НАЛИЧИЕМ ГАЗОВЫХ КАВЕРН

Аннотация—Проведен математический анализ термического сопротивления поверхности раздела для начального участка идеализированной тепловой трубки, когда полость с газом располагается между твердым телом и стационарным жидким металлом. Термическое сопротивление выражается в зависимости от теплопроводности твердого и жидкого металла, числа полостей с газом на единицу кажущейся площади и коэффициента смачиваемости. В последней части статьи приводится качественное сопоставление теории с экспериментальными данными Блювена и других. При сделанных допущениях наблюдается хорошее соответствие между теорией и опытом.