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THE INFLUENCE OF WALL THICKNESS, THERMAL CONDUCTIVITY AND METHOD OF HEAT INPUT ON THE HEAT TRANSFER PERFORMANCE OF SOME RIBBED SURFACES

P. G. BARNETT

Heat Transfer Section, UKAEA, RDL, Windscale Works, Sellafield, Cumberland, England

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Abstract—The average heat transfer coefficient of a surface can be increased by adding turbulence promoters. Use of such devices produces a non-uniform distribution of heat transfer coefficient over the surface. Consequently factors affecting the ease with which heat can reach the areas of high heat transfer coefficient will affect the distribution of surface temperature; examples are the thickness and thermal conductivity of the material forming the surface and the method of heat input. Using a computer program to solve the heat conduction equation the magnitude of these effects has been assessed for some surfaces having transverse ribs. The results show that the performance of geometrically identical surfaces, as judged by, say, the peak surface temperature, may differ significantly. The implication is that such factors must be taken into account when, for example, experimental data obtained for a ribbed surface on a thick walled aluminium can are used to predict the peak surface temperature in a reactor of a geometrically similar surface on a thin walled can made of stainless steel.

NOMENCLATURE

c, c_j or $c_j(R/r_0, Z/r_0)$, increase of the can to coolant temperature difference relative to that for a surface of infinite thermal conductivity, equation (9);

e, rib height [m];

f or $f(R, Z)$, distribution of heat transfer coefficient;

F, denotes an unknown function;

h_b, base value of heat transfer coefficient distribution [W/m^2 °C];

h_{exp}, apparent experimental value of heat transfer coefficient, equations (10) and (14) [W/m^2 °C];

h_{av}, average heat transfer coefficient for a surface, equation (1) [W/m^2°C];

h_p, definition of average heat transfer coefficient used in practice, equation (2) [W/m^2 °C];

j, = 1 denotes input of heat by conduction;
= 2 denotes input of heat by electrical generation;

k, thermal conductivity of the can wall material [W/m °C];

n, normal to the surface;

p, rib pitch, equal to $7e$ in the calculations [m];

$P(m)$, $m = 1, 2, \ldots 5$, parameters in the correlations;

Pr, Prandtl number;

q, heat flux, defined by equation (15) [W/m^2];

Q_j, total heat input per unit length of the can by method j [W/m^2];

r_i, inner radius of can wall [m];

r_0, outer, or rib root radius, of can wall [m];

R, coordinate [m];

Re,	Reynolds number;
$s(R, Z)$,	heat generation rate per unit volume $[W/m^3]$;
St_{exp},	experimental value of Stanton number, equation (11);
St_p,	Stanton number based on h_p;
$t = r_0 - r_i$,	can wall thickness [m];
T,	temperature [°C];
T_b,	bulk coolant temperature [°C];
T_{exp},	wall temperature at mid point between ribs in experiment [°C];
T_j or $T_j(R, Z)$,	temperature at (R,Z) for Q_j [°C];
T_w,	average wall temperature [°C];
U,	$= e/r_0$;
V,	$= r_0 h_p/k$;
w,	width of rib, equal to e in the calculations [m];
X,	$= r_0 h_{exp}/k$;
Y,	$= t/e$;
Z,	coordinate [m];
δT,	value of ΔT_j for a material of infinite thermal conductivity [°C];
ΔT_j,	defined by equation (5) [°C].

INTRODUCTION

THE IMPROVEMENT in heat transfer performance obtained by adding transverse ribs of square cross section to a smooth cylindrical surface has led to the consideration of such ribbed surfaces for use in Advance Gas Cooled Reactors. A wide variety of ribbed surfaces has been studied experimentally, e.g. Wilkie [1], so that the optimum rib configuration may be determined. However, different experimental techniques have been used, e.g. Wilkie [1] and White [2], and in some cases significantly different results have been obtained for geometrically similar surfaces. The experimental techniques commonly used are to manufacture the heat transfer surface either from stainless steel and heat it by the passage of an electric current, or from aluminium and input the heat by conduction to the inner surface. In the reactor the surface would be made from stainless steel and the heat input by conduction and radiation from the nuclear fuel. The possibility that the different experimental techniques might account for the apparent variations in performance led to the series of calculations presented in this paper. The implications for predicting peak surface temperature in a reactor from experimental data are also discussed.

DESCRIPTION OF MODEL

The heat transfer surfaces considered are formed by adding transverse ribs of square cross section to a smooth circular cylinder. The flow of coolant is along the cylinder so that (R, θ, Z) cylindrical polar coordinates are appropriate and the problem is independent of θ. The rib height, width and pitch are denoted by e, w and p respectively. Attention is restricted to the particular values $p/e = 7{\cdot}0$ and $w/e = 1{\cdot}0$ which are relevant to the AGR. The outer radius, inner radius and thickness of the cylinder are denoted by r_0, r_i and t respectively. Typical values for e, w, p and r_0 are 0·28, 0·28, 2·03 and 7·62 mm respectively.

The distribution of heat transfer coefficient over a ribbed surface has been estimated by Kattchee and Mackewicz [3] using a mass transfer technique; it may be written $h_b \,.\, f(R, Z)$ where h_b is a base value of the distribution and $f(R, Z)$ is a dimensionless shape function. The base value h_b used by Kattchee and Mackewicz is the heat transfer coefficient of the smooth surface. It was selected by them because the surface studied had both smooth and ribbed sections so the level of the local ribbed surface heat transfer coefficient relative to the smooth value for the same coolant flow rate was immediately available. However, this link with the smooth surface is only an incidental feature of the calculations.

The distribution $f(R, Z)$ has no features, such as symmetry about some plane Z = constant, which lead to an obvious choice of surface length in the calculations. Since some limit must be introduced a distance equal to two rib pitches

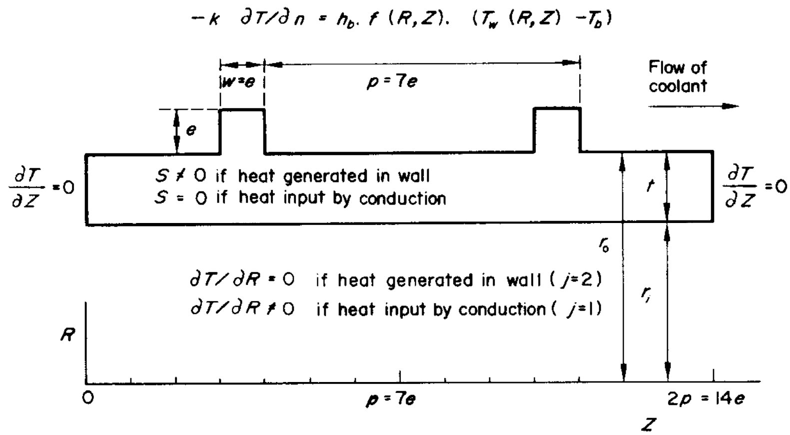

FIG. 1. Model used for the heat conduction calculations.

is considered, Fig. 1, and the, possibly incorrect, boundary condition $\partial T/\partial Z = 0$ applied when $Z = 0$ and $Z = 2p = 14e$. The importance of this assumption will be discussed later. The boundary condition at $R = r_i$ will be $\partial T/\partial R = 0$ if heat is generated in the can wall and ribs, but will be non-zero if the input of heat is by conduction.

The HEATRAN computer program, Collier [4], can be used to solve the heat conduction problems presented in Fig. 1 for different distributions of the volume generation of heat, $s(R, Z)$, in the can wall and ribs. The program requires a subdivision into triangles of the area in which heat conduction takes place. Figure 2 shows a typical distribution of nodal points and

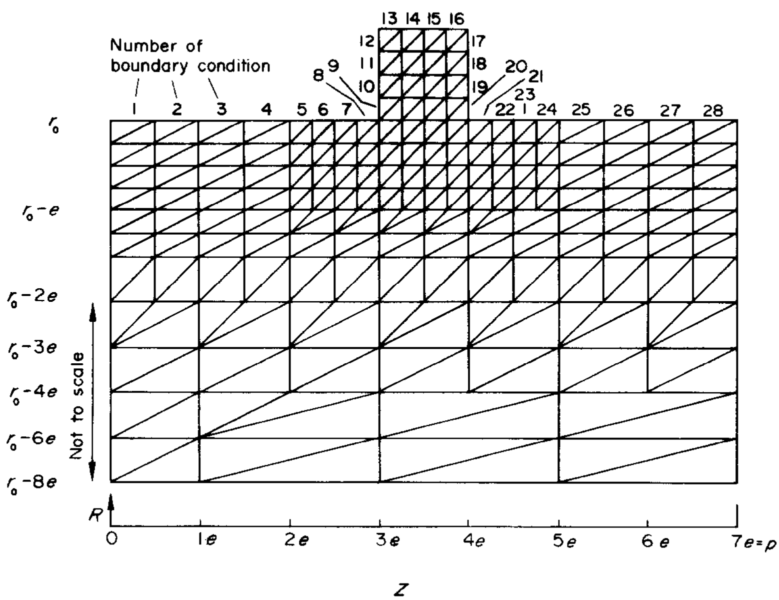

FIG. 2. Typical distribution of nodes and associated triangles.

connections used in the calculations. If, for example, the value of $r_i = r_0 - 2e$ is under consideration then the nodes having $R < r_0 - 2e$ would not be used. The computer program requires a constant heat transfer coefficient on any side of a triangle forming part of the boundary. To meet this requirement the heat transfer coefficient distribution measured by Kattchee and Mackewicz [3] has been simplified to that shown in Fig. 3.

definition of average heat transfer coefficient given by equation (1) is not the one most commonly used. In practice the surface area of the ribs is ignored when calculating $\int da$ and the area under the rib at the rib root radius, r_0, included. Thus the average used in practice, h_p, is given by

$$h_p = \int_{\text{rib pitch}} h_b \,.\, f(R, Z)\, da/2\pi r_0 p \tag{2}$$

$$= h_{av}(1 + e[2r_0 + w + e]/r_0 p). \tag{3}$$

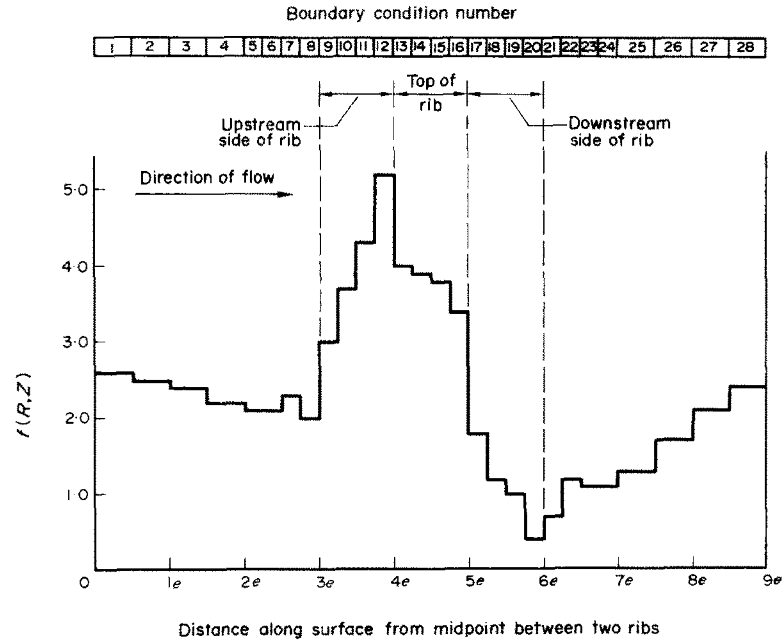

FIG. 3. Distribution of ribbed surface heat transfer coefficient.

The average heat transfer coefficient, h_{av}, can be calculated from

$$h_{av} = \int_{\text{rib pitch}} h_b \,.\, f(R, Z)\, da \Big/ \int_{\text{rib pitch}} da \tag{1}$$

where da is an element of surface area. This is also the heat transfer coefficient which a surface made of infinite conducting material will appear to have, no matter where the temperature is measured, because all parts of the surface will be at the same temperature. However, the

For the particular surfaces under consideration $w = e$ and $p = 7e$ giving

$$h_p = h_{av}(1 + \tfrac{2}{7} + 2e/7r_0). \tag{4}$$

The values of h_p and h_{av} vary slightly with e/r_0 over the range of interest as shown in Table 1.

When the input of heat is solely by conduction to the inner surface of a can the volume generation of heat in the can wall and ribs, $s(R, Z)$, will be zero. However, when the can is heated by

Table 1

$\frac{e}{r_0}$	$\frac{h_{av}}{h_b}$	$\frac{h_p}{h_b}$
0·020	2·302	2·973
0·035	2·305	2·987
0·050	2·309	3·002
0·065	2·312	3·016

passing an electric current through it $s(R, Z)$ will be non-zero and also non-uniform. The equations of heat conduction and electrical conduction are identical so the HEATRAN program can be used to solve the electrical conduction problem first and from the distribution of electrical potential the variations in heat generation for the heat conduction calculation may be determined. In the electrical conduction calculations a given potential drop per rib pitch is assumed. Thus the boundary conditions are $T = 0$ when $Z = 0, T = 100$ (say) when $Z = 2p$, and $\partial T/\partial n = 0$ over ribbed surface and when $R = r_i$. An example of the distribution of heat generation obtained is given in Fig. 4; the numbers show the ratio of the local volume generation of heat relative to that which would arise if the ribs were not present.

DIMENSIONAL ANALYSIS OF THE MODEL

Let Q_j denote the total heat input per unit length of the surface with $j = 1$ indicating that the heat input is by conduction at $r = r_i$ and $j = 2$ indicating internal electrical generation in the can wall and ribs. The temperature in the can at the point (R, Z) is denoted by $T_j(R, Z)$ and T_b is the bulk coolant temperature (assumed constant over the short distance of two rib pitches). Defining

$$\Delta T_j = T_j(R, Z) - T_b \tag{5}$$

we may write

$$\Delta T_j = F(f, Q_j, R, Z, r_0, e, w, p, t, h_p, k) \tag{6}$$

where F denotes an unknown function and k is the thermal conductivity of the material forming the surface. Dimensional analysis using dimensions of heat, length, time and temperature simplifies equation (6) to

$$[\Delta T_j \,.\, r_0 \,.\, h_p/Q_j] = F(f, [R/r_0], [Z/r_0], [e/r_0], [w/e], [p/e], [t/e], [r_0 \,.\, h_p \,./k]). \tag{7}$$

Note that the same indicial equations are obtained for the dimensions of heat and time so the 11 independent variables combine to form 8 dimensionless groups. If the heat transfer surface could be constructed from material

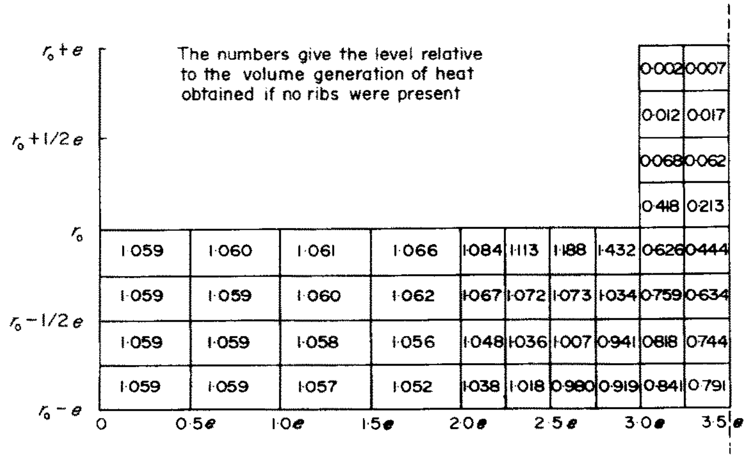

FIG. 4. Distribution of heat generation in a can having transverse ribs; $e/r_0 = 0{\cdot}035$, $t/e = 1{\cdot}0$.

having an infinite thermal conductivity then all parts of the surface would be at the same temperature; in particular $[r_0 . h_p/k]$ would be zero and ΔT_j would be independent of t,R,Z and the method of heat input. Denoting this value of ΔT_j by δT we have

$$[\delta T . r_0 . h_p/Q_j] = F(f, [e/r_0], [w/e], [p/e]). \quad (8)$$

Because the δT values are independent of experimental technique they form a useful base for comparing the effects of $[r_0 . h_p/k]$, $[t/e]$ and method of heat input at various points in the can wall or on the heat transfer surface. The results of the HEATRAN calculations will be presented in terms of the quantity $c_j(R/r_0, Z/r_0)$ defined by

$$c_j(R/r_0, Z/r_0) = (\Delta T_j - \delta T)/\delta T. \quad (9)$$

Thus $c_j(R/r_0, Z/r_0)$ gives the increase of the local can to coolant temperature difference relative to that obtained for a surface of infinite thermal conductivity. Note that the "apparent" experimental value of the average heat transfer coefficient will depend upon the position at which the temperature is measured. Thus

$$h_{\text{exp}}(R/r_0, Z/r_0) = h_p/[1 + c_j(R/r_0, Z/r_0)] \quad (10)$$

and

$$St_{\text{exp}}(R/r_0, Z/r_0) = St_p/[1 + c_j(R/r_0, Z/r_0)] \quad (11)$$

where St_p is the Stanton Number based on h_p.

RESULTS

Thin cans, heat input by conduction

The calculations for thin walled cans with heat input by conduction to the inner surface are relevant to an AGR. The thermal conductivity of the nuclear fuel is usually an order of magnitude less than that of the stainless steel can, so the contribution of the fuel to the flow of heat in the Z direction is ignored. Figure 5 shows the distribution of $100\, c_1(R/r_0, Z/r_0)$ obtained for the particular case of $e/r_0 = 0{\cdot}035$, $t/e = 1{\cdot}5$ and $r_0 h_p/k = 2{\cdot}1$ (or $r_0 h_b/k = 0{\cdot}7$). It can be seen that the hottest point on the heat transfer surface is about one rib height upstream of the mid-rib position. The peak surface temperature is important in reactor assessments because it affects can endurance and reflects through to the peak fuel temperature. Although in the particular example shown the peak outside surface temperature is not associated with the peak on the inside surface it is thought reasonable to present information for the peak outside surface temperature for can endurance calculations and use a uniform temperature rise through the can wall to give a slightly pessimistic peak fuel temperature.

On completion of the calculations it was found that a plot of c_1 at one rib height upstream of the mid-rib position against the group $(h_p e^2/kt) = (h_p r_0/k)\ (e/r_0)\ (e/t)$ for various e/t

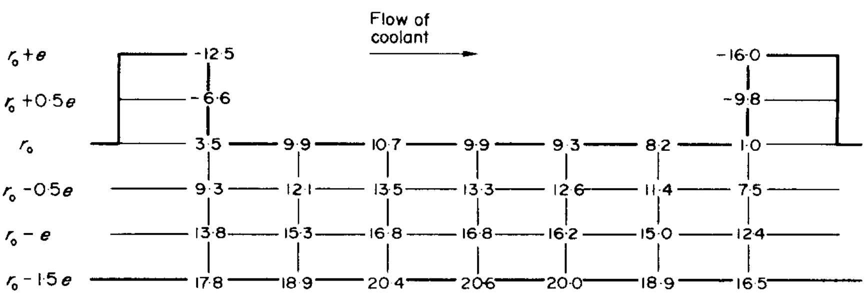

FIG. 5. Distribution of 100 $c_1(R/r_0, Z/r_0)$ for $e/r_0 = 0{\cdot}035$, $t/e = 1{\cdot}5$ and $r_0 . h_p/k = 2{\cdot}1$.

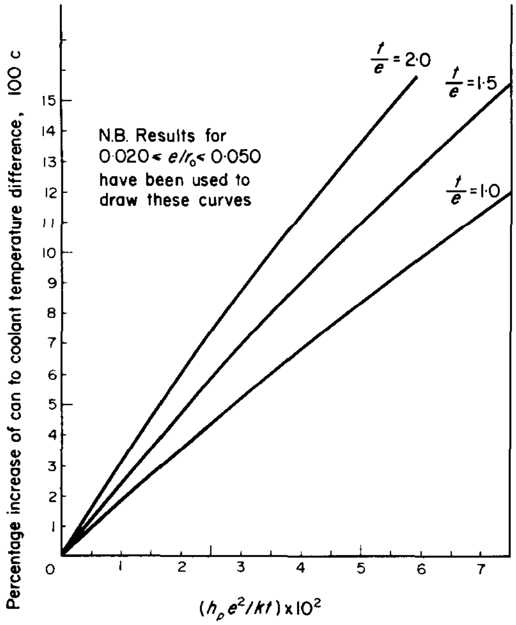

FIG. 6. Variation of 100.c on the surface 2e downstream of a rib for $j = 1$, i.e. heat input by conduction.

was not sensitive to e/r_0. Figure 6 shows the calculated results in this form for $0{\cdot}020 \leqslant e/r_0 \leqslant 0{\cdot}050$, $1{\cdot}0 \leqslant t/e \leqslant 2{\cdot}0$ and $0{\cdot}0 \leqslant h_p r_0/k \leqslant 2{\cdot}11$ (or $0{\cdot}0 \leqslant h_b r_0/k \leqslant 0{\cdot}7$). The results have been correlated in terms of $r_0 h_p/k$ and are represented to within 0·001 by the following equation.

$$c = U \,.\, V[P(1) + P(2)U + P(3)V + P(4)U \,.\, V] \times [1 + P(5)Y^{-2}] \qquad (12)$$

where $P(1) = 156{\cdot}08$ $\quad Y = t/e$
$P(2) = 57{\cdot}64$ $\quad U = e/r_0$
$P(3) = -2{\cdot}29$ $\quad V = r_0 h_p/k$
$P(4) = -226{\cdot}05$
$P(5) = 0{\cdot}16288.$

Thick cans, heat input by conduction

Thick walled cans made of aluminium have been used by several experimenters studying the behaviour of ribbed surfaces, e.g. Wilkie [1]. Input of heat to these surfaces has been effected by using an internal heater bar. The thermal conductivity of aluminium is an order of magnitude greater than that of stainless steel so the values of $r_0 h_p/k$ may be significantly different from the reactor. In fact the range covered in Wilkie's experiments is $0{\cdot}0 \leqslant r_0 h_p/k \leqslant 0{\cdot}12$ (or $0{\cdot}0 \leqslant r_0 h_b/k \leqslant 0{\cdot}04$). The thick walls of the aluminium cans also contribute significantly to the reduction of temperature differences in the can wall, but the calculated effects for $t/e > 8{\cdot}0$ were found to be the same as those for $t/e = 8{\cdot}0$. Consequently the investigation

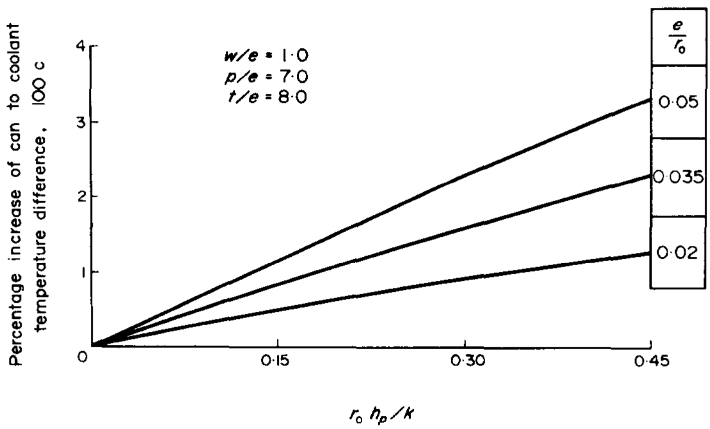

FIG. 7. Variation of 100.c on the surface at the mid rib position for $j = 1$, i.e. heat input by conduction.

was restricted to the one value of $t/e = 8{\cdot}0$ as opposed to the range $10{\cdot}4 \leqslant t/e \leqslant 31{\cdot}5$ covered in the experiments. The results shown in Fig. 7 are for the point on the surface midway between two ribs. This point was selected, rather than the point $2e$ downstream of a rib, because the experimenters usually attempt to locate thermocouples near the mid rib plane and adjust for temperature drop through the can wall to get the temperature at the surface.

Thin cans, internal heat generation

A number of experimenters have used stainless steel cans heated by passing an electric current through the can wall, e.g. White and White [2]; many combinations of t/e and e/r_0 are involved. So that the magnitude of the various effects can be assessed the particular values of t/e and e/r_0 chosen for the HEATRAN calculations are such that adjustments for the various experimental configurations can be obtained by interpolation. The calculated results are shown in Fig. 8 where use is made again of the insensitivity to e/r_0 of c against $h_p e^2/kt$. The value of c plotted refers to that point on the surface midway between the ribs and is selected for reasons given in the previous section. The results for $e/r_0 = 0{\cdot}035$ are of particular interest for reactor applications so these results have been correlated and are generated by the following equation.

$$c = X[P(1) + P(2)X + P(3)X^2] \times [1 + P(4)Y^{-2}] \qquad (13)$$

where

$$P(1) = 5{\cdot}4300$$
$$P(2) = -0{\cdot}34998$$
$$P(3) = 0{\cdot}052572$$
$$P(4) = 0{\cdot}25423$$
$$Y = t/e$$
$$X = r_0 h_{exp}/k$$
$$h_{exp} = q/(T_{exp} - T_b) \qquad (14)$$

and

$$q = Q_j/2\pi r_0. \qquad (15)$$

T_{exp} is the measured temperature of the can surface midway between the ribs and h_{exp} is the "apparent" heat transfer coefficient based on this temperature. Equation (13) is valid for $e/r_0 = 0{\cdot}035$, $1{\cdot}0 \leqslant t/e \leqslant 3{\cdot}0$ and $0{\cdot}0 \leqslant r_0 h_{exp}/k \leqslant 2{\cdot}30$. Note that the group $r_0 h_{exp}/k$ is used instead of $r_0 h_p/k$; it has been chosen so that the value of h_p implied by the experiment may be calculated from

$$h_p = (1 + c)\, h_{exp} \qquad (16)$$

and no iteration procedure is involved.

The results shown in Fig. 8 for $1{\cdot}0 \leqslant t/e \leqslant 2{\cdot}0$ are similar to those in Fig. 6 indicating that the two methods of heat input do not give significantly different results in this range for the

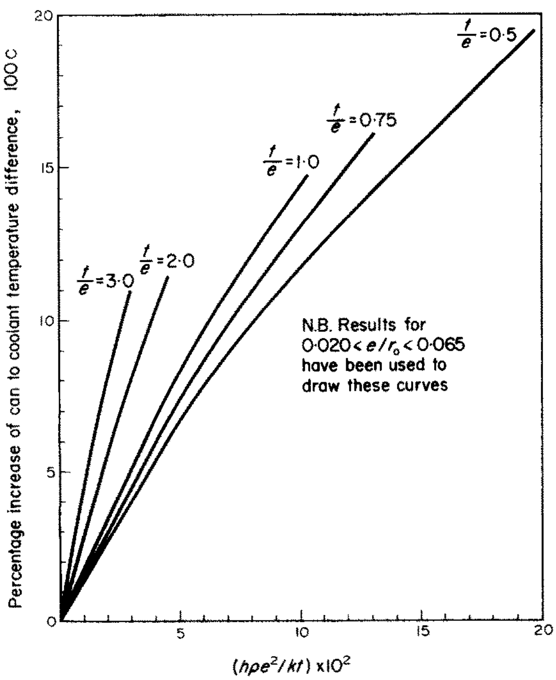

FIG. 8. Variation of 100.c on the surface at the mid-rib position for $j = 2$, i.e. heat input by electrical generation.

mid-rib region of the surface. However, it would be unwise to assume that the same observation would apply for $t/e < 1{\cdot}0$. Figures 6 and 8 could also be presented as plots of c against he/k instead of he^2/kt, but this would lead to confusion of the lines for varying t/e.

Checks on the calculations

To ensure that the information presented is accurate a number of checks have been made. The mesh size shown in Fig. 2 was found satisfactory for $t/e = 1{\cdot}0$. The test was a repeat calculation for the same heat input and boundary conditions, but using additional nodal points at $R = r_0 - \frac{1}{8}e$, $r_0 - \frac{3}{8}e$, $r_0 - \frac{5}{8}e$ and $r_0 - \frac{7}{8}e$ with the values of Z already involved; no significant differences were found. In fact the finer distribution of nodes so obtained was used for all the calculations having $t/e < 1{\cdot}0$.

The effect of the "possibly incorrect" boundary condition $\partial T/\partial Z = 0$ when $Z = 0$ and $2p$ was investigated by repeat calculations for which temperature distribution at $Z = p$ in the first calculation was inserted in the second calculation as the boundary condition at $Z = 0$ and $Z = 2p$. The differences for the nodal points between $Z = 5e$ and $Z = 9e$ were small and no changes were found at $Z = 7e$, the plane mid way between the ribs. It was discovered, however, that significant changes occurred in the planes $Z = 4e$ and $Z = 10e$, i.e. the downstream and upstream faces of the ribs. The results obtained for the repeat calculations were found to be approximately equal to the mean of those at corresponding points in the first calculation. Consequently the numbers presented in Fig. 5 for the downstream side of the rib are the mean of those obtained for $Z = 4e$ and $Z = 11e$ in the first calculation; a similar comment applies to the numbers for the upstream side of the rib.

The distribution of heat transfer coefficient estimated by Kattchee and Mackewicz [3] is not the only one available. Wilkie [5] has also used a mass transfer technique on a ribbed surface and obtained an estimate of the heat transfer coefficient distribution which differs significantly from that of Kattchee and Mackewicz on some parts of the surface. However, calculations using Wilkie's distribution showed that the effect on the value of c near the mid-rib position on the surface was small: typically the difference between the two values of c would be 10 per cent of the larger. Kattchee and Mackewicz's distribution always gave the larger value.

APPLICATION OF RESULTS

Figures 6–8 show how the heat transfer performance of a ribbed surface, as judged by the temperature at or near the mid position between two ribs, varies with e/r_0, t/e, $r_0 h_p/k$ and the method of heat input for the one particular combination of $w/e = 1{\cdot}0$, $p/e = 7{\cdot}0$ and the Kattchee and Mackewicz distribution of heat transfer coefficient. The largest value of $r_0 h_p/k$ shown in each figure is typically $1{\cdot}5$ times the largest value encountered in practice. The implication is that experimental data for one particular surface geometry cannot be applied to a similar geometry without consideration of the experimental techniques, etc, involved. In particular, these effects must be taken into consideration when comparing different sets of data for nominally the same surface or predicting peak surface temperatures in a reactor from experimental data. The sequence of calculations necessary to obtain a reactor prediction is as follows:

(i) The "apparent" experimental value of the heat transfer coefficient, h_{exp}, must be calculated based on the measured surface temperature at the mid-rib position.

(ii) Using Figs. 7 and 8, or equation (13) as applicable, the value of h_p (which corresponds to the heat transfer coefficient of a surface made from an infinite thermal conductivity material) must be found. This is an iterative procedure unless, as in the case of equation (13), the "reverse correlation" has been obtained.

(iii) The value of h_p must be scaled to reactor conditions. This will usually involve a correlation of the form

$$St = F(Re, Pr, T_w/T_b). \quad (17)$$

(iv) The value of $r_0 h_p/k$ for the reactor is calculated and then Fig. 6 or equation (12) will allow the "effective" value of h at a distance $2e$ downstream of a rib to be found from

$$h = h_p/(1 + c). \quad (18)$$

This value, with the heat input, will allow calculation of the peak surface temperature.

CONCLUSIONS

1. The surface temperature may vary significantly over a ribbed surface even though the ribs are physically small compared with the dimensions of the surface.
2. The surface temperature varies with the thickness and thermal conductivity of the surface material and the method of heat input.
3. These factors must be taken into account when predicting the peak surface temperature in a reactor from experimental data or comparing different sets of experimental data for nominally the same surface.

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INFLUENCE DE L'EPAISSEUR DE PAROI, DE LA CONDUCTIVITE THERMIQUE ET DE LA METHODE DE CHAUFFAGE SUR LE TRANSFERT THERMIQUE DE QUELQUES SURFACES ANNELEES

Résumé—Le coefficient de transfert thermique moyen d'une surface peut être augmenté par l'adjonction de promoteurs de turbulence. L'utilisation de tels éléments provoque une distribution non uniforme du coefficient de transfert thermique sur la surface. Des facteurs liés à la facilité avec laquelle la chaleur peut atteindre les zones où le coefficient de transfert thermique est élevé affectent la distribution de la température de surface; des exemples sont l'épaisseur et la conductivité thermique du matériau et la méthode de chauffage. Utilisant un programme de calcul pour résoudre l'équation de la conduction thermique l'importance de ces effets a été déterminée pour quelques surfaces ayant des annelures transversales. Les résultats montrent que les performances de surfaces géométriquement identiques, jugées à partir du pic de la température de surface, peuvent différer de façon significative. Ceci implique que de tels facteurs doivent être pris en compte lorsque par exemple des résultats expérimentaux obtenus pour une surface annelée avec une paroi épaisse d'a'uminium sont utilisés pour prévoir le pic de température de surface dans un réacteur avec une surface géométriquement similaire avec une paroi mince en acier inoxydable.

DER EINFLUSS DER WANDSTÄRKE, DER WÄRMELEITFÄHIGKEIT UND DER ART DER WÄRMEZUFUHR AUF DIE WÄRMEÜBERTRAGUNGSLEISTUNG EINIGER BERIPPTER OBERFLÄCHEN

Zusammenfassung—Der mittlere Wärmeübergangskoeffizient einer Oberfläche kann durch das Anbringen von Turbulenzerzeugern gesteigert werden. Die Anwendung solcher Anordnungen erzeugt über der Oberfläche eine nicht einheitliche Verteilung des Wärmeübergangskoeffizienten. Damit werden jene Faktoren, die den Wärmetransport in Gebiete mit hohem Wärmeübergangskoeffizienten begünstigen, die Temperaturverteilung der Oberfläche beeinflussen. Dies sind zum Beispiel die Wandstärke, die Wärmeleitfähigkeit des Oberflächenmaterials und die Art der Wärmezufuhr. Für einige Oberflächen mit Transversalrippen wurde die Grössenordnung dieser Einflüsse abgeschätzt, wobei ein Rechenprogramm zur Lösung der Wärmeleitgleichung benutzt wurde. Die Ergebnisse zeigen, dass der Wirkungsgrad geometrisch ähnlicher Oberfläche, beurteilt nach der maximalen Oberflächentemperatur, bedeutend differieren kann. Daraus lässt sich folgern, dass solche Faktoren berücksichtigt werden müssen, wenn z.B. experimentelle Ergebnisse für eine berippte Oberfläche einer dickwandigen Aluminiumbrennelementhülle benutzt werden, um die maximale Oberflächentemperatur einer geometrisch ähnlichen Oberfläche einer Hülle aus rostfreiem Stahl in einem Reaktor zu berechnen.

ВЛИЯНИЕ ТОЛЩИНЫ СТЕНКИ, ТЕПЛОПРОВОДНОСТИ И МЕТОДА ПОДВОДА ТЕПЛА НА ХАРАКТЕР ТЕПЛООБМЕНА НЕКОТОРЫХ ОРЕБРЕННЫХ ПОВЕРХНОСТЕЙ

Аннотация—С помощью турбулизаторов можно увеличить средний коэффициент теплообмена поверхности. Использование таких устройств обеспечивает неравномерное распределение коэффициента теплообмена по поверхности. Следовательно, факторы, определяющие скорость переноса тепла в область с высоким коэффициентом теплообмена, влияют также и на распределение температуры поверхности. Такими факторами являются толщина поверхности и теплопроводность материала поверхности, а также метод подвода тепла. Используя программу для решения уравнения теплопроводности, численно исследовалось влияние этих факторов на теплообмен для некоторых поверхностей с поперечными ребрами. Результаты показывают, что характеристики геометрически идентичных поверхностей, если судить, скажем, по максимальной температуре стенки, могут существенно отличаться. Эти факторы необходимо учитывать, когда, например, экспериментальные данные, полученные для оребренной поверхности цилиндра с толстой алюминиевой стенкой, используются для расчёта пиковой температуры поверхности реакторного контейнера с тонкими стенками из нержавеющей стали, имеющими поверхность, геометрически подобную поверхности экспериментального цилиндра.