Control of Flow Distribution by Mixing Headers

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The coolant flow distribution among parallel passages in a nuclear reactor (or boiler or heat exchanger) can be very sensitive to variations in heat input, channel dimensions, etc. In a previous paper this flow sensitivity was defined in terms of certain partial derivatives, which were related by analytical expressions to fluid properties and operating characteristics. Flow sensitivity contributes largely to potential malfunction, reduced efficiency, or failures. The use of valves and orifices was quantitatively evaluated for supercritical water in the earlier paper. The scope of this paper is to consider the utility of mixing headers. These mixing chambers are located along the flow passage as a common receiver for parallel flow from many channels. The headers, in turn, supply subsequent lengths of heated passages in parallel. Analytical expressions are derived for the effect of headers on flow, outlet-fluid enthalpy, and channel-wall temperatures. The limiting cases of minimum and complete mixing in the headers are considered and numerical results for water at supercritical pressures are given to show the marked increase in stability obtained by use of intermediate mixing headers.

Severe flow maldistributions can occur in nuclear reactors in which the coolant decreases appreciably in density or where the physical properties of the coolant are sharply temperature dependent. In such cases small variations in heat flux—either from tube to tube or with time—can cause significant fluctuations in flow, which, in turn, can materially reduce heat transfer efficiency or, in extreme cases, even result in tube burnout. In any case, strong flow variations are unfavorable since they make the heat exchange process difficult to control.

The flow distribution of expansible fluids among parallel heated channels was treated in a previous paper (*1*) in which the variation of flow with heat flux was evaluated in terms of *flow sensitivities*, which are functions of the physical properties of the coolant and such parameters as inlet and outlet temperature, heat flux, hydraulic diameter, and tube length. The paper also discussed the reduction of flow sensitivity through the use of valves and orifices.

The current paper deals with the control of flow sensitivity (and the resultant flow maldistributions) through the use of subdivided tubes jointed to intermediate flow-mixing headers which permit the equalization of temperatures and pressures across the tube bank. Blending the flows from all tubes in the intermediate headers reduces flow sensitivity by preventing the accumulation of adverse effects along the entire length of an abnormally powered tube.

Ideally, equalization of both temperature and pressure should be accomplished in the mixing headers to obtain the maximum reduction in flow sensitivity. In practice, however, the necessity for compactness limits the size of the headers, thereby making thorough lateral temperature equalization difficult. In recognition of this practical difficulty, flow-sensitivity relationships have been derived below for the case in which there is no thermal equalization in the intermediate header, as well as for the case in which perfect thermal equalization is obtained. All derivations pertain to a tube bundle with a single mixing chamber located midway along the flow path. For both cases considered, expressions are also obtained for the variation of outlet temperature and tube-wall temperature with variations in heat flux.

As an illustration of the reduction in sensitivity that can be effected, the flow, outlet-temperature, and wall-temperature sensitivities have been computed for a divided and undivided tube for a typical case in which water at supercritical pressure undergoes an eightfold expansion from inlet to outlet. A comparison of these results shows that even under the overly pessimistic assumption of complete absence of thermal mixing in the header, the flow in a divided tube is still substantially less sensitive to heat flux than the flow in the comparable undivided tube. However, the large reduction in sensitivity that can be obtained in a divided tube by perfect mixing in the intermediate header furnishes incentive for a careful study of the design of efficient mixing headers of compact size.

THEORETICAL DEVELOPMENT

Flow Sensitivity of a Divided Tube

No Temperature Mixing in Intermediate Header. In this case, the intermediate header is assumed to produce pressure equilization only.

The flow through a tube between constant pressure headers will vary with the average heat flux and the fluid inlet temperature.

$$dG = \left(\frac{\partial G}{\partial Q}\right)_{T_{in},\Delta p} dQ + \left(\frac{\partial G}{\partial T_{in}}\right)_{Q,\Delta p} dT_{in} \quad (1)$$

$$\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p} = \left(\frac{\partial G/G}{\partial Q/Q}\right)_{T_{in},\Delta p} + \left(\frac{\partial G/G}{\partial T_{in}}\right)_{Q,\Delta p}\left(\frac{dT_{in}}{dQ/Q}\right)_{\Delta p} \quad (2)$$

A tube heat balance yields

$$\Delta h = h_0 - h_{in} = \frac{4LQ}{GD} \quad (3)$$

Differentiation at constant inlet enthalpy gives

$$\left(\frac{dh_0}{dQ}\right)_{\Delta p,T_{in}} = \frac{4L}{GD} + \frac{4LQ}{G^2D}\left(\frac{dG}{dQ}\right)_{\Delta p,T_{in}} \quad (4)$$

Substituting (3) into (4) and multiplying by Q gives

$$\left(\frac{dh_0}{dQ/Q}\right)_{\Delta p,T_{in}} = \Delta h\left[1 - \left(\frac{dG/G}{dQ/Q}\right)_{\Delta p,T_{in}}\right] \qquad (5)$$

Under the assumption of no temperature equalization in the header, the inlet temperature of the second half will change with the heat-flux variation of the first half of the tube. If the two halves of the tube have the same fractional change in average heat flux, then no distinguishing notation is needed on this change.

$$\left(\frac{dT_{in}}{dQ/Q}\right)_{2/2,\Delta p} = \left(\frac{dT_0}{dQ/Q}\right)_{1/2,\Delta p,T_{in}} \qquad (6)$$

Rearranging and substituting (5) gives

$$\left(\frac{dT_0}{dQ/Q}\right)_{1/2,\Delta p,T_{in}} = \left(\frac{dh_0}{dQ/Q}\right)_{1/2,\Delta p,T_{in}}\left(\frac{dT}{dh}\right)_{1/2,0} = \frac{\Delta h_{1/2}}{(C_p)_{1/2,0}}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{1/2,\Delta p,T_{in}}\right] \qquad (7)$$

where

$$(C_p)_{1/2,0} = \left(\frac{dh}{dT}\right)_{1/2,0}$$

Substituting (6) and (7) into (2), written for the second half, gives

$$\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p} = \left(\frac{\partial G/G}{\partial Q/Q}\right)_{2/2,\Delta p,T_{in}} + \left(\frac{\partial G/G}{\partial T_{in}}\right)_{2/2,\Delta p,Q}\frac{\Delta h_{1/2}}{(C_p)_{1/2,0}}\left[1 - \left(\frac{dG/G}{dQ/Q}\right)_{1/2,\Delta p,T_{in}}\right] \qquad (8)$$

The cyclic properties of partial derivatives are such that

$$\left(\frac{\partial G/G}{\partial T_{in}}\right)_{\Delta p,Q} = -\left(\frac{\partial G/G}{\partial Q/Q}\right)_{\Delta p,T_{in}}\left(\frac{\partial Q/Q}{\partial T_{in}}\right)_{\Delta p,G} \qquad (9)$$

One can write

$$\left(\frac{\partial Q/Q}{\partial T_{in}}\right)_{\Delta p,G} = \left(\frac{\partial Q/Q}{\partial \Delta h}\right)_{\Delta p,G}\left(\frac{\partial \Delta h}{\partial T_{in}}\right)_{\Delta p,G} \qquad (10)$$

Differentiating (3) at constant G gives

$$\left(\frac{\partial \Delta h}{\partial Q/Q}\right)_{\Delta p,G} = \frac{4LQ}{GD} = \Delta h \qquad (11)$$

Substituting (9), (10), and (11) into (8) and recalling that the divided tube is assumed to undergo the same heat-flux variation in both halves, so that $\Delta h_{1/2} = \Delta h_{2/2}$, one gets

$$\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p} = \left(\frac{\partial G/G}{\partial Q/Q}\right)_{2/2,\Delta p,T_{in}} \cdot\left\{1 - \frac{1}{(C_p)_{1/2,0}}\left(\frac{\partial \Delta h}{\partial T_{in}}\right)_{2/2,\Delta p,G} \cdot\left[1 - \left(\frac{\partial G/G}{\partial Q/Q}\right)_{1/2,p,T_{in}}\right]\right\} \qquad (12)$$

Equation (12) gives the means of calculating the fractional change in flow per fractional change in heat flux (flow sensitivity). For the case of pressure equalization, but no temperature equalization, the flow sensitivity of the second half is influenced by that of the first half.

Complete Temperature Mixing in Intermediate Header. If the stream entering the header from the first half of the perverse tube can be completely mixed with a relatively large flow from other tubes, the unstabilizing effect is dissipated before the stream passes on to the second half. In this case both halves operate at fixed inlet temperatures. Equation (1) reduces to

$$\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p,T_{in}} = \left(\frac{\partial G/G}{\partial Q/Q}\right)_{\Delta p,T_{in}} \qquad (13)$$

The flow sensitivities for each half of the tube can be evaluated from the following expressions:

$$\left(\frac{dG/G}{dQ/Q}\right)_{1/2,\Delta p,T_{in}} = -\frac{\left(\frac{\partial \Delta p/\Delta p}{\partial Q/Q}\right)_{1/2,G,T_{in}}}{\left(\frac{\partial \Delta p/\Delta p}{\partial G/G}\right)_{1/2,Q,T_{in}}} \qquad (14)$$

$$\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p,T_{in}} = -\frac{\left(\frac{\partial \Delta p/\Delta p}{\partial Q/Q}\right)_{2/2,G,T_{in}}}{\left(\frac{\partial \Delta p/\Delta p}{\partial G/G}\right)_{2/2,Q,T_{in}}} \qquad (15)$$

A comparison of (12) and (13) shows that for the second half of the tube the flow sensitivity with incomplete mixing is a multiple of the flow sensitivity with complete mixing.

If the design permits a regular variation of heat flux with tube position in the bundle, rather than just an occasional abnormality, perfect mixing in the intermediate header would still result in a fixed inlet enthalpy for the second half of the tube bundle.

A single flow sensitivity cannot describe flow variations in a divided tube; consequently the flow sensitivities of divided and undivided tubes operating under the same conditions, cannot be compared directly. The outlet enthalpy and the wall-temperature sensitivities, however, are of prime importance and are strongly dependent on flow sensitivity. These sensitivities are directly comparable for divided and undivided tubes and furnish a basis for critical comparison of the divided and undivided tubes for a particular design.

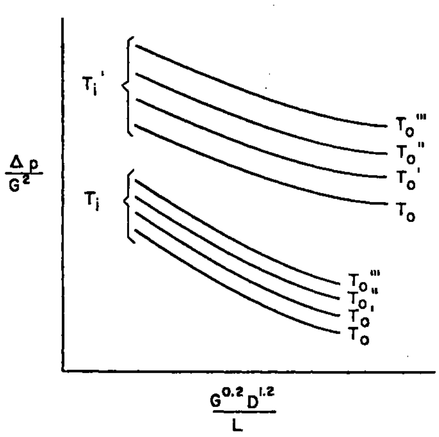

Fig. 1. Generalized pressure-drop chart.

Outlet Enthalpy Sensitivity of a Divided Tube

No Temperature Mixing in Intermediate Header. From Equations (3), (4), and (5) rewritten without the restriction of constant inlet enthalpy one gets

$$\left(\frac{dh_0}{dQ/Q}\right)_{\Delta p} - \left(\frac{dh_{in}}{dQ/Q}\right)_{\Delta p} = \Delta h\left[1 - \left(\frac{dG/G}{dQ/Q}\right)_{\Delta p}\right] \qquad (16)$$

For the second half of a tube without intermediate thermal mixing

$$\left(\frac{dh_{in}}{dQ/Q}\right)_{2/2,\Delta p} = \left(\frac{dh_0}{dQ/Q}\right)_{1/2,\Delta p,T_{in}} \qquad (17)$$

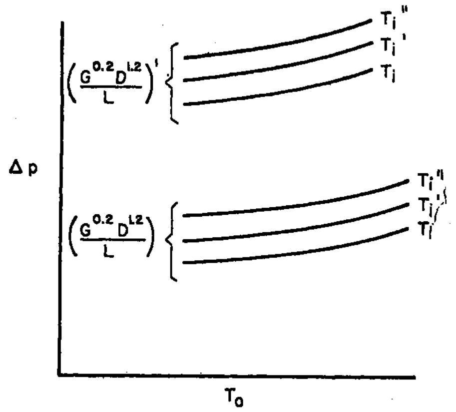

Fig. 2. Cross plot used in computation of flow sensitivity by Equation (12).

Writing Equation (16) for the second half and substituting (17) and (5) into it gives

$$\left(\frac{dh_0}{dQ/Q}\right)_{2/2,\Delta p} = \Delta h_{2/2}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p}\right] + \Delta h_{1/2}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{1/2,\Delta p,T_{in}}\right] \qquad (18)$$

By prior assumption, the enthalpy rise is the same in both halves of the tube, and so

$$\left(\frac{dh_0/\Delta h}{dQ/Q}\right)_{2/2,\Delta p} = 2\left[1-\frac{\left(\frac{dG/G}{dQ/Q}\right)_{1/2,\Delta p,T_{in}}+\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p}}{2}\right] \qquad (19)$$

For an undivided tube the comparable equation is

$$\left(\frac{dh_0/\Delta h}{dQ/Q}\right)_{\Delta p} = \left[1-\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p}\right] \qquad (20)$$

For this case the outlet enthalpy sensitivity of the divided tube can therefore be compared with that of an undivided tube by use of the arithmetic average of the flow sensitivities in the two halves.

Complete Temperature Mixing in Intermediate Header. If the second half of a tube receives completely mixed flow, its inlet enthalpy will not vary and Equation (16) reduces to

$$\left(\frac{dh_0/\Delta h}{dQ/Q}\right)_{2/2,\Delta p,T_{in}} = \left[1-\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p,T_{in}}\right] \qquad (21)$$

Wall-temperature Sensitivity of a Divided Tube

No Temperature Mixing in Intermediate Header. The wall temperature at a point is a function of the average heat flux, the mass velocity, and the bulk temperature of the fluid. The variation in wall temperature may be expressed as

$$\frac{dT_w}{dQ} = \left(\frac{\partial T_w}{\partial G}\right)_{Q,T_b}\left(\frac{dG}{dQ}\right) + \left(\frac{\partial T_w}{\partial Q}\right)_{G,T_b} + \left(\frac{\partial T_w}{\partial T_b}\right)_{Q,G}\left(\frac{dT_b}{dQ}\right) \qquad (22)$$

In turn, the variation in bulk temperature can be related to the heat flux, flow rate, and inlet temperature as

$$\left(\frac{dT_b}{dQ}\right)_{2/2} = \left(\frac{\partial T_b}{\partial Q}\right)_{G,2/2,T_d} + \left(\frac{\partial T_b}{\partial G}\right)_{Q,2/2,T_d}\left(\frac{dG}{dQ}\right)_{2/2} + \left(\frac{\partial T_b}{\partial T_d}\right)_{G,Q,2/2}\left(\frac{dT_d}{dQ}\right)_{2/2} \qquad (23)$$

Rewriting Equation (3) for the section from point of division to the location of maximum wall temperature yields

$$h_b - h_d = \frac{4QL_{db}}{GD} = \Delta h_{db} \qquad (24)$$

Differentiating at constant G and also at constant Q gives

$$\left(\frac{\partial h_b}{\partial Q/Q}\right)_{G,2/2,T_d} = \Delta h_{db} = C_{p,b}\left(\frac{\partial T_b}{\partial Q/Q}\right)_{G,2/2,T_d} \qquad (25)$$

$$\left(\frac{\partial h_b}{\partial G/G}\right)_{Q,2/2,T_d} = -\Delta h_{db} = C_{p,b}\left(\frac{\partial T_b}{\partial G/G}\right)_{Q,2/2,T_d} \qquad (26)$$

$$\left(\frac{\partial T_b}{\partial T_d}\right)_{G,Q,2/2} = \frac{C_{p,d}}{C_{p,b}}\left(\frac{\partial h_b}{\partial h_d}\right)_{G,Q,2/2} = \frac{C_{p,d}}{C_{p,b}} \qquad (27)$$

Because of the earlier assumption that both halves of the tube experience the same heat fluxes and through Equation (5),

$$\left(\frac{\partial T_d}{\partial Q/Q}\right)_{2/2} = \left(\frac{\partial T_d}{\partial Q/Q}\right)_{1/2} = \frac{1}{C_{p,d}}\left(\frac{\partial h_d}{\partial Q/Q}\right) = \frac{\Delta h_{id}}{C_{p,d}}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p,1/2}\right] \qquad (28)$$

Substituting (25), (26), (27), and (28) into (23) gives

$$\left(\frac{dT_b}{dQ/Q}\right)_{\Delta p,2/2} = \frac{\Delta h_{db}}{C_{p,b}}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p,2/2}\right] + \frac{\Delta h_{id}}{C_{p,b}}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p,1/2}\right] \qquad (29)$$

Finally, substitution of (29) into (22) gives

$$\left(\frac{dT_w}{dQ/Q}\right)_{2/2,\Delta p} = \left(\frac{\partial T_w}{\partial Q/Q}\right)_{G,T_b} + \left(\frac{\partial T_w}{\partial G/G}\right)_{Q,T_b}\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p} + \left(\frac{\partial T_w}{\partial T_b}\right)_{Q,G}\frac{1}{C_{p,b}} \cdot\left\{\Delta h_{db}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p,2/2}\right] + \Delta h_{id}\left[1-\left(\frac{dG/G}{dQ/Q}\right)_{p,1/2}\right]\right\} \qquad (30)$$

When the maximum wall temperature is at the outlet, Equation (19) can be used directly with Equation (22) to produce the same result as in Equation (30).

Complete Temperature Mixing in Intermediate Header. In the case of constant inlet enthalpy for the second half of the tube, Equation (23) reduces to

$$\left(\frac{dT_b}{dQ/Q}\right)_{2/2} = \left(\frac{\partial T_b}{\partial Q}\right)_{G,2/2,T_d} + \left(\frac{\partial T_b}{\partial G}\right)_{Q,2/2,T_d}\left(\frac{dG}{dQ}\right)_{2/2} \qquad (31)$$

and Equation (30) becomes

$$\left(\frac{dT_w}{dQ/Q}\right)_{2/2,\Delta p,T_{in}} = \left(\frac{\partial T_w}{\partial Q/Q}\right)_{G,T_b} + \left(\frac{\partial T_w}{\partial G/G}\right)_{Q,T_b}\left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p,T_{in}} + \left(\frac{\partial T_w}{\partial T_b}\right)_{Q,G}\left(\frac{\Delta h_{db}}{C_{p,b}}\right)\left[1 - \left(\frac{dG/G}{dQ/Q}\right)_{2/2,\Delta p,T_{in}}\right] \quad (32)$$

Examination of Equation (32) shows how efficient intermediate mixing reduces wall-temperature sensitivity, as evidenced by the reduced value of flow sensitivity appearing in the second and third terms as well as by the elimination of the effect of the flow sensitivity of the first half of the tube.

METHOD OF CALCULATION

For fluids that have physical properties varying greatly with temperature, the flow losses must be calculated point to point along the heated tube. For a short length containing fluid of fixed physical properties, the Fanning equation can be written

$$\frac{\Delta p}{G^2} = \frac{0.092\mu^{0.2}}{\rho g_c}\cdot\frac{L}{G^{0.2}D^{1.2}} \quad (33)$$

Pressure drops due to acceleration, and entrance and exit effects will depend on the physical properties at ends of the tube divisions.

By point-to-point calculations, integrated for the tube length, it is possible to construct charts of $\Delta p/G^2$ plotted vs. $G^{0.2}D^{1.2}/L$ with families of curves of various outlet temperatures for each inlet temperature. Figure 1 is an example of these plots, which can be made for the whole tube as well as for each half.

Next, one can plot the pressure drop against the value of the heat flux (corresponding then to a particular outlet temperature) at fixed values of $G^{0.2}D^{1.2}/L$ and inlet temperature. The slopes of these lines give the value $(\partial\Delta p/\partial Q)_{G,T_{in}}$. In a similar fashion, $(\partial\Delta p/\partial G)_{Q,T_{in}}$ is obtained.

The partial differential

$$(\partial\Delta h/\partial T_{in})_{\Delta p,G,2/2}$$

needed for Equation (12) is evaluated in the following manner. For fixed values of $(G^{0.2}D^{1.2}/L)$, the values of $\Delta p_{2/2}$ are plotted against $T_{0,2/2}$ in a family of $T_{in,2/2}$ curves (Figure 2). A horizontal line on this plot represents the condition of constant $\Delta p_{2/2}$ and $G_{2/2}$ and intersects the various $T_{in,2/2}$ lines at different values of $T_{0,2/2}$. The values of $T_{0,2/2}$ and $T_{in,2/2}$ at these intersections are converted to values of $\Delta h_{2/2}$ and plotted vs. $T_{in,2/2}$. The slopes of these lines give $(\partial\Delta h/\partial T_{in})_{\Delta p,G,2/2}$.

The usual correlations show film coefficients of heat transfer to be functions of physical properties and of $(G^{0.8}/D^{0.2})$. The ratio of heat flux to film coefficient is equal to the difference between wall temperature and bulk temperature. It is possible to plot $(QD^{0.2}/G^{0.8})$ vs. bulk temperature as a family of curves with wall temperature as parameter. From such a plot are obtained the wall-temperature derivatives needed in some of the foregoing equations.

CALCULATED RESULTS FOR A SUPERCRITICAL WATER BOILER

The material presented above is illustrated by a case that is typical of the results obtained in a series of calculations on water at supercritical pressures, heated in parallel tubes of nominally identical heat input, and flowing in tubes which are connected to common headers. It might be noted that the temperatures and pressures of the example are similar to those of a large central-station supercritical-pressure boiler planned for a utility company (*2*). This boiler has one-pass flow instead of recirculation as in most large boilers and would be subject to problems of flow sensitivity. Dimensions, flow rates, and heat flux are assumed arbitrarily at reasonable values and do not reflect the values for any industrial power plant or any nuclear reactor.

The Typical Case

The physical conditions, flow rate, and heat input are given in Table 1. The headers are assumed large enough to allow calculations on the basis of loss of one half of a velocity head at tube entrance and one velocity head at the tube outlet. It is assumed that the pressure drop between the headers is small compared to 5,000 lb./sq. in. The relevant properties of water have been indicated in an earlier paper (*1*). It should be noted that the water undergoes eightfold expansion in being heated from 515° to 1,150°F.

For the example of Table 1, it is found that the maximum temperature of the tube heating surface is 1,370°F. and the pressure drop is 44.9 lb./sq. in. Heat transfer coefficients and friction factors were evaluated according to calculations of Goldmann (*3*). For present purposes the Goldmann predictions give results substantially the same as do the method of Deissler (*4*) or the usual isothermal-turbulent-flow formulas used with properties based on mean film temperatures.

TABLE 1

p	5,000 lb./sq. in.
T_i	515 °F.
T_0	1,150 °F.
Q	3×10^5 B.t.u./(hr.)(sq. ft.)
G	1.8×10^6 lb./(hr.)(sq. ft.)
D	0.0416 ft.
L	62 ft.

The results of calculation are shown in Table 2, wherein the effectiveness of an intermediate header in reducing flow-sensitivity outlet enthalpy, temperature sensitivity, and tube-wall temperature sensitivity can readily be seen.

TABLE 2

	Undivided tube	Second half of divided tube: No temperature mixing	Second half of divided tube: Complete temperature mixing
$\left(\frac{dG/G}{dQ/Q}\right)_{\Delta p}$	−3.75	−1.70	−0.59
$\left(\frac{dh_0/\Delta h}{dQ/Q}\right)_{\Delta p}$	4.75	2.70	1.59
$\left(\frac{dT_0}{dQ/Q}\right)_{\Delta p}$	6,700°F.	2,700°F.	1,100°F.
$\left(\frac{dT_w}{dQ/Q}\right)_{\Delta p}$	8,000°F.	5,800°F.	3,000°F.

ACKNOWLEDGMENT

The work reported here draws on ideas and results of related studies by several other staff members of Nuclear Development Corporation of America, notably Kurt Goldmann, R. C. Ross, and N. R. Adolph. Elaine Scheer and Gloria Sullivan performed the computations from which these results were taken.

NOTATION

- D = equivalent diameter of heated passage or tube, ft.
- G = mass velocity, lb. mass/(hr.) (sq. ft.)
- Δh = difference in enthalpy between two locations, B.t.u./lb. mass
- L = length of heated passage, ft.
- p = pressure, lb. force/sq. ft.
- Δp = difference in pressure between two locations, lb. force/sq. ft.
- Q = average heat flux along heated passage, B.t.u./(hr.)(sq. ft.)
- T = temperature, °F.

Subscripts

- b = bulk
- d = point of division of tube length
- in = inlet
- o = outlet
- $1/2$ = first half
- $2/2$ = second half

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Manuscript submitted May, 1957; revision received July 12, 1957; paper accepted July 30, 1957.