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Nondimensional Expression for the Relation between Pressure Drop and Flow Rate in Steam Generating Tubes*

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A method to express the relationship between pressure drop and flow rate in steam generating tubes is presented. The characteristics of the pressure drops is represented in nondimensional generalized forms. In a case of uniform heat flux along a tube, it is given by the following equation :

$$\Delta P^* = \frac{1}{2} k \xi^3 + (1-k) \xi^2 + \frac{1}{2} k \xi$$

Here, ΔP^* = nondimensional pressure drop defined in this report, ξ = nondimensional flow rate, v'' = specific volume of saturated steam, v' = specific volume of saturated water, r_0 = subcooling, r = latent heat of evaporation, $k = \{(v''/v') - 1\} r_0 / r$.

The charts in cases of various heat flux distributions along tubes are shown. They can be used to study the influence of each factor such as pressure, subcooling and heat flux distribution and so on characteristics of pressure drop and to estimate the flow stability and the distribution of flow rates in parallel tubes.

1. Introduction

Steam generating tubes used in once-through boilers and boiling water reactors usually consist of an economizer, an evaporator and/or a superheater. The characteristic of pressure drop in each section is different; in the economizer and superheater, pressure drop is not affected by the heat flux; in the evaporator, on the other hand, pressure drop is affected by heat flux density because of the phase change of fluid. Therefore, the characteristic of pressure drop in the steam generating tubes are very complicated. These characteristics are influenced by many factors such as flow rate, heat flux, its distribution along the tube length, pressure, subcooling, tube length, tube diameter which can be selected in a rather wide range in the design of steam generating tubes, accordingly many diagrams have to be plotted to know the influence of each factor. For instance, (1) the characteristic curves

of pressure drop for various heat flux distributions along a tube have to be drawn. (2) The characteristic curves for various heat fluxes have to be drawn to estimate the uniformity of flow rate in parallel tubes. And (3) also many charts are needed to inspect the stability of flow in steam generating tubes.

Ledinegg⁽¹⁾, Profos⁽²⁾, Michel⁽³⁾, Ueda⁽⁴⁾ and Maulbetsch⁽⁵⁾ et al. have studied on static characteristics of pressure drop, flow distribution in parallel steam generating tubes and flow stability. They have shown the influence of each factor on characteristics of pressure drop separately and have not expressed their relationship in generalized forms. Therefore, it is desirable to show the characteristics of pressure drop in generalized forms with which each influence can be estimated.

In this paper, the nondimensional expression for the relationship between pressure drop and flow rate is proposed to satisfy these requirements. That is, flow rate and pressure drop are expressed by nondimensional flow rate ξ and nondimensional pressure drop ΔP^* . ξ is the ratio of actual flow rate G to imaginary flow rate G_0 with which the fluid reaches the saturated liquid at the outlet of tube

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with the given inlet enthalpy, heat flux and pressure, and ΔP^* is the ratio of ΔP to ΔP_0 . Here ΔP is pressure drop corresponding to G and ΔP_0 is one corresponding to G_0 respectively. By using these two nondimensional quantities, the expression of characteristic of pressure drop in steam generating tubes with various geometries and heat flux distributions can be very much simplified. In particular, the expressions in steam generating tubes consisting of an economizer and an evaporator are given as functions of only heat flux and nondimensional quantity $\alpha r_0/r$ determined by inlet enthalpy and pressure.

Using these expressions, nondimensional charts of pressure drop are presented and their applications are shown.

Nomenclatures

- D : inside diameter (m)
 F : cross-sectional area of tube (m²)
 G : flow rate (kg/sec)
 G_0 : imaginary flow rate at which the fluid reaches saturated liquid at outlet of steam generating tubes with given inlet enthalpy, heat flux and pressure (kg/sec)
 i : enthalpy (kcal/kg)
 L : total length of steam generating tubes (m)
 L_1', L_2', L_3' : lengths of economizer evaporator and superheater respectively (m)
 L_1, L_2 : lengths of segments with heat fluxes q_1 and q_2 respectively (m)
 ΔL : difference between L_1 and L_1' (m)
 P : pressure drop (kg/m²)
 P_0 : imaginary pressure drop corresponding to G_0 (kg/m²)
 ΔP^* : nondimensional pressure drop $\Delta P/\Delta P_0$
 q : heat flux (kcal/msec)
 r : latent heat (kcal/kg)
 r_0 : subcooling (kcal/kg)
 v : specific volume (m³/kg)
 x : quality
 α : $(v''/v') - 1$
 θ_1, θ_2 : length of economizer and evaporator for sine curved distribution of heat flux (m)
 π : total length for sine curved distribution of heat flux (m)
 λ : friction factor
 ξ : nondimensional flow rate G/G_0

2. Induction of the nondimensional expression

Let's consider a simple tube system of which the diameter is constant along the tube and the heat flux is uniformly distributed along each tube as a typical example of deriving the nondimensional

expressions.

For simplicity, the following assumptions are used: (1) pressure drop at bends is taken into account in the modified coefficient of frictional pressure drop^{(4) (6)} and the coefficients of pressure drop in the economizer, evaporator and superheater are equal. (2) Relative velocity between steam and water is neglected. (3) Self-evaporation by pressure drop along the evaporator is also neglected. (4) Properties of steam and water are constant for pressure change along the tube system. And (5) specific volume of water in the economizer is equal to that of saturated water. The characteristics of pressure drop derived in accordance with these assumptions agree with that calculated from experimental results⁽⁷⁾.

In a tube system shown in Fig. 1, heat flux in segment 0 ~ 1 is q_1 , that in segment 1 ~ 2 is q_2 (usually $q_1 \neq q_2$) and q_m is mean heat flux. 1' is the incipient point of boiling and 2' is the end point of evaporator. L_T is the total length of the tube system, L_1 and L_2 are the lengths of segments with heat flux q_1 and q_2 respectively. L_1', L_2' and L_3' are the lengths of economizer, evaporator and superheater respectively. ΔL is the difference between the length of economizer and L_1 . Case A shows a tube system in which a superheater is not used and the length of the economizer is longer than L_1 , and case B shows a tube system in which the length of the economizer is shorter than L_1 . And cases C and D show tube systems which have a superheater and the condition of the economizer is the same as in cases A and B.

The imaginary flow rate G_0 , at which the fluid reaches saturated liquid at the outlet of tube system with the given heat flux, pressure and inlet enthalpy, is

$$G_0 = L_T q_m / r_0 \quad \dots\dots\dots (1)$$

and the corresponding pressure drop ΔP_0 is

$$\Delta P_0 = \lambda \frac{L_T}{D} \frac{1}{2g} \left(\frac{G_0}{F} \right)^2 v' \quad \dots\dots\dots (2)$$

The nondimensional flow rate ξ and the nondimensional pressure drop ΔP^* are defined as follows:

$$\xi = G/G_0 \quad \dots\dots\dots (3)$$

$$\Delta P^* = \Delta P/\Delta P_0 \quad \dots\dots\dots (4)$$

The induction of the nondimensional pressure drop is exemplified for case C in Fig. 1.

The condition of the tube length is

$$L_1 + L_2 = L_1' + L_2' + L_3' = L_T \quad \dots\dots\dots (5)$$

The equations of heat balance for segments 0 ~ 1', 1' ~ 2', 2' ~ 2 and total length 0 ~ 2 are

$$G(i_{1'} - i_0) = q_1 L_1 + q_2 \Delta L \equiv G r_0 \quad \dots\dots\dots (6)$$

$$G(i_{2'} - i_{1'}) = q_2 L_2' \equiv G r \quad \dots\dots\dots (7)$$

$$G(i_2 - i_{2'}) = q_2 \{L_T - (L_1' + L_2')\} \quad \dots\dots\dots (8)$$

$$G(i_2 - i_0) = q_1 L_1 + q_2 L_2 = q_m L_T \quad \dots\dots\dots (9)$$

Here, i_0 , i' , i'' and i_2 are enthalpy of feed water, saturated water, saturated steam and liquid at exit of the tube system and subscripts on enthalpy indicate the location.

L_1 , ΔL , L_1' , L_2' and L_3' are obtained in Eqs. (10)~(14) by simple calculation.

$$L_1 = \frac{1 - q_m/q_2}{1 - q_1/q_2} L_T \quad \dots\dots\dots(10)$$

$$\Delta L = \frac{Gr_0}{q_2} - \frac{q_1}{q_2} L_1 = \left\{ \frac{q_m}{q_2} \xi + \frac{1 - q_m/q_2}{1 - q_2/q_1} \right\} L_T \quad \dots\dots\dots(11)$$

$$L_1' = L_1 + \Delta L = \left\{ 1 - (1 - \xi) \frac{q_m}{q_2} \right\} L_T \quad \dots\dots\dots(12)$$

$$L_2' = \frac{Gr}{q_2} = \frac{r}{r_0} \frac{q_m}{q_2} \xi L_T \quad \dots\dots\dots(13)$$

$$L_3' = L_T - (L_1' + L_2') = \left\{ 1 - \left(1 + \frac{r}{r_0} \right) \xi \right\} \frac{q_m}{q_2} L_T \quad \dots\dots\dots(14)$$

The pressure drop in the tube system for flow rate G is the sum of pressure drops in the economizer, evaporator and superheater. The pressure drop in each part is given by the following equation:

$$\Delta P = \lambda \frac{1}{D} \frac{1}{2g} \left(\frac{G}{F} \right)^2 \int_0^L v dL \quad \dots\dots\dots(15)$$

Here, the values of v are different in each part of the tube system. The specific volume in the economizer is assumed to be equal to that of saturated water and to be constant along the tube, so

$$\Delta P_{eco} = \lambda \frac{L_T}{D} \frac{1}{2g} \left(\frac{G_0}{F} \right)^2 v' \left\{ \frac{q_m}{q_2} \xi^3 + \left(1 - \frac{q_m}{q_2} \right) \xi^2 \right\} \quad \dots\dots\dots(16)$$

The specific volume of fluid in the evaporator with the uniform heat flux is

$$v_{eva} = v' + (v'' - v') \frac{L}{L_0} = v' + x(v'' - v') \quad \dots\dots\dots(17)$$

where L_0 is the length on which the steam quality

x reaches 1.0 with the heat flux q_2 , and equals to L_2' in case C. Therefore, the pressure drop in the evaporator is

$$\Delta P_{eva} = \lambda \frac{L_T}{D} \frac{1}{2g} \left(\frac{G_0}{F} \right)^2 v' \frac{r}{r_0} \frac{q_m}{q_2} \left(1 + \frac{\alpha}{2} \right) \xi^3 \quad \dots\dots\dots(18)$$

where $\alpha = (v''/v') - 1$.

In the superheater, enthalpy of superheated steam i_{SH} is proportional to the distance from its inlet for uniform heat flux, and the relationship between i_{SH} and specific volume of superheated steam v_{SH} is approximately linear at constant pressure. Then v_{SH} is shown in Eq. (19),

$$v_{SH} = v'' + \frac{q_2}{G} \frac{dv_{SH}}{di_{SH}} L \quad \dots\dots\dots(19)$$

therefore, the pressure drop in the superheater is

$$\begin{aligned} \Delta P_{SH} = & \lambda \frac{L_T}{D} \frac{1}{2g} \left(\frac{G_0}{F} \right)^2 v' \frac{q_m}{q_2} \left[\left(1 + \frac{r}{r_0} \right) \right. \\ & \times \left\{ \frac{1}{2} \left(1 + \frac{r}{r_0} \right) \frac{r_0}{v'} \frac{dv_{SH}}{di_{SH}} - \frac{v''}{v'} \right\} \xi \\ & + \left\{ \frac{v''}{v'} - \left(1 + \frac{r}{r_0} \right) \frac{r_0}{v'} \frac{dv_{SH}}{di_{SH}} \right\} \\ & \left. + \frac{r_0}{2v'} \frac{dv_{SH}}{di_{SH}} \frac{1}{\xi} \right] \xi^2 \quad \dots\dots\dots(20) \end{aligned}$$

The total pressure drop ΔP for flow rate G is given by the sum of Eqs. (16), (18) and (20) and ΔP is divided by ΔP_0 to get the nondimensional pressure drop equation

$$\begin{aligned} \Delta P^* = \frac{\Delta P}{\Delta P_0} = & \frac{q_m}{q_2} \left\{ \frac{1}{2} \alpha \frac{r_0}{r} - \alpha \left(1 + \frac{r}{r_0} \right) \right. \\ & + \frac{1}{2} \left(1 + \frac{r}{r_0} \right) \frac{r_0}{v'} \frac{dv_{SH}}{di_{SH}} \left. \right\} \xi^3 \\ & + \left[1 + \frac{q_m}{q_2} \left\{ \alpha - \left(1 + \frac{r}{r_0} \right) \frac{r_0}{v'} \frac{dv_{SH}}{di_{SH}} \right\} \right] \xi^2 \\ & + \frac{1}{2} \frac{r_0}{v'} \frac{q_m}{q_2} \frac{dv_{SH}}{di_{SH}} \xi \quad \dots\dots\dots(21) \end{aligned}$$

This equation is a generalized relation between the pressure drop and the flow rate in steam generating tubes. ΔP^* is the third order equation of ξ and the coefficients are determined only by pressure, subcooling and heat flux. The equation is applicable to the region shown by Eq. (22) for case C. Equation (22) is determined by the conditions; (1) an economizer exists in the region with heat flux q_2 ; $\Delta L \geq 0$ and (2) a superheater exists; $\Delta L_3' \geq 0$,

$$\frac{q_1}{q_2} \frac{1 - q_m/q_2}{1 - q_1/q_2} \leq \xi, \quad \xi \leq \frac{1}{1 + r/r_0} \quad \dots\dots\dots(22)$$

The nondimensional pressure drop equation and the applicable range for cases A, B and D are shown in Table 1 which are obtained by the same method mentioned above.

As shown in Table 1, ΔP^* in the steam generating tubes with uniform heat flux in each part is expressed by the third order equation of ξ ,

$$\Delta P^* = C_1 \xi^3 + C_2 \xi^2 + C_3 \xi \quad \dots\dots\dots(23)$$

In the special case of uniform heat flux along

Fig. 1 Steam generating tube

the economizer and evaporator; $q_1 = q_2 = q_m$, ΔP^* can be shown by Eq. (24),

$$\Delta P^* = \frac{1}{2} \alpha \frac{r_0}{r} \xi^3 + \left(1 - \alpha \frac{r_0}{r}\right) \xi^2 + \frac{1}{2} \alpha \frac{r_0}{r} \xi \quad \dots (24)$$

It should be noted that the characteristics of pressure drop are expressed only by the nondimensional quantity $\alpha r_0/r$ which is determined by pressure and inlet enthalpy.

Statistical characteristics of pressure drop in steam generating tubes with nonuniform heat flux along the tube system can be obtained by the same method as mentioned above. That is, nondimensional

Fig. 2 Distribution of heat flux

equations of pressure drop are given in Eqs. (25) ~ (28) for tube systems consisting of an economizer and an evaporator with various heat fluxes shown in Fig. 2.

For linear distribution of heat flux:

$$q = q_i + (q_o - q_i) L / L_T$$

$$\Delta P^* = \xi^3 + \alpha \frac{r_0}{r} \left\{ m \xi (1 - k \xi)^2 + \frac{h}{3} \xi (1 - k \xi)^3 \right\} \quad \dots (25)$$

where $h = (q_o - q_i) / (q_o + q_i)$, $m = q_c / (q_o + q_i)$ and $k = (q_o + q_i) / (q_i + q_c)$

For quadratic curved distribution of heat flux (No. 1): $q = n L^2$

$$\Delta P^* = \frac{3}{4} \alpha \frac{r_0}{r} \xi^{7/3} + \left(1 - \alpha \frac{r_0}{r}\right) \xi^2 + \frac{1}{4} \alpha \frac{r_0}{r} \xi \quad \dots (26)$$

For quadratic curved distribution of heat flux (No. 2): $q = n (L - L_T)^2$

$$\Delta P^* = \xi^2 + \frac{3}{4} \alpha \frac{r_0}{r} \xi (1 - \xi)^{4/3} \quad \dots (27)$$

where n is a constant.

For sine curved distribution of heat flux:

$$q = q_a \sin \theta$$

$$\Delta P^* = \frac{\xi^2}{\pi} \left[\left(1 + \frac{1}{2} \alpha \frac{r_0}{r} \frac{1 - 2\xi}{\xi}\right) \left\{ \pi - \cos^{-1}(1 - 2\xi) \right\} + \cos^{-1}(1 - 2\xi) + \alpha \frac{r_0}{r} \frac{\sqrt{\xi - \xi^2}}{\xi} \right] \quad \dots (28)$$

in this case, the imaginary pressure drop for G_0 is

$$\Delta P_0 = \lambda \frac{\pi}{D} \frac{1}{2g} \left(\frac{G_0}{F} \right)^2 v' \quad \dots (29)$$

These equations (24) ~ (28) are applicable for the range of $x \leq 1$ i.e. $\xi \geq (1 + r/r_0)^{-1}$.

Since the influences of many factors on pressure drop such as tube length, tube diameter, friction factor and heat flux are included in nondimensional quantities ΔP^* and ξ , and also the influences of pressure, subcooling and heat flux distribution are

Table 1 Coefficients of nondimensional equations

	$\Delta P^* = c_1 \xi^3 + c_2 \xi^2 + c_3 \xi \quad (\Delta P^* = \Delta P / \Delta P_0, \xi = G / G_0)$		
	c_1	c_2	c_3
Case A	$\frac{1}{2} \alpha \frac{r_0}{r} \frac{q_m}{q_2}$	$1 - \alpha \frac{r_0}{r} \frac{q_m}{q_2}$	$\frac{1}{2} \alpha \frac{r_0}{r} \frac{q_m}{q_2}$
Case B	$\frac{1}{2} \alpha \frac{r_0}{r} \frac{q_m}{q_2}$	$1 - \alpha \frac{r_0}{r}$	$\alpha \frac{r_0}{r} \left\{ \frac{1}{2} \left(\frac{q_m}{q_2} + \frac{q_1}{q_m} \right) - \frac{q_1}{q_2} \right\} / \left(1 - \frac{q_1}{q_2} \right)$
Case C	$\frac{q_m}{q_2} \left\{ \frac{1}{2} \alpha \frac{r_0}{r} - \alpha \left(1 + \frac{r}{r_0} \right) + \frac{1}{2} \left(1 + \frac{r}{r_0} \right) \frac{r_0}{v'} \frac{dv_{SH}}{d\xi_{SH}} \right\}$	$1 + \frac{q_m}{q_2} \left\{ \alpha - \left(1 + \frac{r}{r_0} \right) \frac{r_0}{v'} \frac{dv_{SH}}{d\xi_{SH}} \right\}$	$\frac{1}{2} \frac{r_0}{v'} \frac{q_m}{q_2} \frac{dv_{SH}}{d\xi_{SH}}$
Case D	$\frac{q_m}{q_2} \left[\frac{1}{2} \alpha \frac{r_0}{r} \left\{ \frac{q_2}{q_1} - \left(1 + \frac{r}{r_0} \right)^2 \right\} + \frac{1}{2} \frac{r_0}{v'} \left(1 + \frac{r}{r_0} \right)^2 \frac{dv_{SH}}{d\xi_{SH}} \right]$	$1 - \alpha \frac{r_0}{r} + \frac{q_m}{q_2} \left\{ \alpha \left(1 + \frac{r}{r_0} \right) - \frac{r_0}{v'} \left(1 + \frac{r}{r_0} \right) \frac{dv_{SH}}{d\xi_{SH}} \right\}$	$\frac{1}{2} \alpha \frac{r_0}{r} \frac{q_1}{q_m} \left(1 - \frac{q_m}{q_2} \right)^2 / \left(1 - \frac{q_1}{q_2} \right) + \frac{1}{2} \alpha \frac{r_0}{v'} \frac{q_m}{q_2} \frac{dv_{SH}}{d\xi_{SH}}$

Region of application		
Case A	$\xi \geq \frac{1}{1 + r/r_0}, \quad \xi \geq \frac{q_1}{q_m} \frac{1 - q_m/q_2}{1 - q_1/q_2}$	Case B $\xi \geq \frac{1}{1 + r/r_0}, \quad \xi \geq \frac{q_1}{q_m} \frac{1 - q_m/q_2}{1 - q_1/q_2}$
Case C	$\xi \leq \frac{1}{1 + r/r_0}, \quad \xi \geq \frac{q_1}{q_m} \frac{1 - q_m/q_2}{1 - q_1/q_2}$	Case D $\xi \leq \frac{1}{1 + r/r_0}, \quad \xi \geq \frac{q_1}{q_m} \frac{1 - q_m/q_2}{1 - q_1/q_2}$

included in the parameter $\alpha r_0/r$ and constants, the expressions can be used to investigate the influence of each factor on characteristics of pressure drop with generalized manner.

3. Nondimensional diagrams of statical characteristics of pressure drop

ΔP^* are the higher order equations of ξ that have maximum values in some conditions. From a point of view of flow stability, it would be desirable that ΔP^* increase monotonously as ξ increases (this is described in this paper as "the flow being stable"). On the other hand when ΔP^* decreases as ξ increases in some regions i.e. the slope of the curve has a negative value in some regions, the flow has the possibility to become unstable (this is described in this paper as "the flow being unstable").

In order to investigate the flow stability and the variation of the pressure drop for the change of flow rate, in this chapter, the diagrams of statical characteristics of pressure drop are presented which have been calculated from the above equations. And the influence of each factor is explained.

Fig. 3 Nondimensional diagram of statical characteristics of pressure drop for steam generating tube with stepwise uniform distribution of heat flux (without superheater), influence of L_1/L_2

3.1 Nondimensional diagrams of statical characteristics of pressure drop for steam generating tubes with an economizer and an evaporator

3.1.1 Stepwise uniform distribution of heat flux

Figures 3 and 4 show curves of pressure drop with stepwise uniform heat fluxes q_1 and q_2 [Eq. (23) and Table 1].

Figure 3 shows ΔP^* curve for $q_1/q_2=1/2$. In the lower part of this figure, the relation between r_0/r and ξ is shown with a parameter x_2 . This figure is used to get the steam quality at outlet of steam generating tubes on the characteristic curves of ΔP^* .

Figure 4 shows ΔP^* curves for $L_1/L_2=2/3$.

In Figs. 3 and 4, it is shown that the larger the value of $\alpha r_0/r$ is, the larger the value of ΔP^* is and ΔP^* increases monotonously with ξ at smaller value of $\alpha r_0/r$, but at larger value of $\alpha r_0/r$, ΔP^* has maximum and minimum values and the value of negative slope increases with $\alpha r_0/r$. This means that the larger the value of $\alpha r_0/r$ is, the

Fig. 4 Nondimensional diagram of statical characteristics of pressure drop for steam generating tube with stepwise uniform distribution of heat flux (without superheater), influence of q_1/q_2

wider the region of unstable flow is.

From Fig. 3 it can be seen that the case, in which heat flux in downstream part is larger than that in upstream part and the length of the former is longer than that of the latter, is favourable for the stability of flow. This conclusion agrees with that of Michel⁽³⁾. And also from Fig. 4 it can be seen that the smaller the heat flux in upstream part, the more stable the flow is for a constant L_1/L_2 . For instance, in the case of $\alpha r_0/r=10$, the negative slope region exists in the case of $q_1/q_2=4/4$ but with the value of q_1/q_2 increased from 4/4 to 1/4 this region vanishes, so that the flow may be stable.

3.1.2 Linear distribution of heat flux

Figure 5 shows the characteristic curves of pressure drop for linear distribution of heat flux according to Eq. (25). ΔP^* curve are shown for various ratios of heat flux of outlet and that of inlet: q_0/q_i . " $q_0/q_i=0$ " means the triangular distribution of heat flux with inlet heat flux q_i and outlet heat flux zero, " $q_0/q_i=\infty$ " means the opposite distribution and " $q_0/q_i=1$ " means uniform distribution of heat flux along the whole tube length. It

Fig. 5 Nondimensional diagram of static characteristics of pressure drop for steam generating tube with linear distribution of heat flux (without superheater)

is shown in Fig. 5 that the value of ΔP^* decreases with a decreasing value of $\alpha r_0/r$ and an increasing value of q_0/q_i and the flow becomes more stable with a decreasing value of $\alpha r_0/r$. Therefore, the flow in steam generating tubes with higher heat flux in downstream part is more stable.

3.1.3 Quadratic curved distribution of heat flux

Figure 6 shows the characteristic curves of pressure drop for quadratic curved distribution of heat flux according to Eqs. (26) and (27). The solid lines are curves showing that heat flux increases toward downstream with zero at inlet and the broken lines are curves showing that heat flux is maximum at inlet and decreases toward downstream.

Fig. 6 Nondimensional diagram of static characteristics of pressure drop for steam generating tube with quadratic curved distribution of heat flux (without superheater)

It is seen that the value of ΔP^* increases with an increasing value of $\alpha r_0/r$ and the flow condition becomes more unstable. Although the gradient of the ΔP^* curve, showing that heat flux is larger in downstream part, is not a negative value even though for the larger value of $\alpha r_0/r$, ($\alpha r_0/r=20$), the gradient of the ΔP^* curves may be negative in the case that the heat flux is larger in upstream part even for the smaller value of $\alpha r_0/r$, ($\alpha r_0/r=5$).

3.1.4 Sine curved distribution of heat flux

Figure 7 shows the characteristic curves of pressure drop for sine curved distribution of heat flux according to Eq. (28). The influence of $\alpha r_0/r$ on the characteristics of pressure drop is the same as in the cases mentioned above.

Now, let's consider the influences of pattern of heat flux distribution on the characteristics of pressure drop. Figure 8 shows the ΔP^* curves for various heat fluxes with the same value of $\alpha r_0/r=10$. It can be seen from this figure that from a point of view of flow stability, quadratic curved distribution with maximum at inlet is the most

Fig. 7 Nondimensional diagram of statical characteristics of pressure drop for steam generating tube with sine curved distribution of heat flux (without superheater)

undesirable case. And the stability is greater in the order: triangular distribution of heat flux with maximum at inlet, sine curve distribution, uniform distribution, triangular distribution with maximum at outlet and quadratic curved distribution with maximum at outlet.

When the enthalpy at inlet is kept constant, the value of $\alpha r_0/r$ generally decreases with an increasing pressure as shown in Fig. 9. Therefore the flow becomes stable with an increasing pressure.

3.2 Nondimensional diagrams of statical characteristics of pressure drop for steam generating tubes with an economizer, an evaporator and a superheater

Figures 10 and 11 show the characteristic curves of pressure drop in the steam generating tube with stepwise distributions of heat flux q_1 and q_2 as

Fig. 8 Nondimensional diagram of statical characteristics of pressure drop for steam generating tube with various distributions of heat flux (without superheater)

Fig. 9 Relation between $\alpha r_0/r$ and pressure

shown in case A in Fig. 1 calculated from Eq.(23). In this case, not only $\alpha r_0/r$ and distribution pattern of heat flux but also v' , r_0 and dv_{SH}/di_{SH} are included as parameters. Therefore the curves for various pressures and subcoolings have to be shown. Figure 10 shows the ΔP^* curves for various values of q_1/q_2 and L_1/L_2 at 80 ata and $r_0=100, 200$. Broken line indicates the steam quality x_2 at outlet. It can be seen from this figure that the value of ΔP^* decreases with a decreasing value of r_0 and the region of negative slope does not exist at lower values of r_0 . This means that the smaller the subcooling is and the shorter the length of econo-

Fig. 10 Nondimensional diagram of static characteristics of pressure drop for steam generating tube with stepwise uniform distribution of heat flux (with superheater)

Fig. 11 Nondimensional diagram static characteristics of pressure drop for steam generating tube with uniform distribution of heat flux (with superheater), influence of pressure

mizer is, the more stable the flow becomes, regardless of distribution pattern of heat flux. And it is seen that the case, in which heat flux in upstream part is smaller than that in downstream part and the length of the former is longer than that of the latter, is favourable for the stability of flow. These tendencies agree with the case mentioned in 3.1.1.

At 80 ata., the characteristic curve indicates the interesting results that the extremals exist not only in the wetted steam region but also in the superheated steam region and 4~5 points of ξ may correspond to certain values of ΔP^* for the larger of r_0 . In this case the flow rate of a certain tube may be very low and then the fluid may be superheated at the outlet, while flow rates of the others may be very high and the fluid at the outlet has a very low quality in parallel steam generating channels.

Figure 11 shows the ΔP^* curves for $r_0=40$ and uniform distribution of heat flux. It is seen that at higher pressure, ΔP^* increases monotonously with ξ and takes no extremals.

4. Application of the nondimensional diagrams of static characteristics of pressure drop

Three examples for the application of the diagrams are shown as follows:

4.1 Flow stability in steam generating tubes

When the characteristic curve of pressure drop in steam generating tubes has a negative slope, the flow rate will fluctuate or excursion of flow rate will occur with the relation of the characteristics of pump head. In this case, it would be necessary to improve the characteristic in the steam generating tube system. Then from the conclusions obtained in Figs. 3~8, the value of $\alpha r_0/r$ should be decreased and the heating surface should be arranged

Fig. 12 Influence of the change of heat flux on ΔP^* and ξ

to give higher heat flux in downstream part. For instance, in Fig. 4, in the case of $\alpha r_0/r=10$ and heat flux being uniformly distributed along the tube, the negative slope region exists. But by changing the value of $\alpha r_0/r$ from 10 to 5 or q_1/q_2 from 1 to 1/4, the flow stability is improved (it may not always be possible to change the value for the arrangement of heating surface).

4.2 Distribution of flow rates in parallel steam generating tubes

Parallel steam generating tubes have been often used in once-through boilers. If the heat flux for each tube is not equal, the flow rate in each tube will not be equal, the flow rate in a certain tube will be very low and the tube will be burnt out. Therefore, from a safety view point it is necessary to estimate the flow rates corresponding to heat flux in each tube. Now let's consider a design point 1 ($\Delta P^*=0.7$ and $\xi=0.8$) in Fig. 12. When heat flux for a tube increases to 1.2 times the normal heat flux, the nondimensional flow rate of this tube G'_0 becomes 1.2 times G_0 for the heat balance: $r_0 G'_0 = (1.2q)/L_T$. Since the pressure drop between headers is generally constant for parallel channels, the nondimensional pressure drop $\Delta P^{*'}_0$ corresponding to G'_0 decreases to 1/1.44 of designed nondimensional pressure drop ΔP^* . Then the equilibrium point moves from 1 to point 1' ($\Delta P^*=0.49$) and the flow rate of this tube G' decreases to 74% of designed flow rate G , i.e.

$$G' = 1.2 \times 0.49 \times G / 0.8 = 0.74G.$$

In the case that the point 2 is the equilibrium point for a normal heat flux, when the heat flux increases to 1.2 times the normal heat flux, in the same manner described above, point 2 moves to point 2' and the outlet quality x_2 increases from 0.3 to about 0.6. As shown above, the change of flow rate can be easily estimated from the diagrams presented in this paper.

4.3 The performance of steam generating tubes at reduced pressure

A boiler is sometimes operated at reduced

pressure. Since the value of $\alpha r_0/r$ decreases with a decreasing pressure at constant inlet enthalpy and the flow stability decreases with a decreasing value of $\alpha r_0/r$, it is necessary to determine the allowable lowest pressure for the operation. Let's consider the case of $q_1/q_2=1/2$, $L_1/L_2=1/3$ and $r_0=100$ in Fig. 3. Since the value of $\alpha r_0/r$ at which the slope of the curve reaches zero is 0.86, then the allowable lowest pressure is determined as 45 ata from this value. Therefore, if the boiler is operated under 45 ata, the flow will become unstable with the relation of the characteristics of pump head.

5. Conclusions

(1) The nondimensional expression for the relation between pressure drop and flow rate in steam generating tubes is derived and the diagrams for many cases are shown that are useful for design of steam generating tube systems. By this expression, the influences of many factors such as tube length, tube diameter, friction coefficient, flow rate, pressure, subcooling and heat flux on the characteristics of pressure drop can be treated in generalized forms.

(2) In the case of smaller value of $\alpha r_0/r$, nondimensional pressure drop ΔP^* is generally small and monotonously increases with ξ and the flow is stable. In the case of larger value of $\alpha r_0/r$, ΔP^* is large and ΔP^* curve has extremals and a negative slope. And the flow is stable in the case that the heat flux is larger in downstream part.

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