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by Andrew A. Schoenberg and Donald R. Packe

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SUMMARY

A problem in the development of single-pass boilers for Rankine space power systems is the potential flow maldistribution of the working fluid among the parallel tubes of the boiler tube bundle. Flow maldistribution can result in undesirable liquid carryover to the turbine. This maldistribution can occur if the pressure drop of an individual tube is the same for more than one value of flow. This multivalued characteristic has long been recognized for boilers with low exit qualities; however, recent tests and the analysis presented herein show that a similar multivalued pressure-drop characteristic can also exist in boilers that operate with superheated vapor at the exit.

The analysis was performed for counterflow liquid-heated single-pass boilers of the type used in Rankine space power systems such as SNAP-8. The analysis takes into account the nonuniform axial distribution of heat flux, variable geometry factors, and the variation of the lengths of the preheat, boiling, and superheat regions. Comparison with data from a single-tube water boiler indicates that the analytical model predicts the pressure drop and its dependence on various boiler-input variables with an average error of less than 10 percent.

With the use of results generated from the analytical model, effects on the pressure-drop function caused by variations in some of the geometric and thermodynamic parameters of the boiler were studied. From these studies, the following causes of the multivalued characteristic were established:

(1) The magnitude of the multivalued effect is strongly influenced by a boiler-tube insert, but the characteristic could exist even with an open tube.

(2) The multivalued characteristic can exist for either a choked nozzle or a constant pressure at boiler discharge.

Methods for eliminating the multivalued characteristic and thus the potential for flow maldistribution are evaluated.

INTRODUCTION

Rankine space power systems such as SNAP-8 (ref. 1) use a single-pass counterflow multitube boiler to vaporize the working fluid that drives the turboalternator. A problem with such boilers can be uneven division of the working-fluid flow among the parallel tubes. A particularly serious form of maldistribution, with attendant liquid carryover to the turbine, can result if pressure drop as a function of flow of the individual tubes is a multivalued function (i.e., if the tube has the same pressure drop for more than one flow). Such multivalued pressure-drop functions, which are characterized by a negative slope of pressure drop as a function of flow for a range of working-fluid flow, have been studied for boiler tubes with low exit quality (refs. 2 to 6). However, for boiler tubes with an extensive superheat region (quality of 1.0 at exit), the problem of multivalued pressure drop as a function of flow has received little attention. A recent test of a 19-tube counterflow boiler (ref. 7) indicates that the pressure-drop function of these boilers can be multivalued even if superheated vapor is being generated. The nature of this characteristic has not been well understood, and indications are that a number of factors related to geometry, heat-flux distribution, and boundary conditions may be influential.

The study reported herein, therefore, was conducted with the purpose of deriving a pressure-drop-against-flow function for a typical counterflow boiler tube in terms of dimensionless variables and parameters. This function would be useful for analyzing the factors that make the function multivalued. A further objective of this analysis was to determine effective methods of making the pressure-drop function single valued and, hence, of eliminating this potential for flow maldistribution in a multitube boiler.

The analysis is limited to a counterflow liquid-heated tube-in-tube boiler and takes into account the axially nonuniform heat-flux distribution, variable-geometry effects, and the lengths of the preheat, boiling, and superheat regions. A check on the accuracy of the analysis was obtained by comparison with data of a single-tube water boiler. With the use of results generated from the analytical model, effects on the pressure-drop function caused by some of the geometric and thermodynamic parameters of the boiler were examined, and from these effects the causes of the multivalued characteristic were established. Finally, methods for eliminating the multivalued characteristic and thus the potential for flow maldistribution were evaluated.

DERIVATION OF BOILER EQUATIONS

A typical tube in a single-pass boiler of a Rankine space power system is shown in figure 1. The heating fluid flows on the outside of the tube, counterflow to the working fluid on the inside of the tube. The wire-wound plug and the spiral wire shown are flow swirl devices intended to promote heat transfer and phase separation by breaking up the larger liquid droplets as well as by centrifuging the denser liquid to the wall.

For purposes of analysis, the boiler tube was divided into three regions:

(1) The all-liquid region, where the working fluid is preheated to the saturation temperature

(2) The boiling or two-phase region, where the liquid is vaporized

(3) The gaseous region, where the vapor is superheated

In all three regions only the contribution of friction to pressure drop was considered. Momentum pressure drop was neglected since it is negligible compared with friction pressure drop for the cases studied. For simplicity, the pressure gradient in the two-phase region was based on homogeneous flow, and a constant friction factor was determined for a vapor quality of 1. The formulation of a pressure gradient included the variation of velocity with local quality and took into account the nonuniform and variable heat-flux distribution and boiling length. The heat-transfer analysis was based on standard counterflow heat-exchanger equations. Constant, average heat-transfer coefficients were assumed in each of the three regions. For purposes of integration of the heat-flux equation in the boiling region, a constant saturation temperature was also assumed. The validity of these assumptions is discussed during the development of the applicable equations.

In the first part of this section, the basic pressure-drop equations used in each region of the boiler are presented, and their dependence on the boiler heat flux and geometry is indicated. In the second part, the equations for the heat flux and the length of the preheat, boiling, and superheat regions are derived. Finally, the equations are synthesized, and the boiler-pressure-drop function is presented in dimensionless form.

Basic Pressure-Drop Functions for the Three Boiler Regions

For the single-phase (preheat and superheat) regions, a form of the standard turbulent friction-loss function (eq. (1)) was used:

$$\frac{dP}{dl} = -\frac{fp}{4A_c}\left(\frac{V^2\rho}{2g}\right) = -\left(\frac{fp}{8A_c^3 g}\right)\frac{w_w^2}{\rho} \tag{1}$$

(All symbols are defined in appendix A.) The friction factor was assumed to incorporate the effect of the spiral wire and the effect of the bends in the spiral flow passages of the plug. The derivative dP/dl was taken with respect to the axial distance along the tube so that in the plug region a multiplication factor dl_{pl}/dl must be used to account for the extra length of the spiral path. An example of how these multipliers are evaluated is given in appendix B. If dP/dl is assumed to be constant in the single-phase regions, equation (1) can be integrated over the preheat and superheat lengths to give equations (2) and (3), respectively:

$$\Delta P_p = \frac{f_p g_p w_w^2 l_p}{\rho_{w,l}} \tag{2}$$

$$\Delta P_s = \frac{f_s g_s w_w^2 l_s}{\rho_{w,v}} \tag{3}$$

where

$$g_s = \frac{p_s}{8gA_s^3}$$

$$g_p = \frac{p_{pl}}{8gA_{pl}^3}\left(\frac{dl_{pl}}{dl}\right)$$

The pressure drop in the two-phase region requires a more elaborate analysis, which includes not only the effect of the variation of length and geometry, but also the local variation in velocity resulting from quality changes. A correlation based on that proposed by W. L. Owens (ref. 8), and which assumes homogeneous flow, was used in this analysis. (A similar analysis for the case of uniform heat flux is given in reference 9.) In the analysis presented herein, several simplifications from Owens' analysis were made. The pressure drop due to change in elevation was neglected since space power boilers will generally operate in zero gravity. As mentioned earlier in this section, the momentum pressure drop was also neglected. Only the frictional component, of the following form, remains:

$$\frac{dP}{dl} = -\frac{f_b}{2Dg}\left(\frac{w_w}{A_c}\right)^2\left[\frac{x}{\rho_{w,v}} + (1 - x)\frac{1}{\rho_{w,l}}\right] \tag{4}$$

The total pressure drop in the boiling region is expressed by

$$\Delta P_b = \int_0^{l_b} \frac{f_b g_b w_w^2}{\rho_{w,v}} \left[x + \frac{\rho_{w,v}}{\rho_{w,l}} (1 - x)\right] dy \tag{5}$$

where

$$\left.\begin{aligned} g_b &= g_p \qquad 0 \leq y \leq (l_{pl} - l_p) \\ g_b &= g_s \qquad l_b \geq y > (l_{pl} - l_p) \end{aligned}\right\} \tag{6}$$

Equation (5) is further simplified because the ratio of the densities $\rho_{w,v}/\rho_{w,l}$ is generally small for the operating pressures of space power system boilers. Hence, it was assumed that a good approximation to the integral can be obtained by neglecting the term multiplied by $\rho_{w,v}/\rho_{w,l}$. The resultant integral takes on the simple form

$$\Delta P_b = \int_0^{l_b} \frac{f_b g_b w_w^2 x}{\rho_{w,v}} dy \tag{7}$$

To permit closed-form integration of equation (6), it was further assumed that f_b and $\rho_{w,v}$ were constant so that they could be taken outside the integral, as shown in equation (8):

$$\Delta P_b = \frac{f_b}{\rho_{w,v}} w_w^2 \int_0^{l_b} g_b x \, dy \tag{8}$$

This simplified form can now be used to calculate the variation in pressure drop due to boiling length l_b, geometry g_b, and the local heat flux or quality x, as well as vapor density $\rho_{w,v}$ and total flow rate w_w. The variation of x as a function of y remains to be determined from the heat-flux analysis. This analysis together with the determination of the lengths of the three regions is presented in the next section.

Derivation of Heat Flux and Lengths of the Three Regions

The heat-flux distribution in a counterflow boiler differs considerably from the more uniform distributions obtained with electrical or constant-temperature heat sources. Figure 2 shows a typical temperature and pressure profile in a counterflow boiler. The working fluid enters subcooled at a temperature T_{wi} and is rapidly heated to the saturation temperature T_{sat} corresponding to the local saturation pressure P_{sat}. In a counterflow boiler, the temperature difference $T_{hb} - T_{sat}$, termed the "pinch-point temperature difference," represents an important limitation on the amount of enthalpy available to vaporize the working fluid. The heating fluid, as shown by the upper curve, experiences little temperature change in the preheat region because of the small amount of enthalpy change required for preheating as compared to boiling. By far the greatest amount of heat-transfer and heating-fluid temperature drop occurs in the boiling region. This region of length L_b begins inside the plug and normally extends over a large part of the tube length. The working-fluid temperature in this region remains at the saturation temperature, which decreases slightly because of the pressure drop in the boiling region.

Of the total tube length, the portion L_s remaining after the preheat and boiling regions serves to superheat the vapor stream. For an actual boiler, the transition from boiling to superheating is not as distinct as indicated in figure 2 becasue of the nonequilibrium coexistnece of liquid droplets and superheated vapor over a considerable part of the tube length. For the purposes of the analysis, however, an abrupt transition was assumed. As in the preheat region, relatively little heat transfer occurs in the superheat region, as indicated by the small drop in heating-fluid temperature. The derivation of the equations for the three regions follows.

The heat-transfer and fluid-exit temperatures in the preheating region can be determined from the standard effectiveness relation for a counterflow heat exchanger (ref. 10):

$$\epsilon_p \equiv \frac{T_{sat} - T_{wi}}{T_{hb} - T_{wi}} = \frac{1 - \exp\left[-N_p(1 - a_p)L_p\right]}{1 - a_p \exp\left[-N_p(1 - a_p)L_p\right]} \tag{9}$$

where

$$N_p = \frac{U_p \pi D_o l_t}{w_w c_{w,l}}$$

$$a_p = \frac{w_w c_{w,l}}{w_h c_h}$$

$$L_p = \frac{l_p}{l_t}$$

Note that N_p is a positive dimensionless parameter commonly referred to as the number of transfer units (NTU) of a heat exchanger which is herein based on the total tube length l_t and the coefficient of heat transfer in the preheat region U_p. All the terms in equation (9) are known or will be specified except the preheat length L_p. Equation (9) can be simplified as follows in order to solve for L_p. Given that N_p is positive and that for most high-quality boilers the parameter a_p is much less than 1, the right side of equation (9) can be expanded in a series:

$$\frac{1 - \exp\left[-N_p(1 - a_p)L_p\right]}{1 - ap \exp\left[-N_p(1 - a_p)L_p\right]} \cong 1 - (1 - a_p) \exp\left[-N_p(1 - a_p)L_p\right] - (1 - a_p)a_p \exp\left[-2N_p(1 - a_p)L_p\right] \ldots \tag{10}$$

Neglecting all second order or higher terms in equation (10), substituting into equation (9), and solving for L_p result in

$$L_p = \frac{1}{(1 - a_p)N_p} \ln\left(\frac{1 - a_p}{1 - \epsilon_p}\right) \tag{11}$$

This equation completes the derivation of the necessary equations for the preheat region. Next, the equations of the boiling region are derived.

The heat flux in forced-convection boiling is a complex function of the flow regimes, temperatures, and surface conditions. A general discussion of these regimes can be found in references 2 and 4. Instead of attempting to predict the complex variations in boiling-side heat-transfer coefficient, an average constant overall coefficient U_b was assumed for the calculation of heat flux in the boiling region. This assumption is justified where the heating-fluid and wall-heat-transfer resistances are controlling. This case occurs in most space power boilers in the regions where contact boiling and, hence, high working-fluid heat-transfer coefficients are maintained (ref. 5). The region of contact boiling can be maintained to local qualities as high as 80 percent, particularly in boilers where the spiral wire helps to maintain annular flow (ref. 11). Thus, it is only in the last part of the boiling region that the boiling coefficient becomes small and, hence, controlling. Figure 3 illustrates the difference between the typical experimental overall heat-transfer coefficient and the coefficient assumed for the analysis. It also shows the resultant difference between typical experimental and computed temperature profiles of the heating and working fluids. The saturation temperature was assumed to be constant at the value where the boiling region begins.

With the assumptions of constant U_b and uniform T_s over the boiler length, the effectiveness relation (ref. 10) can be expressed as

$$\epsilon_b = \frac{T_{hs} - T_{hb}}{T_{hs} - T_{sat}} = 1 - \exp\left(-N_b L_b\right) \tag{12}$$

where

$$N_b = \frac{U_b \pi D_o l_t}{w_h c_h}$$

and

$$L_b = \frac{l_b}{l_t}$$

The following conservation of energy for the whole boiling region can be applied to solve equation (12) for the boiling length in terms of specified variables:

$$w_h c_h (T_{hs} - T_{hb}) = w_w H_w X_o \tag{13}$$

The quality is equal to 1 except in cases where the boiling length would exceed the available tube length. This condition was included to account for the pressure drop in the boiler when the quality at exit is less than 1. Substituting the expression of equation (13) for the terms $T_{hs} - T_{hb}$ in equation (12) and rearranging to solve for the boiling length result in

$$L_b = \frac{1}{N_b} \ln\left(\frac{1}{1 - \epsilon_b}\right) \tag{14}$$

where

$$\epsilon_b = \frac{w_w H_w X_o}{w_h c_h (T_{hs} - T_{sat})}$$

The length of the superheat region L_s, which is required to compute the pressure drop in that region, is the length remaining after the preheat and boiling lengths, as given in dimensionless form in equation (15):

$$\left.\begin{aligned} L_s &= 1 - (L_b + L_p) \qquad (L_b + L_p) < 1.0 \\ L_s &= 0 \qquad L_b + L_p \geq 1.0 \end{aligned}\right\} \tag{15}$$

Note that $L_b + L_p$ cannot become greater than 1 (i.e., exceed the length of the available tube). Hence, if the calculation of L_b from equation (14) exceeds the length constraint, it means that the quality at the boiler tube exit is less than 1. Consequently, the outlet quality X_o which enters into the effectiveness expression must be adjusted so that the end of the boiling region corresponds to the end of the tube. With wet vapor at the tube exit, the appropriate temperature expressions are also adjusted, (i.e., $T_{hs} = T_{hi}$ and $T_{wo} = T_{sat}$). For normal boiler operation where superheat is present, the temperature expressions are given in the following equations:

$$T_{hb} = T_{hs} - \frac{w_w H_w X_o}{w_h c_h} \tag{16}$$

$$T_{hs} = T_{hi} - a_s(T_{wo} - T_{sat}) \tag{17}$$

where

$$a_s = \frac{w_w c_{w,v}}{w_h c_h}$$

$$T_{wo} = T_{hi} - (T_{hi} - T_{sat})(1 - a_s)\exp\left[-N_s(1 - a_s)L_s\right] \tag{18}$$

where

$$N_s = \frac{U_s \pi D_o l_t}{w_w c_{w,v}}$$

Equation (18) was derived from the simplified expression (eq. (9)) of the effectiveness of a counterflow heat exchanger. The derivation of the lengths and terminal temperatures of the various regions is essentially completed. Before the pressure-drop function of the boiler can be expressed in closed form, however, the expression for the boiling-region pressure drop (eq. (7)) must be integrated.

Integration of the Two-Phase Pressure-Drop Equation

An expression for the local quality as a function of the length y is required to integrate equation (7). The amount of vapor generated for any length y, measured from the start of boiling, can be obtained by an energy balance on the heating fluid. The amount of vapor generated is proportional to the temperature drop in the heating fluid as given in equation (19):

$$w_v(y) = \frac{w_h c_h}{H_w}(T_h - T_{hb}) \tag{19}$$

The local temperature difference can be expressed in the form of the effectiveness relation of equation (12), and after appropriate substitution and simplification of terms the expression for $w_v(y)$ becomes

$$w_v(y) = \frac{w_h c_h}{H_w}(T_{hs} - T_{sat})\left[\exp\left(-N_b L_b\right)\right]\left[\exp\left(\frac{N_b y}{l_t}\right) - 1\right] \tag{20}$$

or equivalently,

$$x = \frac{X_o}{\epsilon_b}\left[\exp\left(-N_b L_b\right)\right]\left[\exp\left(\frac{N_b y}{l_t}\right) - 1\right] \tag{21}$$

The variable of integration in equation (21) is y; the length L_b, as given by equation (14), is a constant and corresponds to the upper limit of the integral of equation (7).

The integration of equation (7) can now be performed. The integral is divided into two parts corresponding to the plug and the swirl-wire or open-tube boiling regions as expressed in equation (22a):

$$\Delta P_b = \frac{f_{pl,b} w_w^2 g_p}{\rho_{w,v}} \int_0^{l_{pl} - l_p} x \, dy + \frac{f_{b,\delta} w_w^2 g_s}{\rho_{w,v}} \int_{l_{pl} - l_p}^{l_b} x \, dy \tag{22a}$$

Substituting the expression of equation (21) for x into this equation allows the closed-form integration of equation (22a):

$$\Delta P_b = \frac{X_o}{N_b \epsilon_b} \exp\left(-N_b L_b\right) \left(\frac{f_{pl,b} w_w^2 g_p l_t}{\rho_{w,v}} \left\{ \exp\left[N_b (L_{pl} - L_p)\right] - N_b (L_{pl} - L_p) - 1 \right\} + \frac{f_{b,\delta} w_w^2 g_s l_t}{\rho_{w,v}} \left\{ \exp N_b L_b - N_b L_b - \exp\left[N_b (L_{pl} - L_p)\right] + N_b (L_{pl} - L_p) \right\} \right) \tag{22b}$$

Rewriting the integrated expression in terms of dimensionless parameters results in

$$\Delta P_b = \frac{f_{b,\delta} w_w^2 g_s X_o l_t}{\rho_{w,v} N_b} \left\{ 1 - \left(\frac{1}{\epsilon_b} - 1\right) \ln\left(\frac{1}{1 - \epsilon_b}\right) + \left(\frac{1}{\epsilon_b} - 1\right) \left(r_{s,\delta} - 1\right) \left[\exp\left(N_{pl}\right) - N_{pl} - 1\right] \right\} \tag{23}$$

where

$$r_{s,\delta} \equiv \frac{f_{pl,b} g_p}{f_{b,\delta} g_s}$$

$$N_{pl} \equiv N_b (L_{pl} - L_p)$$

and

$$\epsilon_b = \frac{T_{hs} - T_{hb}}{T_{hs} - T_{sat}} = 1 - \exp\left(-N_b L_b\right) = \frac{w_w H_w X_o}{w_h c_h (T_{hs} - T_{sat})}$$

Equation (23) expresses the pressure drop of the boiling region as the frictional pressure-drop gradient for all gas flow in the swirl-wire or open-tube region, multiplied by $X_o l_t/N_b$ and the expression in braces. This latter expression can be considered as the dimensionless length weighting factor which incorporates the effects of the nonuniform heat-flux distribution. In addition, if $r_{s,\mathscr{s}}$ is 1 (i.e., if the tube has no plug insert and a uniform internal geometry), the latter term in the braces is 0. Hence, this term is the increment of the weighting factor caused by the plug insert.

This equation completes the derivation of the constituents of the boiler pressure drop. Next, these components are combined, and the boiler equations are summarized.

Summary and Synthesis of Boiler Equations

The total pressure drop of the boiler tube is the sum of equations (2), (3), and (23), as expressed in the following equation.

$$\Delta P_t = \frac{f_s g_s w_w^2 l_s}{\rho_{w,v}} + \frac{f_p g_p w_w^2 l_p}{\rho_{w,l}} + \frac{f_{b,\mathscr{s}} w_w^2 g_s}{\rho_{w,v}} \frac{l_t X_o}{N_b}\left\{1 - \left(\frac{1}{\epsilon_b} - 1\right)\ln\left(\frac{1}{1 - \epsilon_b}\right)\right.$$
$$\left. + \left(\frac{1}{\epsilon_b} - 1\right)\left(r_{s,\mathscr{s}} - 1\right)\left[\exp\left(N_{pl}\right) - N_{pl} - 1\right]\right\} \qquad (24)$$

Equation (24) can be expressed in dimensionless form by defining the pressure gradient in the superheat region at design flow times the total tube length as the normalizing factor $\Delta P_{v,d}$:

$$\Delta P_{v,d} = \frac{g_s f_s w_{w,d}^2 l_t}{\rho_{v,d}} \qquad (25)$$

The resultant pressure-drop function becomes

$$\varphi_t \equiv \frac{\Delta P_t}{\Delta P_{v,d}} = \frac{W_w^2}{R_d}\left(L_s + R_d r_{p,s} L_p + \frac{r_{b,s} X_o}{N_b}\left\{1 - \left(\frac{1}{\epsilon_b} - 1\right) \ln\left(\frac{1}{1 - \epsilon_b}\right) + \left(r_{s,\ell} - 1\right)\left(\frac{1}{\epsilon_b} - 1\right)\left[\exp\left(N_{pl}\right) - N_{pl} - 1\right]\right\}\right) \tag{26}$$

where the dimensionless parameters are

$$W_w = \frac{w_w}{w_{w,d}}$$

$$R_d = \frac{\rho_{w,v}}{\rho_{v,d}}$$

$$r_{p,s} = \frac{f_p g_p \rho_{v,d}}{f_s g_s \rho_{w,l}}$$

$$r_{b,s} = \frac{f_{b,\ell}}{f_s}$$

The intermediate variables entering into the pressure-drop function can be expressed in the form of known or computable input variables as follows:

$$\left.\begin{aligned} L_s &= 1 - L_b - L_p \qquad L_p + L_b \le 1 \\ L_s &= 0 \qquad L_p + L_b \ge 1 \end{aligned}\right\} \tag{27}$$

$$L_b = \frac{1}{N_b} \ln\ (1 - \epsilon_b)^{-1} \tag{28}$$

$$\epsilon_b = \frac{w_w H_w X_o}{w_h c_h (T_{hs} - T_{sat})} \tag{29}$$

$$L_p = \frac{1}{(1 - a_p)N_p} \ln \left(\frac{1 - a_p}{1 - \epsilon_p}\right) \tag{30}$$

where

$$\epsilon_p \equiv \frac{T_{sat} - T_{wi}}{T_{hb} - T_{wi}}$$

and

$$a_p = \frac{w_w c_{w,l}}{w_h c_h}$$

$$\left.\begin{array}{ll} N_{pl} = N_b(L_{pl} - L_p) & L_{pl} > L_p \\ N_{pl} = 0 & L_{pl} \le L_p \end{array}\right\} \tag{31}$$

As mentioned previously, X_o in equation (29) is 1 except if either $L_b > 1 - L_p$ or $\epsilon_b \ge 1.0$. In these cases X_o is adjusted so that the constraint $L_b = 1 - L_p$ is satisfied.

The temperature relations required for calculating the effectivenesses are expressed by equations (32) to (35):

$$T_{hs} = T_{hi} - a_s(T_{wo} - T_{sat}) \tag{32}$$

$$T_{hb} = T_{hs} - \frac{w_w H_w X_o}{w_h c_h} \tag{33}$$

$$T_{ho} = T_{hb} - a_p(T_{sat} - T_{wi}) \tag{34}$$

$$T_{wo} = T_{hi} - (T_{hi} - T_{sat})(1 - a_s)\exp\left[-N_s(1 - a_s)L_s\right] \tag{35}$$

The saturation temperature is related directly to the saturation pressure of the boiler P_{sat}. The saturation relation between T_{sat} and P_{sat} is expressed as

$$T_{sat} = \frac{A}{B - \ln(P_{sat})} \tag{36}$$

The parameters A and B were adjusted for each working fluid to fit its standard saturation curve. The saturation pressure, P_{sat} was assumed to be that at the pinch point and is evaluated by equation (37):

$$P_{sat} = P_o + \Delta P_b + \Delta P_s \tag{37}$$

The boiler exit pressure P_o depends on the exit boundary condition. For studying the potential for flow maldistribution among parallel tubes connected to a plenum, the most appropriate exit boundary condition is a constant exit-plenum pressure, which is independent of the flow variations in any one tube. On the other hand, for evaluating the pressure drop for uniform flow distribution in the tube bundle or for comparison with pressure-drop data from a boiler operating in a power loop with a turbine, the appropriate exit boundary condition is a nozzle-type restriction. A choked nozzle characteristic, as given by equation (38), was assumed for the latter situation:

$$P_o = K_{tn} w_w X_o \sqrt{T_{wo}} \tag{38}$$

For exit qualities less than 1, it was assumed that the nozzle pressure was determined by the gas flow rate $w_w X_o$ alone.

The variation of the pressure level in the boiler may cause a significant variation in the vapor density. The variation of the dimensionless density factor R_d was, for simplicity, related to the saturation pressure and temperature by the perfect gas law as expressed in

$$R_d = \frac{\rho_{w,v}}{\rho_{v,d}} \cong \frac{P_{sat}}{R_g T_{sat} \rho_r} \tag{39}$$

where

$$\rho_r \equiv \frac{P_{sat,d}}{T_{sat,d}R_g}$$

was chosen so that R_d is 1 at the reference design point.

Conceptually, equations (24) to (39) constitute a complete mathematical model of the boiler operation. Because of the nonlinearity and complexity of the equations, a closed-form solution of the pressure drop in terms of the primary input variables T_{wi}, w_w, T_{hi}, and w_h cannot be obtained. Some insight into the interactions of the various equations can be gained by representing them in block diagram form, as shown in figure 4. The boundary conditions are a specified working-fluid flow at inlet and a choked nozzle at exit. Superheat was assumed to be present at the exit. In the upper left of the figure, the pressure-drop equations are represented. The primary input variables to this section are the preheat length L_p and the effectiveness ϵ_b of the boiling region. These variables are generated by the boiler heat-transfer equations as represented in the lower part of the figure. The major independent inputs to this section are the flow rates W_w and W_h and the inlet temperatures T_{wi} and T_{hi}. The major dependent input to these equations is the saturation temperature T_{sat} which is specified by the equations of the exit boundary condition and the pressure-drop relations as represented in the upper right of the figure. The major inputs to this latter section are the working-fluid flow, the pressure drop in the boiling and superheat regions, and the nozzle boundary condition. The interaction of the saturation pressure with the pressure drop, the staturation temperature, the density, and the outlet pressure is indicated by the various loops shown in figure 4.

This discussion completes the derivation of the boiler equations and the defintion of their interaction. Before proceeding with the analysis of the theoretical pressure-drop function and its dependence on the various input variables and parameters, it is desirable to estimate the accuracy of the theory. This estimation was made by comparison with experimental data from a single-tube water boiler.

COMPARISON OF THEORY WITH WATER-BOILER DATA

The data presented herein were obtained under NASA contract NAS3-9420 from a counterflow boiler with water as both the heating and working fluids. This boiler is described in detail in appendix C. Tests were conducted around the design point (listed in table I) to establish the variation of boiler pressure, temperature, and pressure drop in response to variation in the working- and heating-fluid inlet temperatures and flows. The range of the variations was wide enough to determinethe dependence of the pressure drop on the input variables and, hence, to permit a good comparison with theory.

The theoretically predicted variations of the boiler pressure drop were calculated by using the equations derived in the previous section, with the parameters adjusted for the water boiler. The equations were programmed on a digital computer. A listing of the FORTRAN IV program and the calculation of the numerical values of the water-boiler parameters are given in appendix B.

The comparison of the measured and predicted pressure drops as a function of the working-fluid flow is shown in figure 5(a). The pressure drops are shown in the dimensionless form normalized by the constant $\Delta P_{v,d} = 9.7$ psia (67 000 N/m^2). The flow rate is shown as a fraction of the design flow. Note the excellent agreement between the predicted and the measured data in both the level and variation of the pressure drop for flow rates below design. For flows higher than design, the overall variation of pressure drop was satisfactorily predicted. The largest error was in the transition from superheated vapor to wet vapor, which occurs at the exit of the boiler as shown in figure 5. In the actual boiler, the transition is much less sharp than for the analytic model, because of the coexistence of droplets and superheated vapor at the exit. This nonequilibrium condition was encountered in the boiler testing as indicated by (1) high and low readings of thermocouples in the superheat region and (2) the decrease in superheat temperature that occurred in an adiabatic, coiled-tube section connected at the exit of the boiler. Under these conditions it could be expected that the predicted variation in exit superheat and quality would differ somewhat from that measured. However, the trend of the variation is predicted reasonably well, as shown in figures 5(c) and (d). Shown in figure 5(b) is the expected and actual variation in the boiler-outlet pressure as a function of the flow with the assumption of a choked nozzle at the exit. In the transition region, the pressure against flow characteristic deviates from the linear relation expected for adiabatic gas flow.

A further check on the validity of the pressure-drop analysis is the ability to predict the variation of pressure drop caused by changes in the heating-fluid flow and inlet temperature, as well as the subcooled inlet temperature of the working fluid. The agreement of the pressure drops both in level and variation is excellent, as shown in figure 6. The predicted values generally fall about 10 percent below the measured values. This difference was expected because of the neglected effects in the simplified homogeneous two-phase flow model used in the pressure-drop prediction.

This comparison of theory and data shows that the analytic model accurately predicts both the pressure-drop level and its variations in the single-tube boiler. The analytic model should therefore be useful in the analysis of the pressure-drop characteristics that cause the maldistribution in the multitube configuration. Although the theory is shown to predict boiler performance for only the particular data available, the analysis might be valid over a much broader range. Applicability to other ranges of boiler operation and design points (such as boilers designed for low pressure drop or high heating-fluid flow rates) has not been evaluated. There appears to be a paucity of data in the literature for total performance.

EXAMINATION OF BOILER PRESSURE-DROP CHARACTERISTICS

The analysis was performed to establish the general characteristics of the counterflow-boiler pressure-drop function and how the various parameters and input variables influence this characteristic. In view of the complexity of the function and the 20 or more variables involved, the analysis was limited to the effects of the parameters $r_{s,\ell}$, l_{pl}, and U_b and the input variables T_{wi}, w_w, w_h, and T_{hi}.

To make the analysis more applicable to present-day space power systems, the reference values for the boiler parameters were chosen to match those of a seven-tube mercury boiler presently being constructed for the SNAP-8 system. This boiler is predicted to have a multivalued pressure-drop function which is to be made single valued by insertion of a nozzle at the entrance of each tube. The influence of the nozzle on the pressure-drop characteristic is treated in the analysis. A brief description of the mercury boiler and how it differs from the water boiler is given in appendix D. In this analysis both choked-nozzle and constant-pressure boiler-tube discharge conditions are considered. The pressure-drop function for a choked-nozzle discharge appropriate when considering the boiler as a whole is computed by using the FORTRAN IV program of appendix B with constants adjusted for the mercury boiler. The pressure-drop function for constant exit pressure appropriate when considering one tube acting in a multitube configuration is computed by using the FORTRAN IV program given in appendix E.

Components of Pressure-Drop Function

Figure 7(a) illustrates the typical shape and composition of the multitube boiler pressure-drop-against-flow function. The boundary condition at the exit is a choked nozzle. The total dimensionless pressure drop φ_t (solid curve) is the sum of the component pressure drops (dashed curves) of the preheat, boiling, and superheat regions. The pressure drop due to an inlet orifice is not included. For all but the highest flows, the preheat pressure drop is negligibly small so that only the boiling-region and superheat-region pressure-drop components must be considered. The latter exhibit a complex variation with flow, which can be explained by considering the variation of the lengths of the regions in relation to the plug and tube lengths. These lengths are shown in figure 7(b). At very low flow rates, the preheat and boiling regions are so short that substantial superheating occurs in the plug with the attendant high velocities and a gas pressure drop in the plug. This pressure drop at first increases with flow. However, at the same time, the preheat and boiling lengths increase to gradually supplant the superheat region in the plug and, thus, produce a net decrease in the superheat pressure drop. The boiling pressure drop, in the meantime, increases rapidly so that the total pressure drop continues to rise until a flow of 0.6 is reached. At this point the superheat region has been moved completely out of the plug, and the boiling pressure drop has reached a maximum. As the flow rate is increased further, the high-quality end of the boiling region is pushed out of the plug region, which causes a definite decrease in both the boiling and total pressure drops. This negative $\partial\varphi_t/\partial W_w$ persists until approximately design flow for the case shown. The superheat pressure drop in the meantime is increasing gradually as a result of the increasing flow but is overcome by a decreasing superheat length as the boiling and preheat regions continue to expand. (In other words, above design flow the superheat length begins to shrink so rapidly that a decrease in superheat pressure drop occurs despite the increase in flow.) The total pressure-drop function shows a corresponding increase in the negative slope at these flow rates. As the boiler runs out of superheat length, the presssure drop reaches a minimum. Any further increase in flow causes a gradual increase in the boiler pressure drop. This increase is, however, not very rapid because of the decreasing exit quality and length of the boiling region. If the preheat region expands rapidly enough, a second region of negative slope could result at this point. This region would correspond to the situation widely treated in the literature (refs. 2 to 4 and 9). This second region of negative slope is, however, not likely to occur in space power boilers because of the greater liquid-pressure-drop gradient in the plug region. Furthermore, in these boilers, inlet orifices are usually employed to add an overriding positive slope above design flow.

As shown in the preceding analysis, the plug insert is primarily responsible for the negative $\partial\varphi_t/\partial W_w$ characteristic. Removal of the plug does not necessarily solve the problem, however, as can be seen in figure 7(c). Here the $r_{s,\delta}$ parameter has been set equal to 1, which effectively eliminates the plug insert (see eq. (19)). Yet the pressure-drop function exhibits a negative-slope region, even though this slope as well as the pressure-drop level is much reduced. Thus, two distinct mechanisms are at work. One is the result of the boiling and superheat regions moving out of the plug insert. The other occurs when the superheat length is rapidly eliminated from the tube as the boiling region expands with flow.

Effect of Boiling Pressure Level and Its Dependence on the Exit Boundary Condition

The simple explanation given in the preceding section is not completely adequate in describing the magnitude of the negative $\partial\varphi/\partial W_w$. This slope is substantially influenced by the variation of the vapor density as the flow rate is changed. This variation is caused by the choked nozzle assumed at the exit of the boiler tube bundle which results in a linear increase in exit pressure with flow as long as superheat remains at the exit. The higher pressure causes a higher vapor density, which acts to reduce the pressure drop (see density parameter R_d in eq. (26)). In contrast, if the pressure in the boiler were fixed, the $\partial\varphi/\partial W_w$ would be less negative. Such a situation is actually of primary interest in evaluating the possibility of a single tube operating a different flow but at the same pressure level and pressure drop as other tubes in a bundle. Such a tube is best described by a constant exit pressure corresponding to the exit-plunum pressure as determined by the total flow through the boiler. Such a boundary condition describes only the simplest possible maldistribution in a tube bundle, which in general can take on a multiplicity of flow-distribution modes. Both the simplest mode and these more complex modes can exist if the single-tube pressure-drop-against-flow function is multivalued. Hence, attention was focused on the pressure-drop characteristic of a simple single tube with constant exit pressure to evaluate the actual potential for maldistribution. The characteristic is evaluated with the assumption that the heating fluid in the tube bundle is unmixed (i.e., each tube has its individual supply of heating fluid). This assumption represents the worst case, in that the slope $\partial\varphi/\partial W_w$ is more negative for the case of unmixed flow than for mixed flow, where the variations of heating-fluid temperature in response to working-fluid flow changes are minimized.

The pressure-drop function φ_t as a function of W_w for various pressure levels is shown in figure 8 (solid curves). The tube bundle with a choked-nozzle characteristic (dashed curve) was added for contrast. The difference between the dashed and the solid curves, as well as the spacing between the solid curves, indicates the substantial effect of the pressure level on density. The general multivalued characteristic is evident at all pressure levels, but as expected, the slopes of the curves are less than the choked-nozzle characteristic shown. Of particular interest is the single-tube pressure-drop as a function of flow around the design condition ($P_o/P_{o,d} = 1.0$). By coincidence, the pressure-drop function for this pressure level is at a maximum at design flow ($W_w = 1.0$). This operating point is unstable for the single tube operating in a tube bundle (i.e., constant pressure at exit) and would probably lead to wide maldistribution of flow. Under such conditions, a mixture of superheated vapor and liquid would emerge from the boiler exit plenum. Similar results would be obtained if operation at other tube-bundle pressure-drop levels were to be attempted, as evidenced by the multivalued nature of the single-tube functions at all exit pressure levels.

Eliminating Negative-Slope Region in Pressure-Drop Function

The most direct way of eliminating the negative-slope region in the pressure-drop-against-flow function is to increase the liquid flow resistance. This resistance can be increased by narrowing the flow passages or by increasing the spiral pitch in the plug (i.e., increasing the $r_{s,\ell}$ parameter). This procedure, however, also increases the resistance to flow in the plug boiling region (increases the $r_{s,\ell}$ parameter) which negates any improvement resulting from the increased liquid flow resistance. Designing a tapered flow passage or using a plug with a tight pitch in the preheat region and loose pitch in the boiling section has been attempted, but these more complicated cases will not be treated herein. A much simpler approach, which is to be used in the latest design of the SNAP-8 boiler, is to insert an orifice at the entrance of each tube. The only limitations to this method are the allowable pumping loss and the possibility of plugging if the orifice is made too small.

The effect of the inlet orifice on the pressure-drop characteristic is illustrated in figure 9. The dimensionless parameter r_i is the ratio of the orifice pressure-drop parameter $\Delta P_{v,d}$. The solid curves represent the characteristics of a tube with constant exit pressure (i.e., a single tube operating in a large bundle); the dashed curves are for the boiler with a choked nozzle at the exit. The solid curves show that an orifice of $r_i = 0.5$ is required to prevent a multivalued pressure drop for the single tube (constant exit pressure). This value is therefore the minimum required to eliminate the potential for maldistribution in a tube bundle. An orifice of greater resistance ($r_i > 1.0$) is required to eliminate the negative $\partial\varphi_t/\partial W_w$ region for the multitube boiler with a choked nozzle (dashed curves). In general, the amount of orificing is dependent not only on the boundary condition, but also on the plug geometry and heat flux as well.

Effect of Plug Geometry

The plug length L_{pl} and its flow resistance per unit length, as defined by the $r_{s,\ell}$ parameter, have a substantial effect on the pressure-drop function. This effect is illustrated in figure 10. The pressure drops shown include the component due to an orifice of $r_i = 0.5$. As the $r_{s,\ell}$ parameter increases (fig. 10(a)), the pressure drop and the peak in the pressure-drop curve increase. Thus, although an inlet orifice of $r_i = 0.5$ was sufficient to eliminate the maldistribution for the boiler with $r_{s,\ell} = 36.0$, a boiler with $r_{s,\ell} = 100.0$ would require an inlet orifice of $r_i \cong 0.8$. The dashed curve in figure 10 represents the part of the pressure drop caused by an orifice with $r_i = 0.5$; higher values of r_i produce proportionately higher pressure drops and steeper positive slopes. The effect of the plug length on the shape of the pressure-drop function is illustrated in figure 10(b). The increase in pressure drop is substantial (about 40 percent at design flow) when the plug length is increased from 15 to 25 percent of the tube length. As with the $r_{s,J}$ parameter, an increase in the plug length requires an increase in the size of the liquid-pressure-drop orifice to eliminate the negative-slope region. Conversely, shortening the plug tends to reduce the amount of required orificing. It is not desirable, however, to remove the plug completely because of heat-transfer considerations.

Effect of Heat-Flux Distribution on Pressure-Drop Function

It has been demonstrated that the density and plug-geometry factors affect the shape and magnitude of the pressure-drop function. A third category of parameters, associated with the heat-flux distribution and rate, must yet be considered. It has already been demonstrated in the section comparison of theory with water-boiler data, that the pressure drop varies with the input variables that influence only the heat transfer (i.e., the heater-loop flow and inlet temperature, see figures 6(a) and (b). In addition to changing the pressure-drop level, these variables also change the shape of the $\varphi_t(W_w)$ function, as shown in figures 11(a) and (b).

Figure 11(a) shows that as the heating-fluid flow W_h increases the pressure drop in the boiler increases at all power loop flow rates. This increase results from the increase rate of vaporization and shorter boiling and preheat lengths associated with the larger average temperature difference. The variation of length is indicated indirectly by the shift in the minimum point of the pressure-drop curve as W_h increases. This minimum point indicates the working-fluid flow for which the superheat length becomes zero. As expected, the working-fluid flow at which the boiler runs out of superheat length increases with increasing heating-fluid flow. This increase has a desirable effect on the shape of the pressure-drop function in that it extends the region of positive slope. At a heating flow of twice design $(W_h = 2)$, the negative slope emerges only at flow rates above 1.6 of design working-fluid flow.

A much smaller liquid-flow resistance (larger orifice) would be needed to eliminate this negative slope than is required at design heating-fluid flow $(W_h = 1.0)$. Thus, an increase in heating-fluid flow, if the additional pumping power can be justified, will generally produce an improvement in the pressure-drop characteristic. In the limit as W_h becomes large, the heating-fluid temperature drop becomes negligibly small; hence, the curve $W_h = 100.0$ represents a boiler with practically uniform heating-fluid temperature. For this case, the region of steep negative slope in the superheat component of the pressure drop (see fig. 7(b)) represents a high working-fluid flow (relative to design). This causes the liquid-pressure-drop resistance to become large enough, even without an orifice, to prevent the total pressure-drop function from having a region of negative $\partial\varphi/\partial W_w$.

The effect of the heating-fluid inlet temperature, as shown in figure 11(b), is basically the same as that of the working-fluid flow. The temperature variation is expressed in the form of a dimensionless parameter $(T_{hi} - T_{sat,d})/(T_{hi,d} - T_{sat,d})$. As T_{hi} increases, the average temperature difference between the heating fluid and the saturation temperature T_{sat} increases and produces a greater pressure drop and a more positive $\partial\varphi/\partial W_w$ at $W_w = 1.0$.

The third factor which can change the heat flux and hence the pressure-drop characteristic is the variation of heat-transfer coefficients. Figure 11(c) shows the effect of variation in the boiling coefficient. Note the curious effect that better heat-transfer produces a less desirable characteristic as far as potential maldistribution is concerned. In contrast, for a U_b of one-fifth of design ($U_b = 100$), the rate of vaporization is so slow that no peak in the pressure-drop function occurs. Such a boiler, however, would generate vapor without superheat at a quality of 60 percent at design liquid flow rate. Thus, decreasing the heat-transfer coefficient does not appear to be a good method of eliminating the undesirable pressure-drop characteristic.

A question arises at this point as to why increasing the heat flux by increasing heating-fluid flow and inlet temperature helps to reduce the negative-slope region in the pressure-drop curve, while the increase in heat flux caused by an increased boiling coefficient produces the opposite effect. The basic difference is apparent in the equation of the boiling length (eq. (14)) as expressed in terms of the primary variables:

$$L_b = \left(\frac{w_h c_h}{U_b \pi D_o l_t}\right) \ln\left[\frac{1}{1 - \dfrac{w_w H_w X_o}{w_h c_h (T_{hs} - T_{sat})}}\right]$$

Increasing U_b shortens the boiling length but does not affect the effectiveness term inside the parentheses. In contrast, increasing heating-fluid flow or temperature decreases the effectiveness. (Note that $T_{hs} \cong T_{hi}$.) In the other direction, as the effectiveness approaches 1, the boiling length becomes large and very sensitive to working-fluid flow. This situation leads to a very rapid expansion of the boiling length and a corresponding decrease of the superheat length as the working-fluid flow increases. It is this effect that is responsible for the negative slope of the pressure-drop function of a tube without the plug insert.

The three variables W_h, T_{hi}, and U_b are, of course, not the only ones that influence the pressure-drop function through changes in heat flux. They are, however, the major ones. The subcooled-working-fluid inlet temperature T_{wi} has little effect on the pressure-drop function, both for the water boiler (see fig. 6(c)) and the mercury boiler (not shown here but similar). This condition is primarily caused by the high N_p values (see table II) which make the preheat length insensitive to changes in the inlet temperature (see eq. (10)). For smaller N_p values (e.g., for low values of U_p), the pressure drop would become more sensitive to the amount of subcooling. This situation is not typical, however, in liquid-metal space power boilers, particularly those that have plug inserts to improve the heat transfer.

CONCLUDING REMARKS

A theoretical boiler model was developed and verified by comparison with experimental data. An analysis based on this model has led to a number of conclusions regarding (1) the factors involved in producing the multivalued function of pressure drop against working-fluid flow for a boiler tube and (2) the possible approaches to boiler design that would make the multivalued function single valued and thus eliminate this potential for flow maldistribution in a tube bundle. These design approaches are directed toward eliminating the negative-slope region of the function (i.e., the region where pressure drop decreases with increasing flow).

The analysis has indicated that this negative-slope region is the result of a combination of performance factors. In a tube with a plug insert, the primary factor is the rapid decrease in the boiling pressure drop in the plug region as the working-fluid flow increases. Another factor, which is present in a boiling tube even without a plug insert, is the decline of the superheat length and pressure drop as the boiling region expands with increasing flow rate. Still another factor is the variation of vapor density with changes in boiler pressure level, where the pressure level is determined by the boiler-exit boundary conditions. The boundary condition for a single tube operating in a large tube bundle is that exit pressure be nearly independent of the flow in the individual tube (i.e., constant pressure). For this boundary condition, the pressure drop in the single tube decreases less with increased flow than for a boundary condition where exit pressure increases directly with flow (i.e., choked nozzle).

The analysis investigated the effect on the negative-slope region of the variation in some of the geometric and thermodynamic parameters of the boiler. These parameters show varying degrees of effectiveness in eliminating the negative-slope region. Minimizing the plug length or its resistance reduces the peak in the pressure-drop function as well as the steepness of the negative-slope region. Such a reduction of the plug is limited, however, by heat-transfer considerations. Increasing the heating-fluid flow or inlet temperature, and thereby the average temperature difference, also helps to make the slope of pressure drop against working-fluid flow less negative. In contrast, increasing the heat-transfer coefficient in the boiling region increases the maximum in the pressure-drop function and also increases the negative slope. Adding an orifice at the entrance of each tube is a simple method of assuring that pressure drop always increases with increasing flow. The amount of orificing required increases with the steepness of the negative slope.

All the proposed methods of improving the pressure-drop characteristic, except the plug modification, require increased pumping power or higher heating temperatures and thus result in a system performance penalty. The reduction of the plug length or plug-flow resistance to the minimum consistent with good heat-transfer performance appears to be the best method of improvement. Nevertheless, even with the plug completely removed, a negative slope in the pressure-drop function may still exist and hence require an orifice to eliminate the potential for flow maldistribution.

Lewis Research Center,
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APPENDIX A

SYMBOLS

Mathematical symbol	FORTRAN variable	Description
A	AGAS	constant in eq. (36) adjusted to fit saturation-temperature-against-pressure curve for working-fluid cross-sectional area of boiler tube
A_c		cross-sectional area of boiler tube
A_{pl}		cross-sectional area of plug flow passage (perpendicular to flow), ft^2; m^2
A_s		cross-sectional area of tube in superheat region, ft^2; m^2
a_p	APH	ratio of working-fluid to heating-fluid heat capacity in preheat region, $w_w c_{w,l}/w_h c_h$
a_s	ASH	ratio of working-fluid to heating-fluid heat capacity in superheated region, $w_w c_{w,v}/w_h c_h$
B	BGAS	constant in eq. (36) adjusted to fit saturation-temperature-against-pressure curve for working fluid
c_h	CH	specific heat of heating fluid, Btu/(lb)(^{o}F); J/(kg)(^{o}K)
$c_{w,l}$	CPP	specific heat of working-fluid liquid, Btu/(lb)(^{o}F); J/(kg)(^{o}K)
$c_{w,v}$	CPV	specific heat of working-fluid vapor, Btu/(lb)(^{o}F); J/(kg)(^{o}K)
D		hydraulic diameter, ft; m
D_i		inside diameter of boiling tube, ft; m
D_o	DBL, DPH	outside diameter of boiling tube used for heat-transfer calculations, ft; m
f		friction factor
$f_{pl,b}$		friction factors in plug boiling region
$f_{b,s}$		friction factors in swirl-wire region

Mathematical symbol	FORTRAN variable	Description
f_p, f_b, f_s		friction factors in preheat, boiling, and superheat regions
$f_{pl,p}$		friction factors in plug preheat region
$f_{pl,v}$		friction factors in plug for all vapor flow
G_p, G_b, G_t	RPHDP, RBLDP, and RTDP	dimensionless equivalent length factors for calculating the preheat, boiling and, total pressure drops
g		acceleration of gravity, ft/sec^2; m/sec^2
g_p, g_b, g_s		geometry factors in pressure-drop equations of preheat, boiling, and superheat regions, respectively
H_w	HV	heat of vaporization of working fluid, Btu/lb; J/kg
$h_{w,s}$		heat-transfer coefficient of working fluid in superheat region
K_{tn}	TNK	choked turbine nozzle constant at boiler exit, $P_{od}/w_{w,d}X_o\sqrt{T_{wo,d}}$
l		generalized length along tube, ft; m
l_b, L_b	EBOIL	boiling length, ft; m (dimensionless, l_b/l_t)
l_p, L_p	ELPH	preheat length, ft; m (dimensionless, l_p/l_t)
l_{pl}, L_{pl}	EPLUG	plug length, ft; m (dimensionless, l_{pl}/l_t)
l_s, L_s	ELSH	superheat length, ft; m (dimensionless, l_s/l_t)
l_t, L_t	ALT	total boiler tube length, ft; m
N		number of 90^o turns for spiral plug
NTU		number of transfer units
N_b	BNTU	NTU of boiler based on U_b
N_p	PHNTU	NTU of boiler based on U_p
N_{pl}	PLNTU	NTU based on boiling region in plug, $N_b(L_{pl} - L_p)$
N_s		NTU of boiler based on U_s
n		number of turns per inch
P		pressure, psia; N/m^2 abs

Mathematical symbol	FORTRAN variable	Description
P_i	PIN	working-fluid inlet pressure, psia; N/m^2 abs
P_o	POUT	working-fluid outlet pressure, psia; N/m^2 abs
$P_{o,d}$		working-fluid outlet pressure (design value), psia; N/m^2 abs
P_{sat}	PSAT	working-fluid saturation pressure, psia; N/m^2 abs
$P_{sat,d}$		working-fluid saturation pressure (design value), psia; N/m^2 abs
	PSATO	working-fluid saturation pressure used for initial estimate in iteration, psia; N/m^2 abs
$\Delta P_p, \Delta P_b, \Delta P_s$		pressure drops in preheat, boiling, and superheat regions, psi; N/m^2
ΔP_t	DP	total boiler-pressure drop, psi; N/m^2
$\Delta P_{v,d}$	AVPRS	normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, psi; N/m^2
ΔP_i		orifice pressure drop at tube inlet at design flow, psi; N/m^2
p		wetted perimeter, ft; m
p_{pl}		wetted perimeter in plug flow passage, ft; m
p_s		wetted perimeter in superheat open-tube region, ft; m
R_d	ROW	dimensionless vapor density, $\rho_{w,v}/\rho_r$
Re_s		Reynolds number in superheat region
Re_{pl}		Reynolds number in plug region (liquid)
R_g	RGAS	working-fluid gas constant
$r_{b,s}$	RBSP	ratio of boiling to superheat friction factors
r_i	PDI	ratio of orifice pressure drop to $\Delta P_{v,d}$
$r_{p,s}$	RPH	ratio of preheat-plug-region pressure gradient to superheat pressure gradient

Mathematical symbol	FORTRAN variable	Description
$r_{s,j}$	RPS	ratio of pressure gradients for vapor at rated flow in plug region to those of vapor at rated flow in swirl-wire region
T_{hb}		heating-fluid temperature at point where boiling starts, ^{o}F; ^{o}K
T_{hi}	THL	heating-fluid inlet temperature, ^{o}F; ^{o}K
$T_{hi,d}$		heating-fluid inlet temperature (design value), ^{o}F; ^{o}K
T_{ho}		heating-fluid outlet temperature, ^{o}F; ^{o}K
T_{hs}	THLSH	heating-fluid temperature at point where superheating starts, ^{o}F; ^{o}K
T_{sat}	TSAT	working-fluid saturation temperature
$T_{sat,d}$		working-fluid saturation temperature (design value)
T_{wi}	TPO	working-fluid inlet temperature, ^{o}F; ^{o}K
$T_{wi,d}$		working-fluid inlet temperature (design value), ^{o}F; ^{o}K
T_{wo}	TPL	working-fluid outlet temperature, ^{o}F; ^{o}K
T_{ws}		working-fluid saturation temperature at start of superheating, ^{o}F; ^{o}K
U_b	UBL	overall average heat-transfer coefficient in boiling region, Btu/(ft^2)(hr)(^{o}F); J/(m^2)(hr)(^{o}K)
$U_{h,d}$		overall average heat-transfer coefficient in boiling region (design value), Btu/(ft^2)(hr)(^{o}F); J/(m^2)(hr)(^{o}K)
U_p	UPH	overall average heat-transfer coefficient in preheat region, Btu/(ft^2)(hr)(^{o}F); J/(m^2)(hr)(^{o}K)
U_s	USH	overall average heat-transfer coefficient in superheat region
V		velocity of working-fluid flow, ft/hr; m/hr
w_h, W_h	WH	heating-fluid flow (dimensional, lb/hr (kg/hr); dimensionless, $W_h = w_h/w_{h,d}$)

Mathematical symbol	FORTRAN variable	Description
$w_{h,d}, W_{h,d}$	WHDG	heating-fluid design flow (dimensional, lb/hr (kg/hr); dimensionless)
$w_v(y)$		local vapor flow along boiling length y, lb/hr; kg/hr
w_w, W_w	WP	working-fluid flow (dimensional, lb/hr (kg/hr); dimensionless), $W_w = w_w/w_{w,d}$
$w_{w,d}, W_{w,d}$	WPDG	working-fluid design flow (dimensional, lb/hr (kg/hr); dimensionless)
X_o	QUAL	outlet quality at end of boiling region
x		local quality along boiling length y, lb/hr; kg/hr
y		length along boiling region, ft; m
$\partial(\Delta P)/\partial w_w$		slope of boiler-pressure-drop-against-working-fluid-flow function
$\partial\varphi_t/\partial W_w$		slope of dimensionless pressure-drop function
ϵ_b	BBOIL	effectiveness of boiling region
ϵ_p	BPHT	effectiveness of preheat region
ϵ_s		effectiveness of superheat region
μ		viscosity
ρ		density, lb/ft^3; kg/m^3
ρ_r	ROWRE	working-fluid vapor density at design saturation pressure used to normalize eq. (39), lb/ft^3; kg/m^3
$\rho_{v,d}$		working-fluid vapor density at design flow in superheat region, lb/ft^3; kg/m^3
$\rho_{w,l}$	ROWL	working-fluid liquid density, lb/ft^3; kg/m^3
$\rho_{w,v}$		working-fluid vapor density, lb/ft^3; kg/m
φ_t	RD	boiler dimensionless pressure-drop function, $\Delta P_t/\Delta P_{v,d}$

APPENDIX B

DIGITAL PROGRAM FOR BOILER MODEL WITH CHOKED-NOZZLE DISCHARGE AND CALCULATION OF WATER-BOILER CONSTANTS

Computer Program

The equations of the analytical model were programmed on a digital computer and solved simultaneously by iteration. (See program listing at the end of this appendix.) The iteration consists simply of adjusting the saturation pressure until the boiler-outlet pressure corresponds to the working-fluid flow under the conditions specified by the choked-nozzle boundary condition. A secondary iteration loop is provided to converge on outlet quality for the cases where the quality at the boiler exit is less than 1. The main body of the program is applicable to any counterflow boiler. However, numerical values and data statements in the listing apply specifically to the water boiler. The dimensionless parameters and the flows, pressures, and temperatures at design conditions required for the evaluation of the boiler equations are listed in table I. The method of calculating the pressure-drop parameters ($P_{v,d}$, $r_{p,s}$, $r_{s,\ell}$ and $r_{b,s}$) is shown in a later part of this appendix. The other parameters in the table can be calculated directly from the fluid properties and the specified design values of the input variables. The heat-transfer coefficients for evaluating N_p, N_b, and N_s were calculated by using the Dittus-Bolter equation for forced convection. The convective coefficient on the working-fluid side was based on the hydraulic diameter. The coefficient in the plug preheat region was multiplied by the factor $(f_{pl,p}/f_{smooth\ tube})^{0.28} = 1.19$, as developed from reference 12, to account for the improvement resulting from spiral flow. The convective coefficient in the superheat region was multiplied by a similar factor, $(h_{w,s}/h_{smooth\ tube}) \cong 2.0$, as obtained from figure 10 of reference 12 for a wire-coil-pitch to coil-diameter ratio of 5. The average boiling coefficient, based on previous experience with a water boiler, was known to be very high in comparison with the heating-fluid coefficient and the wall conductivity. For purposes of calculation, this boiling coefficient was assumed to be 6000 Btu per hour per square foot per ^{o}F (34 000 J/(sec)(m^2)(^{o}K)).

Calculation of Pressure-Drop Parameters for Single-Tube Water Boiler

The fluid properties used for the calculations were taken at the design pressures and temperatures and were assumed to be constant.

Calculation of $\Delta P_{v,d}$. - The normalizing factor for the pressure-drop function is defined as

$$\Delta P_{v,d} = f_s\left(\frac{p_s}{8gA_s^3}\right)\frac{w_{w,d}^2}{\rho_{v,d}} l_t$$

which represents the pressure gradient in the superheat region at design flow multiplied by the total tube length. The friction factor f_s is based on the flow geometry of the superheat region, which includes not only the spiral wire attached to the inside tube wall, but also, in the center of the tube, a 0.25-inch- (0.625-cm-) diameter rod used to support the internal instrumentation.

The flow cross-sectional area is

$$A_s = \frac{\pi}{4}\left(D_i^2 - D_{rod}^2\right) = \frac{\pi}{4}\left[(0.0462)^2 - (0.208)^2\right] = 0.1944 \text{ in.}^2 \ (1.25 \text{ cm}^2)$$

The wetted perimeter is

$$p_s = \pi\left(D_i + D_{rod}\right) = \pi(0.0462 + 0.0208) = 0.21 \text{ ft } (0.064 \text{ m})$$

The Reynolds number for the annular flow passage at design vapor flow is

$$Re_s = \frac{4w_{w,d}}{p_s \mu} = \frac{4 \times 45}{0.210 \times 0.0364} = 23\ 600$$

The smooth-tube friction factor at $Re = 23\ 600$ is approximately 0.025. The friction-factor multiplier to account for the swirl wire is based on the data of reference 12. Figure 15 of this reference gives the friction-factor multiplier as a function of wire-coil-pitch to wire-diameter ratio s/D for various values of wire diameter. These data were extrapolated to the value of $D_{wire}/D_{tube} = 0.12$ at $s/D = 5$ of the water-boiler tube, which gave a value of $f_s/f_{smooth\ tube} \cong 17.0$. Now $\Delta P_{v,d}$ can be calculated.

$$\Delta P_{v,d} = f_s\left(\frac{p_s}{8gA_s^3}\right)\frac{w_{w,d}^2 l_t}{\rho_{v,d}}$$

$$\Delta P_{v,d} = \frac{1}{144\ \frac{\text{lb/ft}^2}{\text{psi}}} \left\{ 17.0(0.025)\left[-\frac{0.21}{8(32.2)(0.00135)^3} \right] \frac{(1.56\times10^{-4})10.0}{0.155} \right\} = 9.7\ \text{psi}\ (66\,930\ \text{N/m}^2)$$

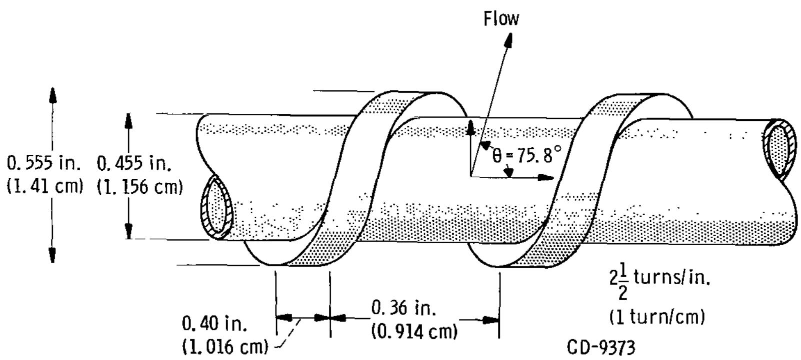
Calculation of $r_{p,s}$. - The definition of $r_{p,s}$ is

$$r_{p,s} = \frac{d_{pl}}{dl} \frac{f_p}{f_s} \frac{\rho_{v,d}}{\rho_{w,l}} \frac{p_{pl}}{p_s} \left(\frac{A_s}{A_{pl}} \right)^3$$

The values of f_p, A_{pl} and p_{pl} were computed on the basis of flow along the spiral path. The factor dl_{pl}/dl modifies the pressure gradient to correspond to the axial tube length of 1.

$$\frac{dl_{pl}}{dl} = \frac{1}{\cos\theta} = \frac{1}{\cos 75.80} = 4.09$$

The calculations are based on the geometry shown in the following sketch:



The wetted perimeter of the area normal to flow is

$$p_{pl} = 2\left[(\sin\theta)\ 0.030 + 0.00417\right] = 0.0664\ \text{ft}\ (0.02024\ \text{m})$$

The cross-sectional area normal to flow is

$$A_{pl} = \sin 75.8 \times 0.00417 \times 0.030 = 0.000121\ \text{ft}^2\ (10.89\times10^{-6}\ \text{m}^2)$$

The Reynolds number for design water flow is

$$Re_{l,pl} = \frac{4w_{w,d}}{p_{pl}\mu} = \frac{4.45}{0.0664(0.635)} = 4270$$

The smooth-tube friction factor at an Re of 4270, with the assumption of turbulent flow, was approximately 0.038. The friction-factor multiplier which accounts for the bends in the spiral flow passage was calculated by use of a simplified form of the relation for equivalent length-diameter ratio for bends greater than 90^{o} (ref. 13). (Symbols are the same as used in ref. 13.)

$$\frac{1}{D} = R_t + (N - 1)\left(R_1 + \frac{R_b}{2}\right)$$

where D is the hydraulic diameter, R_l is the resistance due to the length of one 90^{o} bend, R_b is the bend resistance due to one 90^{o} bend, the total resistance $R_t = R_l + R_b$, and N is the total number of 90^{o} bends in a coil. The resistance due to the bending of flow can be extracted from the preceding expression.

$$\frac{l_{eq}}{D} = (N + 1)\frac{R_b}{2}$$

The number of 90^{o} turns N for the spiral plug is

$$N = 4nl_{pl} \cong 258$$

where N is the number of turns per inch (per cm). When this number is large enough so that $(N + 1) \cong N$, the following expression for equivalent length in terms of plug length l_{pl} due only to the bending of flow can be written:

$$l_{eq} \cong \frac{4nl_{pl}R_bD}{2}$$

The total equivalent length of the plug can now be expressed as

$$l_{pt} = 4.09\, l_{pl} + \frac{4nl_{pl}R_bD}{2}$$

$$= 4.09\, l_{pl}\left(1 + \frac{4nR_bD}{4.09 \times 2}\right)$$

Given the hydraulic diameter $D = 4A_c/p_s = 0.0073$ feet (0.00226 m) and $R_b = 8$ for $r/D = 2.88$, the equivalent length becomes

$$l_{pt} = 4.09\, l_{pl} \left(1 + \frac{120.8 \times 0.0073}{8.18}\right) = (4.09 \times 1.86) l_{pl}$$

All the numbers required for calculating $r_{p,s}$ are now known:

$$r_{p,s} = \frac{\partial l_{pl}}{\partial l} \frac{f_p}{f_s} \frac{\rho_{v,d}}{\rho_l} \frac{p_p}{p_s} \left(\frac{A_s}{a_{pl}}\right)^3 = 4.09 \frac{0.038 \times 1.86}{0.025 \times 17} \frac{0.155}{59.4} \frac{0.0664}{0.21} \left(\frac{0.00135}{0.000121}\right)^3 = 0.78$$

<u>Calculation of $r_{s,\partial}$ parameter.</u> - The definition of $r_{s,\partial}$ parameter is

$$r_{s,\partial} = \frac{f_{pl,b}}{f_{b,s}} \frac{dl_{pl}}{dl} \frac{p_p}{p_s} \left(\frac{A_s}{A_{pl}}\right)^3$$

The friction factor $f_{pl,b}$ for boiling in the plug is assumed to correspond to a single-phase gas Reynolds number of approximately 100 000 at design flow. The friction-factor multiplier is the same used for f_p in the previous section (i.e., 1.86). The friction factor in the swirl-wire region was assumed to be equal to f_s. All the other parameters are already known; hence, the value of $r_{s,\partial}$ can be calculated:

$$r_{s,\partial} = \frac{0.019 \times 1.86}{0.025 \times 17}\, 4.09\, \frac{0.0664}{0.21} \left(\frac{0.00135}{0.000121}\right)^3 = 150.0$$

The $r_{b,s}$ parameter defined as $f_{b,\partial}/f_s$ is 1 since $f_{b,\partial}$ was assumed to be equal to f_s.

PROGRAM LISTING

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$IBFTC BLCN    LIST
C     THIS IS A FORTRAN IV PROGRAM FOR COMPUTING THE PRESSURE DROP
C     OF A SINGLE-PASS COUNTERFLOW BOILER.
C     THE EXIT BOUNDARY CONDITION IS A CHOKED NOZZLE
C     THE PROGRAM IS GENERAL EXCEPT FOR THE INITIALIZATION SECTION AND
C     THE DATA CARDS.  THE NUMERICAL CONSTANTS ARE SPECIFIED IN ENGLISH
C     UNITS , ALTHOUGH ANY CONSISTENT UNITS CAN BE EMPLOYED.
C     (ENGLISH UNITS ARE IN FT  HR  LB  BTU  PSI  DEG-F ) .
C     THE NUMERICAL VALUES ARE FOR THE SINGLE TUBE WATER BOILER
C     THE DATA - AWP - SPECIFIES VALUES  OF WORKING FLUID FLOW (IN UNITS
C     OF FRACTION OF DESIGN FLOW ) FOR WHICH THE PRESSURE DROP FUNCTION
C     IS TO BE EVALUATED.
      DIMENSION  AWP(25)
      DATA AWP/1.0,0.02,.06,0.1,0.2,0.3,0.4,0.5,0.6,0.7,0.8,0.9,1.1,
     11.2,1.3,1.4,1.6,1.8,2.0,0.95,1.05,1.15,1.25,1.35,1.5  /
30    FORMAT(2I12,1F12.7,I12 )
31    FORMAT(6F12.7)
32    FORMAT (49H     WPO          PSATO         WH          TP-        ,
     125H     THL          PO         / 6F12.7  )
33    FORMAT(20H1INPUT PARAMETERS     ,6HICASE=,I5/7H KCMAX=,I5,
     1 6H DK=   ,E12.5,                  6H KPR=   ,I5     )
34    FORMAT(1H ,9(E12.5))
35    FORMAT(50H1 WH/FNBL    WP/FNPH     RPH/FNSH     RPS/RBSP    ,
     160H UPH/TNK   USH/AVPRS  UBL/A1        ALT/A2       EPLUG/A3      )
36    FORMAT (1H ,7(E12.5))
37    FORMAT (50H    APH           ASH          PHNTU      BNTU          ,
     160H SHNTU       PLNTU       ROW                                   )
  106 FORMAT (50H   WP/PIN     QUAL /WH       PSAT/TPO       POUT/THL   ,
     160H    DP/ELPH   TPL/ELSH   TSAT/EBOIL DTSH/EBPL   RTDP/BBOIL    )
  107 FORMAT (1H , 9(E12.5))
  108 FORMAT (1H , 9(E12.5))
C   BEGIN INITIALIZATION FOR WATER BOILER
C   FLUID PROPERTIES - WATER  BOILER DESIGN VALUES
      CPP = 1.0
      CPV = 0.474
      CH = 1.0
      HV = 875.
      USH = 108.
      UPH = 630.
      UBL = 880.
      WPDG = 45.
      WHDG = 770.
      AGAS = 8520.0
      BGAS = 15.426
      RGAS =0.596
      TABS = 460.
      ROWRE = 0.190
      ROWL = 59.4
C   PRESSURE DROP PARAMETERS
      AVPRS = 9.7
      RPH = 0.78
      RPS = 150.
      RBSP = 1.0
      RPHOP = ROWRE*RPS/ROWL
C  ITERATION PARAMETERS
      KCMAX = 500
      DK = 1.0
      ERMIN = 0.001
C   BOUNDARY CONDITIONS AT INLET   - WP SPECIFIED
C   BOUNDARY CONDITION AT OUTLET   -  CHOKED NOZZLE
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      ICASE = 0
      TNK = 0.455
C BOILER GEOMETRY
      ALT = 10.
      EPLUG = 0.215
      DBL = 0.049
      DPH = 0.049
      TUBNU = 1.0
C  END INITIALIZATION FOR WATER BOILER
C  BEGIN GENERALIZED INITIALIZATIONS
C INTERMEDIATE CONSTANTS - NTU VALUES AND RATIOS OF HEAT CAPACITY FLOWS
      A1 = WPDG*CPV/(CH*WHDG)
      A2 = (HV*WPDG)/(CH*WHDG)
      A3 = (CPP*WPDG)/(WHDG*CH)
      FNBL = 3.1416*TUBNU*UBL*DBL*ALT/(CH*WHDG)
      FNPH =3.1416*TUBNU*UPH*DPH*ALT/(CPP*WPDG)
      FNPL = FNBL
      FNSH = FNBL*USH/(A1*UBL)
C  END GENERALIZED INITIALIZATIONS
C  PREPARE FOR ITERATIONS AND READ INPUT VARIABLES
1     CONTINUE
      READ(5,30) ICASE,KCMAX,DK,KPR
      WRITE(6,33) ICASE, KCMAX,DK,KPR
      READ (5,31) WPO,PSATO,WH,TPO,THL,PO
      WRITE(6,32)WPO,PSATO,WH,TPO,THL,PO
      WP =WPO/WPDG
      WH =WH/WHDG
      PSAT=PSATO
      ASH =0.0
      DTSH =0.0
      PIN =PSATO
      WRITE (6,35)
      WRITE (6,34) WH,WP,RPH,RPS,UPH,USH,UBL,ALT,EPLUG
      WRITE (6,34) FNBL,FNPH,FNSH,RBSP,TNK,AVPRS,A1,A2,A3
C  MAIN BODY OF PROGRAM - VARY WORKING FLUID FLOW
      DO 54 KRW=1,25
      WP = AWP(KRW)
C  INITIALIZE FOR ITERATION
      KPRC= KPR
      KCNTR =0
      QUAL = 1.0
      DTSH = 0.0
      PSAT = PSATO * WP
      PIN = PSAT
C  BEGINNING OF ITERATION LOOP TO ADJUST OUTLET PRESSURE TO CORRESPOND
C  TO SPECIFIED FLOW.
   10 CONTINUE
C  COMPUTE HEAT TRANSFER PARAMETERS
      ASH = WP*A1/WH
      BNTU = FNBL/WH
      PHNTU = FNPH/WP
      APH = A3*WP/WH
122   CONTINUE
      TSAT=AGAS/(BGAS-ALOG(PSAT)) - TABS
      THLSH = THL -      ASH*DTSH
      BBOIL =(A2*WP/(WH*(THLSH-TSAT)) )*QUAL
C  TEST WHETHER AVAILABLE POWER IS ENOUGH TO VAPORIZE LIQUID.
      IF(BBOIL-0.99)11,12,12
   12 CONTINUE
C HEATING ENERGY LOW, REDUCE QUALITY AT OUTLET
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      DTSH = 0.0                                                        BLCN1200
      QUAL = QUAL - 0.001                                               BLCN1210
      PSAT = 0.999*PSAT                                                 BLCN1220
      GO TO 122                                                         BLCN1230
C  HEATING ENERGY SUFFICIENT                                            BLCN1240
C     COMPUTE LENGTHS OF REGIONS AND NTU OF PLUG                        BLCN1250
   11 CONTINUE                                                          BLCN1260
      BPHT =(TSAT-TPO)/(THLSH-(A2*WP/WH)*QUAL - TPO )                   BLCN1270
      ELPH = ALOG(ABS((1.0-APH)/(1.0-BPHT)))/(PHNTU*( 1.0-APH))         BLCN1280
      IF (QUAL -1.0) 112,113,112                                        BLCN1290
112   CONTINUE                                                          BLCN1300
      ELSH = -0.00001                                                   BLCN1310
      EBOIL = 1.0 - ELPH                                                BLCN1320
      GO TO 111                                                         BLCN1330
113   CONTINUE                                                          BLCN1340
      RLBL = ALOG(1.0/(1.0-BBOIL))                                      BLCN1350
      EBOIL = RLBL/BNTU                                                 BLCN1360
      ELSH = 1.0-ELPH-EBOIL                                             BLCN1370
111   CONTINUE                                                          BLCN1380
      EBPL =EPLUG -ELPH                                                 BLCN1390
C  PRINT VARIABLES EVERY KPR ITERATIONS                                 BLCN1400
      IF(KPRC)41,41,42                                                  BLCN1410
41    KPRC=KPR                                                          BLCN1420
      KSPR= 0                                                           BLCN1430
      WRITE(6,106)                                                      BLCN1440
      WRITE(6,107) WP,QUAL ,PSAT,POUT,DP,TPL,TSAT,RBLD,RTDP             BLCN1450
      WRITE(6,108)PIN,WH,TPO,THL,ELPH,ELSH,EBOIL,EBPL,BBOIL             BLCN1460
42    KPRC=KPRC-1                                                       BLCN1470
C TEST WHETHER PREHEAT REGION EXTENDS OUTSIDE PLUG                      BLCN1480
      IF(EBPL)13,13,14                                                  BLCN1490
   13 CONTINUE                                                          BLCN1500
C  PREHEAT REGION EXTENDS BEYOND PLUG - SET BOILING AND SUPERHEAT PLUG  BLCN1510
C  PARAMETERS TO ZERO                                                   BLCN1520
      PLNTU = 0.0                                                       BLCN1530
      ESHP = 0.0                                                        BLCN1540
      GO TO 15                                                          BLCN1550
   14 CONTINUE                                                          BLCN1560
      ESHP = EBPL - EBOIL                                               BLCN1570
C  TEST FOR SUPERHEAT LENGTH, ESHP IN PLUG                              BLCN1580
      IF(ESHP) 114,114,115                                              BLCN1590
115   CONTINUE                                                          BLCN1600
C  BOILING REGION ENDS BEFORE END OF PLUG - CALCULATE PLUG NTU          BLCN1610
      EBPL = EBOIL                                                      BLCN1620
      PLNTU = FNPL*EBPL/WH                                              BLCN1630
      GO TO 15                                                          BLCN1640
114   CONTINUE                                                          BLCN1650
C  NO SUPERHEAT IN PLUG - CALCUALATE REGULAR BOILING PLUG NTU.          BLCN1660
      ESHP = 0.0                                                        BLCN1670
      PLNTU = FNPL*EBPL/WH                                              BLCN1680
   15 CONTINUE                                                          BLCN1690
C  TEST FOR OPERATION WITH EXIT QUALITY LESS THAN ONE                   BLCN1700
C  ELSH LESS THAN ZERO -ADJUST EXIT QUALITY. ELSH = ZERO - EXIT QUALITY BLCN1710
C  HAS BEEN ADJUSTED, CONTINUE NORMAL CALCULATION. ELSH GREATER THAN    BLCN1720
C  ZERO - SOME SUPERHEAT , CONTINUE NORMAL CALCULATION                  BLCN1730
      IF(ELSH)17,16,16                                                  BLCN1740
   17 CONTINUE                                                          BLCN1750
C  BOILER OPERATION WITH QUALITY LESS THAN ONE                          BLCN1760
      ELSH = 0.0                                                        BLCN1770
      ASH = 0.0                                                         BLCN1780
      KQUAL = 100                                                       BLCN1790
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171    CONTINUE
C  ITERATE ON QUAL FOR A MAXIMUM OF 100 TIMES UNTIL BOILING LENGTH
C  MATCHES END OF TUBE
       EBOIL = 1.0 - ELPH
       QBBL = 1.0 - EXP(0.0- EBOIL*BNTU)
C  TEST FOR CONVERGENCE
       IF (ABS(QBBL-BBOIL)-0.001   ) 111,111,28
28     CONTINUE
       QUAL = QUAL*QBBL/BBOIL
       KQUAL = KQUAL - 1
       IF (KQUAL) 22,22,172
172    CONTINUE
       IF (QUAL - 1.0) 174,174,173
 173   CONTINUE
       QUAL = 1.0
       GO TO 122
 174   CONTINUE
       THLSH = THL
       BBOIL =(A2*WP/(WH*(THLSH-TSAT)) )*QUAL
       BPHT =(TSAT-TPO)/(THLSH-(A2*WP/WH)*QUAL - TPO )
       ELPH = ALOG(ABS((1.0-APH)/(1.0-BPHT)))/(PHNTU*( 1.0-APH))
       GO TO 171
   16  CONTINUE
C  OPERATION WITH SUPERHEAT - NORMAL CALCULATION
       SHNTU = FNSH*ELSH /WP
       DTSH = (THL-TSAT)*(1.0-(1.0-ASH)*EXP(SHNTU*(ASH-1.0)))
       TPL = TSAT +DTSH
       GO TO 18
C COMPUTE PRESSURE DROPS
   18  CONTINUE
       ROW =PSAT/(ROWRE*RGAS*(TSAT+TABS ))
       PVT = WP*WP/ROW
       RLBL = ALOG(1.0/(1.0-BBOIL))
C COMPUTE RATIOS OF PRESSURE DROP FOR EACH BOILER SECTION
       RSHDP = 1.0-ELPH-EBOIL  +ESHP*(RPS-1.0)
       PPHD = ELPH - EPLUG
C  TEST FOR PREHEAT LENGTH OUTSIDE PLUG
       IF (PPHD) 60,60,61
60     CONTINUE
C  PREHEAT REGION INSIDE PLUG
       RPHDP = RPH*ELPH  *ROW
       GO TO 62
C  PREHAT REGION OUTSIDE PLUG
61     RPHDP = ROW*(RPH*EPLUG+RPHOP*PPHD)
62     CONTINUE
       RB =(1.0-BBOIL)/BBOIL
       RPLDP =(RBSP*RB*(RPS-1.0)*(EXP( PLNTU)-PLNTU-1.0)/BNTU )*QUAL
       RSPDP =(RBSP*(1.0-RB*RLBL)/BNTU)*QUAL
       RTDP = RPHDP + RPLDP +RSPDP+RSHDP
       DP = PVT*RTDP*AVPRS
       RD = RTDP*PVT
       RBLDP = RPLDP+RSPDP
C  COMPUTE INLET AND OUTLET PRESSURES BASED ON CALCULATED PRESSURE DROP
       PIN = PSAT +RPHDP*PVT*AVPRS
       POUT = PIN -DP
C  WRITE INTERMEDIATE RESULTS EVERY KPR ITERATIONS
       IF(KSPR) 43,43,44
43     KSPR = 1
  102 FORMAT (50H  PRESOUT     PRESDROP   PR VAP TOT   PDROP   PHEAT   ,
     150H B RRSPLG  PRDR BSP   PRDRBOIL    PRDRSHT                   )
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  103 FORMAT (1H ,8(E12.5))                                                BLCN2400
      WRITE (6,102)                                                        BLCN2410
      WRITE (6,103) POUT,DP,PVT,RPHDP,RPLDP,RSPDP,RBLDP,RSHDP              BLCN2420
 44   CONTINUE                                                             BLCN2430
C COMPUTE BOILER FLOW BASED ON CHOKED NOZZLE AND COMPARE TO                 BLCN2440
C SPECIFIED FLOW WP                                                        BLCN2450
       WPPR = TNK*POUT/SQRT(TPL +TABS )                                    BLCN2460
      WPERR = WP*QUAL - WPPR                                               BLCN2470
C  TEST FOR CONVERGENCE ERROR IN FLOW LESS THAN ERMIN                      BLCN2480
      IF (ABS(WPERR)-ERMIN ) 20,20,19                                      BLCN2490
   19 CONTINUE                                                             BLCN2500
C  ERROR STILL UNACCEPTABLE - CONTINUE ITERATION IF KCMAX NOT EXCEEDED     BLCN2510
      KCNTR = KCNTR +1                                                     BLCN2520
      IF(KCNTR-KCMAX) 21,21,22                                             BLCN2530
   21 CONTINUE                                                             BLCN2540
C CASE I FLOW WP IS SPECIFIED, ITERATE ON PSAT    ,ICASE =0                BLCN2550
      PSAT = PSAT+DK*WPERR                                                 BLCN2560
      PIN = PSAT +RPHDP*PVT*AVPRS                                          BLCN2570
      GO TO 10                                                             BLCN2580
   22 CONTINUE                                                             BLCN2590
C  NO CONVERGENCE - KCMAX EXCEEDED OR KQUAL = 0                            BLCN2600
  104 FORMAT (52H   NO CONVERGENCE    -KCMAX EXCEEDED                     )BLCN2610
      WRITE (6,104)                                                        BLCN2620
      GO TO 20                                                             BLCN2630
C  NORMAL OUTPUT FOR SUCCESSFUL CONVERGENCE                                BLCN2640
   20 CONTINUE                                                             BLCN2650
  105 FORMAT (7H0 AFTER ,I4,23H ITERATIONS, FOR CASE =,I4,                 BLCN2660
     150H THE FOLLOWING BOILER VARIABLES HAVE RESULTED             )       BLCN2670
      WRITE(6,105) KCNTR,ICASE                                             BLCN2680
      WRITE  (6,37)                                                        BLCN2690
      WRITE  (6,36) APH,ASH,PHNTU,BNTU,SHNTU,PLNTU,ROW                     BLCN2700
      WRITE(6,106)                                                         BLCN2710
      WRITE(6,107) WP,QUAL ,PSAT,POUT,DP,TPL,TSAT,DTSH,RTDP                BLCN2720
      WRITE(6,108)PIN,WH,TPO,THL,ELPH,ELSH,EBOIL,EBPL,BBOIL                BLCN2730
      WRITE (6,102)                                                        BLCN2740
      WRITE (6,103) POUT,RD,PVT,RPHDP,RPLDP,RSPDP,RBLDP,RSHDP              BLCN2750
54    CONTINUE                                                             BLCN2760
C  GO TO READ NEXT SET OF DATA                                             BLCN2770
      GO TO 1                                                              BLCN2780
      END                                                                  BLCN2790
$DATA
          0          500   10.0               100
 45.          70.          770.          140.          410.          85.
```

APPENDIX C

DESCRIPTION OF WATER BOILER

The boiler is single-pass, counterflow type with single tube-in-shell construction, as shown in figure 1(b). It uses pressurized liquid water as the heating fluid in the shell, and water as the boiling fluid inside the tube. The boiling tube is standard stainless-steel tubing 5/8 inch (1.59 cm) in outside diameter with a 0.035-inch (0.089-cm) wall. The shell is standard stainless-steel tubing 7/8 inch (2.22 cm) in outside diameter with a 0.049-inch (0.125-cm) wall. The length of the heated section of the boiler is 10 feet (3.05 m). Inside the boiling tube is an insert, to induce swirl flow, which consists of two sections. At the boiler inlet is a plug 2.15 feet (0.655 m) long machined to provide a spiral annular passage is 0.050 inch (0.127 cm) high and 0.360 inch (0.915 cm) wide. From the end of the machined plug to the boiler exit, the tube contains a 0.067-inch- (0.17-cm-) diameter wire helically coiled to approximately 3 turns per inch (1.18 turns/cm) which is attached to the inner surface of the boiling tube. In this region, concentric with the axis of the boiler, there is also a standard stainless-steel tube 0.25 inch (0.635 cm) in outside diameter to support thermocouples. At the boiler exit, a valve serves to simulate a choked-nozzle boundary condition. At the boiler entrance, another valve actuated by a feedback controller on inlet flow, provides for precise regulation of the working-fluid liquid flow.

The inlet and outlet pressures of the working fluid were measured by two Bourdon tube gages, each with an estimated accuracy of ±0.2 psi (±1380 N/m^2) (error, < 1/2 percent). The heating- and working-fluid inlet and outlet temperatures were measured with Chromel-Alumel thermocouples with an expected accuracy of $\pm 1.5^{\circ}$ F ($\pm 0.83^{\circ}$ K) (error, < 1 percent). The heating-fluid flow was measured by a turbine flowmeter with an accuracy of ±15 pounds per hour (±6.8 kg/hr) (error, < 2.5 percent). The working-fluid flow was also measured by a turbine flowmeter with an accuracy of ±2 pounds per hour (±0.91 kg/hr) (error, < 5 percent).

APPENDIX D

DESCRIPTION OF MERCURY BOILER

The SNAP-8 boiler is a single-pass, counterflow heat exchanger. The mercury working fluid is distributed to seven parallel tubes, heated by liquid-metal NaK (the eutectic of sodium and potassium) flowing in a shell. The boiler tubes are 30 feet (9.14 m) long and double-wall construction where the inside wall is of tantalum, and the outside wall is an elliptically shaped stainless-steel tube. The small space between the two tubes is filled with stagnant NaK. The double-wall construction provides material compatibility and higher reliability. The only effect this special construction had on the analysis was to increase the relative magnitude of the wall heat-transfer resistance. Spiral plug inserts extend 4.5 feet (1.37 m) inside each tube. Sixteen parallel spiral grooves are cut into the plug to improve heat transfer and dispersion of the liquid. Analytically, the only difference between this multipassage plug and the single-passage plug of the water boiler is that the flow resistance to the mercury is less (i.e., the $r_{s,\ell}$ parameter is 36 as compared with 150 for the water boiler). Only this dimensionless parameter is significantly different in the two boilers. The other parameters at design conditions, as listed in table II, are similar. This similarity in dimensionless parameters indicates that the pressure-drop characteristics should also be similar. The pressure-drop parameters for the design point of the mercury boiler, listed in the table, were evaluated by the methods described in appendix B. Representative values of the NTU parameters were calculated by using average constant coefficients of heat transfer estimated from the design calcualtions for the boiler (not shown here). An average overall heat transfer of 500 Btu per hour per ^{o}F per square foot (2830 J/(sec)(^{o}K)(m^2) was assumed for both the preheat and boiling regions and was based on an effective heat-transfer tube diameter of 1 inch (2.54 cm). The coefficient of heat transfer in the superheat region was based solely on the working-fluid coefficient for design vapor flow. This coefficient was multiplied by the factor $h_{wire}/h_{plain\ tube} = 1.5$ (see ref. 12 for s/D ratio of 36 to account for the spiral wire insert). The resultant U_p, neglecting the heating-fluid and wall resistance was 80 Btu per hour per ^{o}F per square foot (454 J/(sec)(^{o}K)(m^2)). The rest of the parameter values listed in table II can be calculated directly from the input variables and from the constants enumerated in the initialization part of the computer program listed in appendix E.

```
$IBFTC BLCP    LIST
C     THIS IS A FORTRAN IV PROGRAM FOR COMPUTING THE PRESSURE DROP
C     OF A SINGLE-PASS COUNTERFLOW BOILER.
C     THE PROGRAM IS GENERAL EXCEPT FOR THE INITIALIZATION SECTION AND
C     THE DATA CARDS.  THE NUMERICAL CONSTANTS ARE SPECIFIED IN ENGLISH
C     UNITS , ALTHOUGH ANY CONSISTENT UNITS CAN BE EMPLOYED.
C     (ENGLISH UNITS ARE IN FT  HR  LB  BTU  PSI  DEG-F ) .
C     THE NUMERICAL VALUES ARE FOR THE SEVEN TUBE MERCURY BOILER
C     THE EXIT BOUNDARY CONDITION IS A CONSTANT PRESSURE
C     THE DATA - AWP - SPECIFIES VALUES  OF WORKING FLUID FLOW (IN UNITS
C     OF FRACTION OF DESIGN FLOW ) FOR WHICH THE PRESSURE DROP FUNCTION
C     IS TO BE EVALUATED.
      DIMENSION  AWP(25)
      DATA AWP/1.0,0.02,.06,0.1,0.2,0.3,0.4,0.5,0.6,0.7,0.8,0.9,1.1,
     11.2,1.3,1.4,1.6,1.8,2.0,0.95,1.05,1.15,1.25,1.35,1.5  /
30    FORMAT(2I12,1F12.7,I12 )
31    FORMAT(6F12.7)
32    FORMAT (49H      WPO          PSATO         WH          TP-        ,
     125H     THL          PO         / 1H  ,6F12.5   )
33    FORMAT(20H1INPUT PARAMETERS     ,6HICASE=,I5/7H KCMAX=,I5,
     1 6H DK=   ,E12.5,                6H KPR=  ,I5    )
34    FORMAT(1H ,9(E12.5))
35    FORMAT(50H1 WH/FNBL   PDI/FNPH     RPH/FNSH     RPS/RBSP     ,
     160H UPH/TNK   USH/AVPRS  UBL/A1       ALT/A2      EPLUG/A3      )
36    FORMAT (1H ,7(E12.5))
37    FORMAT (50H    APH          ASH          PHNTU     BNTU          ,
     160H SHNTU      PLNTU      ROW                                  )
  106 FORMAT (50H   WP/PIN     QUAL /WH       PSAT/TPO      POUT/THL  ,
     160H    DP/ELPH    TPL/ELSH   TSAT/EBOIL RBLD/EBPL    RTDP/BBOIL   )
  107 FORMAT (1H , 9(E12.5))
  108 FORMAT (1H , 9(E12.5))
C   BEGIN INITIALIZATION FOR MERCURY BOILER
C   INPUT DATA FLOWS GIVEN FOR SEVEN TUBES
C    BOILER GEOMETRY - MERCURY 7 TUBE BOILER - DESIGN VALUES
      ALT = 30.0
      EPLUG = 0.15
      DPH = 0.083
      DBL = 0.083
      TUBNU = 7.0
      CPP = .033
      CPV = .025
      CH = 0.21
      HV = 123.1
      USH = 80.
      UBL = 500.
      UPH = 500.
      WPDG = 11500.0
      WHDG = 48000.0
      AGAS = 12100.0
      BGAS = 13.5
      RGAS = 0.0535
      TABS = 460.
      ROWRE = 3.6
      ROWL = 760.
C   PRESSURE DROP PARAMETERS
      AVPRS = 32.4
      RPS = 36.
      RPH = 0.28
      RBSP = 1.0
      RPHOP = ROWRE*RPS/ROWL
```

```
C ITERATION PARAMETERS
      KCMAX = 500
      DK = 1.0
      ERMIN = 0.001
      POUTD = 270.
          PSMIN = 1.0
C  BOUNDARY CONDITIONS AT INLET  - WP SPECIFIED
C  BOUNDARY CONDITION AT BOILER EXIT - CONSTANT PRESSURE
      ICASE = 0
C  END INITIALIZATION FOR MERCURY BOILER
C  BEGIN GENERALIZED INITIALIZATIONS
C INTERMEDIATE CONSTANTS - NTU VALUES AND RATIOS OF HEAT CAPACITY FLOWS
      A1 = WPDG*CPV/(CH*WHDG)
      A2 = (HV*WPDG)/(CH*WHDG)
      A3 = (CPP*WPDG)/(WHDG*CH)
      FNPH =3.1416*TUBNU*UPH*DPH*ALT/(CPP*WPDG)
      FNBL = 3.1416*TUBNU*UBL*DBL*ALT/(CH*WHDG)
      FNPL = FNBL
      FNSH = FNBL*USH/(A1*UBL)
C  END GENERALIZED INITIALIZATIONS
C  PREPARE FOR ITERATIONS AND READ INPUT VARIABLES
1     CONTINUE
      READ(5,30) ICASE,KCMAX,DK,KPR
      WRITE(6,33) ICASE, KCMAX,DK,KPR
      READ (5,31) WPO,PSATO,WH,TPO,THL,PO
      WRITE(6,32)WPO,PSATO,WH,TPO,THL,PO
      WP =WPO/WPDG
      WH =WH/WHDG
      PSAT=PSATO
      ASH =0.0
      DTSH =0.0
      PIN =PSATO
       PDI = 0.0
      WRITE (6,35)
      WRITE (6,34)WH,PDI,RPH,RPS,UPH,USH,UBL,ALT,EPLUG
      WRITE (6,34) FNBL,FNPH,FNSH,RBSP,TNK,AVPRS,A1,A2,A3
C  MAIN BODY OF PROGRAM - VARY WORKING FLUID FLOW
      DO 54 KRW=1,25
      WP = AWP(KRW)
C  INITIALIZE FOR ITERATION
      KPRC = KPR
      KCNTR =0
      QUAL = 1.0
      DTSH = 0.0
      PSAT = POUTD +60.
      PIN = PSAT
C  BEGINNING OF ITERATION LOOP TO ADJUST SATURATION PRESSURE TO
C  CORRESPOND TO SPECIFIED FLOW
   10 CONTINUE
C     COMPUTE HEAT TRANSFER
      ASH = WP*A1/WH
      BNTU = FNBL/WH
      PHNTU = FNPH/WP
      APH = A3*WP/WH
122   CONTINUE
      TSAT=AGAS/(BGAS-ALOG(PSAT)) - TABS
      THLSH = THL -     ASH*DTSH
      BBOIL =(A2*WP/(WH*(THLSH-TSAT)) )*QUAL
C  TEST WHETHER AVAILABLE POWER IS ENOUGH TO VAPORIZE LIQUID.
      IF(BBOIL-0.99)11,12,12
```

```
   12 CONTINUE
C HEATING ENERGY LOW, REDUCE QUALITY AT OUTLET
      DTSH = 0.0
      QUAL = QUAL - 0.001
      GO TO 122
C  HEATING ENERGY SUFFICIENT
C     COMPUTE LENGTHS OF REGIONS AND NTU OF PLUG
   11 CONTINUE
      BPHT =(TSAT-TPO)/(THLSH-(A2*WP/WH)*QUAL - TPO )
      ELPH = ALOG(ABS((1.0-APH)/(1.0-BPHT)))/(PHNTU*( 1.0-APH))
      IF (QUAL -1.0) 112,113,12
112   CONTINUE
      ELSH = -0.00001
      EBOIL = 1.0 - ELPH
      GO TO 111
113   CONTINUE
      RLBL = ALOG(1.0/(1.0-BBOIL))
      EBOIL = RLBL/BNTU
      ELSH = 1.0-ELPH-EBOIL
111   CONTINUE
      EBPL =EPLUG -ELPH
C  PRINT VARIABLES EVERY KPR ITERATIONS
      IF(KPRC)41,41,42
41    KPRC=KPR
      KSPR= 0
      WRITE(6,106)
      WRITE(6,107) WP,QUAL ,PSAT,POUT,DP,TPL,TSAT,RBLD,RTDP
      WRITE(6,108)PIN,WH,TPO,THL,ELPH,ELSH,EBOIL,EBPL,BBOIL
42    KPRC=KPRC-1
C TEST WHETHER PREHEAT REGION EXTENDS OUTSIDE PLUG
      IF(EBPL)13,13,14
   13 CONTINUE
C  PREHEAT REGION EXTENDS BEYOND PLUG - SET BOILING AND SUPERHEAT PLUG
C  PARAMETERS TO ZERO
      ESHP = 0.0
      PLNTU = 0.0
      GO TO 15
   14 CONTINUE
      ESHP =EBPL - EBOIL
C  TEST FOR SUPERHEAT LENGTH, ESHP IN PLUG
      IF(ESHP) 114,114,115
115   CONTINUE
C  BOILING REGION ENDS BEFORE END OF PLUG - CALCULATE PLUG NTU
      EBPL = EBOIL
      PLNTU = FNPL*EBPL/WH
      GO TO 15
114   CONTINUE
C  NO SUPERHEAT IN PLUG - CALCUALATE REGULAR BOILING PLUG NTU.
      ESHP = 0.0
      PLNTU = FNPL*EBPL/WH
   15 CONTINUE
C  TEST FOR OPERATION WITH EXIT QUALITY LESS THAN ONE
C  ELSH LESS THAN ZERO -ADJUST EXIT QUALITY. ELSH = ZERO - EXIT QUALITY
C  HAS BEEN ADJUSTED, CONTINUE NORMAL CALCULATION. ELSH GREATER THAN
C  ZERO - SOME SUPERHEAT , CONTINUE NORMAL CALCULATION
      IF(ELSH)17,16,16
   17 CONTINUE
C  BOILER OPERATION WITH QUALITY LESS THAN ONE
      ELSH = 0.0
      ASH = 0.0
```

```
      KQUAL = 100                                                       BLCP1800
C  ITERATE ON QUAL FOR A MAXIMUM OF 100 TIMES UNTIL BOILING LENGTH      BLCP1810
C  MATCHES END OF TUBE                                                  BLCP1820
171   CONTINUE                                                          BLCP1830
      EBOIL = 1.0 - ELPH                                                BLCP1840
      QBBL = 1.0 - EXP(0.0- EBOIL*BNTU)                                 BLCP1850
C  TEST FOR CONVERGENCE                                                 BLCP1860
      IF (ABS(QBBL-BBOIL)-0.001   ) 111,111,28                          BLCP1870
28    CONTINUE                                                          BLCP1880
      QUAL = QUAL*QBBL/BBOIL                                            BLCP1890
      KQUAL = KQUAL - 1                                                 BLCP1900
      IF (KQUAL) 22,22,172                                              BLCP1910
172   CONTINUE                                                          BLCP1920
      IF (QUAL - 1.0) 174,174,173                                       BLCP1930
173   CONTINUE                                                          BLCP1940
      QUAL = 1.0                                                        BLCP1950
      GO TO 122                                                         BLCP1960
174   CONTINUE                                                          BLCP1970
      THLSH = THL                                                       BLCP1980
      BBOIL =(A2*WP/(WH*(THLSH-TSAT)) )*QUAL                            BLCP1990
      BPHT =(TSAT-TPO)/(THLSH-(A2*WP/WH)*QUAL - TPO )                   BLCP2000
      ELPH = ALOG(ABS((1.0-APH)/(1.0-BPHT)))/(PHNTU*( 1.0-APH))         BLCP2010
      GO TO 171                                                         BLCP2020
   16 CONTINUE                                                          BLCP2030
C  OPERATION WITH SUPERHEAT - NORMAL CALCULATION                        BLCP2040
      SHNTU = FNSH*ELSH /WP                                             BLCP2050
      DTSH = (THL-TSAT)*(1.0-(1.0-ASH)*EXP(SHNTU*(ASH-1.0)))            BLCP2060
      TPL = TSAT +DTSH                                                  BLCP2070
      GO TO 18                                                          BLCP2080
C COMPUTE PRESSURE DROPS                                                BLCP2090
   18 CONTINUE                                                          BLCP2100
      ROW =PSAT/(ROWRE*RGAS*(TSAT+TABS ))                               BLCP2110
      PVT = WP*WP/ROW                                                   BLCP2120
      RLBL = ALOG(1.0/(1.0-BBOIL))                                      BLCP2130
C COMPUTE RATIOS OF PRESSURE DROP FOR EACH BOILER SECTION               BLCP2140
      RSHDP = 1.0-ELPH-EBOIL  +ESHP*(RPS-1.0)                           BLCP2150
      PPHD = ELPH - EPLUG                                               BLCP2160
C  TEST FOR PREHEAT LENGTH OUTSIDE PLUG                                 BLCP2170
      IF (PPHD) 60,60,61                                                BLCP2180
60    CONTINUE                                                          BLCP2190
C  PREHEAT REGION INSIDE PLUG                                           BLCP2200
      RPHDP = (RPH*ELPH +PDI) * ROW                                     BLCP2210
      GO TO 62                                                          BLCP2220
C  PREHAT REGION OUTSIDE PLUG                                           BLCP2230
61    RPHDP = ROW*(RPH*EPLUG +RPHOP*PPHD +PDI)                          BLCP2240
62    CONTINUE                                                          BLCP2250
      RB =(1.0-BBOIL)/BBOIL                                             BLCP2260
      RPLDP =(RBSP*RB*(RPS-1.0)*(EXP( PLNTU)-PLNTU-1.0)/BNTU )*QUAL     BLCP2270
      RSPDP =(RBSP*(1.0-RB*RLBL)/BNTU)*QUAL                             BLCP2280
      RTDP = RPHDP + RPLDP +RSPDP+RSHDP                                 BLCP2290
      DP = PVT*RTDP*AVPRS                                               BLCP2300
      RD = RTDP*PVT                                                     BLCP2310
      RBLDP = RPLDP+RSPDP                                               BLCP2320
C  COMPUTE PIN AND PSAT BASED ON CALCUALTED PRESSURE DROP               BLCP2330
      PIN = POUTD + DP                                                  BLCP2340
      PSATR = PSAT                                                      BLCP2350
      PSAT = PIN - RPHDP*PVT*AVPRS                                      BLCP2360
      PER = PSAT - PSATR                                                BLCP2370
      POUT = POUTD                                                      BLCP2380
  102 FORMAT (50H  PRESOUT     PRESDROP   PR VAP TOT   PDROP  PHEAT   ,  BLCP2390
```

```
     150H B RRSPLG  PRDR BSP   PRDRBOIL    PRDRSHT                )       BLCP2400
C  WRITE INTERMEDIATE RESULTS EVERY KPR ITERATIONS                        BLCP2410
      IF(KSPR) 43,43,44                                                   BLCP2420
43    KSPR = 1                                                            BLCP2430
      WRITE (6,102)                                                       BLCP2440
  103 FORMAT (1H ,8(E12.5))                                               BLCP2450
      WRITE (6,103) POUT,DP,PVT,RPHDP,RPLDP,RSPDP,RBLDP,RSHDP             BLCP2460
44    CONTINUE                                                            BLCP2470
C  TEST FOR CONVERGENCE  - ERROR IN PSAT LESS THAN 0.5  PSI               BLCP2480
      IF (ABS(PER) - 0.5 ) 20,20,45                                       BLCP2490
45    CONTINUE                                                            BLCP2500
C  ERROR STILL UNACCEPTABLE - CONTINUE ITERATION IF KCMAX NOT EXCEEDED    BLCP2510
      PSAT = PSATR + PER/2.0                                              BLCP2520
      KCNTR = KCNTR +1                                                    BLCP2530
      IF(KCNTR-KCMAX) 21,21,22                                            BLCP2540
   21 CONTINUE                                                            BLCP2550
      GO TO 10                                                            BLCP2560
   22 CONTINUE                                                            BLCP2570
C  NO CONVERGENCE - KCMAX EXCEEDED OR KQUAL = 0                           BLCP2580
  104 FORMAT (52H   NO CONVERGENCE    -KCMAX EXCEEDED                    )BLCP2590
      WRITE (6,104)                                                       BLCP2600
      GO TO 20                                                            BLCP2610
   20 CONTINUE                                                            BLCP2620
C  NORMAL OUTPUT FOR SUCCESSFUL CONVERGENCE                               BLCP2630
  105 FORMAT (7H0 AFTER ,I4,23H ITERATIONS, FOR CASE =,I4,                BLCP2640
     150H THE FOLLOWING BOILER VARIABLES HAVE RESULTED            )       BLCP2650
      WRITE(6,105) KCNTR,ICASE                                            BLCP2660
      WRITE  (6,37)                                                       BLCP2670
      WRITE  (6,36) APH,ASH,PHNTU,BNTU,SHNTU,PLNTU,ROW                    BLCP2680
      WRITE(6,106)                                                        BLCP2690
      WRITE(6,107) WP,QUAL ,PSAT,POUT,DP,TPL,TSAT,DTSH,RTDP               BLCP2700
      WRITE(6,108)PIN,WH,TPO,THL,ELPH,ELSH,EBOIL,EBPL,BBOIL               BLCP2710
      WRITE (6,102)                                                       BLCP2720
      WRITE (6,103) POUT,RD,PVT,RPHDP,RPLDP,RSPDP,RBLDP,RSHDP             BLCP2730
54    CONTINUE                                                            BLCP2740
C  GO TO READ NEXT SET OF DATA                                            BLCP2750
      GO TO 1                                                             BLCP2760
      END                                                                 BLCP2770
$DATA
          0          500   10.0               100
11500.        270.       48000.       500.         1300.        360.
```

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TABLE I. - PARAMETERS AND INPUT VARIABLES FOR DESIGN POINT OF WATER BOILER

(a) Parameters

Dimensionless parameter	Definition	Design value
Effectiveness of preheat region, ϵ_p	$(T_{sat} - T_{wi})/(T_{hb} - T_{wi})$	0.823
Effectiveness of boiling region, ϵ_b	$X_o H_w w_w/(T_{hs} - T_{sat}) w_h c_h$	0.575
Number of transfer units (NTU) of boiler based on overall average heat-transfer coefficient in preheat region, N_p	$U_p \pi D_p l_t / w_w c_{w,l}$	21.6
NTU of boiler based on overall average heat-transfer coefficient in boiling region, N_b	$U_b \pi D_b l_t / w_h c_h$	1.76
NTU of boiler based on overall average heat-transfer coefficient in superheat region, N_s	$U_s \pi D_s l_t / w_w c_{w,v}$	3.36
Ratio of working-fluid to heating-fluid heat capacity in preheat region, a_p	$w_w c_{w,l}/w_h c_h$	0.0584
Ratio of working-fluid to heating-fluid heat capacity in super-heated region, a_s	$w_w c_{w,v}/w_h c_h$	0.0277
Preheat length (dimensionless), L_p	l_p/l_t	0.083
Superheat length (dimensionless, L_s	l_s/l_t	0.431
Boiling length (dimensionless), L_b	l_b/l_t	0.485
Plug length, L_{pl}	l_{pl}/l_t	0.215
Ratio of preheat-plug-region pressure gradient to super-heat pressure gradient, $r_{p,s}$	$(f_p/f_s)(\rho_{v,d}/\rho_{w,l})(p_{pl}/p_s)(A_s^3/A_{pl})(dl_{pl}/dl)$	0.78
Ratio of boiling to superheat friction factors, $r_{b,s}$	$f_{b,s}/f_s$	1.0
Ratio of pressure gradients for vapor at rated flow in plug region to those of vapor at rated flow in swirl-wire region, $r_{s,\phi}$	$(f_{pl,b}/f_{b,s})(p_{pl}/p_s)(A_s/A_{pl})(dl_{pl}/dl)$	150.0

(b) Input variables

Variables	Design value
Normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, $\Delta P_{v,d}$, psia (N/m^2 abs)	9.7 (67 000)
Working-fluid outlet pressure, $P_{o,d}$, psia (N/m^2 abs)	64.5 (445 000)
Working-fluid design flow, $w_{w,d}$, lb/hr (kg/hr)	45 (20.4)
Heating-fluid design flow, $w_{h,d}$, lb/hr (kg/hr)	770 (349.0)
Working-fluid inlet temperature, $T_{wi,d}$, ^oF (^oK)	140 (333)
Heating-fluid inlet temperature, $T_{hi,d}$, ^oF (^oK)	410 (483)
Total boiler tube length, l_t, ft (m)	10 (3.048)
Working-fluid vapor density at design flow in superheat region, $\rho_{v,d}$, lb/ft^3 (kg/m^3)	0.155 (2.48)
Working-fluid liquid density, $\rho_{w,l}$, lb/ft^3 (kg/m^3)	59.4 (950)
Hydraulic diameter D in the boiling, preheat, and superheat regions, ft (m)	0.049 (0.015)
Working-fluid saturation temperature, $T_{sat,d}$, ^oF (^oK)	320 (433)
Working-fluid saturation pressure, $P_{sat,d}$, psia (N/m^2)	89 (615 000)

TABLE II. - COMPARISON OF MERCURY AND WATER BOILERS

PARAMETERS AND DESIGN INPUT VARIABLES

(a) Parameters

Parameter	Mercury boiler (1 of 7 tubes)	Water boiler
	Design value	
Effectiveness of preheat region, ϵ_p	0.908	0.823
Effectiveness of boiling region, ϵ_b	.698	0.575
Number of transfer units (NTU) of boiler based on overall average heat-transfer coefficient in preheat region, N_p	72.1	21.6
NTU of boiler based on overall average heat-transfer coefficient in boiling region, N_b	2.72	1.76
NTU of boiler based on overall average heat-transfer coefficient in superheat region, N_s	15.2	3.36
Ratio of working-fluid to heating-fluid heat capacity in preheat region, a_p	0.0376	0.0584
Ratio of working-fluid to heating-fluid heat capacity in superheated region, a_s	0.0285	0.0277
Preheat length (dimensionless), L_p	0.0336	0.083
Superheat length (dimensionless), L_s	0.525	0.431
Boiling length (dimensionless), L_b	0.441	0.485
Plug length, L_{pl}	0.15	0.215
Ratio of preheat-plug-region pressure gradient to superheat pressure gradient, $r_{p,s}$	0.28	0.78
Ratio of pressure gradients for vapor at rated flow in plug region to those of vapor at rated flow in swirl-wire region, $r_{s,s}$	36.0	150.0
Ratio of boiling to superheat friction factors, $r_{b,s}$	1.0	1.0

(b) Input variables

Input variable	Mercury boiler (1 of 7 tubes)	Water boiler
	Design value	
Working-fluid inlet temperature, $T_{wi,d}$, ^oF (^oK)	500 (533)	140 (333)
Heating-fluid inlet temperature, $T_{hi,d}$, ^oF (^oK)	1300 (978)	410 (483)
Working-fluid flow, $w_{w,d}$, lb/hr (kg/hr)	1644 (747)	45 (20.4)
Heating-fluid flow $w_{h,d}$, lb/hr (kg/hr)	6860 (3120)	770 (349.0)
Working-fluid outlet pressure, $P_{o,d}$, psia (N/m^2 abs)	270 (186×10^6)	64.5 (445 000)
Normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, $\Delta P_{v,d}$, psia (N/m^2 abs)	32.4 (2.24×10^5)	9.7 (67 000)
Total boiler tube length, l_t, ft (m)	30 (9.14)	10 (3.048)
Working-fluid vapor density at design flow in superheat region, $\rho_{v,d}$, lb/ft^3 (kg/m^3)	3.6 (57.7)	0.155 (2.48)
Working-fluid liquid density, $\rho_{w,l}$, lb/ft^3 (kg/m^3)	760 (12 200)	59.4 (350)
Working-fluid saturation temperature, ^oF (^oK)	1100 (866)	319 (433)
Hydraulic diameter D in preheat, boiling, and superheat regions, ft (m)	0.083 (0.025)	0.049 (0.015)

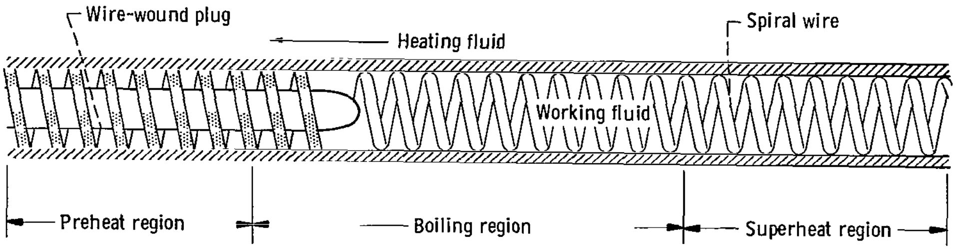

Figure 1. - Typical boiler tube of counterflow-heated Rankine space power boiler.

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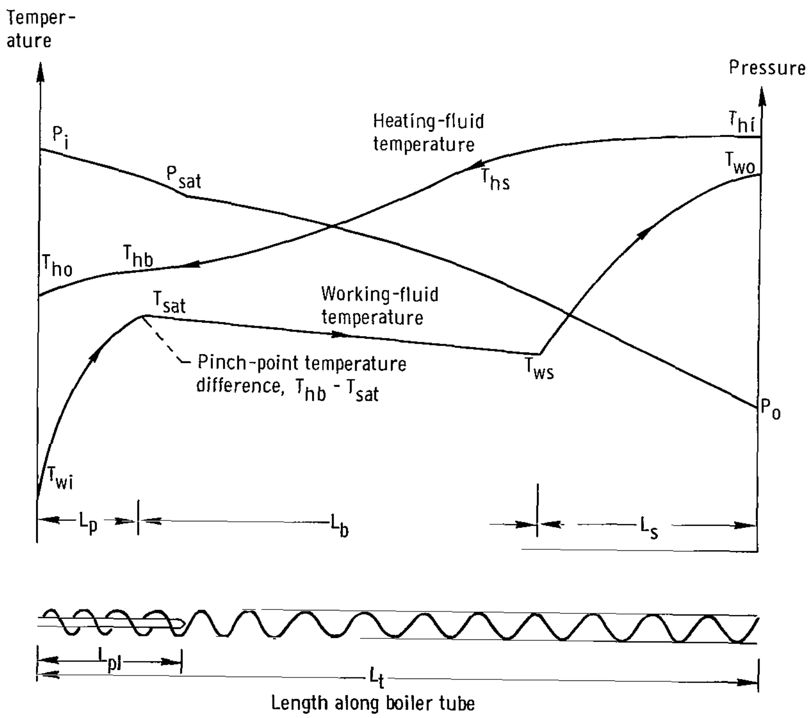


Figure 2. - Typical temperature and pressure profile of single-pass counterflow boiler.

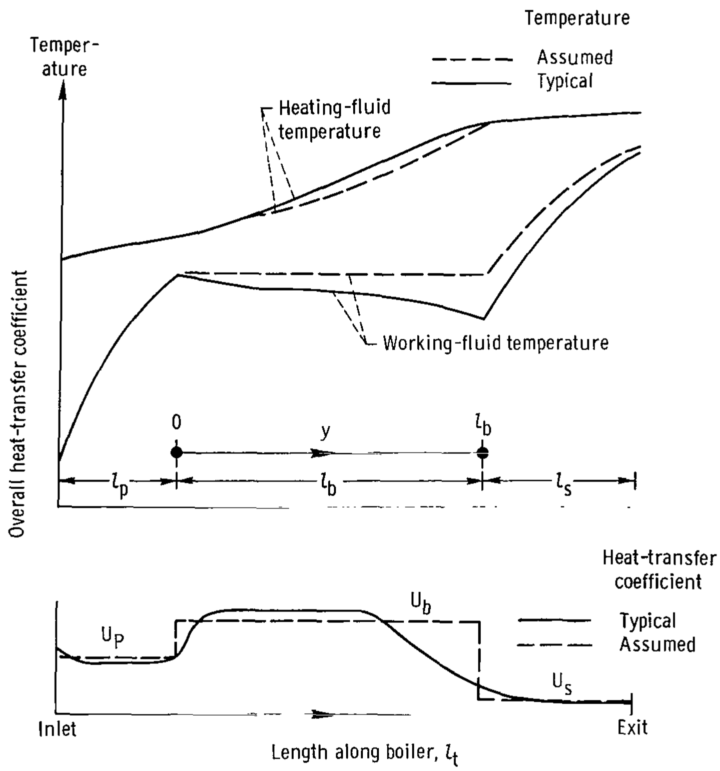


Figure 3. - Temperature and overall heat-transfer-coefficient profile showing difference in actual and assumed characteristic.

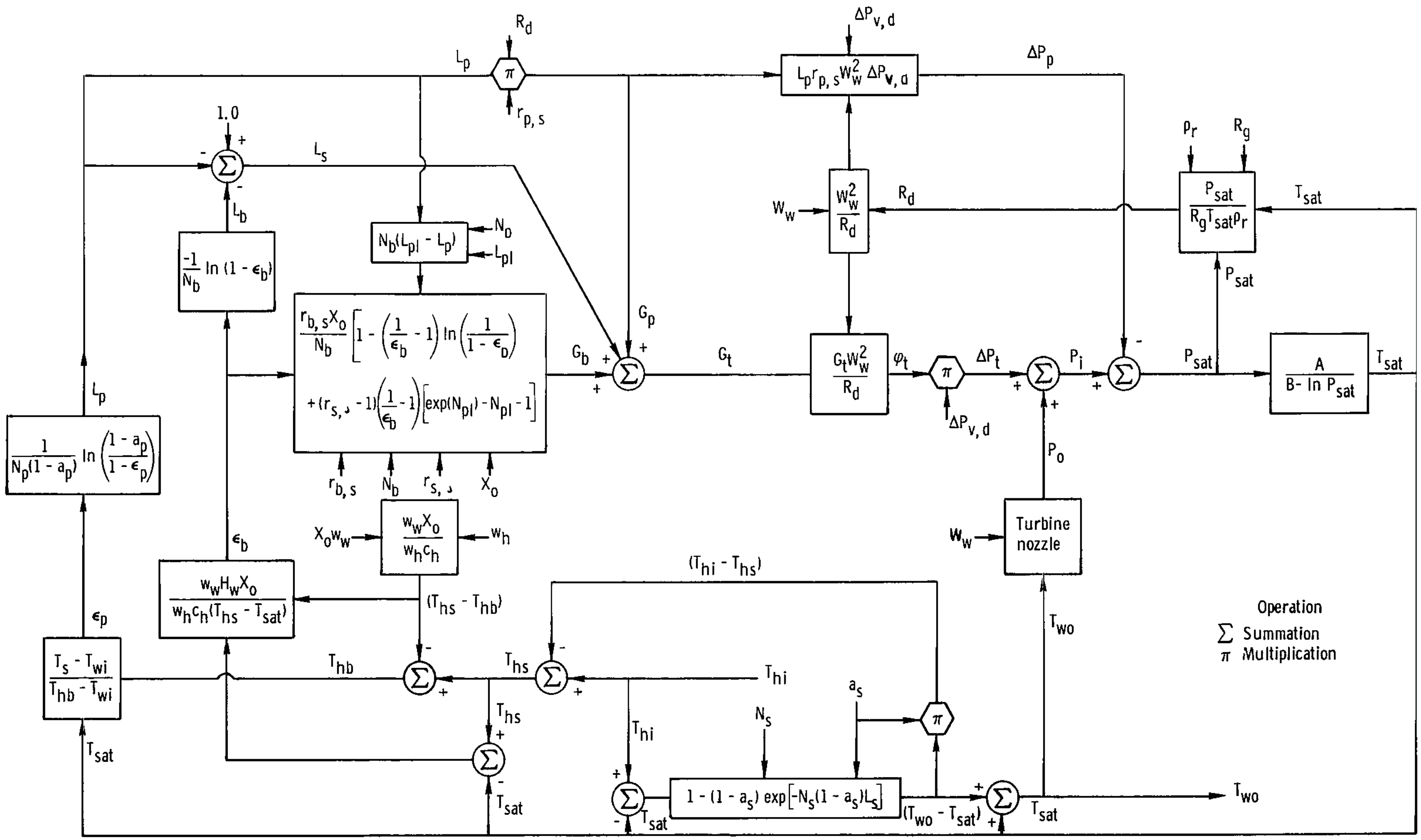


Figure 4. - System representation of boiler equations for superheat at exit. Specified inlet flows, W_w and w_h; specified inlet temperatures, T_{wi} and T_{hi}; boundary condition at exit, choked nozzle.

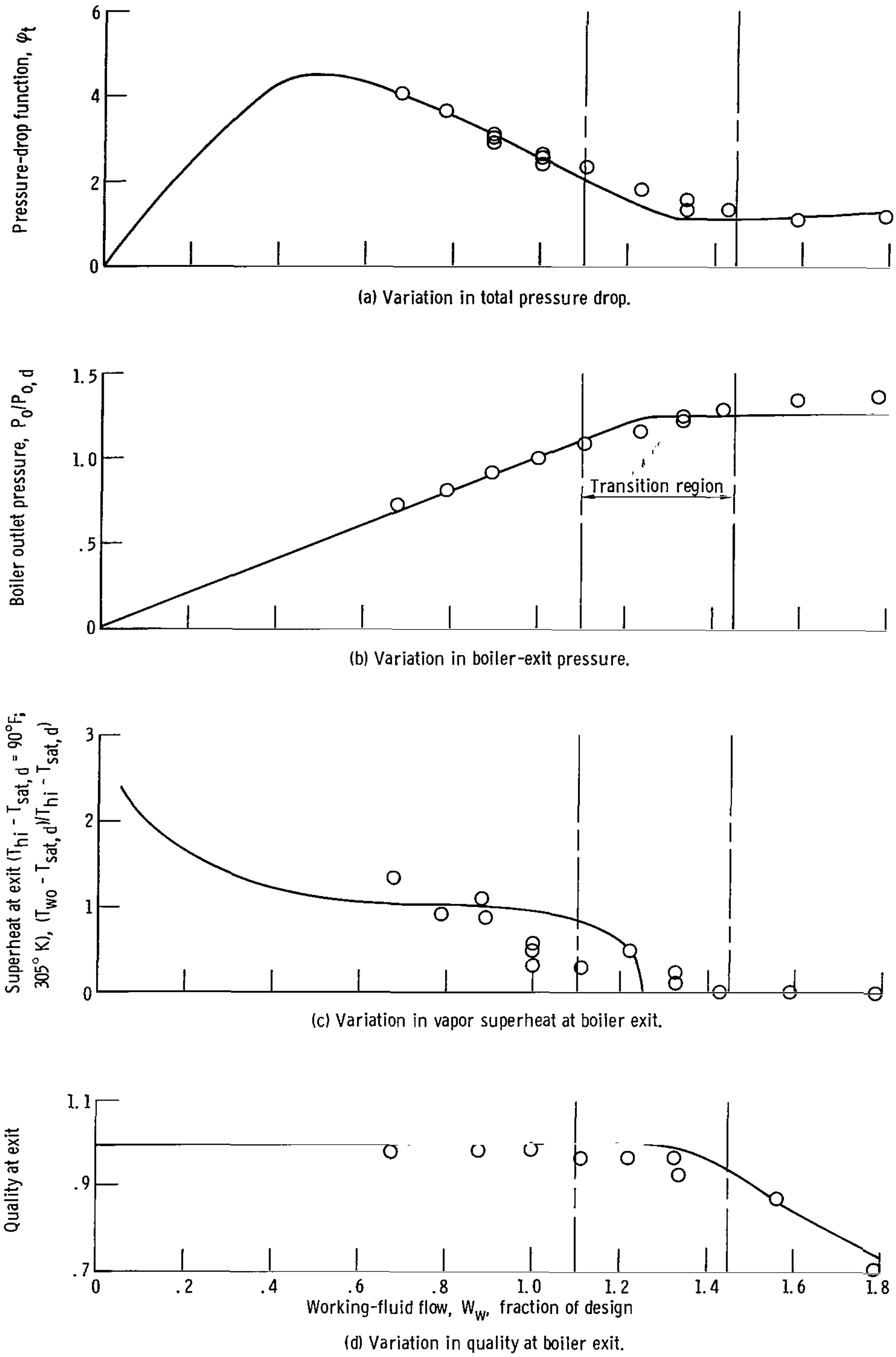
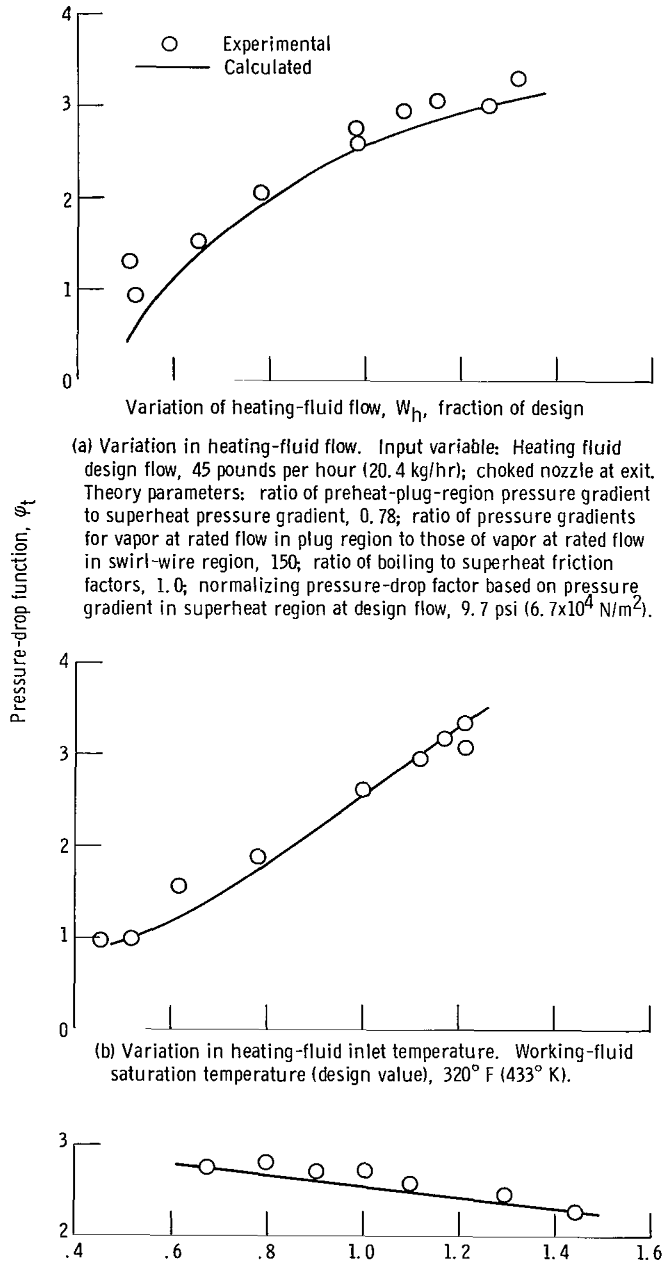


Figure 5. - Comparison of theory with water-boiler data for variation in working-fluid flow. Working-fluid design flow, 45 pounds per hour (20.4 kg/hr); ratio of pressure gradients for vapor at rated flow in plug region to vapor at rated flow in swirl-wire region, 150; ratio of preheat-plug-region pressure gradient to superheat pressure gradient, 0.78; ratio of boiling to superheat friction factors, 1.0; normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, 9.7 psi (6.7×10^4 N/m^2); working-fluid inlet temperature (design value), 140° F (333° K); heat-fluid inlet temperature (design value), 410° F (483° K); heat-fluid design flow, 770 pounds per hour (349 kg/hr); choked nozzle at boiler exit set for 64.5 psia (445 000 N/m^2 abs) at design flow.

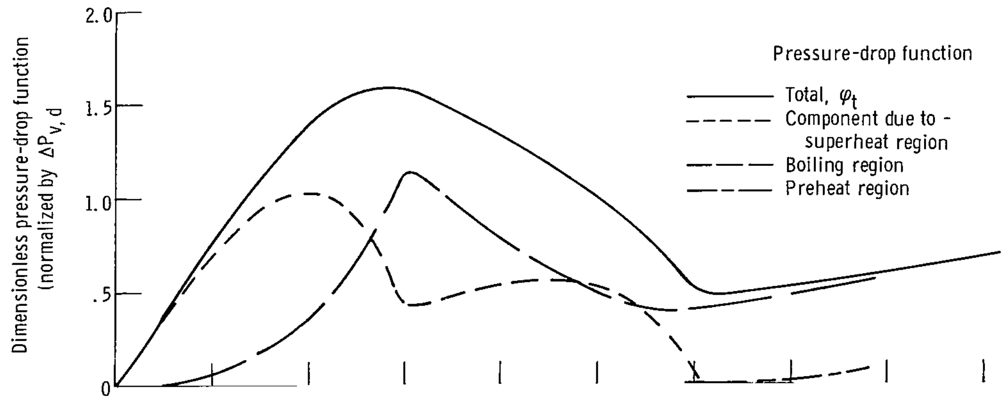


(a) Variation in heating-fluid flow. Input variable: Heating fluid design flow, 45 pounds per hour (20.4 kg/hr); choked nozzle at exit. Theory parameters: ratio of preheat-plug-region pressure gradient to superheat pressure gradient, 0.78; ratio of pressure gradients for vapor at rated flow in plug region to those of vapor at rated flow in swirl-wire region, 150; ratio of boiling to superheat friction factors, 1.0; normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, 9.7 psi (6.7×10^4 N/m^2).

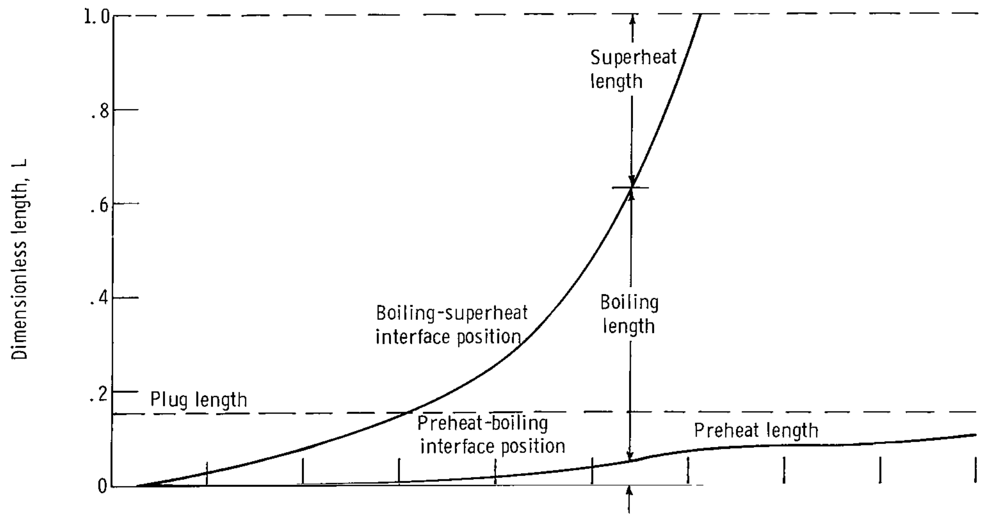
(b) Variation in heating-fluid inlet temperature. Working-fluid saturation temperature (design value), 320° F (433° K).

(c) Variation in working-fluid inlet temperature. Working-fluid saturation temperature (design value), 320° F (433° K).

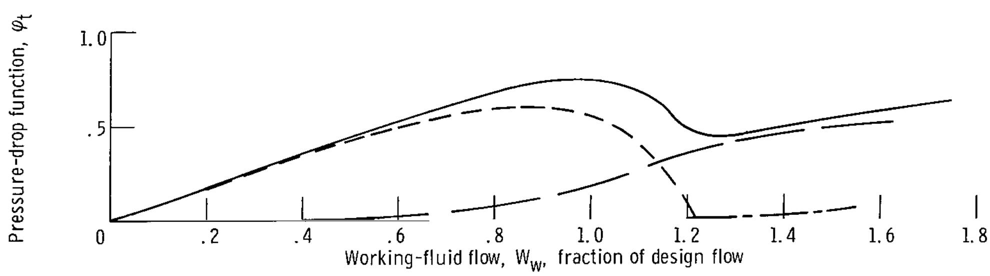
Figure 6. - Comparison of data with predicted pressure drop as function of input variables. Heating-fluid inlet temperature (design value), 410° F (483° K); working-fluid inlet temperature (design value), 140° F (333° K); dimensionless heating-fluid design flow, 1.0; dimensionless working-fluid design flow, 1.0.



(a) Tube with multipassage plug insert. Ratio of pressure gradients for vapor at rated flow in plug region to those of vapor at rated flow in swirl-wire region, 36; dimensionless plug length, 0.15.



(b) Position of interfaces.



(c) Tube without plug. Ratio of pressure gradients for vapor at rated flow in plug region to those of rated flow in swirl-wire region, 1.0.

Figure 7. - Pressure drop and interface positions as function of working-fluid flow showing components due to superheat, boiling, and preheat regions. SNAP-8 mercury-boiler parameters: ratio of boiling to superheat friction factors, 1.0; ratio of preheat-plug-region pressure gradient to superheat pressure gradient, 0.28; normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, 32.4 psi (2.24×10^5 N/m^2); heating-fluid design flow, 6860 pounds per hour (3120 kg/hr); working-fluid design flow, 1644 pounds per hour (747 kg/hr); working-fluid outlet pressure (design value), 270 psia (1.86×10^6 N/m^2 abs); heating-fluid inlet temperature (design value), 1300° F (978° K); working-fluid inlet temperature (design value), 500° F (533° K); choked nozzle at exit; no orifice at inlet.

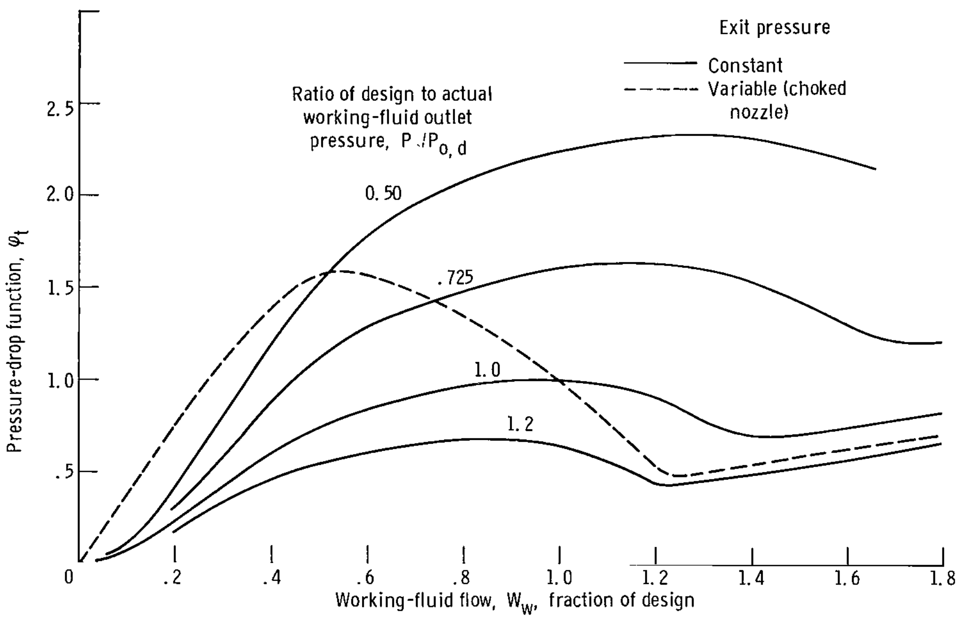


Figure 8. - Boiler-tube pressure drop as function of working-fluid flow showing effect of constant-exit-pressure boundary condition. Working-fluid outlet pressure (design value), 270 psia (1.86×10^6 N/m^2).

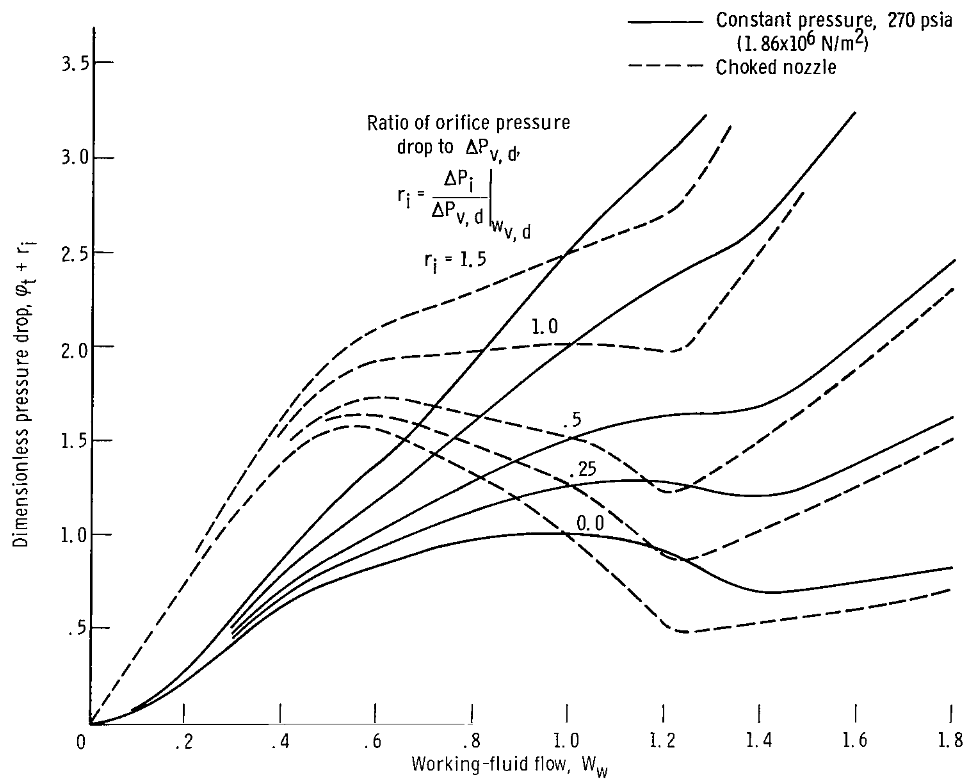


Figure 9. - Pressure drop as function of working-fluid flow showing effect of various inlet orifices (r_i parameter) and difference between choked nozzle and constant exit pressure. Working-fluid design flow, 1649 pounds per hour (747 kg/hr); dimensionless heating-fluid design flow, 1.0; ratio of boiling to superheat friction factors, 1.0; ratio of preheat-plug-region pressure gradient to superheat pressure gradient, 0.28; ratio of pressure gradients for vapor at rated flow in plug region to those of vapor at rated flow in swirl-wire region, 36; normalizing pressure-drop factor based on pressure gradient in superheat region at design flow, 32.4 psia (2.24×10^5 N/m^2 abs); heating-fluid inlet temperature (design value), 1300° F (978° K); working-fluid inlet temperature (design value), 500° F (533° K).

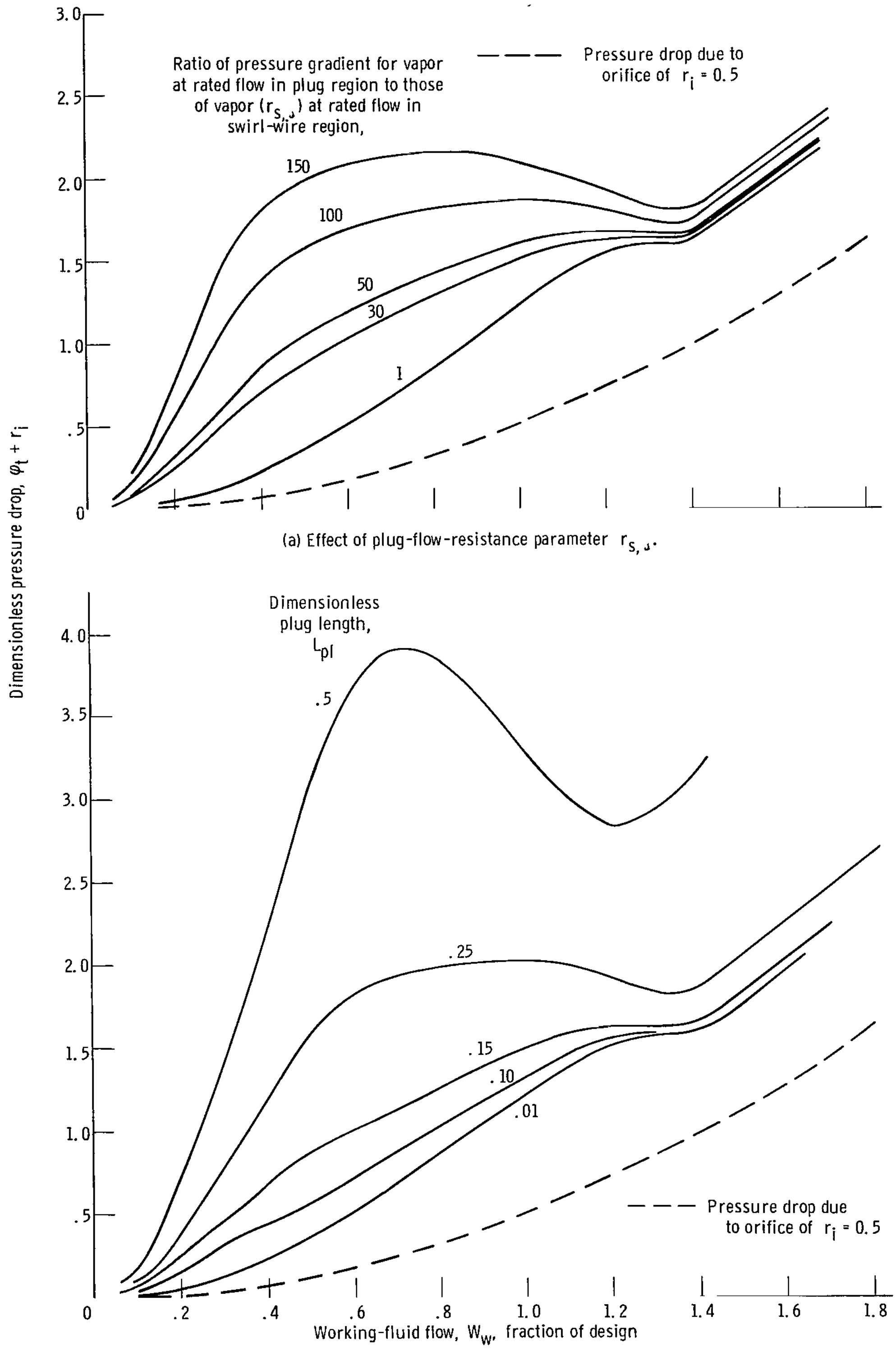
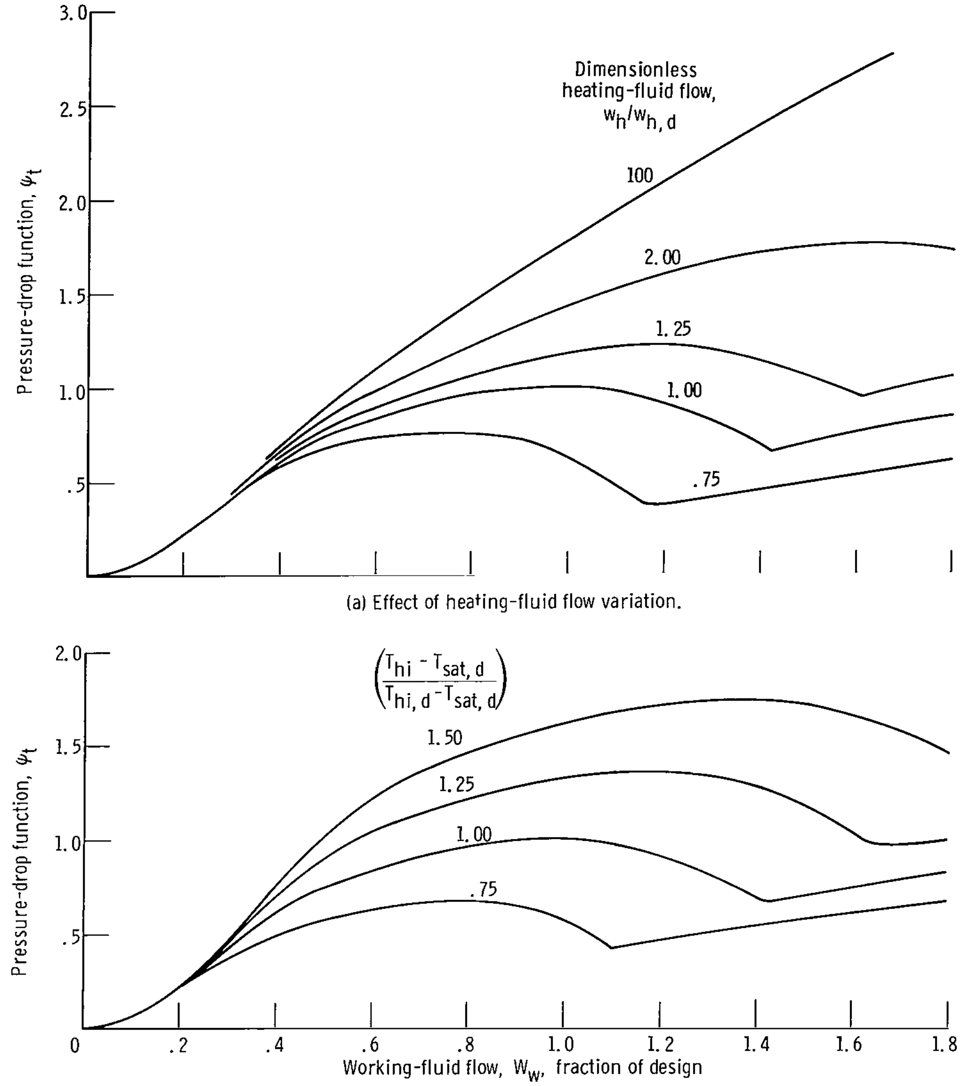


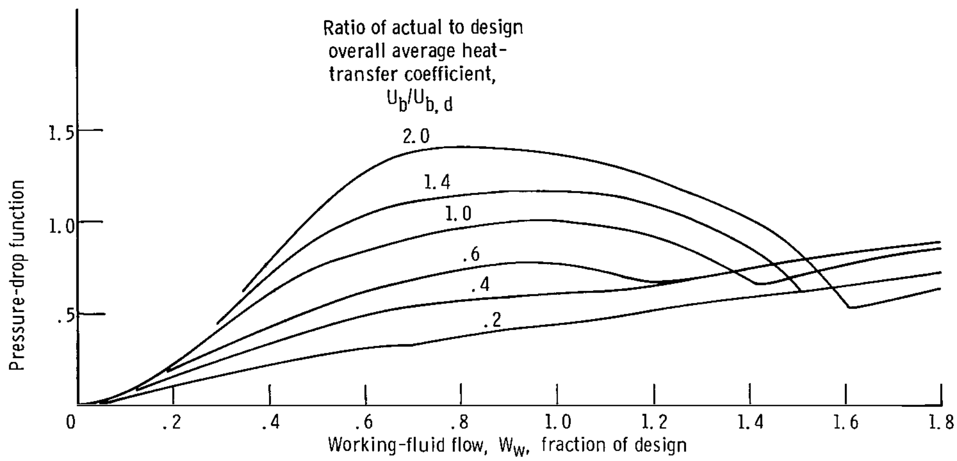
Figure 10. - Effect of plug insert parameters on boiler pressure-drop function. Orifice, r_i = 0.5; constant pressure at tube-exit, 270 psia (1.86×10^6 N/m^2).



(a) Effect of heating-fluid flow variation.

(b) Effect of heating-fluid inlet-temperature variation. Working-fluid saturation temperature (design value), 1100° F (866° K).

Figure 11. - Boiler pressure-drop function as function of working-fluid flow showing effect of heating fluid parameters. Heating-fluid design flow, 6860 pounds per hour (3120 kg/hr); heating-fluid inlet temperature (design value), 1300° F (978° K); working-fluid inlet temperature (design value), 500° F (533° K); working-fluid outlet pressure (design value), 270 psia (constant) (1.86×10^6 N/m^2); working-fluid design flow, 1644 pounds per hour (747 kg/hr); constant exit pressure.



(c) Effect of boiling-region heat-transfer-coefficient variation; overall average heat-transfer coefficient design value, 500 Btu/(°F)(hr)(ft^2); 2830 J/(°K)(sec)(m^2).

Figure 11. - Concluded.

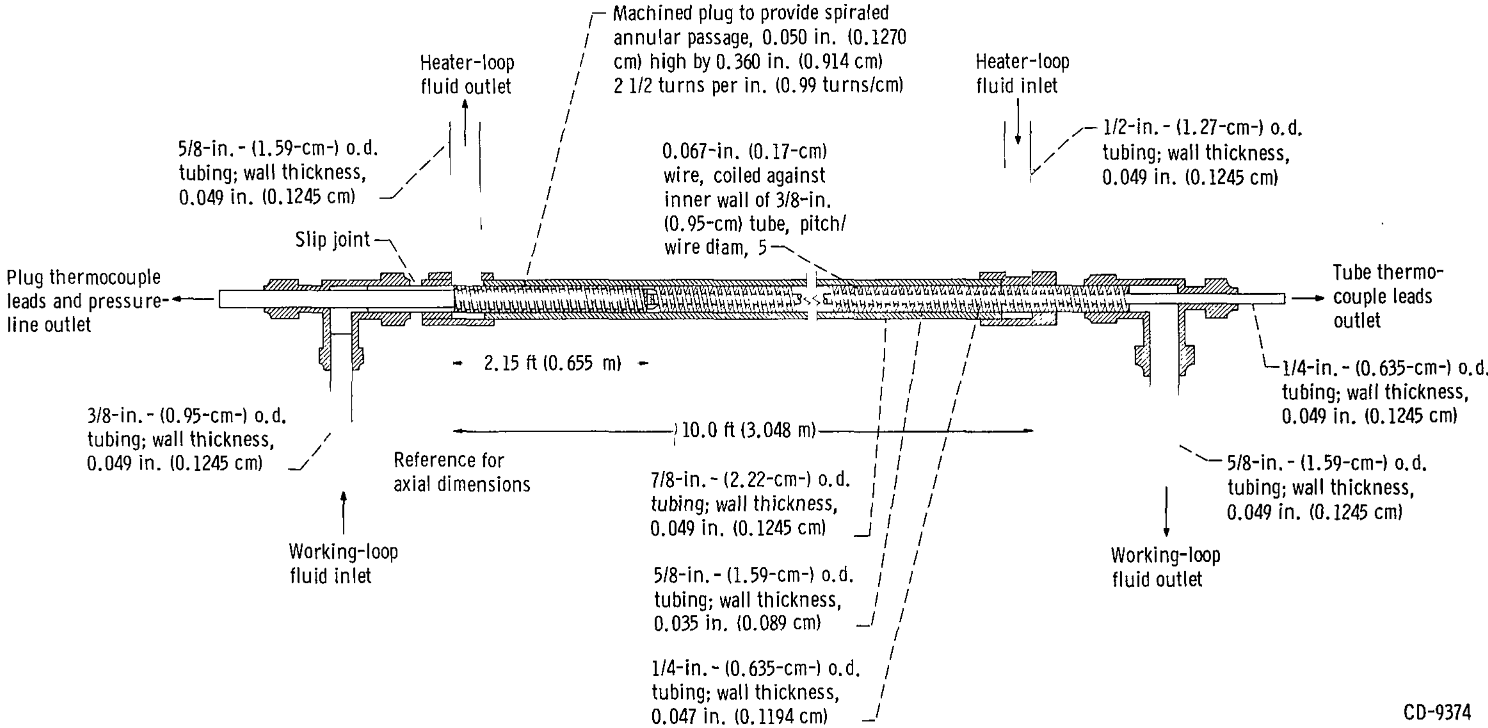


Figure 12. - Water-boiler cross section.

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