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FUSION REACTOR FIRST-WALL COOLING FOR

VERY HIGH ENERGY FLUXES

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ABSTRACT

A study has been made of the feasibility of a protective first wall shield between the plasma and the containment vessel for early experimental controlled thermonuclear fusion machines. The proposed first-wall shield is a water-cooled array of thin-walled tubes designed to take very high local energy fluxes originating from the neutral beam injectors. Detailed computer calculations reveal that heat flux capabilities of 3300 W/cm^2 are possible with first-wall shield sections made up of tubes 1 meter long of Ta-10W alloy (with tube inner diameters of 10 mm and tube wall thickness of 0.5 mm) with a structural safety factor of about 4. If the structural safety factor can be lowered to 3, the heat flux capability can be increased to 4200 W/cm^2 . Required pumping powers on the order of 1 MW per square meter of first wall area exposed to these high energy fluxes are predicted for flow in the non-boiling regime.

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If operation in the subcooled nucleate boiling regime can be achieved without flow oscillations or instabilities, the required pumping power is shown to decrease by about an order of magnitude (because the same heat flux capability can be achieved at lower flow rate).

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1. Introduction

This paper presents the second part of a study on highly cooled, first wall shields for experimental controlled thermonuclear fusion machines. (The first wall shield is to be placed between the fusion plasma and the containment vessel.) In the first part of this study (Ref. 1) heat flux capabilities of over 3000 W/cm^2 were predicted for a water-cooled first wall shield composed of thin-walled tubes of Ta-10W alloy (tube length = 1.0 m, diameter = 10 mm, and wall thickness = 0.5 mm). An adequate structural safety factor (defined as the yield stress divided by the sum of the thermal and hoop stresses) of between 3 and 5 was estimated for this case.

There were, however, many approximations made in the study of Ref. 1 in order to simplify the hand calculations, including the assumption of constant fluid and tube material properties, no sub-cooled nucleate boiling, and a uniform axial heat flux. For this

second part of the study a computer code was written which eliminates the above approximations. In addition, the code permits us to assess the sensitivity of the maximum heat flux capability of these highly-cooled walls to errors or uncertainties in the various material properties and in the empirical equations for the subcooled boiling regime.

In Section 2 the computer model is described and a comparison is made with data from some very high heat flux arc-jet nozzle experiments. In Section 3 the results for the maximum heat flux capability for Ta-10W tubes are presented. Section 4 contains a discussion of the effect of a non-uniform axial heat flux distribution and Section 5 presents some results of the sensitivity analysis.

2. Computer Model - Comparison with Experiments

The computer model permits the calculation of the axial variations of the fluid bulk temperature and pressure, the tube inner and outer wall temperatures, and the maximum local hoop and thermal stresses for any circumferential and/or axial heat flux variation. The equations used in both the non-boiling (NOB) regime and the fully-developed subcooled nucleate boiling (SCB) regime are described in Appendix A. Version c of the code was used for all the results presented except in Section 5 where specifically stated otherwise.

Since many empirical equations are required to calculate the effective heat transfer and friction coefficients, the critical heat flux, and the effect of heat flux non-uniformities, particularly

in the SCB regime, we felt it to be desirable to compare the predictions of the code with some experimental data. A search of the literature revealed that most published data for water coolant are for heat fluxes well below 1000 W/cm^2 , where the majority of present day applications lie. This is far below the values of interest in this study as shown in Figure 1.

Fortunately, we found a set of NASA data for a simulated arc-jet nozzle with heat fluxes up to $13,000 \text{ W/cm}^2$ (Ref. 2). These data are for a very non-uniform axial heat flux distribution as shown by the dashed curve of Fig. 2, where the peak heat flux is at the simulated nozzle throat. The simple parabolic fit at $\frac{x}{L}$ of 0.0, 0.5, and 1.0 shown by the solid curve has been used in our calculations.

Figure 3 shows the data points from the NASA tests for two mass flow rates along with our computed theoretical curves. In the transition region from non-boiling to fully-developed subcooled nucleate boiling our theoretical model can be seen to be conservative. We intentionally have chosen to model the SCB region by the asymptotic curve for fully-developed subcooled nucleate boiling; that is to say, we have chosen not to include the gradual departure from the non-boiling curve as the fluid enters the partial subcooled nucleate boiling regime.

On the log-log plot of Fig. 3, the non-boiling curve and the fully-developed SCB asymptotic curve are essentially straight lines as shown. Near the intersection of these two curves, the theoretical model appears to be conservative by about 20%.

The predicted critical heat flux values for the higher mass flow rate are shown as a band $\pm 5\%$ wide; this is due to the variability in the experimental conditions between different runs. The experimental burnout results also fall within about a $\pm 5\%$ band of values. It can be seen that here again the computer model is conservative by a minimum of 20%.

We can conclude from the above comparison that the relatively simple computer model we have developed for this study probably yields somewhat conservative values of the heat flux capability of tubular walls for a given set of water flow conditions and tube dimensions.

3. Maximum Heat Flux Capability for Axially Uniform Heat Fluxes

Based on the results of the first part of this study (Ref. 1), it was decided to examine the case of the Ta-10W tubes in greater detail. The reference tube geometry and reference water flow conditions are shown in Table 1. The water and tube material property data are not shown because they are inserted into the code for several temperatures, and a least-squares fit is used to predict their variation with temperature.

The code was first run for an axially-uniform incident heat flux of $q_i = 3300 \text{ W/cm}^2$ (on one side of the tubular wall only) for a series of values of the mass flow rate parameter $G \equiv \rho u$. The non-uniform circumferential heat flux variation is approximately $q(\theta) \approx q_i \cos \theta$ ($-90^\circ \leq \theta \leq 90^\circ$).

From Figure 4 we see that the flow passes through several regimes. At very low G , the water is in the SCB (subcooled nucleate boiling)

regime from the inlet to the axial position where bulk boiling begins. As G is increased, there is no longer any bulk boiling in the tube, the entire flow being in the SCB regime where the film temperature drop is almost constant. At a certain value of G (about 7×10^4 kg/m² -s for this example), the flow velocity becomes so large that the flow changes to the NOB (non-boiling) regime throughout the tube. At still higher values of G , the pressure drop becomes so large that SCB occurs near the exit of the tube (at about a $G = 10 \times 10^4$ kg/m² -s). At a very slightly larger value of G , the predicted critical heat flux is exceeded at the tube exit. If burnout did not occur, a slightly higher G would result in bulk boiling at the exit due to the very low exit pressures at these high mass flow rates.

Two of the key performance parameters which strongly influence the value of G at which we choose to operate our cooling system are the required pumping power, P_p , and the structural safety factor, SF. These are shown on Figures 5 and 6, respectively, for three high values of the incident heat flux: 2400, 3300 and 4800 W/cm².

Figure 5 shows the dramatic increase in the required pumping power per unit of first wall area, P_p/A_{fw} , as G increases. While the pumping power required is lowest in the SCB regime, the actual performance in the SCB regime and the true values of P_p/A_{fw} are subject to rather large uncertainties at present due to the uncertainty in the form of the pressure gradient equation in the SCB regime. This will be discussed in detail in Section 5. In addition, the flow is more likely to encounter thermal-hydraulic oscillations and/or

instabilities in this flow regime. As a result, an operating condition with somewhat higher than the minimum pumping power may have to be chosen until better data are available for the SCB regime at these high heat fluxes and mass flow rates.

The minimum structural safety factors (SF) shown in Figure 6 all occur near the tube inlet on the side of the tubular wall receiving the energy flux. A cross-plot of these safety factors for $G \approx 4 \times 10^4$ kg/m² -s in Figure 7 shows that a maximum heat flux capability of about 4200 W/cm² is possible at a structural SF = 3.0. If a more conservative structural SF of 4.0 is chosen, then a $q_{\max}$ on the order of 3300 W/cm² would be permissible.

The major uncertainties in this particular study with application to an experimental fusion machine lie in the prediction of the neutron damage effects on the tube material properties including the yield stress. Since insufficient data are presently available for a meaningful evaluation of neutron damage effects, we have chosen to use the unirradiated materials data. We then should base our choice of the maximum allowable heat flux on an adequately conservative structural safety factor, probably on the order of 4.

4. Effect of Non-Uniform Axial Heat Flux

Several runs were made using the conditions of one of the NASA test series (Ref. 2) to assess the first-order effects of axially non-uniform heat fluxes. Some representative results are listed in Table 2 for the case where the uniform and non-uniform heat flux

profiles have the same average.

For the cases with average heat flux of 800 W/cm^2 there is no subcooled boiling, and the tabulated results show that there is essentially no difference in the fractional pressure drop, $\Delta p/p_{in}$, or the pumping power, P_p , between the uniform heat flux case and the highly peaked heat flux profile of Fig. 2. That is, within the limits of our model, which does not include thermal lags or inlet development length effects, the very slight differences in $\Delta p/p_{in}$ and P_p in this non-boiling case are due solely to small property variations with temperature.

The higher heat flux case (with an average heat flux of 3600 W/cm^2) shows an important effect of the non-uniform heat flux distribution which manifests itself when part or all of the flow is in the SCB regime. For this higher heat flux case both $\Delta p/p_{in}$ and P_p/A_{fw} are about 60% higher for the non-uniform distribution. This is due to the fact that there is sub-cooled boiling with the non-uniform heat flux distribution and hence a larger pressure drop in the last third of the tube.

The major difference between the uniform and non-uniform heat flux cases is in the maximum tube wall temperatures. For both the low heat flux and high heat flux cases, the non-uniform profile results in a temperature drop across the wall at the tube exit, $\Delta T_{w(ex)}$, that is two and a half times that of the equivalent case with uniform heat flux. It follows that the maximum thermal stress is two and a half times higher in the non-uniform heat flux case.

5. Sensitivity Analysis

A preliminary set of runs for the reference data in Table 1 showed that the maximum heat flux capability is relatively insensitive to the choice of inlet pressure for a fixed structural safety factor and a fixed fractional pressure drop over a range of inlet pressures from about $40. \times 10^5$ to $80. \times 10^5$ Pa (about 600 to 1200 psia). At lower inlet pressures bulk boiling occurs in so many cases that these pressures are not considered further. Since we wish to minimize the hoop stress so as to maximize the allowable thermal stress component and thereby maximize the allowable heat flux, a low inlet pressure of $40. \times 10^5$ Pa (about 600 psia) has been used throughout the remainder of this study.

In order to find the optimum tube wall thickness, δ_w , for this inlet pressure, computer runs were made for an incident heat flux to one side of the tubular wall of 3300 W/cm^2 and all the reference conditions listed in Table 1, except tube wall thickness. Fig. 8 shows that the optimum δ_w is about 0.25 mm and yields a maximum structural safety factor (SF) of about 5.8. For our reference case, we chose $\delta_w = 0.5 \text{ mm}$ as a practical compromise value for ease in manufacturing and handling in assembly and disassembly, and for some margin of safety against sputtering and erosion. This choice can be seen to lower the SF to about 4.1 and to result in a thermal stress which is about four times the hoop stress. If experience shows that a tube wall thinner than 0.5 mm is practical, it would allow us to raise the SF or the heat flux or both.

Fig. 9 shows the effects on the SF of uncertainties or changes with time and neutron bombardment of the various tube material properties. The properties considered are the yield stress σ_y , the coefficient of thermal expansion α , the modulus of elasticity E , and the tube wall thermal conductivity, k_w . These results can be used to estimate the minimum SF likely to occur during the life of the first wall shield.

The final figures show the sensitivity of our results to various empirical constants and equations used to model the SCB (subcooled nucleate boiling) flow regime.

Three models of the pressure gradient in the SCB regime were used in three versions of the code (refer to Appendix A for details):

Version a: Owen & Schrock's model for low pressures (100 to 400 psia) with the length of the SCB regime repetitively estimated based on local conditions. (Ref. 3 p. 91)

Version b: Owen & Schrock's model for low pressures (100 to 400 psia) but with the length of the SCB regime now held constant at the value estimated at the beginning of the SCB regime.

Version c: Tong's recommendation of a modified Mendler-Martinelli-Nelson model for medium pressures (800 to 1850 psia) with all parameters based on local conditions (Ref. 3).

Up to now we have used Version c of the computer code exclusively,

because it appears to us to be the more reasonable model for our flow conditions where the fractional pressure drops may easily exceed 50%. However, our selected pressure of $40. \times 10^6$ Pa (~ 600 psia) falls between the pressure ranges for the two basic models. Consequently, it was decided to investigate the difference between the results of the two models.

In Fig. 10 results are shown for the three versions of the code for $G = 5.5 \times 10^4$ kg/m² -s (and the other reference conditions of Table 1). The relatively small effect on the effective heat transfer coefficient, h_{eff} , and the much larger effect on the effective friction coefficient, f_{eff} , are illustrated. Also shown is a calculation for an artificial non-boiling (NOB) case; i.e. a case where the possibility of subcooled boiling is ignored in the calculation.

Version a of the Owen & Schrock empirical pressure gradient equation (and to a lesser extent, Version b,) shows a very rapid increase in SCB pressure gradient and hence f_{eff} as bulk boiling is approached. This behavior seems unrealistic because pressure gradients over 13 times the non-boiling pressure gradient are predicted near the end of the SCB regime. Version c predicts values only about 2 to 3 times the NOB values. Version c seems more reasonable because the pressure gradients in the very low quality bulk-boiling regime are only about 1.5 to 3.0 times the NOB values (Ref. 3), and we expect a smooth transition from the SCB regime to the

bulk-boiling regime. This probably means that the Owen & Schrock empirical equation should be used only for high degrees of subcooling near the beginning of the SCB regime.

The behavior of Versions a and b of the code is illustrated even more dramatically by the runs shown on Figure 11 at a slightly higher G ($G = 6.0 \text{ kg/m}^2 \text{ -s}$). Version a predicts bulk boiling at about 88 cm (of the 100 cm tube length); Version b predicts 36°C subcooling at the exit and a very rapidly falling pressure; and Version c predicts about 176°C subcooling at the exit and a rate of pressure decrease only about twice the NOB case. (The subcooling, ΔT_{sub} , and other temperature differences referred to in this study are defined in Figure 12.)

For the conditions of Figure 11, Version b predicts a critical heat flux at the exit, $q_{\text{crit(ex)}}$, of 2600 W/cm^2 , less than the 3300 W/cm^2 incident on the tubular wall, while Version c predicts a q_{crit} of 9600 W/cm^2 . In addition, the required pumping power predictions vary over a factor of four for the runs of Fig. 11. These large differences due to the SCB pressure gradient models indicate the necessity of experimental verification of predictions before the design and construction of a high heat flux structure of this type is undertaken.

At the present time we feel that the pressure gradient model of Version c recommended by Tong (Ref. 3) is probably the most believable one for our high flow rates and high fractional pressure drops.

In the SCB regime, in addition to the uncertainties in the empirical equations for the pressure gradient, there are also uncertainties in the

form of the empirical equation for the difference between the inner tube wall temperature and fluid saturation temperature, $\Delta T_{\text{sat}} \propto T_{\text{wi}} - T_{\text{sat}}$.

We have chosen to use the Jens and Lottes form (Ref. 4), since it seems to agree best with available data in the fully-developed SCB regime in our pressure range ($40 - 80 \times 10^5 \text{ Pa} \approx 600 \text{ to } 1200 \text{ psia}$).

We have made a preliminary sensitivity analysis of this Jens and Lottes equation by multiplying the equation by factors ranging from 0.6 to 1.4. The resulting film temperature drop, ΔT_{film} , varied only $\pm 10\%$ for the $\pm 40\%$ variation in ΔT_{sat} . This shifts the fully-developed SCB asymptotic curve a small amount and also changes its slope somewhat (the nearly vertical line on the log-log plot of Fig. 3). Consequently, some experimental verification of the Jens and Lottes empirical equation in our very high heat flux regime would be highly desirable, but it is not as crucial as the SCB pressure gradient verification.

The final empirical equation we have examined is the Rousar correlation (Ref. 5) for the critical heat flux of water coolant in the highly subcooled flow regime where $u\Delta T_{\text{sub}} \gtrsim 850^\circ\text{C} \cdot \text{m/s}$. This correlation is based on some very careful high-heat-flux experiments with water coolant run at the Aerojet-General Corp. for uniform axial and circumferential heat fluxes. It has been shown in Section 2 to be about 20% conservative (i.e. on the low side) compared to the NASA data for an axially non-uniform heat flux profile. Consequently, we can use this empirical equation with some reasonable confidence, but experimental verification is recommended for applications with axially and circumferentially non-uniform heat flux profiles.

6. Conclusions

This two-part study of the feasibility of a high-velocity, water-cooled first wall shield to handle very high energy fluxes from a fusion plasma has indicated the following:

- (a) With the use of a high-temperature alloy such as Ta-10W, heat flux capabilities as high as 3300 W/cm^2 seem entirely feasible in 1 meter long tubular wall sections (with tube inner diameter of 10 mm and wall thickness of 0.5 mm) with a structural safety factor of 4.0.
- (b) The pumping power requirements are considered reasonable for an early experimental fusion machine even in the non-boiling flow regime (less than 1 MW per square meter of first wall area exposed to the high energy flux).
- (c) Lower flow rates and pumping powers are required in the subcooled nucleate boiling regime (pumping power as much as an order of magnitude lower). However, uncertainties in the empirical equations used to model the subcooled boiling regime at these high mass flow rates and high heat fluxes, particularly the pressure gradient equations, indicate that some experimental verification of these predictions would be necessary before using the results in design work.

- (d) Even higher heat flux capabilities are possible with lower structural safety factors. For example, if the safety factor can be lowered from 4.0 to 3.0, the maximum allowable heat flux on the 1 meter long tube is increased from 3300 W/cm^2 to about 4200 W/cm^2 . However, the establishment of the required structural safety factor for this fusion reactor application must await much more detailed data on neutron damage and plasma sputtering and erosion.

APPENDIX A

Details of the Analysis

This Appendix discusses the details of the computer model for the case of the first wall shield, which is assumed to be made up of an array of closely packed tubes with an incident heat flux on one side only. (Appropriate changes were made in the computer code or the calculation of the cases of Ref. 2, which had circumferentially uniform heat flux.)

(1) Tube wall material and water coolant property data are input at several temperatures, and a least squares fit is used to provide a continuous variation of each property with wall (or bulk fluid) temperature.

(2) The total stress in the tube wall is estimated at each axial location as the sum of the local hoop stress and thermal stress:

$$\sigma_{TOT} = \sigma_{HOOP} + \sigma_{TH} \quad (A1)$$

$$\text{where } \sigma_{HOOP} = \frac{pD_i}{2\delta_w} \quad (A2)$$

$$\text{and } \sigma_{TH} = \frac{\alpha E}{2(1-\nu)} \Delta T_w \quad (A3)$$

These relations are valid for thin-walled tubes where $\delta_w/D \ll 1$.

(3) The temperature drop across the tube wall is calculated for the local value of heat flux at the point on the circumference of the tube facing the incoming energy flux, $q_{peak}(x) = q(x, \theta = 0^\circ)$:

$$\begin{aligned}\Delta T_w &= T_{w0} - T_{wi} \\ &\approx \frac{q_{\text{peak}}(x) \delta_w}{k_w}\end{aligned}\quad (\text{A4})$$

where $q(x, \theta) \approx q_1(x) \cos \theta$ $(-90^\circ \leq \theta \leq 90^\circ)$

and $q_1(x)$ is simply the axial variation of the incident heat flux.

(4) The local bulk temperature rise of the water coolant is calculated from the energy balance equation:

$$\Delta T_B = T_B - T_{B(in)} = \frac{\bar{q} \pi D_1 x}{\dot{m} C_p} \quad (\text{A5})$$

$$\text{where } \bar{q} = \frac{1}{2\pi x} \int_0^x \int_0^{2\pi} q(x, \theta) d\theta dx \quad (\text{A6})$$

$$\dot{m} = \frac{G \pi D_1^2}{4} = \text{constant}$$

$G = \rho u$ and u is the average flow velocity.

(5) The film temperature drop at any point along the tube length is calculated from the local axial heat flux at the $\theta = 0^\circ$ point:

$$\Delta T_{\text{film}} = T_{wi} - T_B = C_{\text{ASYM}} \frac{q_{\text{peak}}(x)}{h} \quad (\text{A7})$$

(6) The C_{ASYM} factor is a correction for the nonuniform circumferential heat flux distribution. A value of 1.3 has been used in this study as a first approximation; this is based on the analytical model of Reynolds (Int. J. Heat Mass Trans., 6, pp 445-454, 1963). Further

work is in progress to improve this correction factor.

(7) The convective heat transfer and friction coefficients have been evaluated for the NOB (non-boiling) flow regime using the following correlations:

$$h = 0.023 \frac{k}{D_i} Re_B^{0.8} Pr_B^{0.33} \left(\frac{\mu_B}{\mu_w} \right)^{0.14} \quad (A8)$$

$$f = 0.184 Re_B^{-0.2} \left(\frac{\mu_w}{\mu_B} \right)^{0.25} \quad (A9a)$$

Valid for $2000 \lesssim Re_B \lesssim 10^6$; $Pr_B \gtrsim 0.5$

$$\text{where } Re_B \equiv \frac{GD_i}{\mu_B}$$

$$\text{and } Pr_B \equiv \frac{\mu_B C_p}{k_B}$$

For $Re_B > 10^6$ the friction coefficient is calculated by

$$f = 0.0984 Re_B^{-0.155} \left(\frac{\mu_w}{\mu_B} \right)^{0.25} \quad (A9b)$$

(8) The pressure drop in the NOB flow regime is calculated for each axial step using the usual equation where gravity and inertia effects are negligible:

$$\Delta p = - \frac{f G^2}{2 \rho D_i} \Delta x. \quad (A10)$$

(9) The overall pumping power required for a single tube of length L is calculated from:

$$P_p = - \frac{\pi D_1^2 G}{4\rho} \Delta p_{TOT} \quad (A11)$$

(10) For the SCB (subcooled nucleate boiling) flow regime, the computer model uses the fully-developed SCB asymptotic curve (the nearly vertical line on the log-log plot of Fig. 3).

(11) At each axial location the code estimates the inner wall temperature for ONB, the onset of nucleate boiling.

$$T_{w1}(ONB) = T_{sat} + \Delta T_{sat} \quad (A12)$$

where the ΔT_{sat} used is the correlation of Jens and Lottes (Ref. 4) given here in SI units:

$$\Delta T_{sat} = 0.791 \left(q_{peak}(x) \right)^{0.25} \exp \left[-1.609 \times 10^{-7} p \right]. \quad (A13)$$

(12) The calculated value of $T_{w1}(ONB)$ is compared to the actual T_{w1} . If $T_{w1} < T_{w1}(ONB)$, the code continues with the NOB calculations. If $T_{w1} \geq T_{w1}(ONB)$, the code enters the SCB calculation routine at this axial location and T_{w1} is set equal to $T_{w1}(ONB)$.

(13) Three different versions of the code have been used, each incorporating a different model for the pressure gradient variation in the SCB regime.

(14) Versions a and b use the empirical equation of Owen and Schrock for $100 \lesssim p \lesssim 400$ psia ($7 \times 10^5 \lesssim p \lesssim 28 \times 10^5$ Pa) given in Tong (Ref. 3, p 91)

$$\left(\frac{dp}{dx}\right)_{SCB} = \left(\frac{dp}{dx}\right)_{NOB} \left[0.97 + .028 \exp \left(6.13 \frac{\Delta x_{SCB}}{\Delta L_{SCB}} \right) \right] \quad (A14)$$

where Δx_{SCB} is the axial distance from the ONB point to the present location, and ΔL_{SCB} is the total length of the SCB regime from the ONB to the OBB (onset of bulk boiling).

(15) In both Version a and b of the code, ΔL_{SCB} is estimated from an iterative solution of the following set of equations:

$$\Delta L_{SCB} = \frac{p_{OBB} - p_{ONB}}{K_{SCB} \left(\frac{dp}{dx}\right)_{NOB}} \quad (A15)$$

$$\text{where } K_{SCB} = \int_0^{\Delta L_{SCB}} \left[0.97 + .028 \exp \left(6.13 \frac{\Delta x_{SCB}}{\Delta L_{SCB}} \right) \right] dx = 3.06$$

$$p_{OBB} \approx p_{REF} \exp \left[\frac{T_{B(OBB)} - T_{REF}}{50.46} \right] \quad (A16)$$

$$T_{B(OBB)} = T_{B(ONB)} + \frac{4\bar{q}\Delta L_{SCB}}{GC_p D_i} \quad (A17)$$

(16) In Version a of the code, this set of three equations in ΔL_{SCB} , p_{OBB} and T_{OBB} is solved at each local value of x using local values of $T_{B(ONB)}$ and p_{ONB} .

(17) In Version b of the code, these three equations are solved only once, at the ONB point. Then ΔL_{SCB} is held constant at this value

for the remainder of the calculations in the SCB flow regime.

(18) For Version c of the code, the empirical equations recommended by Tong (Ref. 3, p 91-94) for the intermediate pressure range $800 \lesssim p \lesssim 1850$ psia ($54 \times 10^5 \lesssim p \lesssim 126 \times 10^5$ Pa) are used.

This is essentially a modification of a method suggested by Mendler, using the Martinelli-Nelson empirical equations for the bulk-boiling pressure gradients at an effective quality of 4.2%.

(19) We assume for Version c of the code:

$$\frac{\left(\frac{dp}{dx}\right)_{SCB}}{\left(\frac{dp}{dx}\right)_{NOB}} = 1 + \left[\frac{T_B - T_{B(ONB)}}{T_{sat} - T_{B(ONB)}} \right] \left[\phi_{LO}^2 - 1 \right] \quad (A18)$$

$$\text{where } \phi_{LO}^2 = \phi_{LO(M-N)}^2 \left[\frac{\phi_{LO}^2}{\phi_{LO(M-N)}^2} \right] \quad (A19)$$

using the notation of Tong.

In SI units, the empirical Martinelli-Nelson equation is:

$$\frac{\phi_{LO}^2}{\phi_{LO(M-N)}^2} = 1.26 - 5.80 \times 10^{-8} p + 161.65 \left(\frac{1}{G} \right) + 5.52 \times 10^{-5} \left(\frac{p}{G} \right) \quad (A20)$$

The following empirical fit to the data for ϕ_{LO}^2 at a quality of 4.2% has been used (SI units):

$$\frac{1}{\phi_{LO}^2} \approx 0.050 + 0.00426 \frac{p}{10^5} . \quad (A21)$$

The SCB pressure gradient is evaluated at each axial step using local values for T_B , $T_B(\rho_{NB})$, p and $T_{sat}(p)$.

(20) An effective local friction coefficient in the SCB regime is calculated at each step from:

$$f_{eff} = - \left(\frac{dp}{dx} \right)_{SCB} \left(\frac{2\rho D_1}{G^2} \right) \quad (A22)$$

The pumping power, P_p , is still calculated from Eq. (A11).

(21) An effective local heat transfer coefficient in the SCB regime is calculated from:

$$h_{eff} = C_{ASYM} \frac{q(x, \theta=0^\circ)}{(T_{wt} - T_B)} \quad (A23)$$

(22) If the OBB point is reached, i.e. $T_B \geq T_{sat}$, the calculation is terminated and all the data up to that point are printed out for axial increments of 0.01 L.

(23) For the prediction of the critical ("burnout") heat flux, the empirical equation of Rousar (Ref. 5) has been used:

$$q_{crit} = D_1 + D_2 \Delta T_{SUB}$$

where $\Delta T_{SUB} = T_B(x) - T_{sat}(x)$ and the constants for water are:

$$D_1 = 8.34 \times 10^6 \text{ (W/m}^2\text{)}$$

$$D_2 = 8.11 \times 10^3 \text{ (J/m}^3\text{ }^\circ\text{C)}$$

This equation is probably valid for $u\Delta T_{\text{SUB}} \gtrsim 850.^\circ\text{C} \cdot \text{m/s}$ according to Rousar.

(24) The code prints out all of the following for axial increments of 0.01 L:

- (a) The local pressure
- (b) The wall and coolant bulk temperatures
- (c) Actual and critical heat fluxes
- (d) Hoop, thermal and yield stress
- (e) Structural safety factor
- (f) Re , Pr , h_{eff} , f_{eff}
- (g) All tube material and coolant property data.

In addition, the overall fractional pressure drop, pumping power and thermal power are printed out.

(25) The code also incorporates the option of ignoring all subcooled boiling. This provides us with the (artificial) non-boiling performance to use in evaluating the effect of SCB on various parameters.

TABLE 1

Reference Case Parameters

Tube material	Ta-10W (tantalum + 10% tungsten)
Tube length, L	1.0 m
Tube inner diameter, D_i	10.0 mm
Tube wall thickness, δ_w	0.5 mm
Bulk inlet pressure, p_{in}	40.0×10^5 Pa (~ 600 psia)
Bulk inlet temp., $T_{B(in)}$	20. $^{\circ}$ C
Mass flow rate parameter, $G \equiv \rho u$	Variable
Heat flux profile	Uniform axially Non-uniform circumferential (Incident heat flux, q_i , uniform on one side of the tubular wall only).
Incident heat flux, q_i	Variable

TABLE 2
EFFECT OF NON-UNIFORM AXIAL HEAT FLUX

	"Low" Heat Flux Runs		"High" Heat Flux Runs	
	Uniform q	Non-Uniform q	Uniform q	Non-Uniform q
$q_{inlet} \left(\frac{W}{cm^2} \right)$	800	340	3,600	1,530
$\bar{q} \left(\frac{W}{cm^2} \right)$	800	800	3,600	3,600
$q_{exit} \left(\frac{W}{cm^2} \right)$	800	2,000	3,600	9,000
$q_{crit(ex)}$	9,730	9,720	9,650	9,560
$T_{B(ex)} (^{\circ}C)$	30.7	30.7	33.0	33.0
$T_{wi(ex)} (^{\circ}C)$	76	134.	206	262
δ_w (mm)	0.36	0.36	0.36	0.36
$\Delta T_w(ex) (^{\circ}C)$	37	92	170	425
$G(kg/m^2 - s)$	6.1×10^4	6.1×10^4	6.1×10^4	6.1×10^4
$p_{ex} (10^6 \text{ Pa})$	1.88	1.88	1.90	1.85
$\Delta p/p_{in}$	.0605	.0615	.048	.077
$\frac{P_p}{A_{FW}} \left(\frac{MW}{m^2} \right)$	1.08	1.10	0.85	1.38
NOB Portion	100%	100%	100%	First 68%
SCB Portion	0%	0%	0%	Last 32%

Fig. 1 Heat Flux Ranges for Various Applications

Fig. 2 Heat Flux Distribution for NASA Experiments

Fig. 3 Comparison of Theoretical Prediction to NASA Experiments

Fig. 4 Film Temperature Drop vs. Mass Flow Parameter

Fig. 5 Pumping Power vs. Mass Flow Parameter

Fig. 6 Structural Safety Factor vs. Mass Flow Parameter

Fig. 7 Allowable Heat Flux vs. Structural Safety Factor

Fig. 8 Optimum Tube Wall Thickness

Fig. 9 Sensitivity of Structural Safety Factor
to Material Property Variations

Fig. 10 Heat Transfer Coefficient and Friction Factor for the Various Subcooled Boiling Heat Transfer Models

Fig. 11 Pressure Drop for the Various Subcooled Boiling Heat Transfer Models

Fig. 12 Flow Regime and Temperature Definitions

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