



## **Boiling inside tubes: post-dryout heat transfer in vertical tubes**

Endorsed by  
The Institution of Mechanical Engineers  
The Institution of Chemical Engineers

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## BOILING INSIDE TUBES: post-dryout heat transfer in vertical tubes

### 1. NOTATION AND UNITS

|                 |                                                                                                             | <i>Units</i>               |                                                        |
|-----------------|-------------------------------------------------------------------------------------------------------------|----------------------------|--------------------------------------------------------|
|                 |                                                                                                             | <i>SI</i>                  | <i>British</i>                                         |
| $Bo$            | boiling number, see Equation (6.1)                                                                          | –                          | –                                                      |
| $c_p$           | specific heat capacity at constant pressure                                                                 | J/kg K                     | Btu/lb °F                                              |
| $D$             | tube internal diameter                                                                                      | m                          | ft                                                     |
| $Fr_l$          | Froude number, see Equation (6.2)                                                                           | –                          | –                                                      |
| $g$             | acceleration due to gravity                                                                                 | m/s <sup>2</sup><br>(9.81) | ft/h <sup>2</sup><br>(4.17 × 10 <sup>8</sup> )         |
| $g_c$           | constant of proportionality in Newton's second law of motion                                                | –<br>(1.0)                 | lbf h <sup>2</sup> /lb ft<br>(4.17 × 10 <sup>8</sup> ) |
| $h$             | specific enthalpy                                                                                           | J/kg                       | Btu/lb                                                 |
| $\dot{M}$       | gas (vapour) flow rate                                                                                      | kg/s                       | lb/h                                                   |
| $\dot{m}$       | mass flux                                                                                                   | kg/m <sup>2</sup> s        | lb/ft <sup>2</sup> h                                   |
| $Pr$            | Prandtl number                                                                                              | –                          | –                                                      |
| $\dot{q}$       | heat flux                                                                                                   | W/m <sup>2</sup>           | Btu/h ft <sup>2</sup>                                  |
| $\dot{q}_{rad}$ | radiation heat flux                                                                                         | W/m <sup>2</sup>           | Btu/h ft <sup>2</sup>                                  |
| $Re$            | Reynolds number                                                                                             | –                          | –                                                      |
| $T$             | absolute temperature                                                                                        | K                          | °R                                                     |
| $U$             | overall heat transfer coefficient                                                                           | W/m <sup>2</sup> K         | Btu/h ft <sup>2</sup> °F                               |
| $U'$            | combined heat transfer coefficient for heating fluid, tube wall and fouling, referred to inner surface area | W/m <sup>2</sup> K         | Btu/h ft <sup>2</sup> °F                               |
| $x$             | quality                                                                                                     | –                          | –                                                      |
| $x_c$           | critical quality                                                                                            | –                          | –                                                      |
| $\alpha$        | heat transfer coefficient (inside tube)                                                                     | W/m <sup>2</sup> K         | Btu/h ft <sup>2</sup> °F                               |
| $\alpha'$       | heat transfer coefficient for low quality region (inside tube)                                              | W/m <sup>2</sup> K         | Btu/h ft <sup>2</sup> °F                               |

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|                 |                                                 |                                                               |                                                                      |
|-----------------|-------------------------------------------------|---------------------------------------------------------------|----------------------------------------------------------------------|
| $\Delta h_v$    | specific enthalpy of vaporisation               | J/kg                                                          | Btu/lb                                                               |
| $\varepsilon$   | non-equilibrium parameter, see Equation (3.3)   | —                                                             | —                                                                    |
| $\varepsilon_g$ | homogeneous void fraction                       | —                                                             | —                                                                    |
| $\varepsilon_w$ | emissivity of tube wall                         | —                                                             | —                                                                    |
| $\eta$          | dynamic viscosity                               | kg/s m                                                        | lb/ft h                                                              |
| $\lambda$       | thermal conductivity                            | W/m K                                                         | Btu/h ft °F                                                          |
| $\rho$          | density                                         | kg/m <sup>3</sup>                                             | lb/ft <sup>3</sup>                                                   |
| $\sigma$        | surface tension                                 | N/m                                                           | lbf/ft                                                               |
| $\sigma_s$      | Stefan–Boltzmann radiation constant             | W/m <sup>2</sup> K <sup>4</sup><br>(56.7 × 10 <sup>-9</sup> ) | Btu/h ft <sup>2</sup> °R <sup>4</sup><br>(1.713 × 10 <sup>-9</sup> ) |
| $\psi$          | parameter related to $\psi_o$ , see Section 5.1 | —                                                             | —                                                                    |
| $\psi_o$        | parameter defined by Equation (5.3)             | —                                                             | —                                                                    |

## Extra subscripts

|     |                              |
|-----|------------------------------|
| $a$ | actual conditions            |
| $e$ | at thermodynamic equilibrium |
| $f$ | at film temperature          |
| $g$ | gas or vapour                |
| $H$ | heating fluid                |
| $l$ | liquid                       |
| $s$ | under saturation conditions  |
| $w$ | at wall temperature          |

## 2. PURPOSE AND SCOPE OF THIS DATA ITEM

This is part of a series of Data Items on heat transfer in boiling of single component fluids (that is pure fluids and not mixtures) inside tubes as follows.

ESDU 85041<sup>21</sup> discusses the problems and gives a procedure for estimating heat transfer rates in saturated wet-wall boiling.

ESDU 86032<sup>22</sup> gives a procedure for estimating the critical heat flux (often referred to as CHF) in upward flow in uniformly heated vertical tubes. The phenomenon at the critical heat flux has been termed 'dryout', 'burnout', or 'boiling crisis'.

ESDU 88001<sup>26</sup> gives a procedure for estimating the critical heat flux in uniformly heated horizontal tubes.

This Data Item considers the problem of calculating the heat transfer rate in the dry wall region when the critical heat flux has been exceeded. This is usually known as post-dryout heat transfer. Attention is here restricted to post-dryout heat transfer in upflow in uniformly heated vertical tubes. Comparatively little is known about the situation in inclined or horizontal tubes. However, a major difference in horizontal flow is that, unlike the case of vertical flow, there is no sudden transition from wetted wall flow to post-dryout flow; instead the transition occurs over a significant length of the tube. Attention is also restricted here to cases where the dryout, and therefore the post-dryout region, occurs when most of the liquid has been evaporated (that is at high quality). Further details of this restriction are given in Section 3.

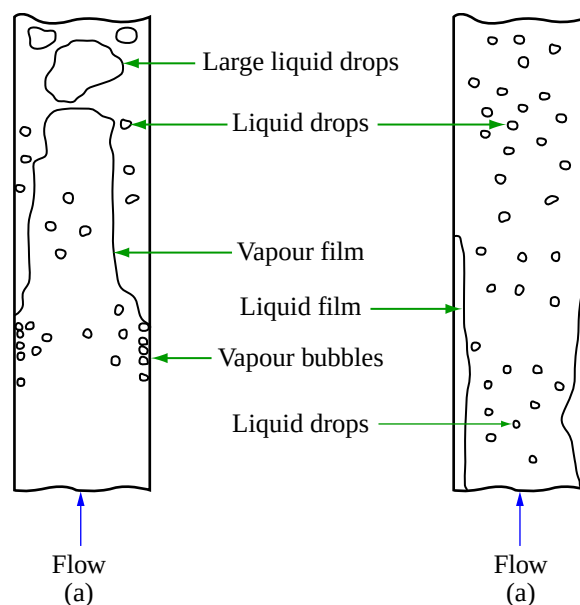
Normally the liquid is in contact with the heating surface when a liquid is heated and vapour is formed, and the boiling heat transfer coefficient is high. This is the situation in normal saturated wet-wall boiling, see ESDU 85041<sup>21</sup>. However, if the heat flux is increased sufficiently, a dramatic change in the type of heat transfer takes place. At some heat flux the liquid is no longer in contact with the heating surface; the boiling heat transfer coefficient is then quite low and one of the following situations will arise.

- (1) If the surface is heated by a source of constant heat flux (as in electrical heating or in a nuclear reactor) the temperature of the heated surface increases very rapidly. A fossil fired boiler is, to a good approximation, a constant heat flux situation.
- (2) If the surface is heated by a hot fluid approximating to a source of constant temperature, as in a heat exchanger or a waste heat boiler, the rate of heat transfer drops dramatically.

This phenomenon of a sudden change in the heat transfer is variously termed burnout, dryout, boiling crisis, or critical heat flux phenomenon. The value of the critical heat flux can be calculated from ESDU 86032<sup>22</sup> (for vertically upwards flow) or ESDU 88001<sup>26</sup> (for horizontal flow).

## 3. DISCUSSION ON POST-DRYOUT HEAT TRANSFER

When the critical heat flux (CHF) has been exceeded, it is useful to consider what the flow looks like. This is best done by initially considering the flow before the CHF has been exceeded. Conventionally, CHF has been divided into two categories: that occurring at sub-cooled or low quality conditions, and that occurring at relatively high quality conditions. Quality here is defined as the fraction of the total mass flow rate that is vapour. Thus a low quality or sub-cooled flow contains relatively little vapour, whereas the behaviour of a high quality flow is dominated by the relatively large amount of vapour present. The boundary between low and high quality flow is uncertain – ranging from 2% or 3% for systems at atmospheric pressure up to possibly 30% for systems where the pressure approaches the critical pressure. The significant difference between the two types of critical heat flux is the two-phase flow pattern present at the time, as shown in Sketch 3.1 and the associated discussion.



**Sketch 3.1 Typical flows around the critical heat flux point:**  
**(a) low quality flows – inverted annular flow**  
**(b) high quality flows – dispersed droplet flow**

- (1) At sub-cooled or low quality conditions, prior to CHF, the flow is almost always bubbly flow, as shown in the lower part of Sketch 3.1(a). As CHF is reached, the heat transfer mechanism breaks down, usually because the high density of bubbles near the heated surface prevents sufficient liquid reaching the heated surface to cool it. Therefore, after the dramatic change in heat transfer, the flow is commonly of the inverted annular form. There is then an unstable core of liquid separated from the heated wall by a thin film of vapour. The heat transfer in this region after the critical heat flux has been exceeded – the post-dryout region – occurs now by conduction and by radiation through this vapour film. Because the liquid core is unstable it touches the wall at intervals, and this causes partial breakdown to form some very large liquid drops.

- (2) At high quality conditions, the flow prior to CHF is almost always annular flow, as shown in the lower part of Sketch 3.1(b). The main characteristics of annular flow can be seen from this sketch: the liquid flows partly as a film on the heated walls of the channel and partly as drops entrained into the gas which flows along the centre of the channel. The liquid film is not smooth, but has large disturbance waves on it. As CHF is reached, the heat transfer mechanism breaks down because the liquid film has been evaporated away and the walls of the tube are left dry. The heat transfer in the post-dryout region then occurs by convection to the flowing vapour, which rapidly becomes superheated. The liquid drops remaining in the flow are very close to the local saturation temperature. Heat is therefore transferred from the superheated vapour to the relatively cold drops, and so the drops slowly evaporate. This heat transfer region is sometimes known as the 'liquid deficient' region or dispersed post-dryout region. In the constant heat flux case, the wall temperature can become very large – well above the Leidenfrost temperature – so that the drops cannot wet the wall. The consequence of this is that, as stated previously, the heat transfer occurs by convection to the vapour. In the fluid heated case, the wall temperature is limited by the heating fluid temperature, and so direct heat transfer from the wall to the drops may be more important.

This Data Item is concerned mainly with the second of these two types of post-dryout heat transfer – the region of relatively high quality. This is the relevant region for a well designed piece of industrial equipment. A once-through boiler, for example, will usually be designed so that CHF occurs at high quality. This is because the heat transfer is much more efficient (that is, the heat transfer coefficient is greater over a greater length of the boiler tube) than if CHF occurs at low qualities. Although concerned mostly with the high quality region, this Data Item also includes a checking procedure, in which a method applicable at low quality is used to verify that the flow really is in the high quality region.

One of the most comprehensive reviews on the subject of post-dryout heat transfer is given by Groeneveld and Snoek<sup>24</sup>; however this is rather selective because they concentrate on the calculation methods available. Collier<sup>16</sup> adds details of the experimental information available. He summarises the experimental work and divides the available correlations into three groups: purely empirical correlations, non-equilibrium methods and correlations, and semi-theoretical models. A recent, but more restricted review, is given by Chen<sup>23</sup>.

## 3.1 Purely Empirical Correlations

These are widely used, being convenient to apply, see Dougall and Rohsenow<sup>2</sup>, Groeneveld<sup>7</sup> and Slaughterbeck *et al*<sup>9</sup>. They are mostly based on the simple single-phase Dittus-Boelter equation, but often include extra terms of an arbitrary nature; consequently extrapolation outside the range of applicability can produce absurd results. The equations are based almost exclusively on steam-water data and make no pretence of applicability to other fluid systems.

## 3.2 Non-equilibrium Methods and Correlations

These methods are based on the difference between the actual quality and the thermodynamic quality. The actual quality,  $x_a$ , is the quality based on the actual flow rates of the phases; thus

$$x_a = \frac{\dot{M}_g}{\dot{M}_g + \dot{M}_l}, \quad (3.1)$$

where  $\dot{M}_g$  and  $\dot{M}_l$  are the actual flow rates of the liquid and the vapour phases. The thermodynamic quality,  $x_e$ , is the quality based on the thermodynamic equation

$$x_e = \frac{h - h_{l,s}}{\Delta h_v}, \quad (3.2)$$

where  $h$  is the fluid mean specific enthalpy,  $h_{l,s}$  is the specific saturation enthalpy of the liquid and  $\Delta h_v$  is the specific enthalpy of vaporisation. The maximum value of  $x_d$  is unity, but there is no reason why  $x_e$  cannot be larger than unity, as would certainly be the case for a single-phase flow of superheated vapour. Differences between these two qualities indicate a degree of thermodynamic non-equilibrium in the flow.

The non-equilibrium type of correlation is still empirical but the empirical nature is at least controlled within a physically sensible framework. One common way of proceeding is to use a modified Dittus-Boelter equation to calculate the heat transfer coefficient; however, use of the actual quality means that the final result is effectively constrained within reasonable limits.

It is worth noting that, strictly, these non-equilibrium methods should involve an integration along the tube length, and in this way involve a history effect. However, by various approximations, this type of correlation can often be written in a purely 'local' form; that is, dependent only upon local variables and not on the history of the flow.

There have been many examples of this type of approach, see for example: Dougall and Rohsenow<sup>2</sup>, Lavery and Rohsenow<sup>5</sup>, Forslund and Rohsenow<sup>3</sup>, Hynek *et al*<sup>6</sup>, Plummer *et al*<sup>8, 12</sup>, Rohsenow<sup>19</sup>, Groeneveld and Delorme<sup>11</sup>, Groeneveld and Snoek<sup>24</sup>, Jones and Zuber<sup>13</sup>, Hein and Kohler<sup>20</sup>, Saha<sup>14</sup> and Shah<sup>15</sup>.

### 3.3 Semi-theoretical Models

These physical models or correlations attempt to write equations for the heat transfer and flow processes occurring. In some cases the actual processes occurring are not very complicated, for example the heat transfer from the superheated vapour to a saturated liquid drop is characterised by a fairly well established equation (as long as the drop is assumed spherical). Although attractive, this approach rapidly leads to a complicated set of equations. The equations are usually in the form of set of non-linear differential equations which have to be numerically integrated simultaneously along the length of the flow. Although not suitable for the present purpose, it is probably true that in the long term substantial progress will be made using this type of approach.

Examples of this type of model are those of Bennett *et al*<sup>4</sup>, Iloeje *et al*<sup>10</sup>, and Whalley *et al*<sup>17</sup>. Although these models predict many details of the flow, such as drop velocities, drop sizes and phase temperatures, the only substantial experimental results available for comparison are usually the wall temperatures. In this type of approach there are many problems; for example, the importance of direct heat transfer between the wall and the drops (when the drops touch the wall) is a matter of some debate. Chen<sup>18</sup> reports that such heat transfer is included in some models, but it has not been measured experimentally, and there is no reason to suppose it is important.

### 3.4 Tabular Methods

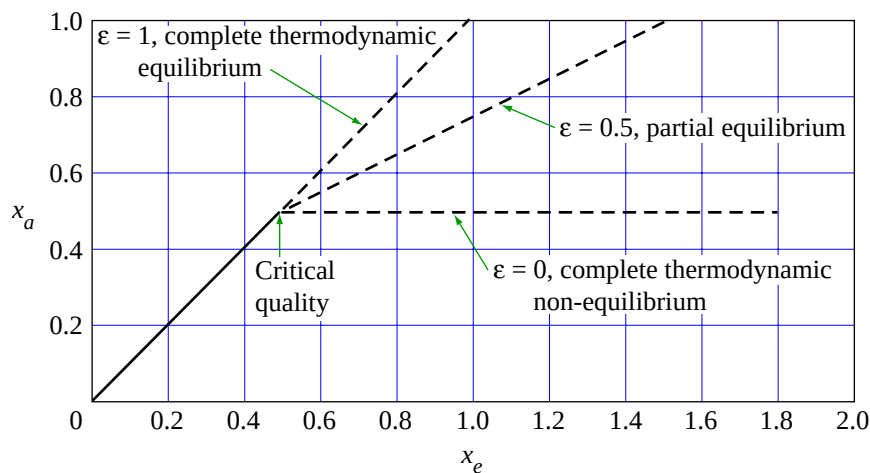
Groeneveld<sup>25</sup> has recently discussed the production of a tabular method for the calculation of post-dryout wall temperature and heat fluxes. Tabular methods are well known for critical heat flux calculations, and have indeed been adopted by ESDU in ESDU 86032<sup>22</sup>. For post-dryout heat transfer it seems highly likely that the tables will only be available for the water-steam system. For various pressures, mass fluxes and qualities, the wall superheat (that is wall temperature minus the saturation temperature) will be tabulated against the heat flux. Thus the defining of four variables enables the fifth to be found from the tables. This is even more complicated than for the critical heat flux calculation where the defining of three variables (pressure, mass flux, quality) enables the fourth (critical heat flux) to be found. It is thus inevitable that because of this extra dimension in the table there will have to be correspondingly more interpolation between the experimental data to produce the table.

## 3.5 Selection of General Method for Use in this Data Item

Here the approach using non-equilibrium methods and correlations (see Section 3.2) has been selected; it has the advantage that it is not completely empirical, but still retains some physical limits to the processes by use of the two qualities. This is very important in the present work because it holds out the promise that a method can be selected or devised which will work for any fluid system, not just for the well researched steam-water system.

## 3.6 Further Details of the Non-equilibrium Method

The relevance of the two qualities in the non-equilibrium method as already described in Section 3.2 is that they help to define two limiting and bounding conditions, see Sketch 3.2.



**Sketch 3.2 Variation of quality (actual and thermodynamic) in post-dryout flow**

- (1) The first bounding condition is that of complete thermodynamic equilibrium. This is when the vapour and the liquid drops are both at the saturation temperature. All the heat input is used to evaporate the liquid drops and none of it is used to raise the vapour temperature. Thus the heat transfer from the heated wall to the vapour can be calculated as if the vapour were flowing in single-phase flow in the channel. Then the only effect of the liquid drops will be to increase the velocity of the vapour slightly. Complete thermodynamic equilibrium tends to occur when the system pressure approaches the thermodynamic critical pressure or the mass flux of the fluid is high.
- (2) The other limiting condition is that of complete non-equilibrium. Thermodynamic equilibrium is maintained up to the point where the critical heat flux is exceeded; at this point the quality is the critical quality,  $x_c$ . However, in the post-dryout region it is possible to assume that all the heat is going to superheat the vapour. Under these conditions, the actual quality,  $x_a$ , remains constant and equal to  $x_c$ . It is of course the actual quality which determines the phase flow rates. Therefore the heat transfer from the heated wall to the vapour can once again be calculated as if the vapour were flowing in single-phase flow in the channel. Complete non-equilibrium tends to occur when the pressure or the mass flux is low.

The important differences between the two limiting cases as far as the heat transfer calculation is concerned are the vapour flow rate and the temperature of the vapour flow. A parameter,  $\epsilon$ , characterising the degree

of non-equilibrium can be defined:

$$\varepsilon = \frac{x_a - x_c}{x_e - x_c} . \quad (3.3)$$

For complete thermodynamic equilibrium the actual quality,  $x_a$ , is always equal to the thermodynamic quality,  $x_e$ , and so from Equation (3.3)  $\varepsilon = 1$ , subject to the limitation that  $x_a$  cannot exceed 1. For complete thermodynamic non-equilibrium the actual quality does not rise after the critical heat flux has been reached, and so  $x_a = x_c$ . Thus, from Equation (3.3) in this case,  $\varepsilon = 0$ . These two limiting cases are shown in Sketch 3.2, together with a more realistic case of partial non-equilibrium defined in this particular case by  $\varepsilon = 0.5$ .

Three major simplifications to the general problem can be made. These are that radiative heat transfer can usually be neglected, that the contribution of direct heat transfer between a drop and the wall is usually small, and that the drops and the superheated vapour usually flow with the same velocity; this latter assumption is usually called homogeneous flow. The inclusion of radiative heat transfer presents no fundamental difficulty, but can complicate the calculation considerably as it can necessitate a further iteration loop. The method for the inclusion of radiation heat transfer is discussed further in Section 5.3. Support for these simplifications is given by Groeneveld and Delorme<sup>11</sup>.

## 4. AVAILABLE DATA

### 4.1 Steam-water Systems

The available data on post-dryout heat transfer are dominated by steam-water data, mostly of the points collected in connection with nuclear reactor safety studies. What is surprising is the quantity of data available: certainly over 20 000 points. One reference alone, Becker *et al*<sup>39</sup>, contains over 10 000 points. Most of these data points are for experimental systems heated electrically at constant heat flux. The number of points for fluid heated systems is small.

In view of the large number of previous studies, the task of deriving a special calculation method for steam-water from this vast amount of data would be formidable. It was considered preferable here to select an appropriate existing non-equilibrium method. The important factors governing the selection of a suitable calculation method are as follows.

- (a) It should be a local conditions type method and not a history-dependent method involving an integral. This requirement is essential for a convenient calculation method.
- (b) It should be, in other ways, convenient to use so that, for example, it does not depend on graphical interpolations and constructions but on algebraic equations.
- (c) Some methods require information about dryout. The need to have this information makes the method more difficult to use, but probably improves its accuracy.
- (d) It should, if possible, be written in non-dimensional form.
- (e) It should have been well tested, preferably by someone other than its originators.
- (f) It should be accurate.
- (g) A reasonable estimate of the errors produced by the method should be available.

The result of applying these selection criteria to the methods listed in Section 3.2 was that the method of Groeneveld and Delorme<sup>11</sup> was probably the most tested, widely used and accurate and it was convenient to use.

This method is therefore set out in Section 5, and is applicable to the high quality region. A check is also included to see that the high quality region is the appropriate one. This check is performed by using a method (Berenson<sup>1</sup>) applicable only to the low quality region: it is included solely to provide a lower bound to the heat transfer coefficient calculations.

## 4.2 Non Steam-water Systems

For other fluids there is a limited quantity of data available for:

|          |                                                                                                                    |
|----------|--------------------------------------------------------------------------------------------------------------------|
| hydrogen | (Hendricks <i>et al</i> <sup>27, 29</sup> ),                                                                       |
| nitrogen | (Lavery and Rohsenow <sup>30</sup> , Forslund and Rohsenow <sup>28</sup> , and Hynek <i>et al</i> <sup>31</sup> ), |
| methane  | (Hynek <i>et al</i> <sup>31</sup> ),                                                                               |
| propane  | (Hynek <i>et al</i> <sup>31</sup> ),                                                                               |
| R-12     | (Cummo <i>et al</i> <sup>32</sup> , Schnittger <sup>37</sup> ),                                                    |
| R-113    | (Ueda <i>et al</i> <sup>33</sup> , Koizumi <i>et al</i> <sup>35</sup> ),                                           |
| R-22     | (Nishikawa <i>et al</i> <sup>36</sup> ).                                                                           |

These data are all for electrically heated constant heat flux experiments, except for Cummo *et al*<sup>32</sup>, in these latter experiments the R-12 was heated by a hot fluid, but even here the heat flux was almost constant.

The difficulties of comparing data and correlations for these fluids are usually concerned with the availability of physical property information. The properties needed are enthalpies (and therefore specific heat capacities), vapour viscosity, vapour thermal conductivity and surface tension. Sometimes these are required far from the saturation line (for a highly superheated vapour) and this can cause problems. For steam-water systems there are, of course, extensive tabulations of the required properties.

The most complete comparison of a calculation method with experimental data has been carried out by Shah<sup>15</sup>. This comparison involved most of the data mentioned above, and was only carried out against the author's own correlation. Since this gave good results for all the fluids for which data are available EXCEPT HYDROGEN, the Shah<sup>15</sup> correlation is used here, see Section 6. For the special case of hydrogen, Shah recommends a particular modification of his correlation which is not quoted here. It can also be noted that the Groeneveld and Delorme<sup>11</sup> method given in Section 5 can also be used with fluids other than water, but it has not been tested as thoroughly as the Shah method; some details are given in Section 7.2.

## 5. PROCEDURE FOR STEAM-WATER SYSTEMS

The recommended calculation method of Groeneveld and Delorme<sup>11</sup> is set out for steam-water. Two versions are given:

- A – calculation of the wall temperature, and
- B – calculation of the heat flux.

In each method it should be checked that the critical heat flux has indeed been exceeded. This is best done by reference to ESDU 86032<sup>22</sup>.

Much of the method has been taken directly from Groeneveld and Delorme: for example the excessive number of significant figures in Equation (5.3) are quoted by these authors.