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THE CONDENSATION OF STEAM ON THE EXTERNAL SURFACES OF THE SHELLS OF HIFAR HEAVY WATER HEAT EXCHANGERS DURING A LOSS-OF-COOLANT ACCIDENT

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ABSTRACT

A study of steam condensation rates on the HIFAR heavy water heat exchangers was undertaken to predict thermohydraulic conditions in the HIFAR containment during a postulated loss-of-coolant accident (LOCA). The process of surface condensation from a mixture of air and steam, and methods for calculating the rate of condensation, are briefly reviewed. Suitable experimental data are used to estimate coefficients of condensation heat transfer to cool surfaces in a reactor containment during a LOCA. The relevance of the available data to a LOCA in the HIFAR materials testing reactor is examined, and two sets of data are compared. The differences between air/H_2O and air/D_2O mixtures are discussed. Formulae are derived for the estimation of the coefficient of heat transfer from the heat exchanger shells to the cooling water, and a method of calculating the rate of condensation per unit area of surface is developed.

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AIR; EXPERIMENTAL DATA; HEAT EXCHANGERS; HEAT TRANSFER; HEAVY WATER; HIFAR REACTOR; LOSS OF COOLANT; REACTOR COOLING SYSTEMS; STEAM; VAPOUR CONDENSATION.

EDITORIAL NOTE

From 27 April 1987, the Australian Atomic Energy Commission (AAEC) is replaced by Australian Nuclear Science and Technology Organisation (ANSTO). Serial numbers for reports with an issue date after April 1987 have the prefix ANSTO with no change of the symbol (E, M, S or C) or numbering sequence.

CONTENTS

1. INTRODUCTION

In a hypothetical loss-of-coolant accident (LOCA) in the materials testing reactor HIFAR, it is postulated that the primary coolant escapes into the plant room from a break or leak in the system. Heat from the decay of fission products in the fuel elements boils the coolant draining from the reactor tank, and D_2O vapour is discharged into the plant room through the break. Any fuel that melts and finds its way down into the plant room heats the escaped coolant on the plant room floor and produces more vapour. Gaseous and volatile fission products from melted or damaged fuel elements mix with vapour and air in the plant room. The destination of these fission products depends on the build-up of pressure in the plant room and on the rate of deposition on surfaces. Both of these factors are affected by the rate at which vapour condenses on the cool surfaces, particularly those of the space conditioners and the shells of the heat exchangers. This work examines the means by which the rate of condensation on the heat exchanger shells may be estimated.

2. SURFACE CONDENSATION FROM AIR/STEAM MIXTURES

When a mixture of air and steam comes into contact with a surface which is at a temperature below the dew point of the mixture (*i.e.* the saturation temperature of the steam corresponding to its partial pressure in the mixture), heat is transferred and steam condenses to form a film of water on the surface. The conduction of heat through this water film governs the rate at which condensation continues [Nusselt 1916]. The residual air forms a gas layer separating the mixture from the water film, across which steam must diffuse before further condensation can occur. A difference of steam partial pressure is necessary for this diffusion and the dew point of the mixture at the surface of the water film is lower than in the bulk of the mixture. The consequent reduction of the temperature difference producing a heat flow through the water film is highly significant; rates of condensation from mixtures containing only a small proportion of air are considerably lower than those from pure steam and large proportions of air result in very low rates of heat transfer. Many textbooks quote the example of a power plant steam condenser in which the overall coefficient of heat transfer at the outlet (where the concentration of air is high) is only one per cent of that at the steam inlet (where the concentration of air is low).

Various analytical methods for calculating the rate of condensation of vapours from mixtures of vapour and non-condensable gas, with either natural or forced convection, are described in the literature [Colburn and Hougen 1934; Kern 1950; Sparrow and Lin 1963; Rohsenow and Hartnett 1973]. All are complicated, indirect methods requiring the simultaneous solution of at least two equations, and none has been verified by experiment for mixtures containing more than 3.5 vol.% of non-condensable gas. These analytical methods are therefore unsuitable for calculations which have to be repeated a great many times, as in the stepwise analysis of the transient processes consequent upon a LOCA, or for mixtures containing a large proportion of non-condensable gas. The only alternative methods for predicting rates of condensation from air/steam mixtures involve a direct reference to experimental data and the application of an empirical formula. Because of the complex nature of the heat transfer process, any empirical formula must be highly specific, *i.e.* the conditions of the experiments on which it is based must closely resemble those of the application.

The scope of experimental work on condensation from air/steam mixtures has traditionally been confined almost exclusively to steam atmospheres containing traces of air, because in the great majority of industrial applications the air is an unavoidable contaminant which does not occur in large concentrations. The data of Othmer [1929], which have for long been regarded as setting the standard, cover air concentrations between 1 and 3.5 vol.%. Very few data are to be found for the range of air concentrations likely to occur in reactor containments under postulated accident conditions; however, the experiments of Uchida *et al.* [1964] and Thompson [1970] were directly related to reactor accident analysis. The data of Uchida *et al.* (hereafter referred to as Uchida) cover air concentrations between 6 and 97 vol.% and those of Thompson between 15 and 80 vol.%.

There is no doubt that direct reference to experimental data is the most satisfactory means of predicting the rate of condensation from air/steam mixtures on cool surfaces in a reactor containment during a LOCA when a time-step sequential calculation procedure is required. The relevance of the available data to conditions within the HIFAR containment is considered in **section 3**.

3. RELEVANCE OF THE EXPERIMENTAL DATA TO A LOCA IN HIFAR

The principal variables affecting condensation heat transfer from air/steam mixtures are the temperatures of the mixture and cooling surface, the vertical length of the surface, and the convection system (natural or forced). The temperatures control the conduction of heat through the condensate film and, in saturated mixtures, also control the partial pressures which govern the diffusion of steam through the air layer. In unsaturated mixtures, the steam partial pressure is not related to the mixture temperature, and in this case the air/steam ratio and mixture pressure are additional principal variables. The vertical length of the surface affects the mean thickness of the condensate film, but should have only a very small effect on the overall heat transfer, which is controlled by diffusion.

Uchida and Thompson both performed experiments with steam (H_2O) and air mixtures at atmospheric pressure in natural convection systems. Thompson's mixtures were in equilibrium with the boiling liquid and therefore saturated; it is presumed (in the absence of a contrary indication) that those of Uchida were too. Air/steam mixtures within the HIFAR containment following a LOCA would be at a pressure close to atmospheric and, being in the presence of boiling liquid, would also be saturated. In saturated mixtures the total pressure and the mixture temperature determine the air/steam ratio; at the same air/steam ratio, therefore, mixtures in the HIFAR containment and in the experiments would be at much the same temperature.

Thompson's surface was cooled by circulating water at 18°C; Uchida did not report the cooling surface temperature, but it is most likely that it was cooled by water obtained from an atmospheric environment and the temperature would be similar to that of the D_2O heat exchanger shells in the HIFAR plant room. The vertical length of Thompson's surface was nearly the same as that of the HIFAR heat exchangers but that of Uchida's surface was much shorter (only one sixth of the shell length).

The surface length in the Uchida data, and the fact that steam in the HIFAR containment would be D_2O vapour, are the only significant differences between conditions in the HIFAR containment and in the experiments. As discussed below, effects of surface length and the replacement of H_2O vapour with D_2O vapour are expected to be small. There may be small differences in mixture temperature (due to pressure differences) and surface temperature, but these can be assumed to have no effect on heat transfer coefficient, which is the heat transfer rate per unit surface area per unit temperature difference. The relations between condensation heat transfer coefficient and air/steam ratio determined in the experiments of Uchida and Thompson are considered to be directly applicable to conditions after a LOCA in HIFAR, and provide the best available means for predicting the rate of condensation of steam on the heat exchanger shells.

4. COMPARISON OF THE UCHIDA AND THOMPSON DATA

Uchida *et al.* presented their data graphically as a plot of condensation heat transfer coefficient against air/steam mass ratio. Richardson *et al.* [1967] tabulated a series of points from the Uchida data which are used in the computer code CONTEMPT to predict condensation heat transfer coefficients in reactor containments; this table provides a more convenient record of the Uchida data and is reproduced in **appendix A**. Thompson's data are presented graphically as a plot of volumetric steam condensation rate against volumetric steam proportion in the mixture; to permit direct comparison of the Thompson and Uchida data, corresponding values of condensation heat transfer coefficient and air/steam mass ratio have been calculated (**see appendix B**). The comparison is shown in **figure 1**.

It can be seen that curves through the data points intersect at an air/steam mass ratio of about 2.6, near the upper limit of this ratio in the Thompson experiments. At smaller ratios, Thompson's heat transfer coefficients lie below those of Uchida, the difference becoming greatest at a mass ratio of 0.5, where Thompson's coefficient is only 3 per cent of the Uchida coefficient. A lower envelope of the data would follow the Thompson data between 0.3 and 2.6, and return to Uchida's data at higher mass ratios. The mass ratio in a saturated mixture is dependent on the mixture temperature, and the region of relevance to predicted conditions in the HIFAR plant room after a LOCA is more easily determined from **figure 2**, in which condensation heat transfer coefficient is plotted against saturated mixture temperature at one atmosphere mixture pressure. In the various situations considered by Marshall and Rodd [1984], the temperature in the plant room, with heat removal does not rise above 74°C. Up to this temperature, the data of Uchida indicate the lower heat transfer coefficients and should therefore be used in analyses of HIFAR containment conditions.

5. D_2O IN AIR/STEAM MIXTURES

The relation between partial pressures and mass ratio in a mixture of perfect gases depends on the molecular weights of the components. For air/steam mixtures, which approximate to a perfect gas mixture,

$$V = \frac{P_V}{P_T} = \frac{1}{1+\alpha\, m} \quad , \tag{5.1}$$

where $\alpha = M_S/M_A = 0.6219$ for air/H_2O, or 0.6915 for air/D_2O.

For a given mass ratio (m) and total pressure (P_T), the partial pressure of D_2O vapour is therefore lower than that of H_2O vapour. However, because the saturation pressure of D_2O vapour is also lower than that of H_2O vapour at the same temperature, there is very little difference between the temperatures of saturated mixtures of air/D_2O and air/H_2O of equal mass ratio. For a mixture pressure of one atmosphere and air/steam mass ratios less than 42, the difference in temperature is less than 1°C. With a given air/steam mass ratio, the temperature difference producing condensation heat transfer is therefore approximately the same for both air/D_2O and air/H_2O mixtures.

Because of the difference in properties of the liquids, the rate of heat transfer through a D_2O condensate film is about 95 per cent of that through an H_2O film with the same temperature difference. There is likely to be a greater disparity, however, between the rates of heat transfer through the more important air layer. According to the empirical formula given by Gilliland [1934], the mass diffusion coefficient for D_2O vapour in air is 97 per cent of that for H_2O vapour, but when the difference in latent heat of the vapours (which represents the greater part of the enthalpy transfer) is taken into account, the expected rate of heat transfer by diffusion of D_2O vapour through an air layer would be about 90 per cent of that due to diffusion of H_2O vapour with the same temperature difference and same mass concentration gradient.

It appears, then, that if an empirical relation between mass ratio and condensation heat transfer coefficient for air/H_2O mixtures is applied to air/D_2O mixtures, the heat transfer coefficients should be reduced by a factor of between 0.90 and 0.95, depending on the mass ratio. There are, however, no experimental data by which this factor can be verified, and the factor is hardly significant when the probable accuracy of the experiments with air/H_2O mixtures is considered.

6. HEAT TRANSFER FROM THE HEAT EXCHANGER SHELLS TO THE COOLING WATER

The rate of heat transfer from an air/steam mixture in the HIFAR plant room, and hence the rate of condensation of steam, depends on the complete heat transfer system, from air/steam mixture to cooling water. The air layer and condensate film are only parts of this system; the other parts are the shell wall and the convective water film on the inside of the shell. A notional diagram of the heat transfer system is shown in **figure 3**.

The coefficient of heat transfer for heat conducted through the shell wall is the thermal conductivity of the material divided by its thickness. A 0.25 in. thickness of stainless steel has a coefficient of 2500 W/m^2 K.

When the secondary cooling water pumps are running, water flows circumferentially over the shell inside surface as it passes across the tube bundle, and longitudinally as it passes through the segmental baffle cuts. On the shell side of shell-and-tube heat exchangers, there is no critical value of the Reynolds number for heat transfer [McAdams 1954] and the heat transfer coefficient between shell and water can be calculated using the McAdams form of the Dittus-Boelter equation as applied to channels of non-circular cross-section:

$$h_i = 0.023 \, \frac{k}{D_e} \, Re^{0.8} \, Pr^{0.4} \quad . \tag{6.1}$$

This formula is preferable to any of the commonly used formulae for the shell side heat transfer coefficient in shell-and-tube heat exchangers, because the latter are applicable to the tube surface and not to the whole shell. Separating the terms of the McAdams formula that are dependent on channel geometry,

$$\begin{aligned} h_i &= 0.023 \left\{ \frac{k\, Pr^{0.4}}{\mu^{0.8}} \right\} \left\{ \frac{G^{0.8}}{D_e^{0.2}} \right\} \\ &= 0.023 \left\{ \frac{k\, Pr^{0.4}}{\mu^{0.8}} \right\} w^{0.8} \left\{ \frac{P_w^{0.2}}{1.32\, A} \right\} \quad . \end{aligned}$$

The flow area (A) and wetted perimeter (P_w) vary in the tube bundle region; mean values may be obtained by dividing, respectively, the volume occupied by water and the wetted surface area by the length of the channel in the direction of flow. Although the flow areas and wetted perimeters of the bundle and baffle cut regions differ considerably, the ratio $P_w^{0.2/A}$ is not greatly different and an average value of this ratio for the two regions may be used to obtain a mean forced convection heat transfer coefficient for the whole shell (see **appendix D**):

$$h_i = 0.178\ k \left[\frac{w}{\mu} \right]^{0.8} Pr^{0.4} \quad . \tag{6.2}$$

The temperature rise of the cooling water with the normal flow rate is insignificant and the water properties can be evaluated at the water inlet temperature. For convenience in calculations, the formula may be expressed in the form

$$h_{io} = F_w w^{0.8} \quad , \tag{6.3}$$

where h_{io} is the heat transfer coefficient referred to the external surface of the shell = $(33/33.5)h_i$, and F_w is a function of cooling water inlet temperature and may be obtained from **figure 4** or from the equation

$$F_w = 44.22 + 69.48 \left[\frac{T_{CW}}{100} \right] - 10.66 \left[\frac{T_{CW}}{100} \right]^2 \quad . \tag{6.4}$$

When the flow of secondary cooling water is stopped, heat is transferred by natural convection from the inside surface of the shell to the water contained in the shell. In this case, the temperature of the water and the heat transfer coefficient increase with time. The appropriate formula for the natural convection heat transfer coefficient at a vertical wall of minimum length 0.438 m (the distance between baffles) is given by McAdams [1954] as

$$h_i = 0.13\,k \left[\frac{g\,\rho^2\,\beta\,C\,\Delta T}{\mu\,k} \right]^{1/3} \quad . \tag{6.5}$$

Water properties should be evaluated at the mean film temperature, which is the average of the surface and bulk water temperatures. The temperature difference (ΔT) is that between the shell inner surface and the bulk water temperatures:

$$\Delta T = T_{Si} - T_{CW} \quad .$$

For greater convenience, the heat transfer coefficient may be referred to the external shell surface and expressed in the form

$$h_{io} = F_o\ \Delta T^{1/3}, \tag{6.6}$$

where F_o is a function of the mean film temperature and may be obtained from **figure 5**, or from the equation

$$F_o = 58.45 + 784.8 \left[\frac{T_f}{100} \right] - 923.7 \left[\frac{T_f}{100} \right]^2 + 856.8 \left[\frac{T_f^3}{100} \right]^3 - 316.1 \left[\frac{T_f}{100} \right]^4 . \tag{6.7}$$

To determine F_o, a sufficiently close approximation to T_f can be obtained from

$$T_f \approx T_{CW} + \frac{T_M - T_{CW}}{4 + \sqrt{3m}} \quad .$$

Equation 6.6 may be solved by iteration, as ΔT is dependent on h_{io}.

$$\text{Since } q = U\left[T_M - T_{CW} \right] = h_{io} \left[T_{Si} - T_{CW} \right] \quad ,$$

$$\Delta T = \frac{U}{h_{io}} \left[T_M - T_{CW} \right] \quad ,$$

$$= \frac{T_M - T_{CW}}{1 + h_{io} \left[\frac{1}{h_c} + \frac{t}{k_w} \right]} \quad ,$$

$$h_{io} = F_o \left[\frac{T_M - T_{CW}}{1 + h_{io}\left[\dfrac{1}{h_c} + \dfrac{t}{k_w}\right]} \right]^{1/3} \quad . \qquad (6.8)$$

When the flow of secondary cooling water is less than about 20 kg/s per shell, heat is transferred by a combination of natural and forced convection and the heat transfer coefficient is higher than that due to either process alone. At low flow rates, a conservative estimate is obtained by taking either the forced or the natural convection coefficient, whichever is the greater.

7. CALCULATION OF THE RATE OF CONDENSATION PER UNIT AREA OF SURFACE

Heat is transferred from the air/steam mixture to the surface through an air layer and a condensate film. Since almost all of the thermal resistance is in the air layer, the condensate may be assumed to be at the temperature of the surface and the air at the mean of the mixture and surface temperatures. If the mixture is saturated, its temperature is the dew point temperature, *i.e.* the saturation temperature of the steam at its partial pressure.

The heat transferred comes from three sources:

(i) the latent heat of the condensed steam;

(ii) the cooling of the condensate from the mixture temperature to the surface temperature; and

(iii) the cooling of the air from the mixture temperature to the mean of the mixture and surface temperatures.

In the present case, typical proportions are latent heat 92 per cent, condensate cooling 5 per cent, and air cooling 3 per cent.

Taking the specific heat capacity of air to be 1 kJ/kg K:

$$q = \left\{ \lambda w_S + 4.196\,(T_M - T_S)\,w_S + 0.5\,(T_M - T_S)\,m\,w_S \right\} \times 10^3 \quad ,$$

$$w_S = \frac{q \times 10^{-3}}{\lambda + (4.196 + 0.5\,m)\,(T_M - T_S)} \quad .$$

Since

$$\left[T_M - T_S \right] = \frac{q}{h_c} \quad ,$$

$$w_S = \frac{q \times 10^{-3}}{\lambda + (4.196 + 0.5\,m)\,\dfrac{q}{h_c}} \quad . \qquad (7.1)$$

The heat transfer equation is

$$q = U\left[T_M - T_{CW} \right] \quad . \qquad (7.2)$$

The overall coefficient of heat transfer (U) is obtained from the individual heat transfer coefficients by

$$\frac{1}{U} = \frac{1}{h_c} + \frac{t}{k_w} + \frac{1}{h_{io}} \quad , \qquad (7.3)$$

giving

$$w_S = \frac{(T_M - T_{CW}) \times 10^{-3}}{\lambda\left[\dfrac{1}{h_c} + \dfrac{t}{k_w} + \dfrac{1}{h_{io}}\right] + (4.196 + 0.5\,m)(T_M - T_{CW})\,\dfrac{1}{h_c}} \quad . \qquad (7.4)$$

The source of each of the terms on the RHS of **equation 7.4** is as follows:

T_M	is the independent mixture temperature in unsaturated mixtures; in saturated mixtures it is the saturation temperature of the vapour at its partial pressure (P_V), which is obtained from the total mixture pressure (P_T) and the air/vapour mass ratio (m) by **equation 5.1.**
T_{CW}	is the independent cooling water inlet temperature.
λ	is the latent heat of the vapour at its partial pressure (P_V)
h_c	is the condensation heat transfer coefficient, obtained from Uchida's data, which is tabulated in **appendix A** and plotted **figures 1 and 2.**
$\frac{t}{k_w}$	is the thermal resistance of the heat exchanger shell wall, of wall thickness (t) and thermal conductivity (k_w). In the HIFAR heat exchangers, the value of t/k_w is approximately 0.0004 m^2 K/W.
h_{io}	is the film coefficient of heat transfer from the heat exchanger shell to the cooling water; it is obtained from the cooling water mass flow rate and temperature by **equation 6.3** for flow rates greater than 20 kg/s, or **equation 6.8** for lower flow rates.
m	is the independent air/vapour mass ratio.

8. RECOMMENDATIONS

Since 92 per cent or more of the heat transferred to the surfaces comes from condensation of the vapour, a reasonably good estimate of the rate of condensation per unit area of surface can be obtained by dividing the heat flux by the latent heat of the vapour at its partial pressure. For a closer estimate, allowance should be made for cooling of the condensate and air by using **equation 7.1**, in which the heat flux (q) can be estimated from the heat transfer coefficients of the various processes and the overall temperature difference between the air/steam mixture and the cooling water inside the heat exchanger shells. This leads to **equation 7.4** which gives the rate of condensation

$$w_S = \frac{(T_M - T_{CW}) \times 10^{-3}}{\lambda\left[\frac{1}{h_c} + \frac{t}{k_w} + \frac{1}{h_{io}}\right] + (4.196 + 0.5m)(T_M - T_{CW})\,\frac{1}{h_c}} .$$

The best means for estimating the condensation heat transfer coefficient (h_c) is provided by direct application of the experimental data of Uchida *et al.* **[1964] and Thompson [1970].** It is recommended that the Uchida data be used when the air/steam mixture temperature is below 74°C, and the Thompson data when it is above 74°C. These data, obtained in experiments with air/H_2O mixtures, can be applied to air/D_2O mixtures without significant error.

An estimate of the film coefficient of heat transfer at the inner surface of the heat exchanger shells (h_{io}) can be obtained from **equation 6.3** used in conjunction with **figure 4**:

$$h_{io} = F_W w^{0.8} \qquad (w > 20) \quad ,$$

or from **equation 6.8** used in conjunction with **figure 5**:

$$h_{io} = F_o \left[\frac{T_M - T_{CW}}{1 + h_{io}\left[\frac{1}{h_c} + \frac{t}{k_w}\right]}\right]^{1/3} \qquad (w < 20) \quad .$$

9. CONCLUSION

The rate of condensation of vapour on the external surfaces of the HIFAR D_2O heat exchangers from an air/steam mixture in the plant room can be estimated with reasonable accuracy, using existing data.

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11. NOTATION

A	channel flow area	m^2
C	specific heat of cooling water	J/kg K
C_A	specific heat of air	kJ/kg K
C_c	specific heat of condensate	kJ/kg K
D_e	hydraulic equivalent diameter of heat exchanger shell	m
F_o	natural convection heat transfer factor	$W/m^2K^{1.33}$
F_w	forced convection heat transfer factor	$J/kg^{0.8}s^{0.2}m^2\ K$
g	acceleration due to gravity	m/s^2
G	mass flux of cooling water	$kg/m^2\ s$
h_c	condensation heat transfer coefficient	$W/m^2\ K$
h_i	heat transfer coefficient at shell inner surface	$W/m^2\ K$
h_{io}	heat transfer coefficient at shell inner surface referred to outer surface area	$W/m^2\ K$
k	thermal conductivity of cooling water	W/m K
k_w	thermal conductivity of shell wall material	W/m K
m	mass ratio of air to steam in mixture	
M_A	molecular weight of air	
M_S	molecular weight of steam	
PR	Prandtl number of cooling water	
P_T	total pressure of mixture	bar
P_V	partial pressure of vapour in mixture	bar
P_W	wetted perimeter of heat exchanger shell	m
q	heat flux	W/m^2
Q	condensation rate of steam	ft^3/min
Re	Reynolds number of cooling water flow	
S	total external surface area of heat exchanger shells	m^2
t	thickness of heat exchanger shell wall	m
T_{CW}	temperature of cooling water	°C
T_f	mean film temperature	°C
T_M	temperature of air/steam mixture	°C
T_S	temperature of outer surface of heat exchanger shell	°C
T_{Si}	temperature of inner surface of heat exchanger shell	°C
U	overall coefficient of heat transfer	$W/m^2\ K$
V	concentration of vapour by volume	
w	mass flow rate of cooling water per shell	kg/s
w_S	mass rate of condensation of vapour per unit area of surface	$kg/s\ m^2$
α	ratio of molecular weights, vapour to air	
β	coefficient of thermal expansion of cooling water	1/K
ΔT	temperature difference between shell inner surface and cooling water	K
μ	dynamic viscosity of cooling water	kg/m s
ρ	density of cooling water	kg/m^3
λ	latent heat of evaporation of vapour at its partial pressure	kJ/kg

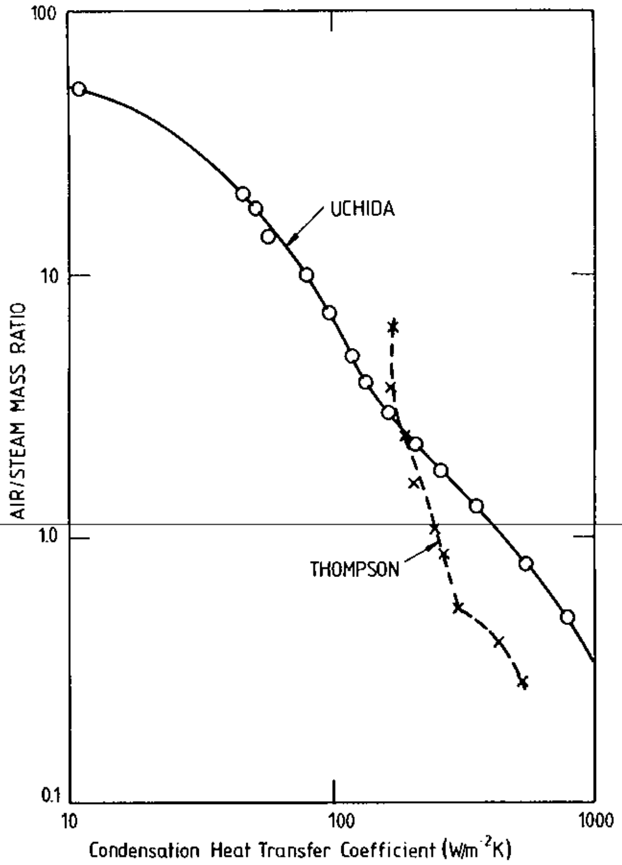


Figure 1 Experimental determinations of the coefficient of heat transfer for condensation on a surface from air/steam mixtures

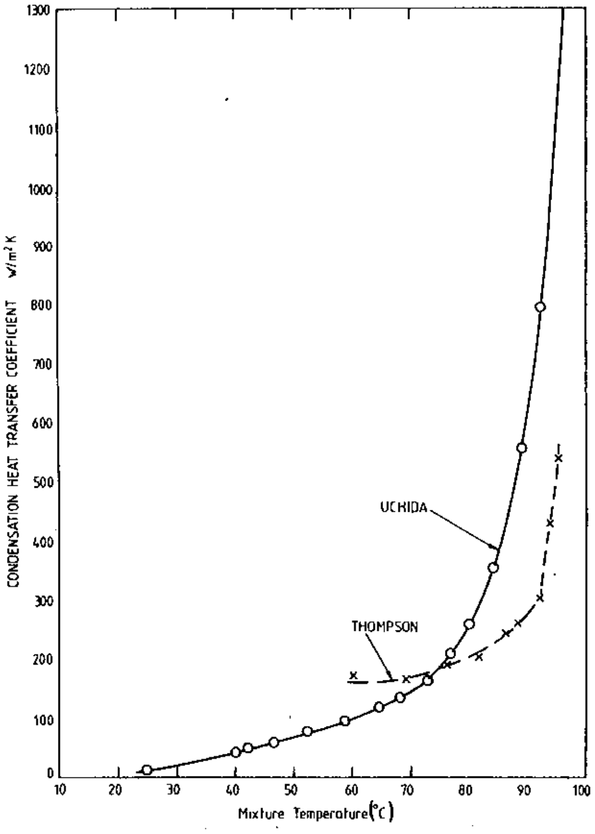
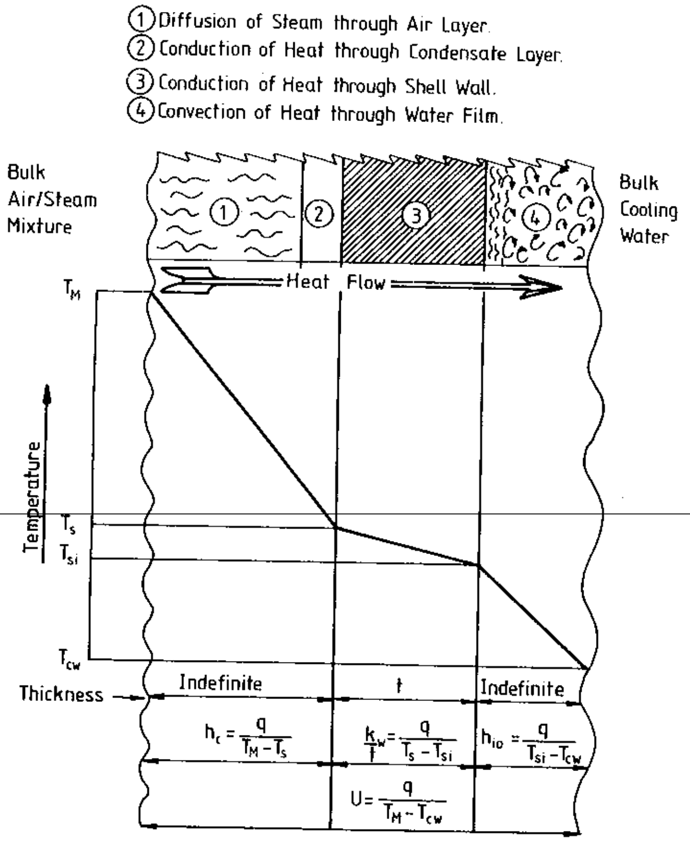


Figure 2 Comparison of the experimental results of Uchida and Thompson



$$\frac{1}{U} = \frac{T_M - T_{cw}}{q} = \frac{T_M - T_s}{q} + \frac{T_s - T_{si}}{q} + \frac{T_{si} - T_{cw}}{q} = \frac{1}{h_c} + \frac{t}{k_w} + \frac{1}{h_{io}}$$

Figure 3 Notional temperature diagram of heat transfer systems

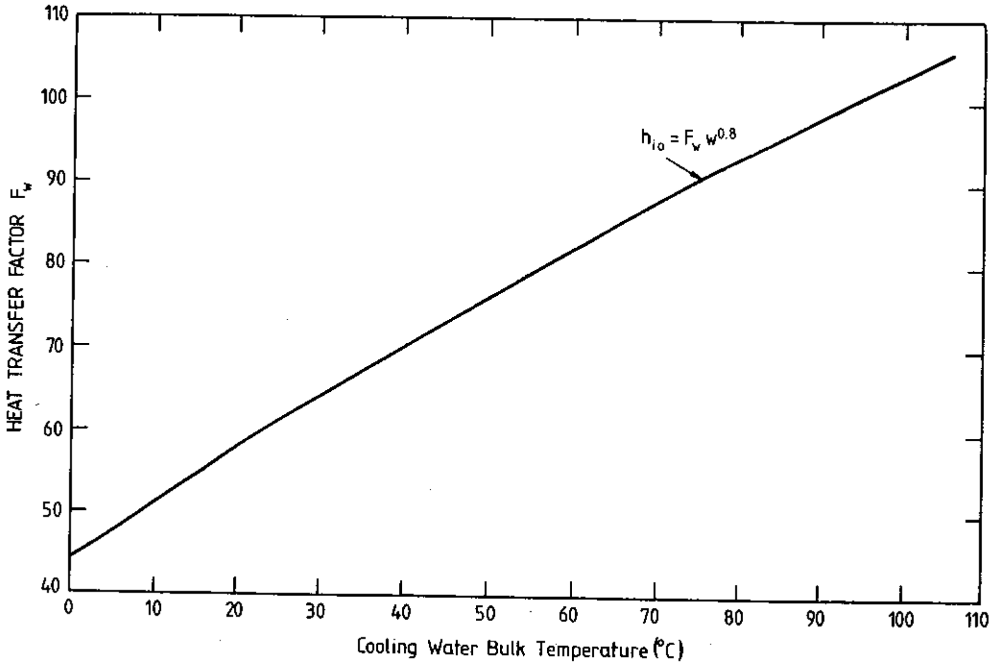


Figure 4 Factor for coefficient of heat transfer between inner surface of heat exchanger shell and cooling water in forced convection

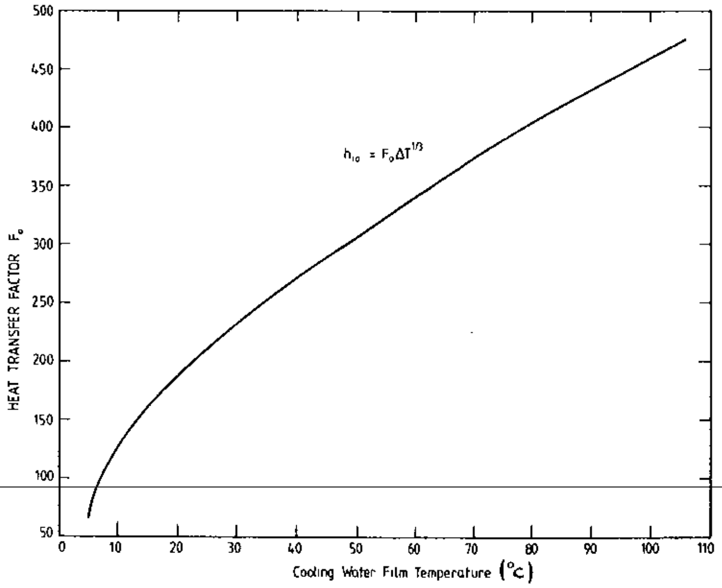


Figure 5 Factor for coefficient of heat transfer between inner surface of heat exchanger shell and cooling water in natural convection

APPENDIX A
RICHARDSON'S TABLE OF UCHIDA'S DATA WITH CORRESPONDING MIXTURE TEMPERATURES

Air Steam Mass Ratio	Condensation Heat Transfer Coefficient (W/m^{2}K)	Corresponding Temperatures of Air/Steam Mixture Saturated at Atmospheric Pressure (°C)
50	11	24.9
20	45	40.8
18	51	42.2
14	57	46.7
10	79	52.6
7	96	58.9
5	119	64.7
4	136	68.4
3	165	72.9
2.3	210	76.9
1.8	261	80.2
1.3	358	84.2
0.8	556	89.1
0.5	795	92.6
0.1	1590	98.3

APPENDIX B
CALCULATION OF CONDENSATION HEAT TRANSFER COEFFICIENT FROM THOMPSON'S DATA

Thompson expressed the measured total steam condensation rate (Q) in ft^3/min of steam at 100°C and atmospheric pressure; this yields the mass rate of condensation per unit area of cooling surface (w_S) by the equation

$$w_S = \frac{Q}{S} \times 2.82 \times 10^{-4} \quad . \tag{B1}$$

The heat transferred to the cooling water is the sum of the enthalpy changes of the steam and air. The air layer presents a much greater resistance to heat flow than does the condensate film; the temperature of the condensate is therefore close to that of the surface. For the calculation of enthalpy changes, the condensate is assumed to be at the surface temperature, and the air at the mean of the mixture and surface temperatures. The heat flux (q) is obtained by multiplying the mass rate of condensation per unit area of cooling surface by the enthalpy change of the steam and air, the mass ratio of air to steam being represented by m:

$$q = w_S\, \lambda + C_c(T_M - T_S) + \frac{m\, C_A}{2}\,(T_M - T_S) \times 10^3 \quad . \tag{B2}$$

The specific heat content of the condensate (C_c) may be taken as 4.196 kJ/kg K and that of the air (C_A) as 1.0 kJ/kg K.

The temperature rise of the cooling water is less than 0.5°C and may be neglected, therefore the overall coefficient of heat transfer (U) is the quotient of the heat flux and the difference between the mixture temperature (T_M) and the cooling water temperature (T_{CW}):

$$U = \frac{q}{T_M - T_{CW}} \quad . \tag{B3}$$

The overall thermal resistance (1/U) is the sum of three resistances in series - those due to the air layer and condensate film ($1/h_c$), the cooling surface wall (t/k_w), and the cooling water convective film ($1/h_{io}$). All heat transfer coefficients are referred to the external cooling surface:

$$\frac{1}{U} = \frac{1}{h_c} + \frac{t}{k_w} + \frac{1}{h_{io}} \tag{B4}$$

Since

$$q = U(T_M - T_{CW}) = h_c(T_M - T_S) \quad ,$$

$$T_M - T_S = \frac{U}{h_c}\,(T_M - T_{CW}) \quad . \tag{B5}$$

The condensation heat transfer coefficient (h_c) is obtained by combining the above equations:

$$h_c = \frac{\lambda + (4.196 + 0.5\, m)(T_M - T_{CW})}{\dfrac{S(T_M - T_{CW}) \times 10^{-3}}{2.82 \times 10^{-4}\, Q} - \lambda\, \dfrac{t}{k_w} + \dfrac{1}{h_{io}}}$$

For Thompson's test apparatus,

S = 0.1459 m_2;
T_{CW} = 18.3°C;
t = 0.002 m (assumed to be 14 SWG tubing);
k_w = 15.9 W/m K;
h_{io} = 14,750 W/m^2 K (30 gal/min of water at 18°C, flowing in 1 in., 14 SWG pipe);

$$h_c = \frac{\lambda + (4.196 + 0.5\, m)(T_M - 18.3)}{\dfrac{0.517(T_M - 18.3)}{Q} - 0.0002\lambda} \qquad (\mathrm{W/m^2\,K})$$

where Q = condensation rate (ft^3/min),
λ = latent heat of condensation (kJ/kg), and
m = air/steam mass ratio.

The following table shows the values derived from Thompson's data.

Volumetric Proportion of steam	Condensation Rate (Q) (ft^3/min)	Air/Steam Mass Ratio (m)	Steam Partial Pressure (bar)	Mixture Temp. T_M (°C)	Condensate Film Temp (°C)	Latent Heat (kJ/kg)	Condensation Heat Transfer Coefficient (W/m^2K)
0.2	1.35	6.43	0.203	60.4	39.3	2357.7	171
0.3	1.62	3.75	0.304	69.4	43.9	2335.4	167
0.4	2.10	2.41	0.405	76.2	47.3	2318.4	191
0.5	2.50	1.61	0.507	81.7	50.0	2304.6	207
0.6	3.17	1.07	0.608	86.3	52.3	2292.7	246
0.65	3.50	0.866	0.659	88.4	53.3	2287.4	264
0.75	4.20	0.536	0.760	92.1	55.2	2277.6	302
0.8	6.00	0.402	0.81	93.9	56.1	2273.1	430
0.85	7.57	0.284	0.861	95.5	56.9	2268.8	540

APPENDIX C
PARTICULARS OF HIFAR HEAVY WATER HEAT EXCHANGERS

Material of construction throughout is stainless steel type 321.

Number of shells	3		
Shell inside diameter	33	in.	(0.838 m)
Shell wall thickness	0.25	in	(0.00635 m)
Shell and tube length between tubeplates	75.25	in.	(1.91 m)
Thickness of tubeplates	1.375	in.	
Number of tubes per shell	1841		
Tube outside diameter	0.375	in.	
Tube wall thickness	0.048	in.	
Tube pitch (60° triangular)	0.5625	in.	
Number of segmental baffles per shell	3		
Baffle spacing	19	in.	(0.483 m)
Baffle plate thickness	0.25	in.	
Baffle cut	23.9%		
Width of tubefield	17.25	in.	(0.438 m)
External shell surface, excluding branches, per shell	7570	in^2	(4.88 m^2)
Volume of secondary water contained, per shell	48 470	in^3	(0.794 m^3)
Volume of stainless steel in shell, tubes, tubeplates and baffles	11 410	in^3	(0.187m^3)
Water equivalent of stainless steel, per shell	10 980	in^3	(0.180 m^3)
Total external surface area of three shells	14.65	m^2	
Total water equivalent of three heat exchangers (filled with secondary water)	2.92	m^3	

APPENDIX D
DERIVATION OF HEAT TRANSFER COEFFICIENT AT INTERNAL SURFACE OF HEAT EXCHANGER SHELL, WITH COOLING WATER FLOWING

Volume of water contained in tube bundle region between baffles = 6354 in.^{3}
Wetted surface in tube bundle region between baffles = 42 025 in.^{2}
Passage length in direction of flow = 17.25 in.

Mean flow area	=	368.3	in.^{2}	=	0.2376	m^2
Mean wetted perimeter	=	2436.2	in.	=	61.88	m
Equivalent diameter	=	0.6047	in.	=	0.01536	m
Flow area through baffle cut	=	156.5	in.^{2}	=	0.101	m^2
~~Wetted perimeter of baffle cut~~	~~=~~	~~61.82~~	~~in.~~	~~=~~	~~1.57~~	~~m~~
Equivalent diameter	=	10.13.	in.	=	0.2572	m
Outside diameter of shell	=	33.5	in.	=	0.85	m
Inside diameter of shell	=	33.0	in.	=	0.838	m

$$h_i = 0.023 \times \frac{k}{D_e}\left[\frac{G\,D_e}{\mu}\right]^{0.8} Pr^{0.4}$$

$$= 0.023 \times \frac{kPr^{0.4}}{\mu^{0.8}}\left[\frac{w^{0.8}}{A^{0.8}\,D_e^{0.2}}\right] ,$$

$$= 0.023 \times \frac{k\,Pr^{0.4}}{\lambda^{0.8}}\left[\frac{w^{0.8}\,P_w^{0.2}}{1.3195\ A}\right] ,$$

$$= 0.0173 \times \frac{k\,Pr^{0.4}}{\mu^{0.8}}\,w^{0.8}\left[\frac{P_w^{0.2}}{A}\right] ,$$

For tube bundle region, $\frac{P_w^{0.2}}{A} = 9.604$.

For baffle cut, $\frac{P_w^{0.2}}{A} = 10.836$.

Mean value of $\frac{P_w^{0.2}}{A} = 10.22$.

$$h_i = 0.178\,k\,\frac{(w)}{(\lambda)^{0.8}}\,Pr^{0.4}\ \text{(SI units)}$$

This is the convective film heat transfer coefficient, referred to the inside surface of the shell. When referred to the outside surface, the coefficient becomes

$$h_{io} = \frac{33}{33.5}h_i = 0.175\,\frac{k\,Pr^{0.4}}{\lambda^{0.8}}\,w^{0.8}\ \text{(SI units)}$$