The Condensation of Vapours of Binary Immiscible Liquids

by

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SUMMARY

Heat transfer data are reported for the condensation of steam-toluene and steam-trichloroethylene eutectic mixtures on 25.4 mm diameter oxidised copper and gold plated horizontal tubes. Data are also presented for the condensation of pure steam, toluene and trichloroethylene on the oxidised copper tube and the film heat transfer coefficients obtained agree to within ± 20% of the Labunstov form of the Nusselt equation.

For the binary immiscible systems the heat transfer coefficients decrease as the temperature difference increases, with the oxidised copper surface giving higher coefficients than the gold. These differences are attributed to the two observed fundamental mechanisms of condensation, a channelling mode on the oxidised copper and a standing drop mode on the gold. Models based on the different mechanisms are proposed and predict the experimental results to within ± 20%.

Finally it is postulated that the temperature dependent mutual solubilities affords an explanation of the formation of the large number of very small droplets observed during the condensation of these eutectic mixtures.

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Chapter 1

Introduction

The condensation of vapours has been extensively studied, both theoretically and experimentally during the past seventy years. A great number of these studies have been concerned with the condensation of pure vapours and in particular steam.

By comparison work on vapour mixtures, particularly those mixtures which form immiscible liquids on condensation, has been uncommon. The probable reason being that steam condensation in particular is a much more important industrial process than that of vapour mixture condensation. However, the condensation of vapours which form immiscible liquid condensates is nevertheless common in industrial practice. For example steam distillation and azeotropic distillation processes commonly give vapours which form immiscible liquid mixtures on condensation, as do certain chemical reactor processes, particularly those associated with the petroleum industry.

To design condensers for the above processes it is necessary to know the values of the condensing heat transfer coefficients. Most of the previous studies on "immiscible liquid condensation" were primarily concerned with the determination and prediction of these heat transfer coefficients.

It is apparent from the earlier investigations that the condensation processes involved are extremely complex. And although several studies have been made the effects of certain important parameters are still not clear.

The principal objectives of the present study were to investigate several of these potentially important parameters. In particular the effects on heat transfer performance of film temperature difference,condenser tube surface properties and condensate flow regimes were studied.

Chapter 2

Literature Survey

2.1 Introduction

Pure single component vapours have been found to condense on a cooled surface in one of two ways. The condensate may form either a continuous film or droplets; these two modes of condensation are termed filmwise and dropwise respectively.

When condensing vapour mixtures, the modes of condensation vary. For miscible liquids the condensate usually forms a film, although Mirkovitch and Missen (1961) have reported systems which form both films and drops. In the case of immiscible liquids the condensate consists of both films and drops of different liquid phases. Thus the mechanism of condensation of vapour mixtures, particularly of immiscible liquids, is much more complex than for pure vapours.

Most of this chapter is devoted to a detailed review of the literature on the condensation of vapours of immiscible liquids. However, a brief survey of filmwise and dropwise condensation is given first. No review is given for the case of vapours of miscible liquids but the interested reader is referred to van Es and Heertjes (1962) for details.

2.2 Laminar film condensation

Nusselt (1916) derived theoretical equations for predicting the heat transfer coefficients obtained during the filmwise condensation of a pure vapour. The equations are

$$h_N = C \left[\frac{k^3 \rho^2 \lambda g}{\mu \Delta T_f B} \right]^{0.25} \qquad (2.1)$$

where: $C = 0.728$ and $B = D_o$ for horizontal tubes and

$C = 0.943$ and $B = L$ for vertical tubes

$\Delta T_f = T_s - T_w$

T_s is the saturation temperature

T_w is the wall temperature.

The other symbols are defined in the nomenclature.

An alternative form of equation (2.1) is

$$h_N \left(\frac{\mu^2}{k\,\rho^2\,g} \right)^{\frac{1}{3}} = P \left(\frac{4\Gamma}{\mu} \right)^{-\frac{1}{3}} \tag{2.2}$$

where: P = 1.47 or 1.51 for horizontal and vertical tubes respectively

Γ is the mass flowrate of condensate per unit width of film.

The main assumptions used to derive the above equations were as follows:

1) The only significant resistance to the condensation process is presented by the liquid film.

2) The condensate flow is laminar.

3) The wall temperature is constant.

4) The fluid properties are constant.

5) Subcooling of the condensate may be neglected.

6) There is no vapour drag on the condensate film.

7) Acceleration of the liquid film is negligible.

8) The temperature gradient through the film is linear.

Many of the later workers have relaxed the restrictions imposed by the above assumptions. Bromley (1952) and Rohsenow (1956) took account of the subcooling and non linear temperature gradient effects, the final equation being

$$h_R/h_N = (1 + 0.68\,\varepsilon)^{0.25} \tag{2.3}$$

where $\varepsilon = C_p\,\Delta T_f/\lambda$.

The above equation is widely used in place of the original Nusselt equation.

Sparrow and Gregg (1959) give a boundary layer treatment of laminar film condensation in which the liquid film acceleration as well as the convective terms were included. Chen (1961), using the integral form of the boundary layer equations, and Koh, Sparrow and Hartnett (1961) using the differential boundary layer equations, took account of the effects of drag due to the initially stationary vapour, as well as the terms included by Sparrow and Gregg. The inclusion of vapour drag terms made a significant difference for low Prandtl number liquids (e.g. liquid metals) but was not significant for liquids with Prandtl numbers greater than one. The agreement between the solutions of the integral and differential forms of the boundary layer equations is excellent.

Chen presents approximate equations for predicting the heat transfer coefficients for a vertical plate and a horizontal tube which are within 1% of the detailed numerical solutions. The equations are, for a flat plate

$$h_c/h_N = \left[\frac{1 + 0.68\,\varepsilon + 0.02\,(\varepsilon^2/Pr)}{1 + 0.85\,(\varepsilon/Pr) - 0.15\,(\varepsilon^2/Pr)}\right]^{0.25} \tag{2.4}$$

and for a horizontal tube

$$h_c/h_N = \left[\frac{1 + 0.68\,\varepsilon + 0.02\,(\varepsilon^2/Pr)}{1 + 0.95\,(\varepsilon/Pr) - 0.15\,(\varepsilon^2/Pr)}\right]^{0.25} \tag{2.5}$$

where $\varepsilon = C_p\,\Delta T_f/\lambda$ and $Pr = C_p\,\mu/k$.

The above equations are valid for liquids with Prandtl numbers larger than 1.0 and for those with Prandtl numbers less than 0.05 provided $\varepsilon \leqslant 2.0$.

Comparing equations (2.3), (2.4) and (2.5), it can be seen that they agree if the Prandtl number is large; in fact if $Pr \geqslant 1.0$ and $\varepsilon \leqslant 0.2$ there is no significant difference between the Chen, Rohsenow and Nusselt equations.

Most common liquids have Prandtl numbers between 1.0 and 10.0, It is therefore apparent that the detailed boundary layer treatments show that Nusselts equation is adequate for such liquids. Large deviations are only expected for low Prandtl number fluids (e.g. liquid metals) and for high condensate subcoolings ($\varepsilon > 0.2$).

All of the above treatments assume the physical properties of the condensate film are constant. Drew has shown (see McAdams (1954)) that if the temperature distribution is linear and it is assumed that the viscosity varies inversely with temperature, then the effects of variable viscosity can be estimated by using Nusselts equation with the viscosity evaluated at a reference temperature given by

$$T_{ref} = T_W + 0.25\ \Delta T_f. \tag{2.6}$$

Voskresenskiy (1948) and later Labuntsov (1957) incorporated a linear variation in the condensate thermal conductivity as well as the above viscosity variation. Labuntsov showed that if the physical properties in Nusselts equation are evaluated at the vapour saturation temperature a simple correction can be applied to take account of the conductivity and viscosity variations across the film. Thus

$$h_L = h_N\ \phi_L \tag{2.7}$$

and
$$\phi_L = [(k_w^{\ 3}\ \mu_s)/(k_s^{\ 3}/\mu_w)]^{1/6} \tag{2.8}$$

where: ϕ_L is the Labuntsov correction factor, k_w and μ_w are the thermal conductivity and viscosity of the condensate evaluated at the wall temperature.

k_s and μ_s are the thermal conductivity and viscosity of the condensate evaluated at the vapour saturation temperature. Poots and Miles (1967) have shown that for the condensation of pure steam the above methods of taking account of variable fluid properties are adequate even at very large temperature differences (i.e. $\Delta T_f = 100^{\circ}C$).

The assumption of a constant wall temperature was investigated in an indirect manner by Fujii et al (1972). They assumed that the heat flux was constant with varying wall temperature. The conclusion of their work was that the difference between the constant heat flux and constant wall temperature cases was insignificant. Van der Walt and Kroger (1971) investigated the problem of variable wall temperature for the case of a vertical flat plate by assuming a wall temperature profile and again the conclusion was that the effects are negligible. The present author (see appendix E) has used the same approach as van der Walt and Kroger applied to the case of a horizontal tube. The conclusion that no significant errors are introduced by the constant wall temperature assumption is again substantiated.

The conclusion from the above brief survey is that Nusselts equation should be adequate for predicting heat transfer coefficients in laminar filmwise condensation for fluids with Pr $\geqslant 1.0$ and small subcoolings ($\varepsilon \leqslant 0.2$). However, it is apparent from comparisons with experimental data that some disagreement exists: for example McAdams (1954) has stated that for most substances Nusselts equation underpredicts the heat transfer coefficients. The discrepancy between theory and experiment is usually attributed to the effects of waves. Kapitsa (1948) has shown that gravity induced waves (capillary waves) cause a reduction in the mean film thickness and hence an increase in the heat transfer coefficient. The conditions for such waves to exist has been shown by Kapitsa (1948) to be when the film Reynolds number exceeds a critical value given by

$$Re_{crit} = 2.43 \left(\frac{\mu^4 g}{\rho \sigma^3} \right)^{-1/11} \qquad (2.9)$$

An empirical correlation for predicting mean heat transfer coefficients, when the condensate film is disturbed by waves, was given by Chun and Seban (1971) as,

$$h_W/h_N = 0.8\ (\Gamma/\mu)^{0.11} \qquad (2.10)$$

The agreement between their experimental data and equation (2.10) written for local coefficients was good.

2.3 Dropwise condensation

Since McAdams (1954) reported that heat transfer coefficients observed during dropwise condensation of steam are several times larger than those obtained for filmwise condensation, a large amount of research, into both the theoretical and experimental aspects of dropwise condensation has been undertaken.

The presently accepted mechanism for the process is as follows. The vapour condenses as discrete drops on the surface; these drops grow by coalescence and condensation until they are large enough to be removed by the action of gravity or other body forces (e.g. vapour shear). When such drops move they coalesce with other drops in their path, thus sweeping an area of the surface clear of condensing drops. This enables the condensation process to restart on the clear area. It is thus apparent that the dropwise condensation process is cyclic in nature.

Several models for predicting the detailed processes involved have been presented, an excellent review of the more important theoretical and experimental contributions in this very active field is given by Merte (1973).

2.4 Condensation of vapours of immiscible liquids

The following review is divided into three main sections. The first covers experimental studies concerned with binary systems, the second work on multicomponent systems and the third models and correlations.

Not all of the published papers are reviewed in detail. Only the key papers or those of particular interest are discussed. However, a summary of the information contained in most papers can be found in Table 1.

TABLE 1

Summary of Various Investigations

Investigator	Condenser Surface	Liquids Studied with Water	Size and Orientation of Surface			Correlation proposed
			Diameter mm	Length m	Orientation	
Kirkbride (1933)	Steel	Benzene Cleaners-naptha	33.4	2.492	Horiz.	$h = (Q_1 h_{N1} + Q_2 h_{N2}) / (Q_1 + Q_2)$
Baker and Mueller (1937)	Oxidised Copper	Benzene Toluene Mixed Heptanes Trichloro-ethylene	33.4	1.129	Horiz.	$h\left(\frac{\mu_1^2}{k'^{3}_{av}\,\rho_{av}^2\,g}\right)^{\frac{1}{3}} = 1.28\left(\frac{C_{pav}\,\mu_1\,\rho_{av}^{0.7}}{k'_{av}\,\lambda_{av}}\right)^{2.38}$ $\times\ (Q_1 / Q)^{-3.28}$
Patterson et al (1937)	Wrought Iron	Heptane	22.2	1.229	Vert.	
	Muntz Metal	Heptane	19.1	1.143	Vert.	
Baker and Tsao (1940)	Oxidised Copper	Benzene Toluene Chloro-	25.4	1.129	Horiz.	$h / (1 - \frac{0.0167}{D_o})$ $= \left(500 / (1 - 0.0025\,V_2)\right) + 80$ and $h = \frac{366(1/D_o)^{\frac{1}{4}}\left(1 - \frac{0.0284}{D_o}\right)}{(1 - 0.0035\,V_2)} + 1.67/D_o$
		benzene Trichloro-ethylene Tetrachloro ethylene	15.9	1.129	Horiz.	
Patton and Feagan (1941)	Oxidised Copper	Turpentine	33.4	1.249	Horiz.	$h = 3000\,\Delta T_f^{-0.5}$ *
Cooper et al (1942)	Copper	Butyl acetate	22.2	0.914	Vert.	
Hazelton and Baker (1944)	Sanded Copper	Benzene Toluene	33.4	1.149	Vert.	$h = 79\left\{(a\lambda_1 + b\lambda_2) / aL\right\}^{\frac{1}{4}}$ for vertical tubes $h = 61\left\{(a\lambda_1 + b\lambda_2) / aD_o\right\}^{\frac{1}{4}}$ for horizontal tubes
		Chloro-	25.4	1.149	Vert.	
		benzene	15.9	1.149	Vert.	
Edwards et al (1948)	Copper	Styrene Butadiene	25.4	0.203	Vert. in tube condensation	$h\left(\frac{\mu_1}{k_1^3\,\rho_1^2\,g}\right)^{\frac{1}{3}} = 0.96\left(\frac{4\Gamma}{\mu_1}\right)^{-0.15}$

TABLE 1 (Continued)

Investigator	Condenser Surface	Liquids Studied with Water	Size and Orientation of Surfaces			Correlation proposed
			Diameter mm	Length m	Orientation	
Tobias and Stoppel (1954)	Polished Yellow Brass	Toluene Benzene Cyclohexane Carbon tetrachloride n-Heptane	25.4	1.372	Vert.	$h = h_H \Big/ \left\{ 1 - \frac{1}{545} \left\{ \left[\frac{\Delta T^3 \Delta\rho}{\mu_1^4 g} \right] \times \left[\frac{m_1 \rho_2 k_2}{m_2 \rho_1 k_1} \right]^{\frac{1}{2}} \right\}^{0.21} \right\}$
Stepanek and Standart (1958)	Copper	Benzene Toluene Dichloroethane Chlorobenzene	10.0	1.000	Horiz.	$h_e = 0.725 \left[\frac{\lambda_H k_1^3 \rho_1^2 g}{\mu_1 \Delta T_f D} \right]^{\frac{1}{4}}$ $\times K_1 (1 + K_2 \Delta T_f)$ **
Akers and Turner (1962)	Polished Brass	Benzene Heptane Carbon-tetrachloride	63.5	0.076	Vert.	$h = (a\lambda_1 h_{N1} + b\lambda_2 h_{N2}) / (a\lambda_1 + b\lambda_2)$ $h \left(\frac{\mu_1^2}{k'^3_{av} \rho_{av}^2 g} \right)^{\frac{1}{3}} = 1.47 \left(\frac{4\Gamma}{\mu_1} \right)^{-\frac{1}{3}}$
Sykes and Marchello (1970)	Oxidised Copper	Toluene Carbon-tetrachloride	34.9	0.610	Horiz.	$h_e = h_{N1} (1 - 0.8R) (\Delta T_f)^{0.67R}$ $h_e = h_{N1} \left(K_3 + \frac{1}{K_4 e^{\beta \Delta T_f}} \right)^{-1}$ ***
Bernhardt et al (1972)	Gold	Freon 112 Freon 113 Perchloroethylene p-xylene	203.2 mm high by 76.2 mm wide plate		Vert.	$h = V_1 h_{N1} + V_2 h_{N2}$

TABLE 1 (Continued)

Investigator	Condenser Surface	Liquids Studied with water	Size and Orientation of Surface			Correlation proposed
			Diameter mm	Length m	Orientation	
Kawaski et al (1972)	Copper	Benzene Toluene Trichloro-ethylene	6.2 9.5 16.0	0.0261 0.0347 0.0397	Horiz.	$Nu = 0.0295\ (Ga.\ Ku.\ Pr)^{\frac{1}{4}}\ (Re_v)^{\frac{1}{7}}$
Ponter and Diah (1974)	Oxidised Copper P.T.F.E.	Benzene Carbon-tetra-chloride 1,1,2 Trichloro ethylene Cyclo-hexane	28.6	0.813	Horiz.	For the copper tube $h_e = h_{N1} / \left\{ 1 - 1.99 \times 10^4 \left[\frac{\Delta\sigma^3\ \Delta\rho}{\mu_{av}^4 g} \right]^{-0.413} \left[\frac{\lambda_{av}\ \mu}{k_{av} \Delta T_f} \right]^{-0.286} \right\}$ For the P.T.F.E. tube $h_e = h_{N1} / \left\{ 1 - 0.047 \left[\frac{\Delta\sigma^3\ \Delta\rho}{\mu_{av}^4\ g} \right]^{0.143} \left[\frac{\lambda_{av}\ \mu_{av}}{k_{av} \Delta T_f} \right]^{-0.359} \right\}$

* As shown by Sykes and Marchello (1970)

$$** \quad K_1 = \left[1 - 4.38 \left(\frac{b}{a} \right)^{0.033} \left(\frac{\rho_1}{\rho_2} \right)^{0.62} \left(\frac{\Delta\sigma}{\sigma_2} \right)^{3.2} \right]^{\frac{1}{2}}$$

$$K_2 = 0.0584 \left(\frac{a}{b} \right) \left(\frac{k_1}{k_2} \right)^{0.5} \left(\frac{\rho_2}{\rho_1} \right)^{1.4} \left(\frac{\Delta\sigma}{\sigma_2} \right)^{1.6}$$

$$*** \quad K_3 = \left[7.6 - 1.8\ (Pr_1 - Pr_2) \right]^{-1}$$

$$K_4 = \left[17.3 \times 10^{-10} Pr_1 \left(1 + \frac{a\lambda_1}{b\lambda_2} \right) \right] / \left[(Oh)_1 \left(\frac{\Delta\sigma}{\sigma_1} \right)^{\frac{1}{2}} \left(\frac{\mu_1}{\mu_2} \right) \left(\frac{M_2}{M_1} \right) \right]^2$$

2.4.1. Experimental studies on binary systems

2.4.1.1 Investigations using horizontal tubes

The primary concerns in most condensation studies are the determination and prediction of heat transfer coefficients. When measuring heat transfer coefficients it is important to use the correct temperature difference. To determine the temperature difference for "immiscible liquid" condensation we must first look at the manner in which the mixed vapours can condense.

The temperature-composition diagram for a totally immiscible binary system is shown in Fig. 2.1. Three possible condensation paths can be envisaged, of which one is exclusive to the hetero-azeotropic mixture (the so called eutectic mixture). Consider the superheated eutectic mixture shown by point E (Fig. 2.1), the condensation path will be one of desuperheating followed by simultaneous condensation of both components at constant temperature (eutectoid temperature T_e). The condensate will consist of two phases, the overall liquid composition being the same as that of the vapour, that is the eutectoid composition. This situation is similar to the condensation of a pure vapour except for the behaviour of the condensate film.

The other two condensation paths are for non-eutectic mixtures. They are:

(1) Condensation of one component preferentially, with the other component acting as an incondensable gas.

(2) Condensation of both components, the condensate composition being dependent on the rates of mass transfer of the two components through the vapour phase.

Which of the above two processes is occurring depends on the vapour-liquid interface temperature. Considering point M (Fig. 2.1), if the interface temperature (T_i) is greater than the eutectoid temperature (T_e) then only one component can condense (in this case component 1). The other component acts as an incondensable gas in this situation.

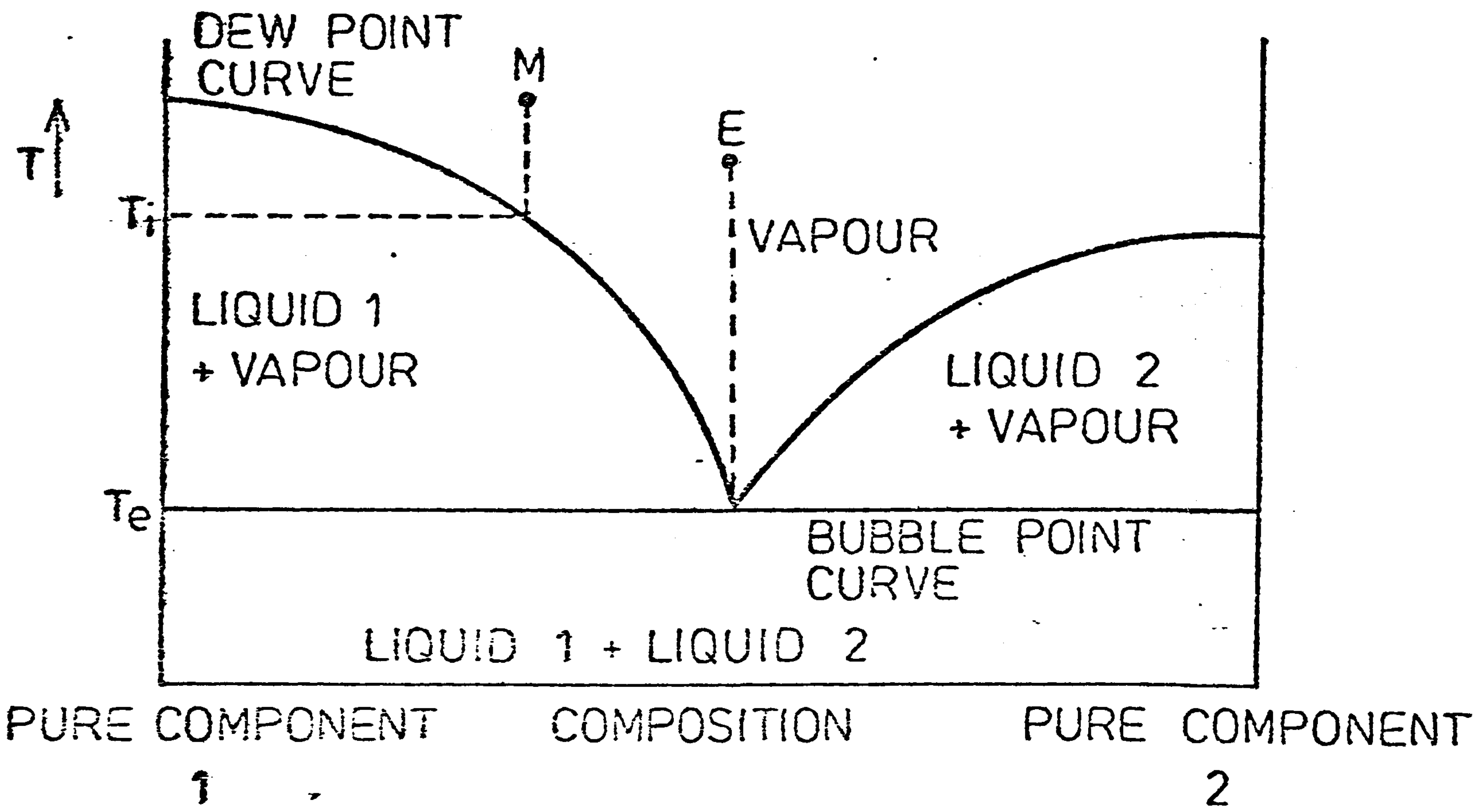

FIG. 2.1 TEMPERATURE COMPOSITION DIAGRAM FOR A TOTALLY IMMISCIBLE BINARY SYSTEM.

If the interface temperature is such that both components can condense and the vapour is to remain in equilibrium with the two phase condensate then the interface temperature must be the eutectoid temperature and the condensate composition is being governed by the rates of mass transfer through the vapour phase.

Hence if mixed vapours condense to form a two phase condensate the vapour liquid interface temperature is the eutectoid temperature. The appropriate film temperature difference is therefore given by the difference between the eutectoid and wall temperature ($T_e - T_W$).

The wall temperature T_W has commonly been determined using one of two methods. In the first, suitably spaced thermocouples are used, the mean surface temperature being calculated by an appropriate averaging technique. The second method uses the condenser tube as a resistance thermometer, the tube surface temperature being calculated using methods first proposed by Jeffries (1936).

The choice of method seems to depend on the personal preference of the investigator, although recently the thermocouple method has been the more commonly used technique. This may be because it is easier to interpret exactly what temperature is being calculated.

The fluids used in the experimental studies (see Table 1 for details) vary widely, for example, benzene, carbon tetrachloride, freon 112, turpentine and styrene have all been used as the organic phase in organic water mixtures.

The surface is usually stated to be oxidised copper. Although Stepanek and Standart (1958) and Kawaski et al (1972) do not indicate in their papers whether they used polished copper or oxidised copper surfaces. Various tube diameters have been used, these varying from 6.2 mm (0.244 in) o.d. to 34.9 mm (1.375 in) o.d.

Most investigators do not condense eutectic mixtures specifically, but the data usually includes some of eutectic composition.

Reviewing the various papers it is clear that the effects of such variables as, film temperature difference, tube diameter, and condensate flow regimes are not well understood. Much of the data are very difficult to compare since they have been taken using various condensate compositions. If the heat transfer coefficient is plotted against composition there appears to be a composition dependency. Therefore the most useful data for comparison are those taken for the eutectic vapour mixtures, since here the condensate compositions obtained by various workers should be the same if identical mixtures are considered. The following discussion will deal with eutectic mixtures unless otherwise stated.

The effects of film temperature difference on heat transfer coefficient was studied in detail by Sykes and Marchello (1970). The conclusion, after comparing the data of several authors, was that variation of the coefficient with temperature difference was dependent on the organic-steam mixture being considered.

Looking at the various studies the data from investigators using the same fluids and tube surface are quite different. As an example consider a steam-toluene mixture. Baker and Mueller (1937) and Sykes and Marchello (1970) condensed this mixture on an oxidised copper tube. The tube diameters were 33.4 mm (1.313 in) and 34.9 mm (1.375 in) o.d. respectively. Thus the systems were almost identical, yet the slopes of a plot of ln h_e vs. ln ΔT_f, as determined by Sykes using least squares methods, were + 0.062 for Baker and Muellers data and − 0.130 for Sykes and Marchellos data.

Recently Ponter and Diah (1974) have presented data for benzene-steam and trichloroethylene–steam mixtures condensing on a 28.6 mm (1.125 in) o.d. oxidised copper tube and their results do not agree with the results of Baker and Mueller (1937) either. They suggested the discrepancy was because the tube surface used by Baker and Mueller was not homogeneous, that is the surface properties and hence the condensation mechanism varied along the tube.

The criterion used by Ponter and Diah (1974) to indicate that the surface was homogeneous appears to be when filmwise condensation of steam is consistently produced over the whole length of the tube. If this is the case their suggested explanation of the discrepancy between the two data sets is complicated by the fact that Baker and Mueller (1937) also reported that they too obtained filmwise condensation of steam on their tube. Thus there is still doubt as to why these data sets are different.

The detailed effects of tube diameter on the heat transfer coefficient are far from certain. Both Kawaski et al (1972) and Baker and Tsao (1940) have stated that the heat transfer coefficient increases as the tube diameter increases, this is contrary to the trend when condensing pure vapours or vapour mixtures of miscible liquids. It thus appears that the tube diameter has a marked effect on the heat transfer coefficient, exactly why there is such an effect is unclear.

The differences in behaviour of the various systems may well be due to the condensation mechanism, since quite different descriptions have been given by various authors. The description given by Baker and Mueller (1937) is as follows, the organic forms a film with the water forming standing drops on the tube surface, these drops were "fairly stable and remained on the tube considerable lengths of time and covered the greater portion of the tube". The mechanism reported by Sykes and Marchello (1970) is quite different, they observed the water drops on the organic film, these drops eventually coalesced, and finally formed a continuous water film which then flowed from the tube, over the organic film. They also observed what they termed secondary drainage, that is the water film shedding from the side of the tube.

Thus we have two quite different descriptions of the condensation process occurring on what apparently are similar tubes with the same fluids condensing. It is unfortunate that Ponter and Diah (1974), Stepanek and Standart (1958) and Kawaski et al (1972) have not reported the mechanism in their experiments, since the effects of the mechanism seem to influence the heat transfer coefficients. It is also possible that the observed tube diameter effects are caused by changes in mechanism. However, one cannot be certain of this explanation in view of the lack of descriptions in the relevant papers.

In a recent review Boyes and Ponter (1972) put forward various ideas as to the hydrodynamic behaviour of organic-water mixtures. These ideas arose from studies carried out with organic-water mixtures on a low energy surface (P.T.F.E.) and a high energy surface (copper). They state that "surface and buoyancy forces play equally dominant roles in influencing hydrodynamic and hence heat transfer behaviour".

In particular the value of the heat transfer coefficient is influenced by the position of the water drops in the organic film. Thus, if the organic phase density is less than the water density it would be expected that the water drops would reside at the tube surface. Therefore, disturbing the film and hence enhancing the heat transfer process by promoting better mixing.

If the organic is the denser phase the water drops would float at the vapour liquid interface and little or no enhancement would be expected. However, a complicating factor is the relative growth rates of the film and the drops, as the condensation rate is increased. If, as might be expected, the drops grow faster than the film (i.e. by coalescence as well as condensation) they could become large enough to penetrate the organic film, and again enhancement of the heat transfer process would be expected.

Boyes and Ponter (1972) also proposed, that it should be possible to increase the heat transfer coefficient by increasing the rate of removal of the water drops, since this would increase the disturbance in the film. Further it was suggested that using a P.T.F.E. coated surface would accomplish this increased rate of removal.

Recently Ponter and Diah (1974) have conducted experiments using both an oxidised copper and P.T.F.E. coated copper tubes. The results obtained from this work tend to support the earlier ideas that greater heat transfer coefficients would be obtainable using P.T.F.E. surfaces. Unfortunately most of the enhancement goes into compensating for the resistance of the P.T.F.E. coating, so that in fact the overall enhancement is not very great. However, it does show that if sufficiently thin P.T.F.E. coatings were used an increase in heat transfer coefficient might be obtained.

From the above discussion there are considerable discrepancies between various data sets. It would appear that the mechanism of the condensation process is important in trying to understand such discrepancies, as are the effects of tube diameter. The reasons why the mechanism is apparently different on supposedly identical tube surfaces is at present unknown, unless of course an oxidised copper surface does not give a consistent oxide layer.

2.4.1.2 Investigations using vertical surfaces

Although there have been several investigations using vertical surfaces, there are relatively few studies which treat eutectic mixtures, the study of Bernhardt et al (1972) being the only one to treat eutectic mixtures exclusively.

Bernhardt et al (1972) studied the condensation of various organic steam mixtures on a vertical gold plated copper plate. From their experiments it is apparent that the heat transfer coefficient increases as the film temperature difference decreases, this is contrary to the conclusion made by Hazelton and Baker (1944) for the condensation of various mixtures on several different diameter sanded copper tubes. They stated that the heat transfer coefficient was independent of the temperature difference. However, this conclusion was made on the basis of comparing data taken at various condensate compositions and film temperature differences. Since they do not appear to have systematically varied the film temperature difference at constant condensate composition it is possible that the effect of temperature difference is being masked by a composition dependency.

The effect of the tube diameter has been shown by Baker and Hazelton (1944) to be similar to the horizontal tube case, that is, the heat transfer coefficient increases as the tube diameter increases.

The mechanism of the condensation process has been investigated in considerable detail by Bernhardt et al (1972). They took high speed cine films of the process and also used conductivity probe and dye techniques to identify the various phases. The description given by the above authors is as follows. The organic phase forms a film in which water drops are suspended, large standing water drops touch the metal surface and also protrude through the film. Thus the bulk of the vapour contacts both liquids and both liquids contact the solid. Very small mobile drops of organic were observed to be present on the surface of the large standing water drops. The origin of these organic drops was uncertain, but the authors recognised the possibility that they are nucleating on the surface of the water drops, the nucleation sites perhaps being microscopic dust particles entrained in the inlet vapours. These dust particles are also used to explain the origin of the small water drops floating on the film. The above description of the condensation mechanism appears to have been the same for all condensation rates and for different fluid systems.

Hazelton and Baker (1944) postulated six condensation mechanisms, of these only three were observed in their experiments they are:

(1) Film drop mechanism - here the organic forms a continuous film on the surface, the water forming drops in and on this film.

(2) Channelling mechanism - in this case both phases form films which flow from the surface in separate rivulets.

(3) The third mechanism is a mixture of the previous two.

From their experiments they found that the mechanism observed depended on tube diameter. For the 15.9 mm (0.625 in) and 25.4 mm (1.000 in) o.d. tubes the mechanisms were predominantly of types (1) and (3) while for a 33.4 mm (1.313 in) o.d. tube the mechanism was of type (2). The authors state that a change of mechanism from types (1) to (3) has no marked effect on the heat transfer coefficient whereas the channelling mechanism (type (2)) consistently produced greater heat transfer coefficients than the other two mechanisms. No explanation as to the cause of these effects was given.

From the description of the condensation process, a channelling mechanism can be assumed for the experiments of Tobias and Stoppel (1954) with a 25.4 mm (1.000 in) o.d. brass tube.

In an attempt to predict the condensation mechanism Akers and Turner (1962) introduced the spreading coefficient concept of Harkins and Feldman,

$$S_{BA} = \sigma_A - \sigma_B - \sigma_{AB} \qquad (2.11)$$

where

S_{BA} is the spreading coefficient for B on A.

σ_A and σ_B are the surface tensions of liquids A and B respectively.

σ_{AB} is the interfacial tension between liquids A and B.

If liquid B spreads on liquid A then S_{BA} is positive. A negative coefficient indicates B will not spread on A. Also if B spreads on A then A cannot spread on B.

To use this concept the above authors first assume one component condenses as a film and then look at the behaviour of the other phase on this film. For the case of organic water systems they describe the mechanism as follows. First assume the organic preferentially wets the surface, then if the spreading coefficient for the organic on water is large ($S_{BA} >> 0$), the organic will spread over any water formed on the film. And thus spherical drops of water within the organic film will be formed. For organic liquids which do not spread on water ($S_{BA} << 0$), the water will form as lenses on the film. The mechanism of condensation is hence dependent on the value of S_{BA}. Also at high condensation rates or high water vapour concentrations the lenses coalesce to form channels which flow over the organic film. At near zero spreading coefficients the mechanism would be a mixture of the above processes.

A difficulty in using equation (2.11) is to assign the correct values of surface tension to the various liquids, for example if the pure liquid surface tensions are used for a benzene-water mixture S_{BA} = 8.9, whereas if the surface tensions are those of the mutually saturated liquids S_{BA} = -1.6. Adamson (1967) states that for low surface tension liquids (e.g. organics) in contact with water the final spreading coefficient will be close to zero or negative. Thus the film-drop mechanism of Akers and Turner (1962) should not be realised in practice. However, in the experiments reported by Akers and Turner (1962) all three of their postulated mechanisms were observed.

The descriptions of the condensation mechanisms given above, although apparently different are in fact quite similar, that is the standing drop type mechanism observed by Bernhardt et al (1972) would be like the type (3) mechanism described by Hazelton and Baker (1944) when the large water drops rolled from the surface. Also if the water channels in Akers and Turners (1962) description were touching the metal surface instead of the organic film as described, this too would be similar to the other mechanisms.

It is apparent from the above discussion that the heat transfer coefficient is dependent on condensation mechanism. This mechanism is influenced by tube diameter, condensate composition, condensation rate and tube surface properties. The manner in which these variables effect the mechanism is uncertain, but it would seem that increasing the tube diameter changes the mechanism from a film-drop to a channelling flow. An increase in the water concentration on the tube also causes a film drop mechanism to revert to channelling flow.

Hazelton and Baker (1944) have stated that changing from film drop to channelling flow increases the heat transfer coefficient. Therefore the increase in heat transfer coefficient with increasing tube diameter can be attributed to a change in mechanism. Whether this is also true for horizontal tubes cannot be said but it seems likely.

2.4.2 Multicomponent systems

Very few papers have been published which deal with multicomponent mixtures of vapours which form immiscible liquid phases, two of the more recent papers being due to Yusofova and Neikducht (1970) and Barnea and Mizrahi (1972).

Yusofova and Neikducht (1970) condensed a steam petroleum mixture ("Shirvanneft") on the inside of a horizontal tube. Vapour velocities up to 15 m/s were used in these experiments. The correlation presented contains six empirical constants and was derived specifically for the particular mixture and experimental conditions studied.

Barnea and Mizrahi (1972) propose that for a kerosene steam mixture $h \alpha (Q/A)^{1.26}$ whereas the Nusselt dependence is $h \alpha (Q/A)^{-\frac{1}{3}}$ The authors point out that the condensation process is extremely complex, since the temperature and composition are continuously changing along the length of the condenser.

The paper does serve to point out the dangers in assuming a Nusselt type dependence for predicting heat transfer coefficients for such complex mixtures.

2.4.3 Models and correlations

Most authors have presented some form of empirical correlation and/or model of the condensation process. The correlations and models can be classified into three basic types:

a) Homogeneous models, these usually use Nusselts equation with the physical properties averaged in some manner.

b) Shared surface models, these assume that the two liquid phases form seperate films which do not interfere with one another.

c) Other models, these usually start with specific assumptions on flow patterns, or are derived empirically using intuitive mechanistic arguments.

Not all of the existing correlations will be mentioned in the following sections since some of them apply only to a single specific system. The correlations not included can, however, be found in table 1.

2.4.3.1 Homogeneous models

The first correlation of this type was presented by Baker and Mueller (1937), the equation is as follows,

$$h\left[\frac{\mu_1^{2}}{k'^{3}_{av}\,\rho_{av}^{2}\,g}\right]^{\frac{1}{3}} = 1.28\left[\frac{C_{pav}\,\mu_1\,\rho_{av}^{0.7}}{k'_{av}\,\lambda_{av}}\right]^{2.38}\left(\frac{Q_1}{Q}\right)^{-3.28} \qquad (2.12)$$

where

k'_{av} is a volume average of the pure liquid thermal conductivities.

C_{pav}, ρ_{av} and λ_{av} are weight averages of the specific heats, densities and latent heats of the pure liquids respectively.

μ_1 is the viscosity of the wall wetting phase.

Q_1 is the heat load of the wall wetting phase.

Q is the total heat load.

The constant in equation (2.12) is not dimensionless and is valid only for the British engineering system of units. This equation is only accurate for Baker and Mueller's own data.

The equation presented by Kawaski et al (1972) to correlate data obtained using vapour crossflow over various diameter tubes is,

$$Nu = 0.0295\ (Ga.\ Ku.\ Pr.)^{\frac{1}{4}}\ Re_v^{\frac{1}{2}}. \qquad (2.13)$$

This may be rewritten as

$$h = 0.0295\left(\frac{k'_{av}\,\rho'^{2}_{av}\,g\,\lambda_{av}}{D_o\,\mu'_{av}\,\Delta T_f}\right)^{\frac{1}{4}}\ Re_v^{\frac{1}{2}}, \qquad (2.14)$$

where: k'_{av}, ρ'_{av}, and μ'_{av} are volume averages of the pure liquid thermal conductivities, densities and viscosities respectively.

λ_{av} is the weight average of the pure liquid latent heats of vapourisation.

Re_v is the vapour crossflow Reynolds number.

This correlation predicts their own data quite well, but has not been tested against other data sets. However, since the correlation is specifically for vapour crossflow it is not applicable to the bulk of the available experimental data, where stagnant or near stagnant vapour conditions have been used.

Akers and Turner (1962) presented the following general equation,

$$h \left(\frac{\mu_1^2}{k_{av}'^3 \rho_{av}^2 g} \right)^{\frac{1}{3}} = 1.47 \left(\frac{4\Gamma}{\mu_1} \right)^{-\frac{1}{3}} \qquad (2.15)$$

where μ_1 is the viscosity of the film forming component

ρ_{av} is the weight average of the pure liquid densities

k_{av}' is the volumetric average of the pure liquid thermal conductivities

Γ is the mass flow rate of the condensate per unit width of condenser surface.

If equation (2.15) is compared with equation (2.2) it can be seen that it is simply Nusselts equation with averaged physical properties. Akers and Turner (1962) stated that their equation was suitable for mechanisms of the film drop or film lens type, since for these the major resistance to heat transfer would be expected to be that of the organic film.

Bernhardt et al (1972) have shown that equation (2.15) predicts the majority of the data from several authors to within an average error of ± 20%.

2.4.3.2 <u>Shared surface models</u>

The first model of this kind was proposed by Kirkbride (1933). His equation is,

$$h = (Q_1 h_{N1} + Q_2 h_{N2}) / (Q_1 - Q_2) \qquad (2.16)$$

where h_{N1} and h_{N2} are the Nusselt coefficients for components 1 and 2 respectively.

Q_1 and Q_2 are the heat loads for components 1 and 2 respectively.

This equation is simply the Nusselt coefficients for the pure liquids weighted on a heat load basis. Akers and Turner (1962) presented equation (2.16) in a slightly different form, that is.

$$h = (a \lambda_1 h_{N1} + b \lambda_2 h_{N2}) / (a \lambda_1 + b \lambda_2) \qquad (2.17)$$

where

a and b are the weight fractions of components 1 and 2 in the condensate.

Akers and Turner recommended that equation (2.17) should be used for channelling flows. They also recommend the equation be multiplied by 0.8 in order to predict their own experimental data.

Bernhardt et al (1972) proposed the following general correlation,

$$h = v_1 h_{N1} + v_2 h_{N2} \qquad (2.18)$$

where

v_1 and v_2 are the volume fractions of components 1 and 2 in the condensate.

This equation weights the Nusselt film coefficients on a volume fraction basis. Bernhardt et al (1972) have shown that equation (2.18) predicts the existing data, for various systems, to within an average error of ± 15.0%.

However, if the above equation is compared with data taken on tube diameters less than 25.4 mm (1.0 in) o.d. it does not predict the data well. This is expected since the equation has the Nusselt diameter dependence and it was shown earlier that the dependence for the "immiscible condensation case" is opposite to that of Nusselts equation.

2.4.3.3 Other models and correlations

Baker and Tsao (1940) presented two empirical equations for evaluating the heat transfer coefficients, namely,

$$h / (1 - \frac{0.0167}{D_o}) = [500 / (1 - 0.0085\ v_2)] + 80 \qquad (2.19)$$

and

$$h = \{366(1/D_o)^{\frac{1}{4}} (1 - \frac{0.0284}{D_o}) / (1 - 0.0035\ v_2)\} + 1.67/D_o \qquad (2.20)$$

where D_o is the tube diameter in feet,

v_2 is the volume fraction of component 2 in the condensate.

The authors state that the above equation should only be used for tube diameters between 12.7 mm (0.5 in) and 38.1 mm (1.5 in) o.d. This warning is justified since both of the above equations exhibit strange behaviour at tube diameters outside the quoted range.

Hazelton and Baker (1944) attempted a theoretical derivation in their study. They used a model based on the channelling mechanism, but found they could not compute the areas occupied by the two liquid films. Eventually they applied the model to the film drop mechanism, the resulting equation for a vertical tube being,

$$h = h_H = 0.943 \left[\frac{k_1\ \rho_1\ (a\lambda_1 - b\lambda_2)\ g}{\mu_1\ L\ a\ \Delta T_f} \right]^{\frac{1}{4}} \qquad (2.21)$$

The above equation failed to correlate their own data, and finally the following empirical correlations were derived. For vertical tubes,

$$h = 79\ [(a\lambda_1 + b\lambda_2) / a\ L]^{\frac{1}{4}} \qquad (2.22)$$

and for horizontal tubes

$$h = 61\ [(a\lambda_1 + b\lambda_2) / a\ D_o]^{\frac{1}{4}} \qquad (2.23)$$

Bernhardt et al (1972) have shown that these equations are capable of predicting a large amount of existing data to within ± 20%, the predictions falling outside these limits often being conservative.

Stepanek and Standart (1958) attempted a theoretical derivation based on simple hydrodynamic and heat transfer models. The hydrodynamic model assumed the action of the floating water droplets on the organic film could be considered the same as the action of a water film of equivalent thickness. It was also assumed that this water film flows at a constant velocity equal to that of the surface of the organic film.

The heat transfer model was formulated on the basis that the complicated drop shape could be replaced by that of a pillbox, its diameter being equal to the maximum drop diameter and its volume being that of the drop. A uniform distribution of drop sizes was also assumed.

The above authors failed to arrive at an analytical solution due to intractable mathematical difficulties. Their analysis did suggest the importance of various parameters, namely the film temperature difference and a surface tension and density ratio effect. Finally empirical correlating techniques were employed to derive an equation to fit their data. The final equation contains several empirical constants, and is valid for horizontal tubes only. The equation is,

$$h_e = 0.725 \left[\frac{\lambda_H \, k_1^{\,3} \, \rho_1^{\,2} \, g}{\mu_1 \, \Delta T_f \, D} \right]^{\frac{1}{4}} K_1 (1 + K_2 \, \Delta T_f) \qquad (2.24)$$

where

$$K_1 = [1 - 4.38 (b/a)^{0.033} (\rho_1/\rho_2)^{0.62} (\Delta\sigma/\sigma_2)^{3.2}]^{\frac{1}{4}}$$

$$K_2 = 0.0584 (a/b) (k_1/k_2)^{0.5} (\rho_2/\rho_1)^{1.4} (\Delta\sigma/\sigma_2)^{1.6}$$

and $\lambda_H = \dfrac{a \, \lambda_1 + b \, \lambda_2}{a}$

This equation was derived from data treating eutectic mixtures only, and thus should be used with great caution for any other mixtures.

Sykes and Marchello (1970) present three correlations in their study, the first being empirical of the form,

$$h_e / h_{N1} = (1.0 - 0.8R) (\Delta T_f)^n \qquad (2.25)$$

where $n = 0.67R$ and R is defined as

$$R = (\Delta r \; \rho_2 / r)^{0.5} \qquad (2.26)$$

$\Delta r = 1 - r$ and $r = \rho_1 / \rho_2$

The exponent n was then determined by least squares techniques using all of the available eutectic data. The value of n was different for each system considered.

The actual temperature dependence of h_e in equation (2.25) is $\Delta T_f^{\,n - 0.25}$, thus if $n > 0.25$ h_e will increase as ΔT_f increases. This condition should be met for all organics with specific gravities < 0.88, for specific gravities > 0.88 n is < 0.25 and h_e should decrease as ΔT_f increases. Comparing the above statements with Sykes and Marchello's (1970) calculated values of n, it is evident that of the seven mixtures considered all but two of them obey the above rules.

The second correlation was derived from a laminar two film model. Here they assumed a film of organic flowing adjacent to the tube wall with the water film flowing over the organic film.

In deriving the equations for this model an algebraic error was made (see Sykes (1968) eqtn.D-12 p 121to D-30 p 126), however, when corrected the conclusion that the model does not agree with available data still holds.

When comparing their earlier models against experimental data Sykes and Marchello (1970) noticed that the eutectic coefficient was in some cases lower than the pure organic coefficient. This led them to postulate the existence of a nucleation barrier to steam condensation, that is, with a film of organic covering the surface, there are no nucleation sites for the condensation of steam. This causes a resistance to heat transfer and hence the overall heat transfer coefficient (from vapour to condenser wall) could conceivably fall below that for the pure organic coefficient.

The final correlation derived from nucleation arguments was as follows,

$$h_e/h_{N1} = \left[K_3 + 1 / (K_4\, e^{\beta\Delta T_f}) \right]^{-1} \tag{2.27}$$

where $K_3 = [7.6 - 1.8\,(Pr_1 - Pr_2)]^{-1}$

$$K_4 = \left[17.3 \times 10^{-10} Pr_1 \left(1 + \frac{a\lambda_1}{b\lambda_2}\right) \right] / \left[(Oh)_1 (\left(\frac{\Delta\sigma}{\sigma_1}\right)^{\frac{1}{2}} \left(\frac{\mu_1}{\mu_2}\right) \left(\frac{M_2}{M_1}\right) \right]^2$$

$Pr = C_p\mu/k$

$Oh = (\mu^2/\rho\, g D_o \sigma)^{\frac{1}{2}}$

$e^{\beta\Delta T_f}$ is the rate of nucleation of water drops on the organic film

$\beta = 0.035\ {}^{\circ}F^{-1}\ (0.063\ K^{-1})$

Of the three models proposed by Sykes and Marchello (1970) the nucleation model was the most successful over a wide range of systems, however, the empirical expression (equation 2.25) did give a better fit for some systems.

The correlation presented by Tobias and Stoppel (1954) was obtained by the use of dimensionless groups deduced from dimensional analysis, the final equation being,

$$h = h_H / \left\{ 1.0 - \frac{1}{545} \left[\left(\frac{\Delta\sigma^3 \Delta\rho}{\mu_1^{\,4}\, g} \right) \left(\frac{m_1\, \rho_2\, k_2}{m_2\, \rho_1\, k_1} \right)^{0.5} \right]^{0.21} \right\} \tag{2.28}$$

where m_1 and m_2 are the mass rates of condensation for components 1 and 2 respectively.

The authors recommend that equation (2.28) should only be used within the composition range 8–98% water.

Recently Marschall and Hickman (1973) presented a purely theoretical study of the problem. They applied the conservation equations to both the liquid and vapour phases in order to solve the condensation problem. The flow model assumed was the laminar two film model of Sykes and Marchello (1970). Marschall and Hickman (1973) state that if the film temperature difference is greater than 15°C then the heat transfer resistance in the vapour boundary layer may be neglected. Thus in this case the heat transfer resistance may be obtained by considering the hydrodynamics of the film only. Since Sykes and Marchello (1970) have previously concluded that the two film model is inadequate, then presumably the above model is also inadequate.

Another recent paper (Salov and Danilov (1975)) uses the two film model to investigate the effects of variable wall temperature when condensing on vertical surfaces or horizontal tubes.

Instead of assuming a constant wall temperature Salov and Danilov (1975) use equations describing the wall temperature variation, derived empirically from experimental data. The equations used are as follows: for a horizontal tube,

$$T_W = \bar{T}_W + C_1 \cos \phi \qquad (2.29)$$

and for a vertical surface,

$$T_W = \bar{T}_W + C_2 + C_3 \left(\frac{x}{L}\right) + C_4 \left(\frac{x}{L}\right)^2 \qquad (2.30)$$

where $\bar{T}_W$ is the mean wall temperature

C_1, C_2, C_3 and C_4 are constants

ϕ is the angle at which T_W is being calculated

x is the height at which T_W is being calculated

L is the length of the vertical surface.

The above authors conclude that assuming a constant wall temperature instead of a variable wall temperature has no significant effect on the calculation of the mean heat transfer coefficient. Whether this conclusion would hold if a different model of the heat transfer process were used is not known.

Broadly the models and correlations above fall into two main areas:

(1) Empirical - these are usually derived from limited experimental data.

(2) Models - these employ some model of the heat transfer process.

The correlations of type (1) are usually restricted to the data sets used in their derivation. Great care is needed if they are used outside these limits.

Correlations of type (2), which use a model of the condensation process should be capable of handling any situation for which the model is valid.

One of the main problems in trying to predict heat transfer coefficients for immiscible systems is in determining the correct condensation mechanism and hence heat transfer model. As we have seen earlier the way in which variables such as tube diameter, tube surface properties, condensation rate and condensate composition influence the condensation mechanism is not clearly understood. In view of the above uncertainties it is not surprising that the previously outlined models and correlations break down under certain circumstances. In fact it is perhaps surprising that equations (2.15), (2.18), (2.22) and (2.23) are so successful.

2.5 Conclusions

(1) The heat transfer coefficient obtained during the condensation of vapours of immiscible liquids depends on the mechanism of condensation.

(2) The condensation mechanism is influenced by the tube diameter, tube surface properties, condensation rate and condensate composition; the way in which these variables effect the mechanism is not clearly understood.

(3) Difficulties in the nucleation of water drops onto the organic film could be important.

(4) Most of the correlations and models proposed are limited to the experimental data or assumed mechanism used in their derivation. Because of the difficulties in determining the condensation mechanism the choice of the appropriate heat transfer model is difficult. Equations (2.15), (2.18), (2.22) and (2.23) appear to be the best general correlations, although they too can be out by over ± 100% in certain circumstances.

Chapter 3

Apparatus and Procedure

3.1 Introduction

It is common industrial practice to use horizontal shell and tube condensers, The design of such units for vapours of immiscible liquids is not well understood, nor are the mechanisms of the condensation process. The present experimental facility has been designed in an attempt to improve this understanding. A single tube horizontal shell and tube condenser was selected for this study because it provides a relatively simple means of obtaining the necessary heat transfer and mechanistic data needed to improve our understanding of the condensation process.

A flowsheet and photograph of the apparatus are shown in Fig. 3.1 and 3.1a respectively, the essential features are:-

(1) Vapouriser circuit

(2) Test section

(3) Condensate circuit

(4) Cooling water circuit

(5) Condenser tube

(6) Total condenser

The above items will now be described in **detail**.

3.2 Apparatus

It was known at the beginning of the present study that the liquids used would be both toxic and in some cases highly inflammable, The first consideration in designing the rig was thus safety. The laboratory in which the apparatus was built is fully flameproofed, and hence all electrical equipment and spark inducing devices had to be either eliminated or flameproofed. Ventillation is provided by large fans which suck air from a set of ducts at floor level. In case of fire the laboratory is protected by an automatic fire

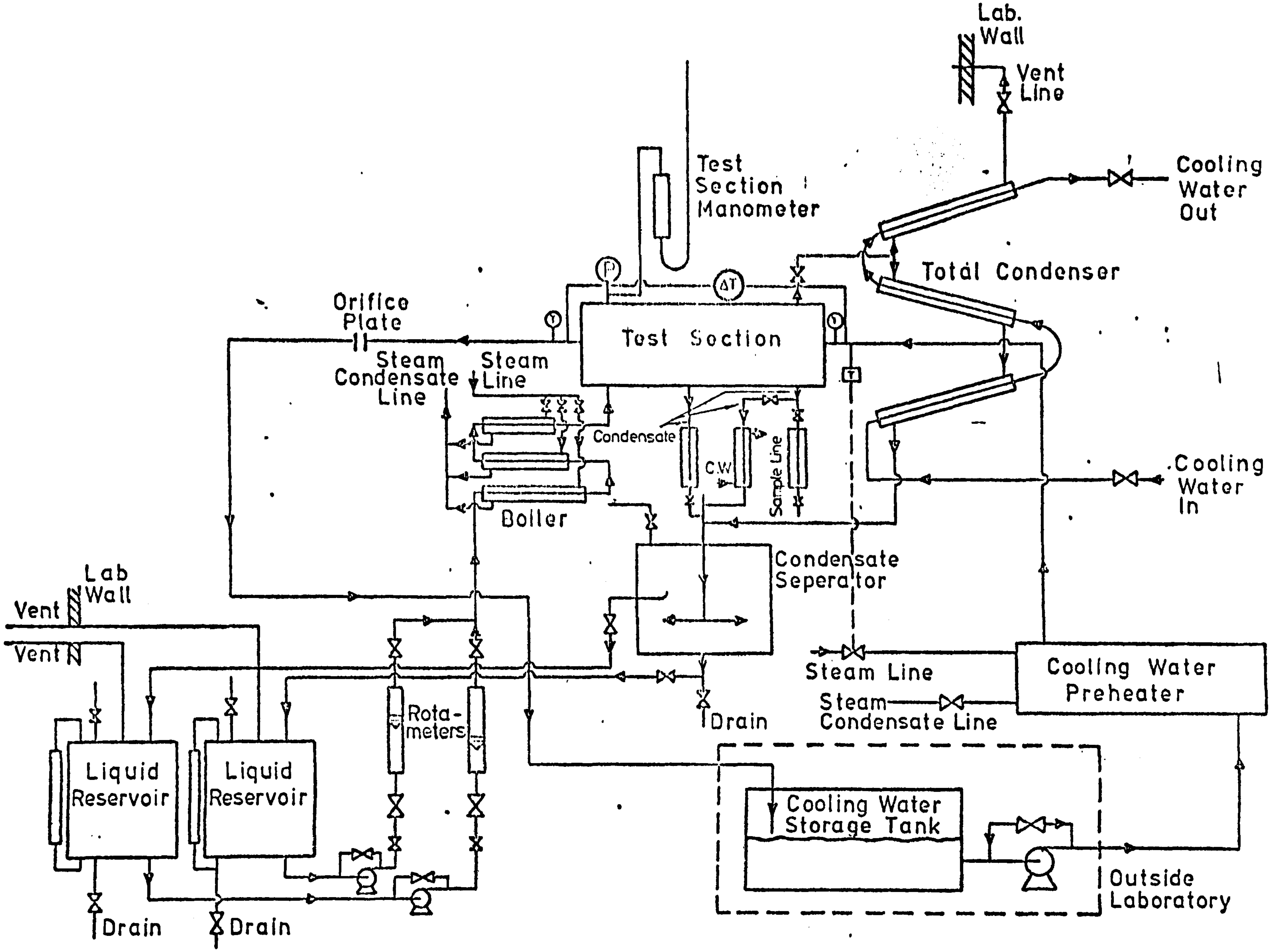

FIG. 3·1. FLOW DIAGRAM OF THE APPARATUS.

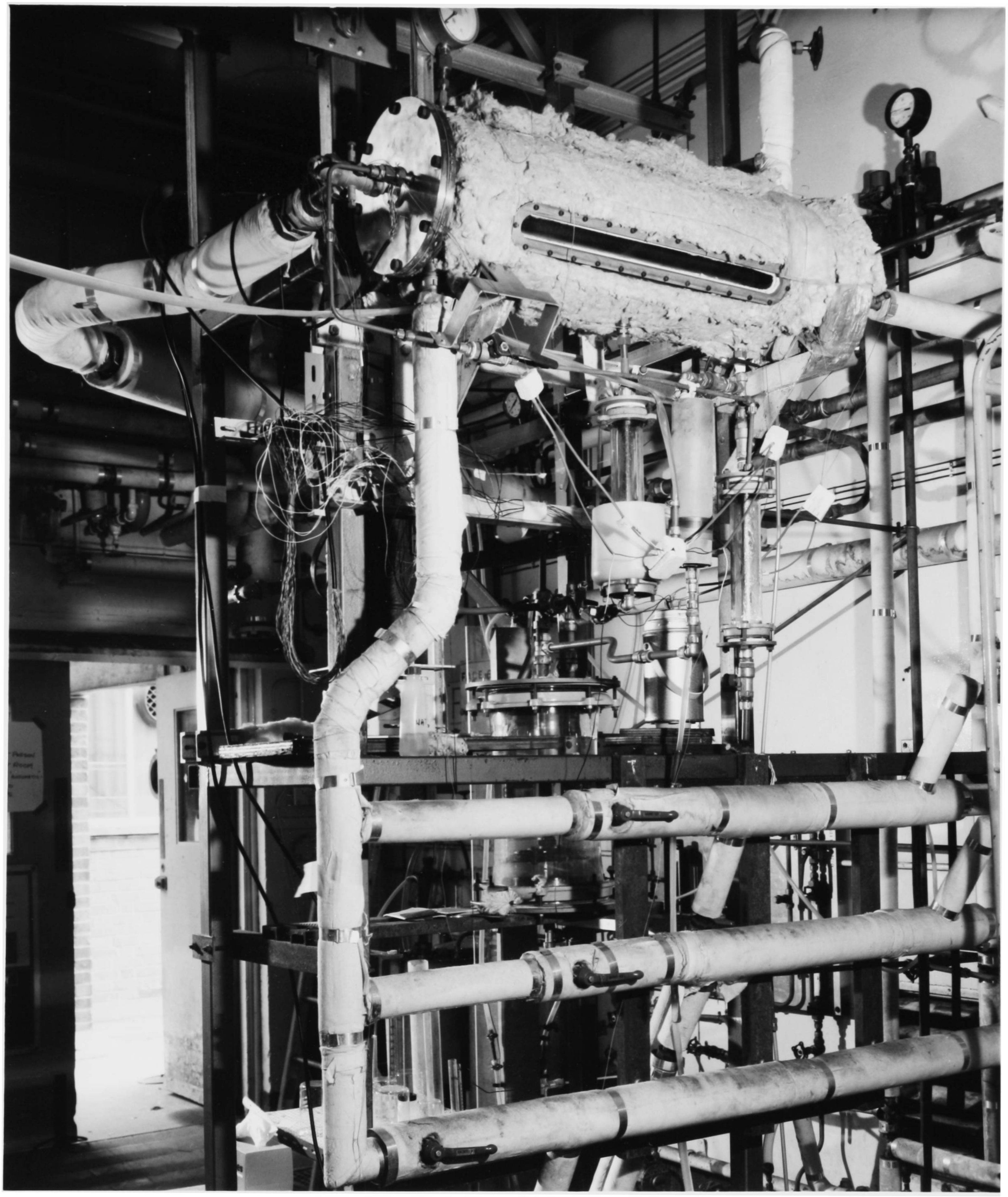

Fig. 3.1a OVERALL VIEW of the APPARATUS

fighting system, which will flood the laboratory with carbon dioxide gas if tripped. Under normal working conditions the system is operated manually, since there is a danger that any personel in the laboratory when the system is triggered will suffocate.

3.2.1 Vapouriser circuit - Liquid is pumped from the two stainless steel reservoirs (approx. capacity 60 litres each) by 0.56 kW compressed air operated gear pumps. The flowrate of each liquid is controlled by a globe valve and is metered by a rotameter (accuracy ± 2.0%).

The vapouriser consists of three jacketted copper tubes. The process fluid flows through the inner (2.54 cm i.d.) tube, whilst steam condensing in the annular space between the inner and outer jacket (3.81 cm i.d.) provides the heat to boil the liquids. Twisted metal tapes were installed in the inner tubes, these ensure good heat transfer and hence total vapourisation of the liquids.

All three tubes are independently heated, their respective lengths being 1.2 m, 0.6 m and 0.5 m, giving a maximum heated length of 2.3 m.

The resulting vapours are then delivered to the test section by a 2.54 cm i.d. copper pipe. The whole of the vapouriser circuit is lagged with fibreglass.

3.2.2 Test Section - The condenser tubing runs through the centre of a 21 cm i.d. stainless steel shell 92 cms in length. The shell has three windows (61 cms x 5.0 cm) spaced 120° apart (see Fig.3.2), these are provided so that observation along the whole length of the condensing surface is possible. Special heat resistant glass was used for these windows.

The incoming vapours enter the shell through a bent copper tube 1.3 cm i.d., (see Fig.3.2) sixteen 0.7 cm diameter holes provide the vapour flow area. The purpose of this tube is to ensure that the vapour flows parallel to the condenser tube.

Condensate is collected from the central portion of the tube in an

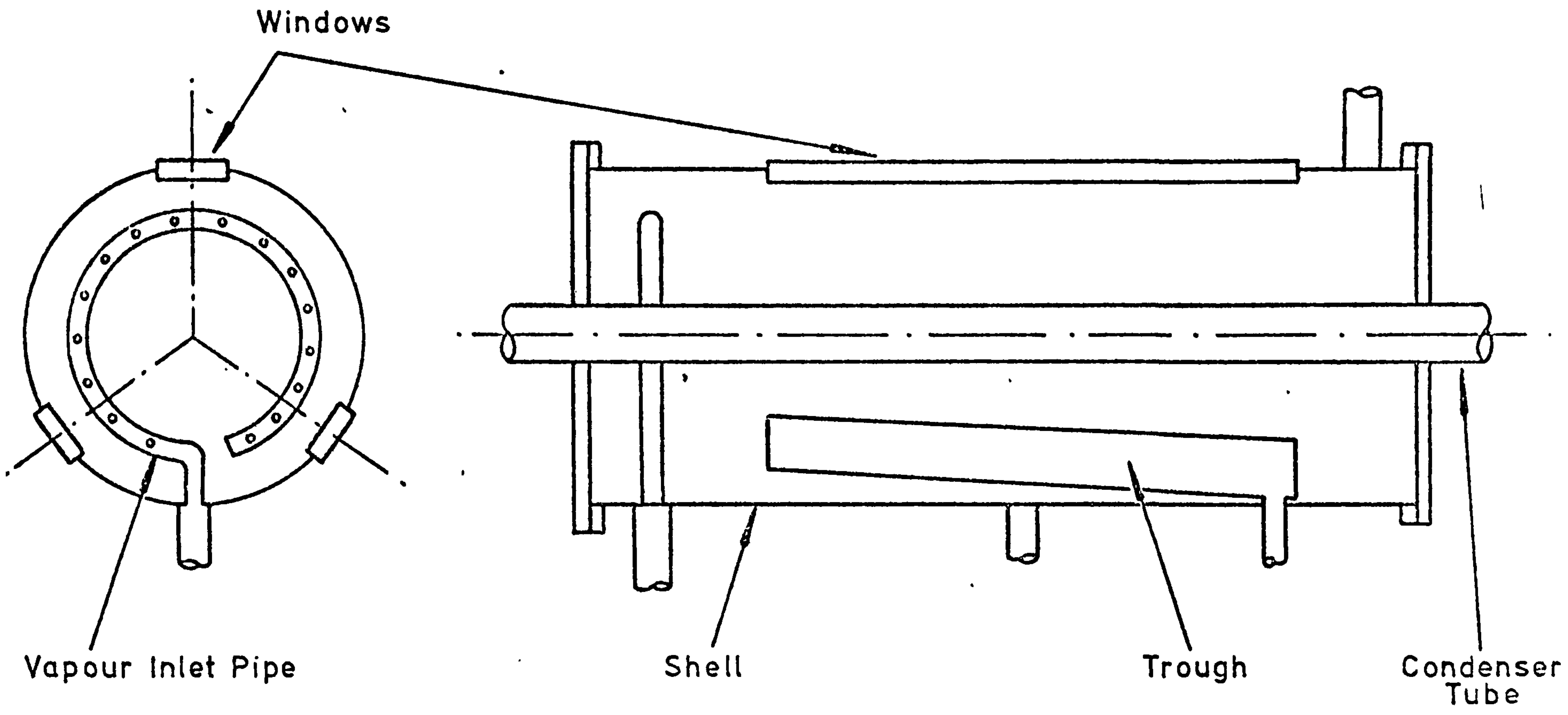

FIG. 3·2. DIAGRAM OF THE TEST SECTION.

inclined trough (61 cms x 5.0 cms), it passes out of the test section via a 1.3 cm i.d. copper pipe. The condensate collected in the shell also drains through a 1.3 cm i.d. copper pipe.

Excess vapours and condensate are removed at the end of the shell opposite to the vapour inlet. The excess vapours pass through a 2.54 cm i.d. copper pipe to the total condenser.

The vapour temperatures at the inlet and outlet ends of the shell are measured by stainless steel sheathed chromel—alumel thermocouples. Test section pressure is determined by a pressure gauge and a water manometer. The shell is also lagged with fibreglass.

3.2.3 Condensate circuit - The condensate passes from the shell drain line into either a sampling vessel or through a water cooled line to the separator. A glass vessel 23 cms o.d. and 38 cms in height is provided as the separator, When running the rig in practice the condensate is drained from the separator into storage tanks where it is allowed to settle before being returned to the main reservoirs.

3.2.4 Cooling Water circuit - Cooling water is pumped from a large storage tank via a 7.5 kW centrifugal pump through a 5.0 cm i.d. copper pipe to the preheater. From the preheater water is delivered to the test section at a set temperature automatically controlled to ± 0.5oC. Heating is provided by condensation of steam in the tubes of a shell and U tube condenser. The unit is approximately 1.5 m long and is rated by the manufacturer at 150 kW.

A thermocouple in the cooling water outlet pipe provides a signal to a feedback control loop which adjusts the steam valve setting. The controller operates in the proportional + integral mode.

From the test section the water flows back to the storage tank through an 8.0 cm i.d. galvanised iron pipe. The water flowrate is determined by a 3.06 cm orifice plate situated in the line. The pressure drop across the orifice plate is measured by an inverted water manometer, the orifice plate was calibrated by measuring the water flowrate (by collecting a known weight in a known time) and noting the corresponding manometer reading.

The cooling water temperature is determined at the inlet and outlet ends of the test section by two stainless steel sheathed chromel-alumel thermocouples, these were calibrated against National Physical Laboratory (N.P.L.) tested mercury in glass thermometers and estimated to be accurate to ± 0.1°C. An independent check on the cooling water temperature rise is made by using a system of four thermocouples arranged to give the difference in temperature at the inlet and outlet ends of the test section. This device was calibrated against two N.P.L. calibrated platinum resistance thermometers and has an estimated accuracy of ± 0.05°C.

All thermocouples are connected to a "Modulog" data logging system, capable of handling up to fifty channels of input data.

3.2.5 Condenser tube - Copper tube 2.54 cm o.d., 1.905 cm i.d. and 122 cms long is used in the test section. Only 61.0 cms of tube are used when taking experimental data. Two such tubes were manufactured, the first had nine thermocouples (copper-constantan) and the second twelve arranged as shown in Fig. 3.3.

The thermocouples were embedded in the tube wall in the following manner. A small copper plug was soldered to the end of a constantan wire and the surface of the plug was copper plated. The plug was then soldered into a hole drilled in the tube wall, the thermocouple leads being taken out through the centre of the tube. Fig. 3.3 shows the details of the above procedure. When all the thermocouples had been installed the plugs were filed flush with the tube surface. A copper wire soldered to one end of the copper tube provides the other thermocouple lead.

In order to provide a condenser with uniform surface properties the following procedures were adopted. The nine thermocouple tube was polished with emergy paper, the final finish being achieved with grade four polishing paper. It was then thoroughly washed with acetone and distilled water.

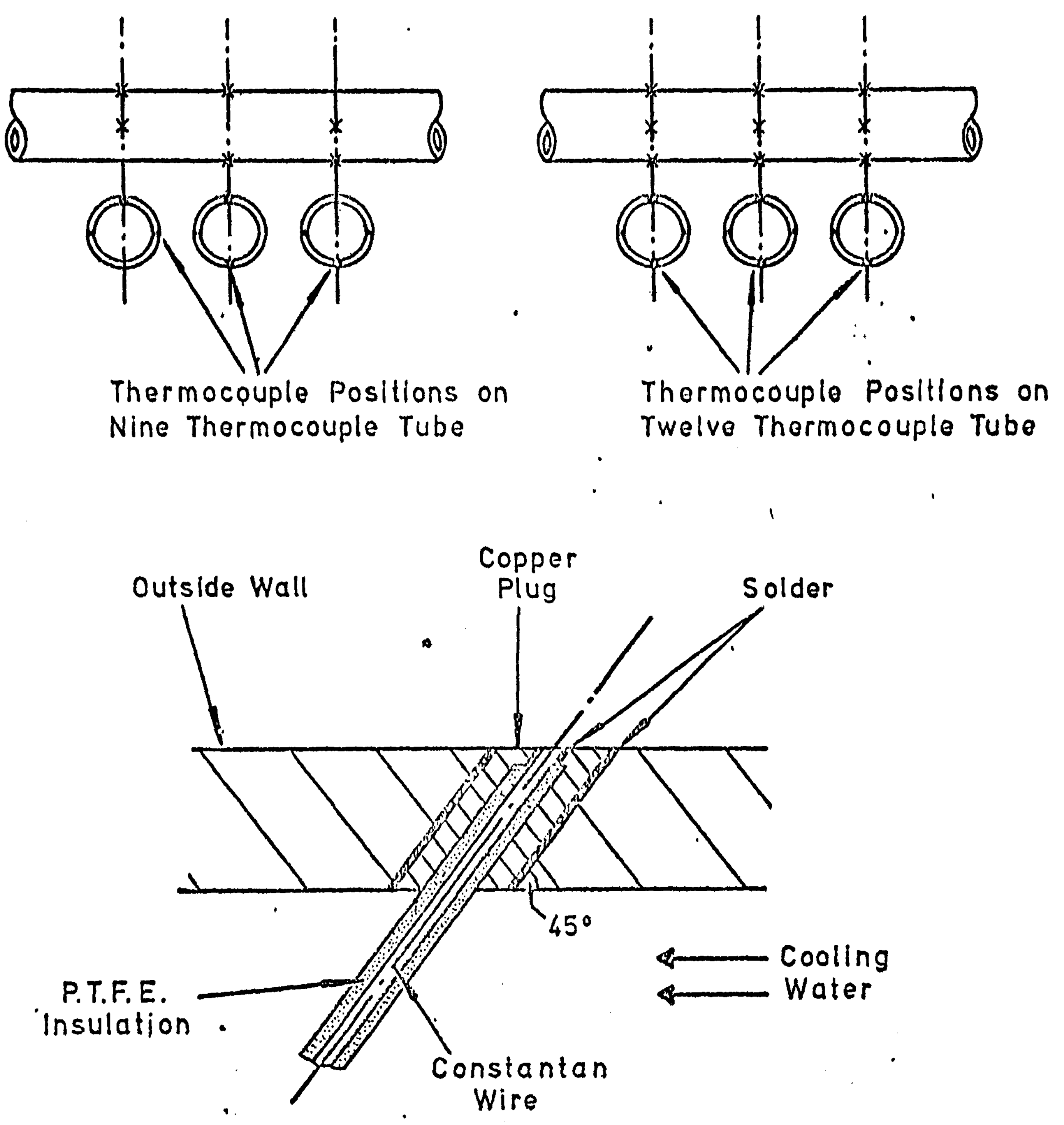

FIG. 3·3. DETAILS OF THERMOCOUPLE INSTALATION.

Before taking any experimental measurements the tube was used as a steam condenser until it consistently produced filmwise condensation over the whole tube length, this process took twenty days.

The twelve thermocouple tube was first copper plated and then gold plated before use. This procedure was adopted because a gold plated surface would not be affected by any of the chemicals used in this study and therefore a reproducable surface would be obtained.

3.2.6 Thermocouple calibration - The thermocouples in the condenser tubes were calibrated in two ways; in the first method the tube was placed in a glass jacket and water from a constant temperature bath was passed through the jacket and tube back to the bath. Four previously calibrated chromel-alumel thermocouples were used to measure the water temperature. All of the condenser tube and water thermocouple outputs were recorded by the "Modulog" data logger, and printed out by an I.B.M. typewriter. The sampling speed was usually set at 1 channel/s but could be increased to 2 ch/s if necessary, the sensitivity of the data logger was to within ± 1μV and the estimated accuracy of the calibration was ± 0.2^{o}C. The second calibration method was an "in situ" procedure developed to check that no drift occurs during operation. In this method the apparatus is operated with only the cooling water supply turned on, since the temperature at the inlet and outlet end of the tube are known. The heat lost through the tube by convection and radiation can be estimated, hence the error in assuming the water temperature is the same as the tube surface temperature can be calculated. The accuracy of this method has been estimated at better than ± 0.2^{o}C.

3.2.7 Total condenser - The excess vapour from the test section flows through a 2.54 cm i.d. copper pipe into the total condenser. This consists of 3.6 m of jacketed copper tube. Vapour flows in the annular space between the jacket (3.81 cm i.d.) and the inner tube (2.54 cm i.d.), while cooling water from the mains flows through the tube. A 1.27 cm i.d. copper tube fixed into the top section of the total condenser acts as a vent line for any incondensable gases.

3.2.8 Liquids used - Demineralised water was used in all experiments. The toluene used was a sulphur free grade obtained from May and Baker, while the trichloroethylene was a purified grade obtained from B D H Chemicals Ltd.

3.3 Procedure

The start up procedure was as follows:

(1) An ice water mixture was placed in the cold junction dewar flasks.

(2) The cooling water supply to the total condenser was turned on

(3) The cooling water supply to the test section was turned on and the automatic controller adjusted to give the required water temperature.

(4) Steam to the preheater and the vapouriser was turned on

(5) The fluids to be used were pumped to the vapouriser at the desired flowrates.

(6) The valve on the vent line was opened slightly and left open. Operating at pressures greater than atmospheric then ensures that any incondensable gases are continuously vented from the system. The effectiveness of this procedure is discussed later (see section 6.2).

When the cooling water inlet temperature, condensate flowrate and several wall temperatures were constant a measurement was made. The time taken from start up to the first measurement was typically two hours. Approximately ten to twenty minutes were required to bring the system back to steady state after a small change in the cooling water temperature was made.

The readings taken were as follows:

(a) Condensate volume, collection time and temperature.

(b) Orifice plate manometer reading and manometer fluid temperature.

(c) Test section pressure gauge and manometer readings.

(d) Thermocouple outputs, these are recorded continuously whilst taking the other readings.

(e) Rotameter readings on the input lines to the vapouriser.

(f), Barometric pressure

Using the above measured values the data were processed using a computer programme. The physical property correlations used are listed in Appendix A.

Chapter 4

Results

4.1 Introduction

In this chapter the experimental results are presented. The first section deals with the pure component data and the second binary mixture data. All of the data presented below in graphical form are tabulated in detail in Appendix B. A detailed error analysis of the results is given in Appendix G. From the analysis it can be seen that the measured heat transfer coefficients are accurate to within ± 15.0% for film temperature differences greater than approximately 4.0°C.

4.2 Pure component data

The pure components used were steam, toluene and trichloroethylene; the tube used was the oxidised copper tube described earlier (Chapter 3 section 3.2.5). All three systems condensed in the filmwise manner.

The steam data are shown in figure 4.1 and tabulated in table B1, toluene data are shown in figure 4.2 and tabulated in table B 2 while the trichloroethylene data are shown in figure 4.3 and tabulated in table B 3.

4.3 Immiscible liquid data

4.3.1 Heat transfer data

The mixtures used were steam-toluene and steam-trichloroethylene. Both mixtures were condensed on the oxidised copper and gold plated tubes.

The steam-toluene data for the oxidised copper and gold plated tubes are shown in figures 4.4 and 4.5, and tabulated in table B 4 and B 5 respectively, while the steam-trichloroethylene data for the oxidised copper tube and gold plated tube are shown in figures 4.6 and 4.7, and tabulated in table B 6 and B 7 respectively.

4.3.2 Observed flow patterns

The flow pattern observed depended on the tube surface being used. For the oxidised copper tube the mechanism for the steam-toluene mixture was as follows. Both phases formed irregular films on the tube surface (see figure 4.8). The organic film contained water drops which adhered to the tube surface. On or in these water drops smaller organic drops were observed. These were moving very rapidly. Also the water film had organic drops in it, however, in this case the drops moved freely with the water film.

The steam-trichloroethylene condensing on the oxidised copper surface gave a flow pattern similar to the one described above (see figure 4.9). However, in this case the rivulets were much more clearly defined forming discrete bands on the condenser surface.

The flow pattern for the steam-toluene mixture condensing on the gold plated tube was as follows. The toluene formed a film, and the water standing drops within this film. Rivulets of water were also observed (see figure 4.10). Small water drops flowed on or in the organic films. Again small organic drops flowed on or in the standing water drops. Further it was observed that when a standing drop drained from the surface a rivulet was formed, this contained flowing organic drops. Another phenomenon was the existance of patches under the water drops, these disappeared when the organic film flowed over them.

The flow regime for the steam-trichloroethylene mixture on the gold plated tube was similar, to that for the steam-toluene mixture, however, in this case their were no continuous rivulets and far more standing drops (Figure 4.11).

When the larger standing water drops drained a track of small water drops was left in their wake.

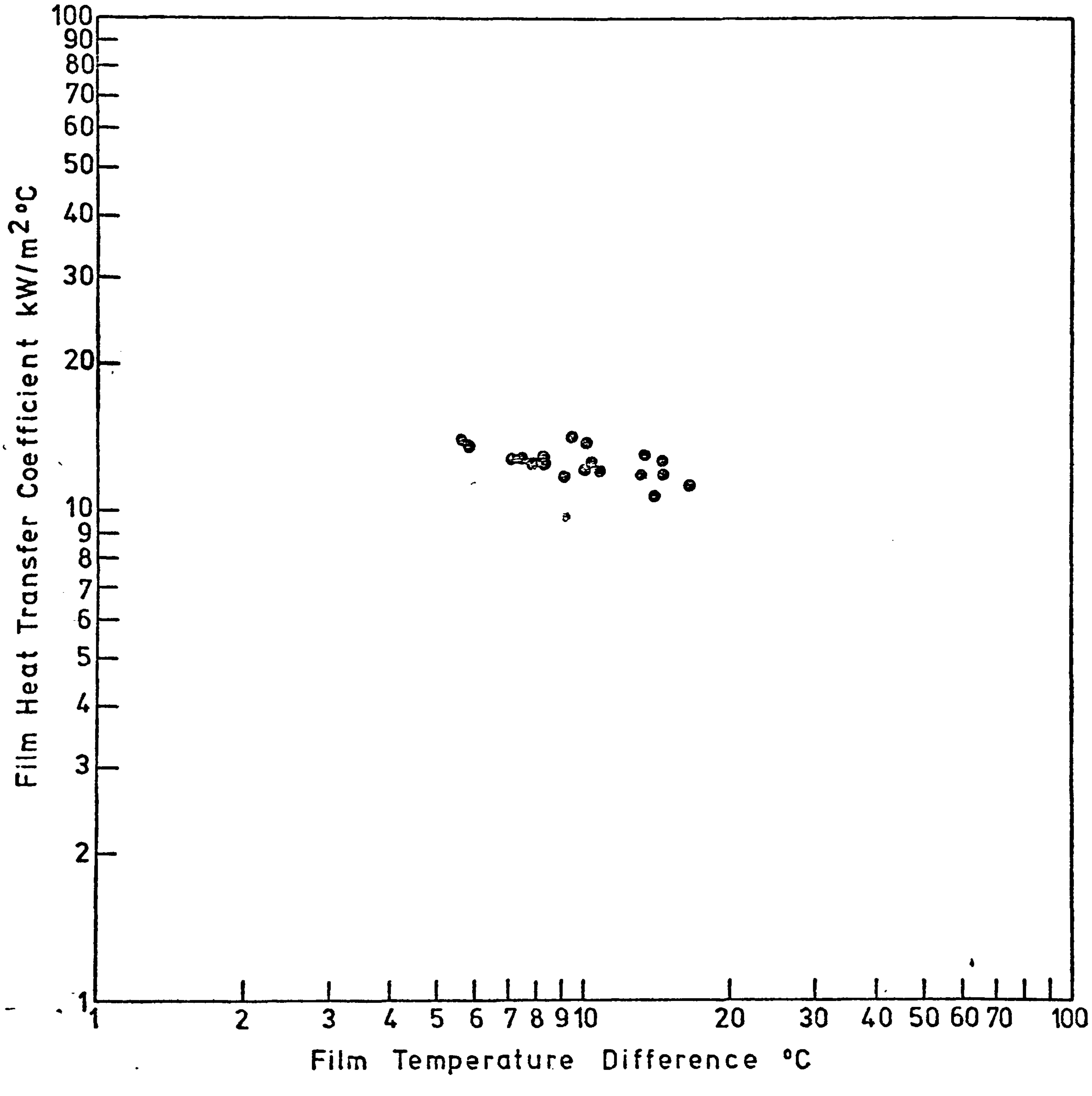

FIG. 4·1. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF PURE STEAM ON AN OXIDISED COPPER TUBE.

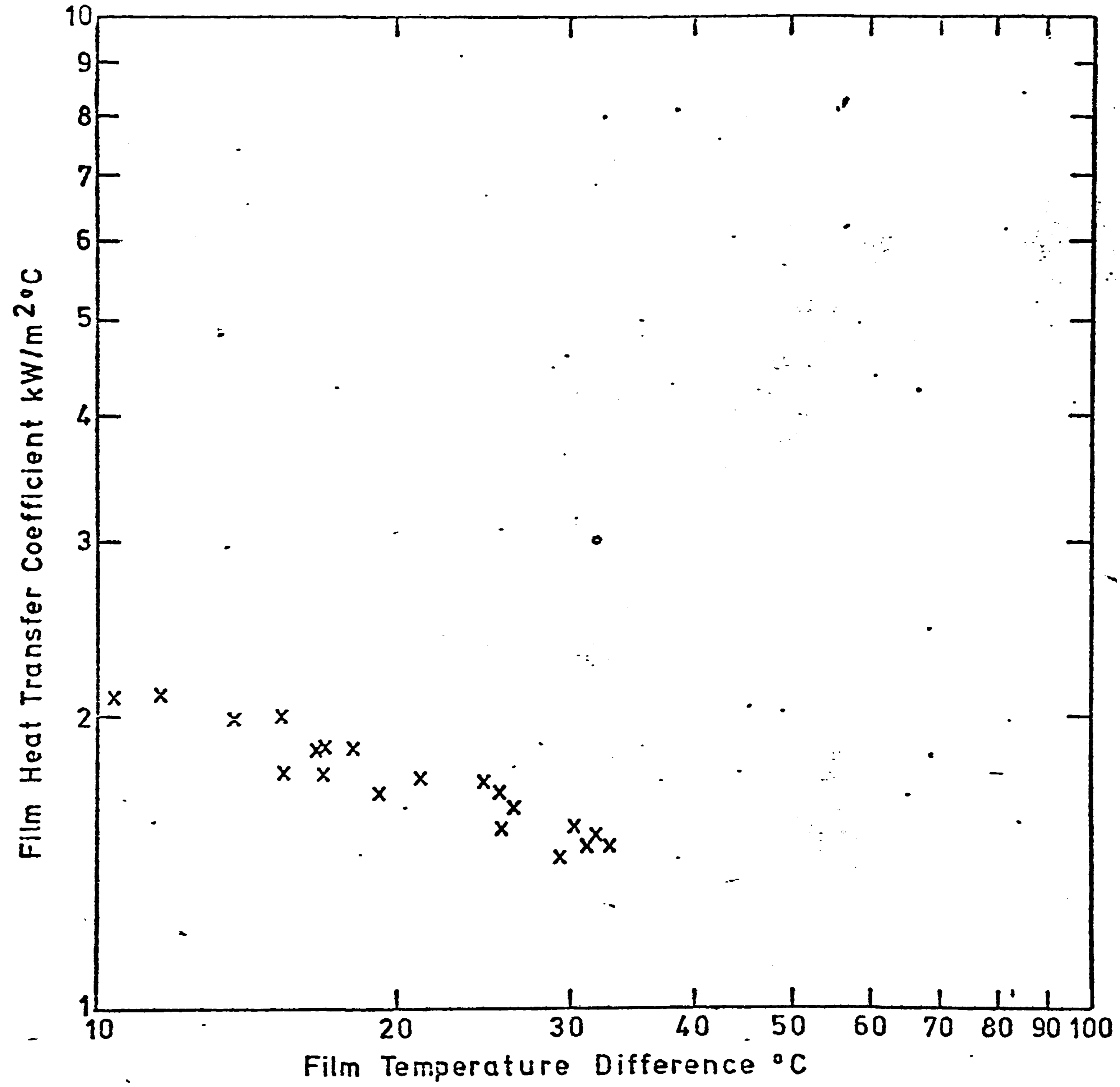

FIG. 4·2. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF PURE TOLUENE ON AN OXIDISED COPPER TUBE.

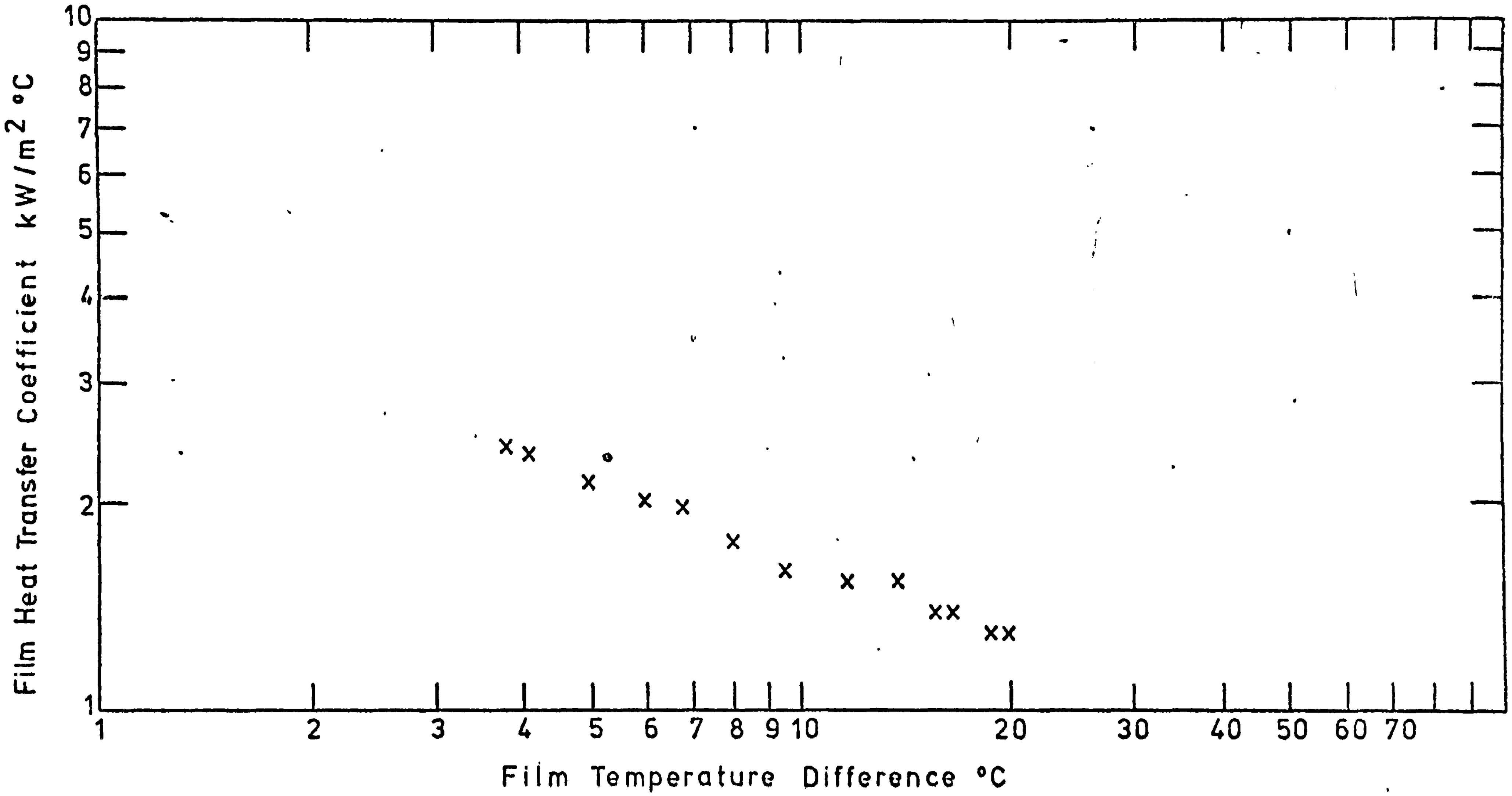

FIG. 4·3. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF PURE TRICHLOROETHYLENE ON AN OXIDISED COPPER TUBE.

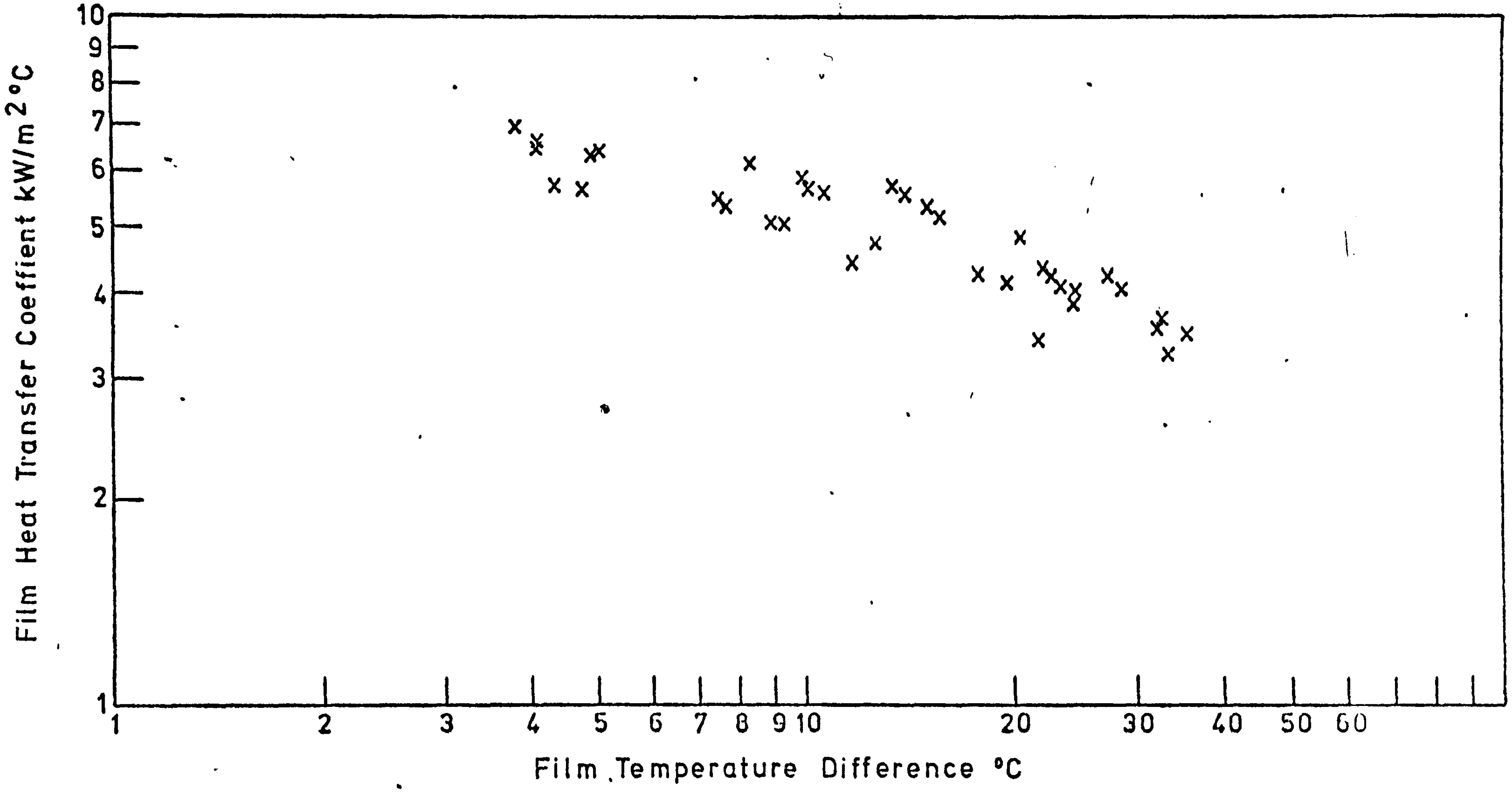

FIG. 4·4. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF STEAM-TOLUENE MIXTURES ON AN OXIDISED COPPER TUBE.

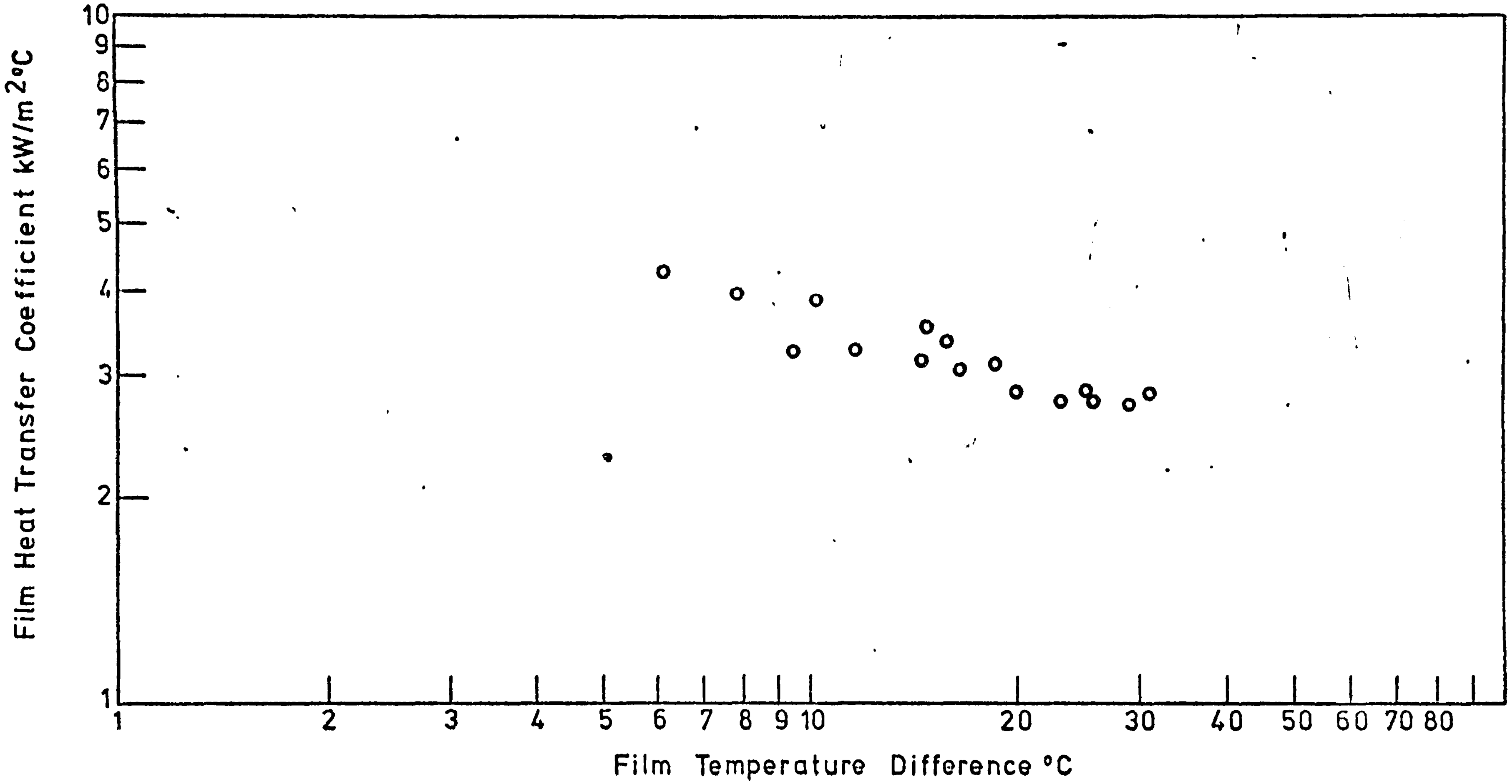

FIG. 4·5. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF STEAM - TOLUENE MIXTURES ON A GOLD PLATED COPPER TUBE.

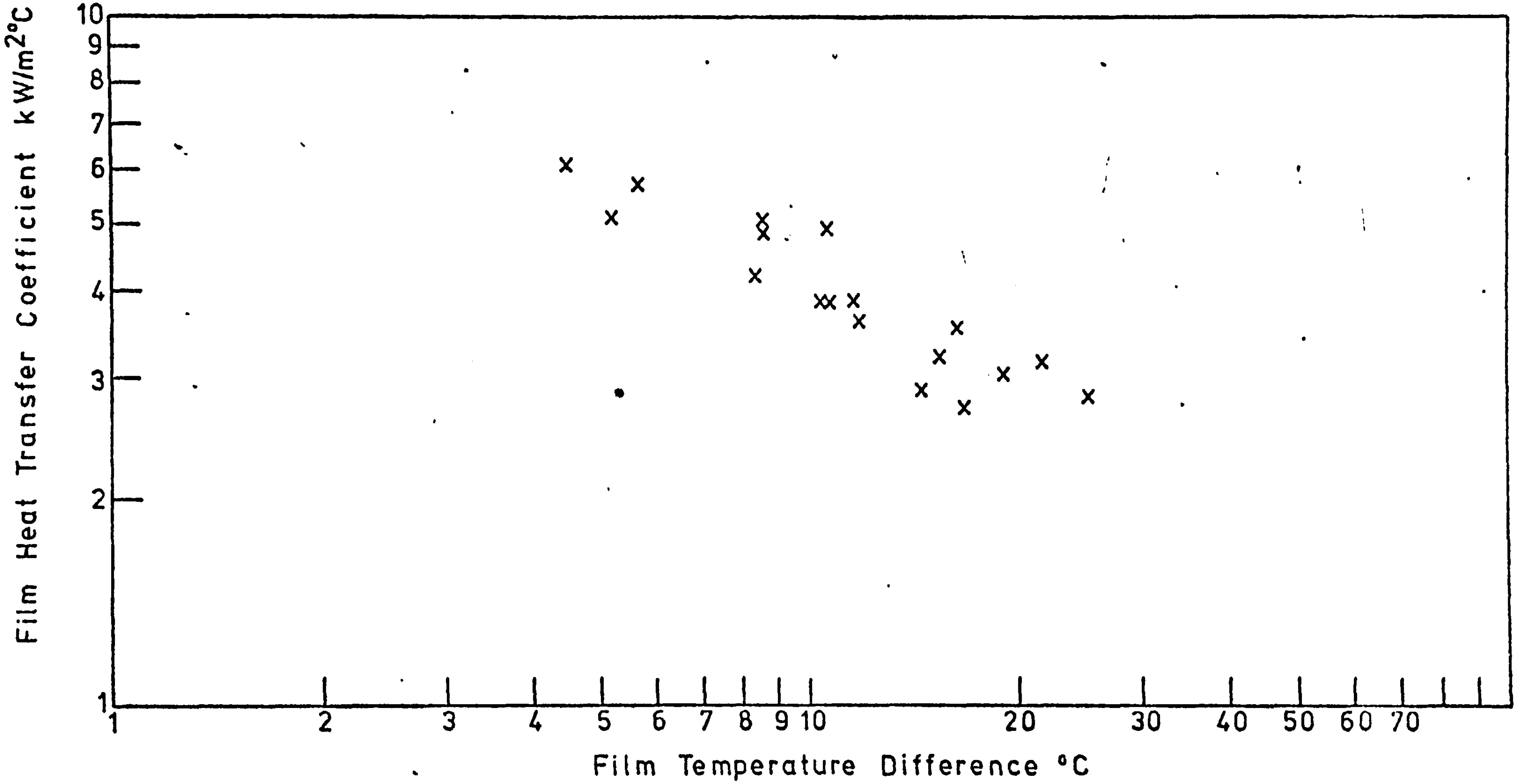

FIG. 4·6. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF STEAM - TRICHLOROETHYLENE MIXTURES ON AN OXIDISED COPPER TUBE.

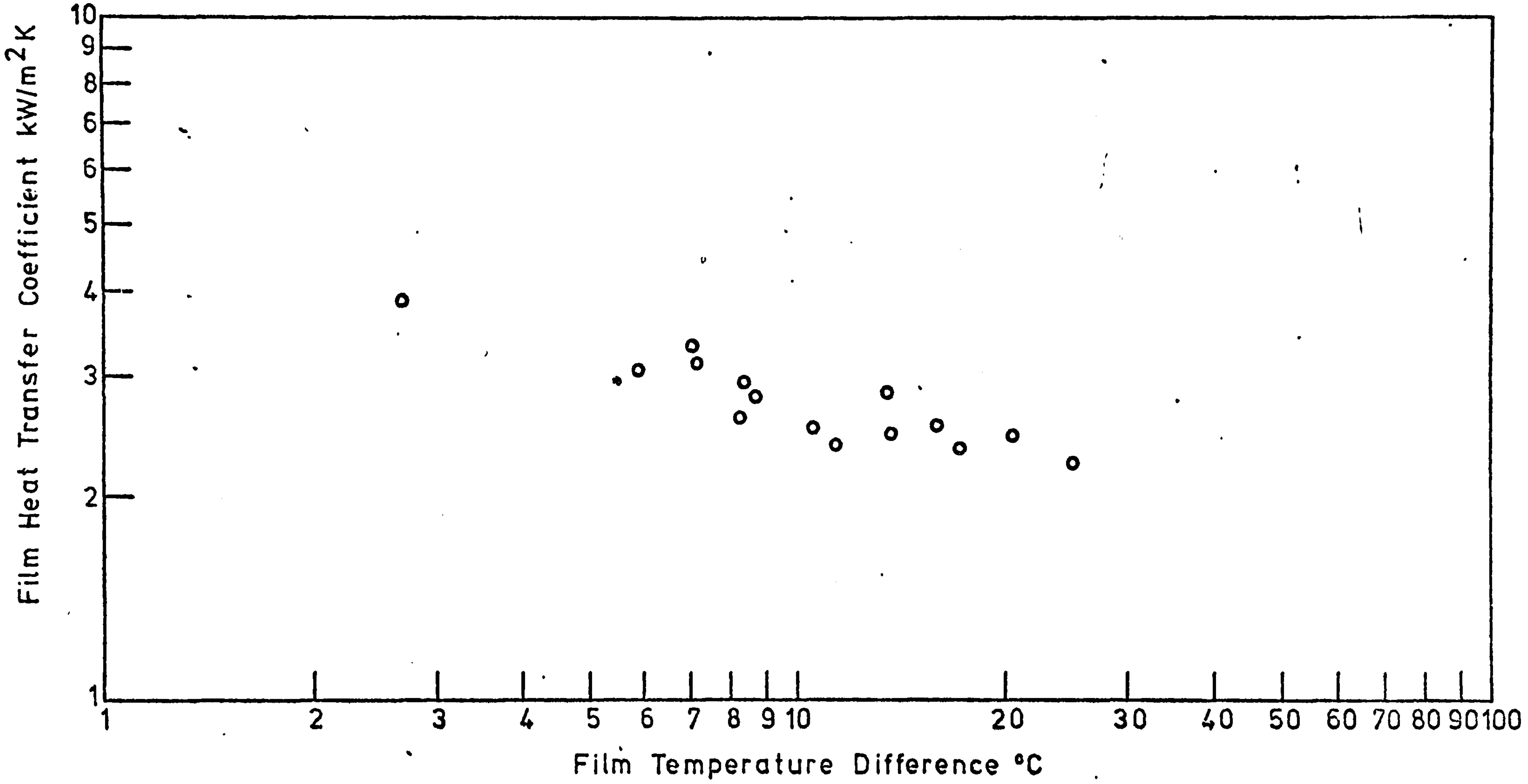

FIG.4·7. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF STEAM-TRICHLOROETHYLENE MIXTURES ON A GOLD PLATED COPPER TUBE.

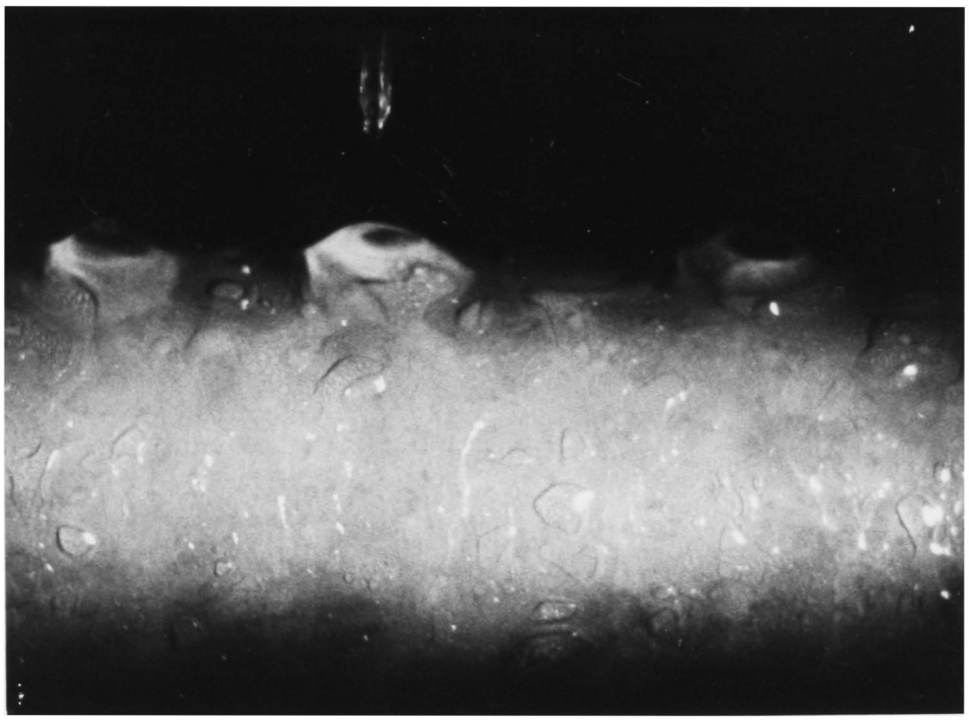

FIG. 4.8 FLOW PATTERN for the CONDENSATION of STEAM-TOLUENE MIXTURES ON AN OXIDISED COPPER TUBE

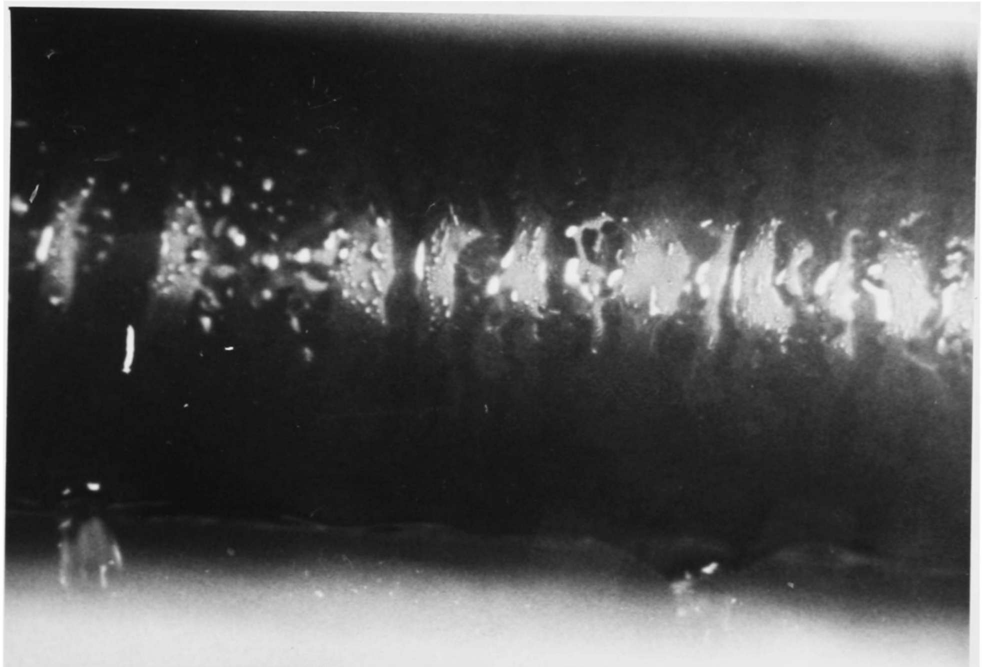

FIG. 4.9 FLOW PATTERN for the CONDENSATION OF STEAM-TRICHLOROETHYLENE MIXTURES ON AN OXIDISED COPPER TUBE

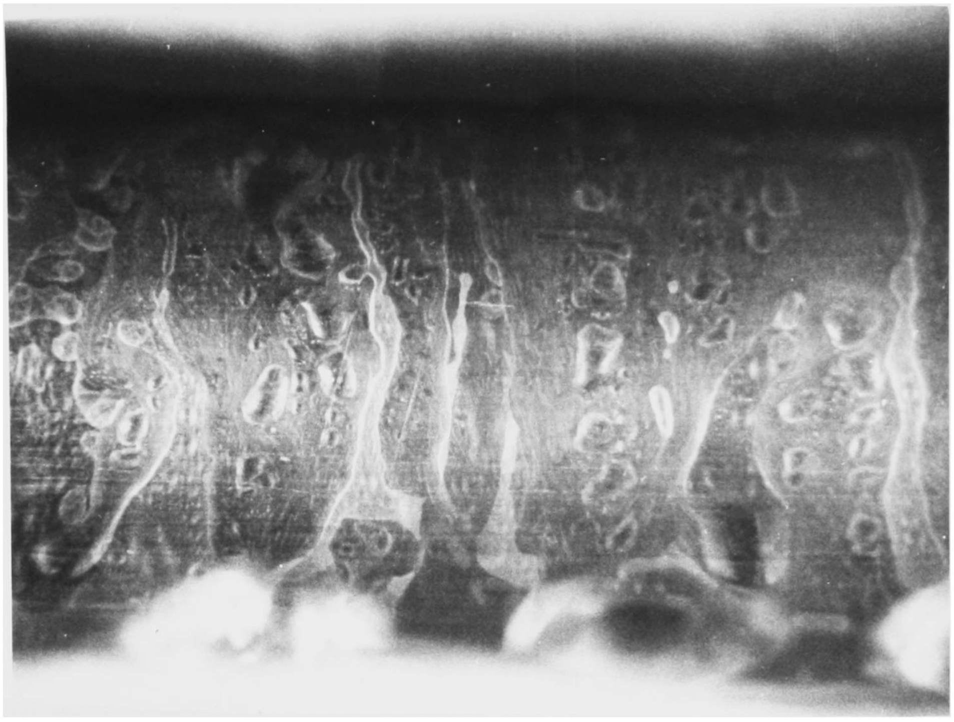

FIG. 4.10 FLOW PATTERN for the CONDENSATION OF STEAM-TOLUENE MIXTURES on a GOLD PLATED TUBE

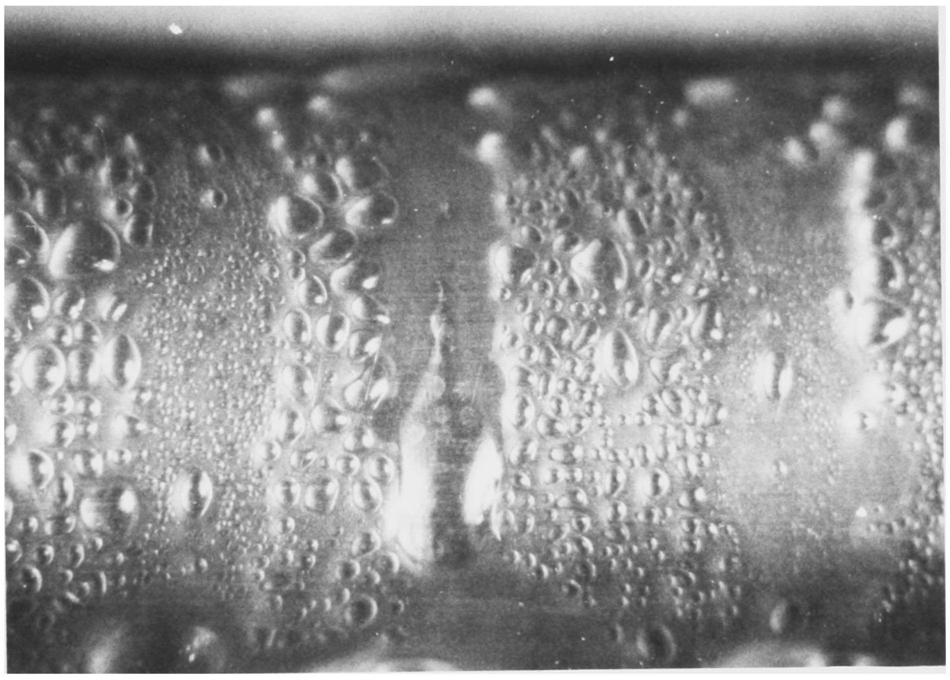

FIG. 4.11 FLOW PATTERN for the CONDENSATION OF STEAM-TRICHLOROETHYLENE MIXTURES ON A GOLD PLATED TUBE

CHAPTER 5

Theory

5.1 Introduction

This chapter considers the prediction of the heat transfer coefficients obtained when condensing vapours of immiscible liquids. In deriving a model the first consideration is the condensation mechanism. Two distinct mechanisms have been observed in this study (Chapter 4 section 4.3.2), a channelling mechanism and a standing drop mechanism.

Two models will be presented, the first assumes a channelling flow the second a standing drop mechanism. The agreement between the models and experimental data is discussed, along with the limitations, in chapter 6.

5.2 Channelling model

The following assumptions are made in order to simplify the model:

1. Both liquids form Nusselt type films and these obey the usual Nusselt assumptions (see Chapter 2 section 2.2)

2. The temperature drop through the two films is equal and is given by $\Delta T_f = T_e - T_W$.

The total amount of heat transferred through the tube wall is

$$Q = Q_1 + Q_2 \tag{5.1}$$

where: $Q_1 (= h_{N1} A_1 \Delta T_f)$ is the heat transferred through film 1, Q_2 $(= h_{N2} A_2 \Delta T_f)$ is the heat transferred through film 2,

h_{Ni} is the Nusselt coefficient for film i and

A_i is the area occupied by film i.

From the above definitions,

$$Q = (h_{N1} A_1 + h_{N2} A_2) \Delta T_f, \tag{5.2}$$

but since $Q = hA \Delta T_f$, where h is the overall heat transfer coefficient from the vapour liquid interface to the tube wall and A is the total area, then

$$h = A_1 h_{N1}/A + A_2 h_{N2}/A \tag{5.3}$$

Let the volumes of liquid 1 on the tube surface be V_1, then the area occupied by this liquid is

$$A_1 = V_1/\delta_1 \text{ thus giving } A_2 = (V - V_1)/\delta_2, \tag{5.4}$$

where δ_i is the film thickness of film i and V is the total volume of liquid on the tube surface.

From (5.4) it is apparent that

$$A_1/A = \frac{1/\delta_1}{1/\delta_1 + (\frac{1}{v_1} - 1)/\delta_2} \tag{5.5}$$

where $v_1 = V_1/V$ is the volume fraction of liquid 1 on the tube surface. Since the two liquid films are laminar then,

$$h_{N1} = k_1/\delta_1 \text{ and } h_{N2} = k_2/\delta_2. \tag{5.6}$$

Combining(5.3), (5.5) and (5.6) gives

$$h = h_{N1}/(1+Y) + h_{N2}Y/(1+Y), \tag{5.7}$$

$$\text{where } Y = \frac{k_1 h_{N2}}{k_2 h_{N1}} \left(\frac{1}{v_1} - 1 \right) \tag{5.8}$$

To use equation (5.7) the volume fraction v_1 must be determined. For the channelling flow mechanism, since there is no liquid hold up it is reasonable to assume that v_1 is the volume fraction of component 1 in the condensate. Hence using this fraction the value of the heat transfer coefficient can be calculated. The agreement between equation (5.7) and the experimental data is discussed in Chapter 6.

If it is assumed that the thickness of the two films are equal then equation (5.8) becomes

$$Y = \frac{1}{v_1} - 1, \tag{5.9}$$

and hence equation 5.7 reduces to

$$h = v_1 h_{N1} + (1-v_1) h_{N2} \tag{5.10}$$

Thus it is apparent that the equation presented by Bernhardt et al (1972), without derivation is a special case of equation (5.7). For the case of steam trichloroethylene and steam-toluene mixtures the predictions given by equations (5.7) and (5.10) do not differ by more than ± 5%.

5.3 Standing drop model

For this model the condensation mechanism is assumed as follows. The phase which best wets the wall forms a film adjacent to the wall. The other phase forms drops, these drops adhere to the tube surface until they reach a certain critical size when they roll from the surface.

It is further assumed that only drops protruding through the film take part in the heat transfer process, the temperature drop across the film and drops is ΔT_f and the film is a Nusselt type film.

The heat transferred through a drop is given by,

$$Q_d = h_d \ \Delta T_f \ \frac{\pi d^2}{4} \tag{5.11}$$

where h_d is the heat transfer coefficient for the drop.

Fatica and Katz (1949) have shown that the heat transfer coefficient through a drop of base diameter d and contact angle θ is

$$h_d = k\, f\,(\theta)/d, \qquad (5.12)$$

where k is the thermal conductivity of the liquid in the drop and $f(\theta)$ is a function of the contact angle.

To calculate the heat transferred through all drops of diameter d it is necessary to know the dropsize distribution. Since there is no information available concerning distribution functions for the case of immiscible liquid condensation , it will be assumed that the distribution function is of the same form as that used for dropwise condensation studies. A commonly used distribution (Tanaka (1975)) is given by

$$N = c\, d^n, \qquad (5.13)$$

where N is the probability density of time averaged dropsize distributions and c and n are constants. The number of drops in unit area of diameter d in the size range d + dd is hence Ndd and thus the total amount of heat transferred through drops of diameter d is given by

$$Q_d^1 = \frac{kf\ \ (\theta)}{d} \cdot \Delta T_f \cdot \frac{\pi\, d^2}{4} \cdot N\, dd\, A, \qquad (5.14)$$

where A is the total heat transfer area.

The total heat transferred through all drops in the size range d_{max} to d_{min} is given by integrating equation (5.14), on substituting equation (5.13) this becomes

$$Q_D = \int_{d_{min}}^{d_{max}} \frac{\pi k\ f\ (\Theta)\ \Delta T_f cA}{4}\ d^{n+1}\ dd. \qquad (5.15)$$

The total heat transferred through the wall is

$$Q = Q_D + Q_F \qquad (5.16)$$

where $Q = h\ \Delta\ T_f A$ and Q_F is the heat transferred through the film i.e. $h_{NF}\ \Delta T_f A_F$. Combining 5.15 and 5.16 gives

$$h\ \Delta T_f = \frac{\pi f(\Theta)\Delta T_f ck}{4} \int_{d_{min}}^{d_{max}} d^{n+1}\ dd + \frac{A_F}{A}\ h_{NF}\Delta T_f. \qquad (5.17)$$

Equation (5.17) can be written

$$h = \frac{\pi k\ f\ (\Theta)c}{4} \int_{d_{min}}^{d_{max}} d^{n+1}\ dd + (1-\alpha)\ h_{NF}, \qquad (5.18)$$

where α is the fractional area occupied by the drops. From equation (5.13) it is apparent that,

$$\alpha = \frac{\pi c}{4} \int_{d_{min}}^{d_{max}} d^{n+2}\ dd \qquad (5.19)$$

Integrating equation (5.18) and (5.19) and combining gives

$$h = k\ f\ (\theta)\ a\left(\frac{n+3}{n+2}\right) \left[\frac{d_{max}^{n+2} - d_{min}^{n+2}}{d_{max}^{n+3} - d_{min}^{n+3}} \right] + (1- a)\ h_{NF}. \qquad (5.20)$$

Thus to find h, the unknowns in equation (5.20) have to be determined, that is d_{max}, d_{min}, ∂ and n must be known.

The maximum drop diameter is the diameter of the critical drops, that is the diameter of those drops which are just large enough to roll from the surface. Fatica and Katz (1949) give the following equation for d_{max}.

$$d_{max} = \left[\frac{\sigma_{ij}(\cos \theta_R - \cos \theta_A)}{\rho\ g \sin\ a' f(\theta)_1} \right]^{\frac{1}{2}}, \qquad (5.21)$$

where σ_{ij} is the interfacial tension between the two liquids, θ_R and θ_A the advancing and receeding contact angles, ρ the density of the drops, a' the angle of inclination of the surface and $f(\theta)_1$ a function of the contact angle given by

$$f\ (\theta)_1 = (\pi\theta/90 - \sin 2\theta)\ (1/8 \sin 2\theta)$$

The minimum drop diameter is the diameter of that drop having a height equal to the film thickness, that is

$$d_{min} = 2\,\delta \sin \theta /\ (1 - \cos \theta) \qquad (5.22)$$

At the present time there is little or no information available on the contact angle values and the drop size distributions. However, to try and test the validity of the model a contact angle of 70^{o} (the value for water on gold) will be used and the constant n determined by fitting the steam-toluene and steam-trichloroethylene data obtained (this study) on the gold plated tube. Since d_{max} in equation (5.21) cannot be determined, the observed experimental value of approximately 3.0 mm will be used. The final equation derived in the above manner (see Appendix C for details) is,

$$h = 13.362\ \alpha \left[\frac{0.003 - d_{min}}{0.003^2 - d_{min}^2} \right] + (1-\alpha)\ h_{NF} \qquad (5.23)$$

The units used in the above equation must be in S.I. That is the constant 13.362 has the units W/m°C.

Chapter 6

Discussion and Conclusions

6.1 Introduction

In this chapter the results presented in chapter 4 are discussed in detail. The chapter is divided into five main sections. In the first four the topics discussed are the pure component data, immiscible liquid data, comparisons between theory and data and nucleation phenomenon respectively. The fifth section presents the conclusions and recommendations.

6.2 Pure component data

It was decided early in the present study that pure components would be condensed; this would not only add to existing data but also serve to test the reliability of the experimental apparatus and operating procedures.

As was stated previously (chapter 2 section 2.2) the first attempt to predict condensing film heat transfer coefficients was made by Nusselt (1916) (equation (2.1)). Since then several authors have improved the Nusselt analysis by relaxing the various assumptions made in the original analysis. One of the assumptions which so far appears to be relatively rigid (particularly for horizontal tubes) is the constant wall temperature assumption. Salov and Danilov (1975) have shown that the wall temperature variation around horizontal tubes for both pure vapours and vapour mixtures of immiscible liquids is given by the empirical equation,

$$T_W = \bar{T}_W + C_1 \cos \phi \qquad \text{(Eqtn. 2.29)}$$

Salov and Danilov (1975) used equation (2.29) to show that when condensing immiscible liquid mixtures the variation in wall temperature made no significant difference to the mean heat transfer coefficients obtained, they used the laminar two film model to show this. However, they did not study the case of pure components, this case is analysed in detail in appendix E.

From the curves in appendix E it is apparent that if the parameter

ω is sufficiently large the variation in heat transfer coefficient from the isothermal case is quite large, a 9% increase in local coefficient occuring at the top of tube with $\omega \simeq 0.3$. However, the mean coefficient is only 5% larger with values of $\omega \simeq 0.3$. The values of ω of practical interest are usually less than 0.3 and hence the increase in mean coefficient will not be greater than 5.0%.

It is apparent from the above discussion and chapter 2 section 2.2 that Nusselts equation should be adequate for predicting the pure component data.

The pure component data obtained in this study are compared with the Labuntsov (1957) form of the Nusselt equation i.e.,

$$h_L = \phi_L \, h_N \qquad \text{(Eqtn. 2.7)}$$

the only correction taken into account in deriving equation (2.7) being the variation in physical properties across the condensate film.

The comparison between equation (2.7) and the steam data is shown in figure 6.1; as can be seen the agreement is excellent. Figure 6.2 shows the comparison for the toluene data, here the predictions are lower than the data, the maximum deviation occuring at low film temperature differences. The trichloroethylene data is compared against equation (2.7) in figure 6.3, the agreement is again good, however, the predictions at film temperature differences above about 7.0°C are high by up to 20%.

It can be seen from figures 6.1, 6.2 and 6.3 that the present data agree with Nusselts equation to within ± 20.0% for all three components. Thus Nusselts analysis is adequate for predicting condensation heat transfer coefficients for the above fluids, as would be expected (see chapter 2 section 2.2).

The above comparisons thus show that the method of operating the

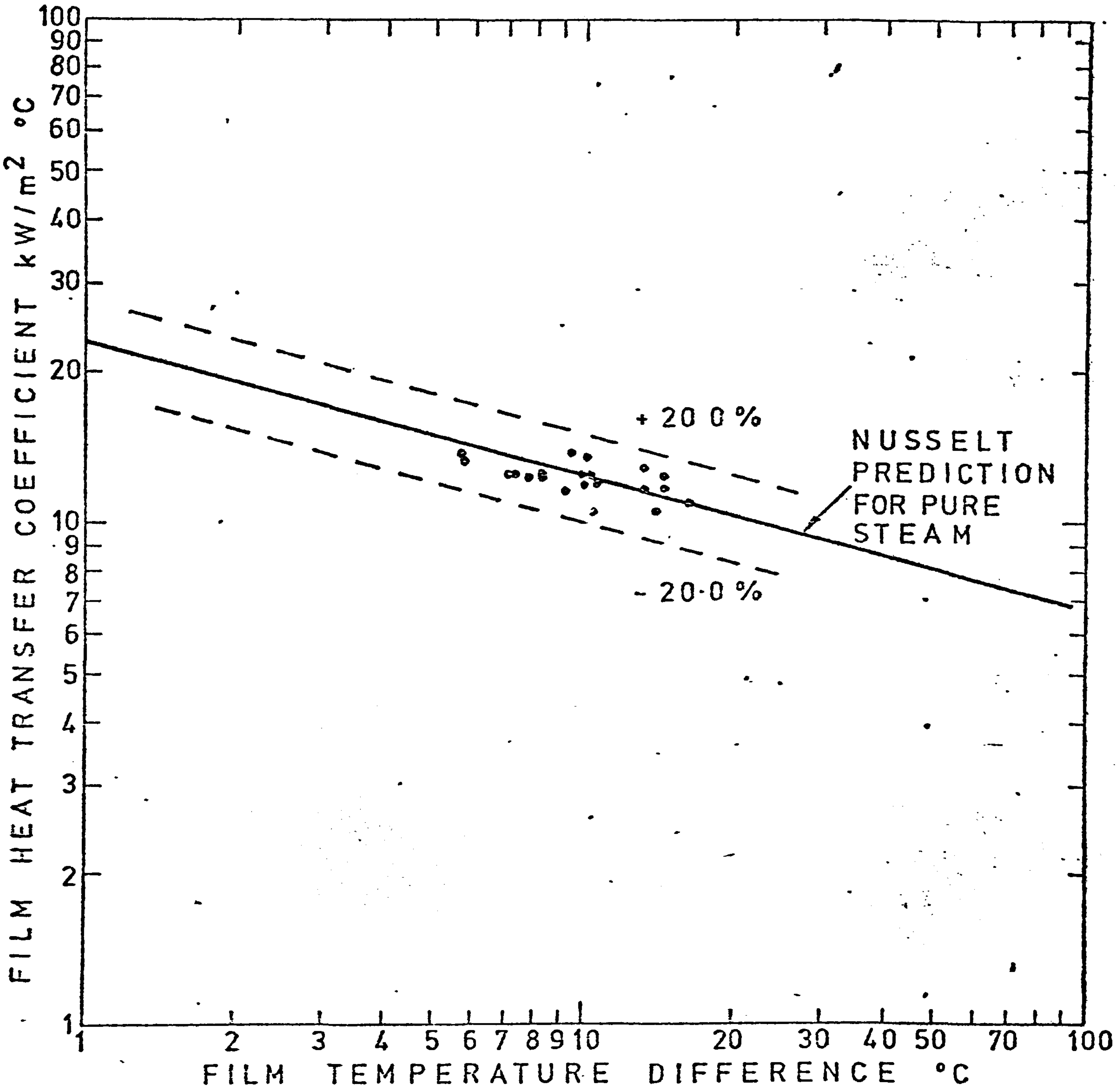

FIG. 6·1. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF PURE STEAM ON AN OXIDISED COPPER TUBE

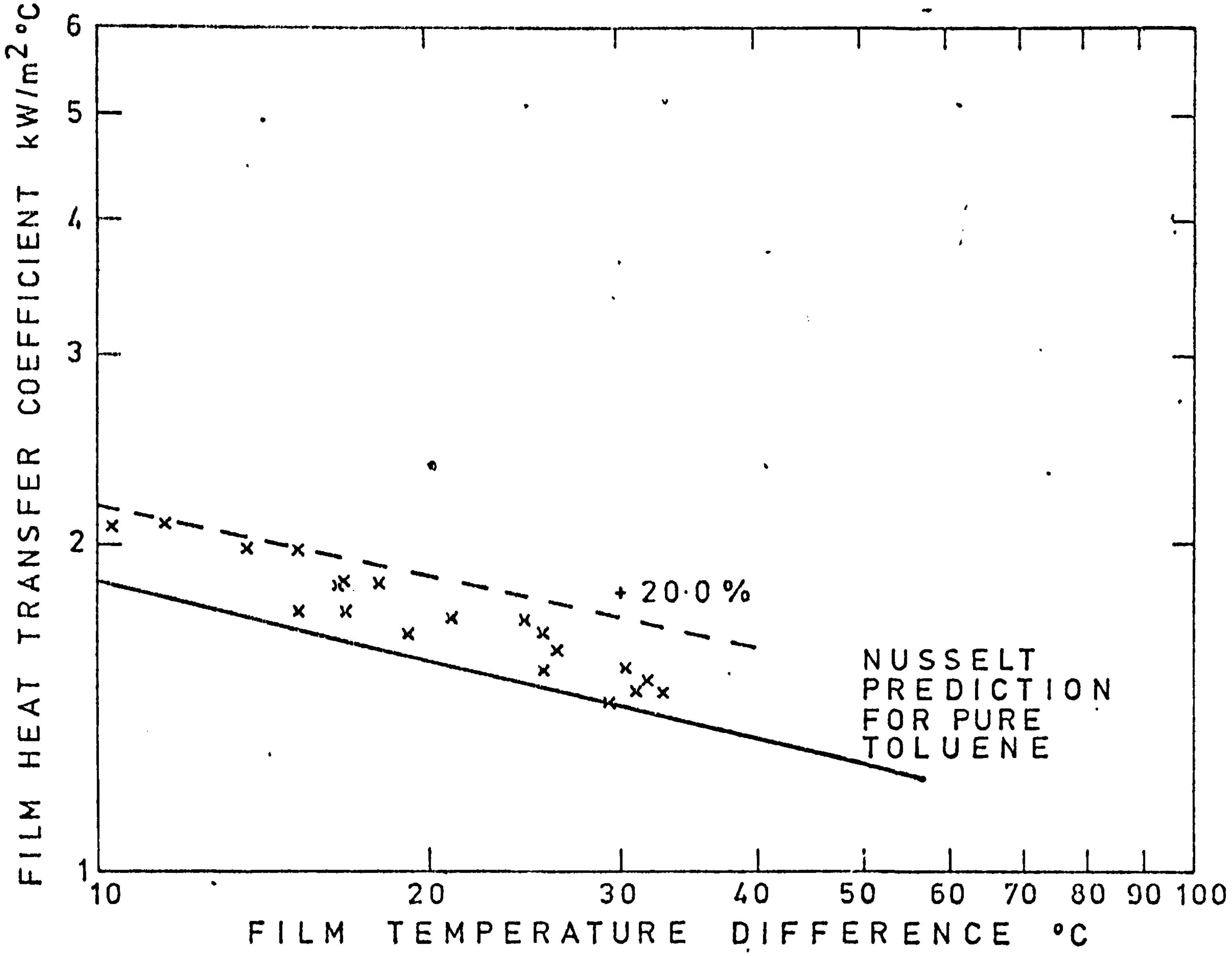

FIG. 6·2. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF PURE TOLUENE ON AN OXIDISED COPPER TUBE

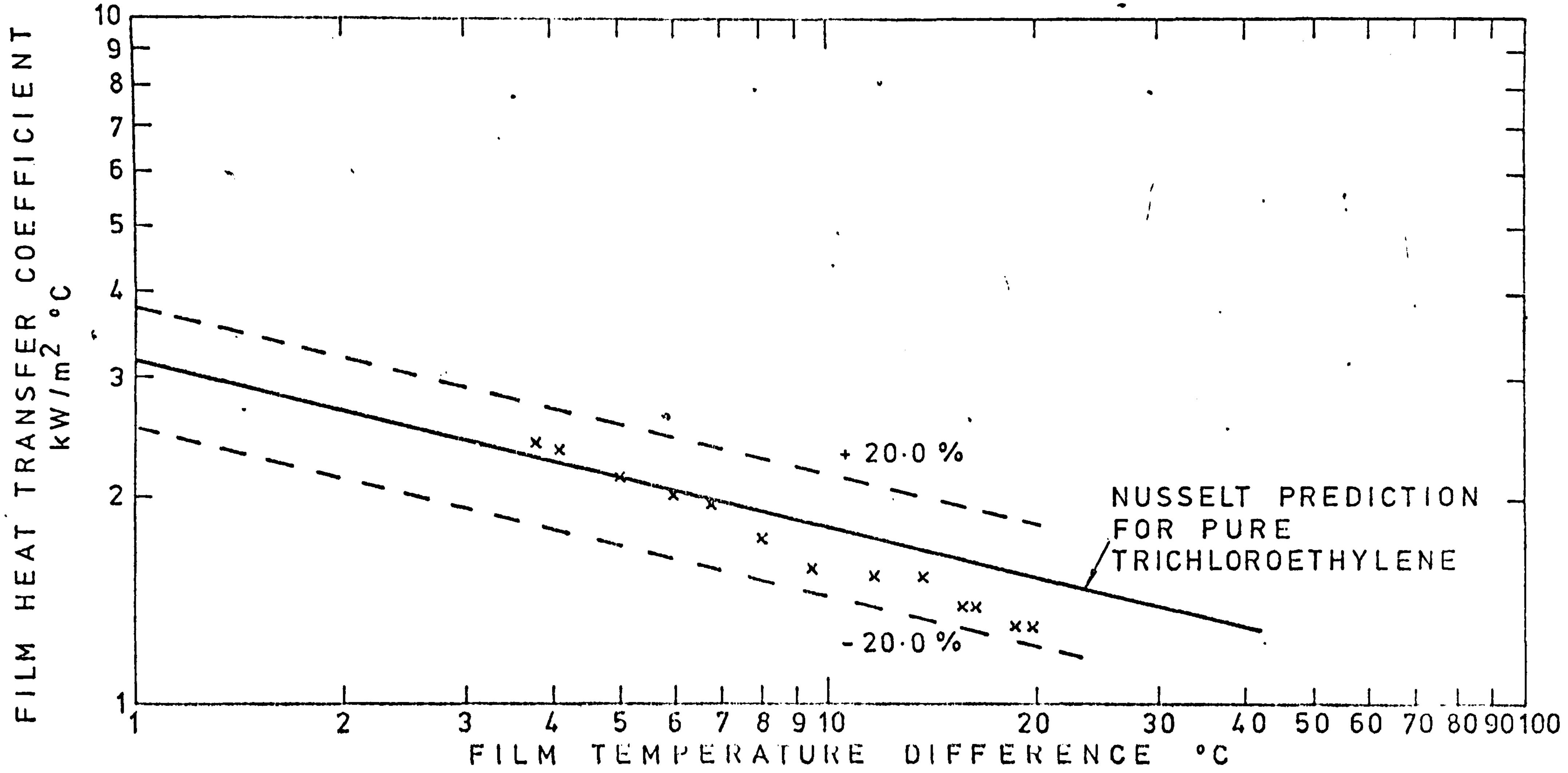

FIG. 6·3. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF PURE TRICHLOROETHYLENE ON AN OXIDISED COPPER TUBE experimental apparatus, particularly the method of purging incondensable gases, was satisfactory. That is no major deviations from the Nusselt predictions exist, as would have been expected if incondensable gases had been present.

6.3 Immiscible liquid data

6.3.1 Effect of film temperature difference

From the literature survey (chapter 2) the effect of film temperature difference on film heat transfer coefficients is not certain. The data obtained in this study however, show an unmistakeable trend, that is as the temperature difference increases the heat transfer coefficient decreases. Figure 6.4 shows that this trend occurs for all combinations of mixtures and surfaces used.

6.3.2 Effect of the tube surface

It is apparent from the results (chapter 4) that the flow regimes on the oxidised copper and gold plated tubes are different. On the oxidised copper surface the water-toluene and water-trichloroethylene mixtures flow from the tube in essentially the same manner, that is in rivulets flowing side by side (channelling flow). This mechanism is very similar to the channelling mechanism postulated by Hazelton and Baker (1944).

The flow mechanism for the water-trichloroethylene mixture on the gold surface was of the standing drop type previously described by Bernhardt et al (1970) for the condensation of various organic-steam mixtures on a vertical gold plated surface. Here the water formed irregular shaped standing drops in the flowing organic film. These drops rolling from the surface after reaching a certain critical size. It was also observed (from cine films) that there were many tiny water droplets flowing with the organic film. Whether they are in or on the surface of the film is not known.

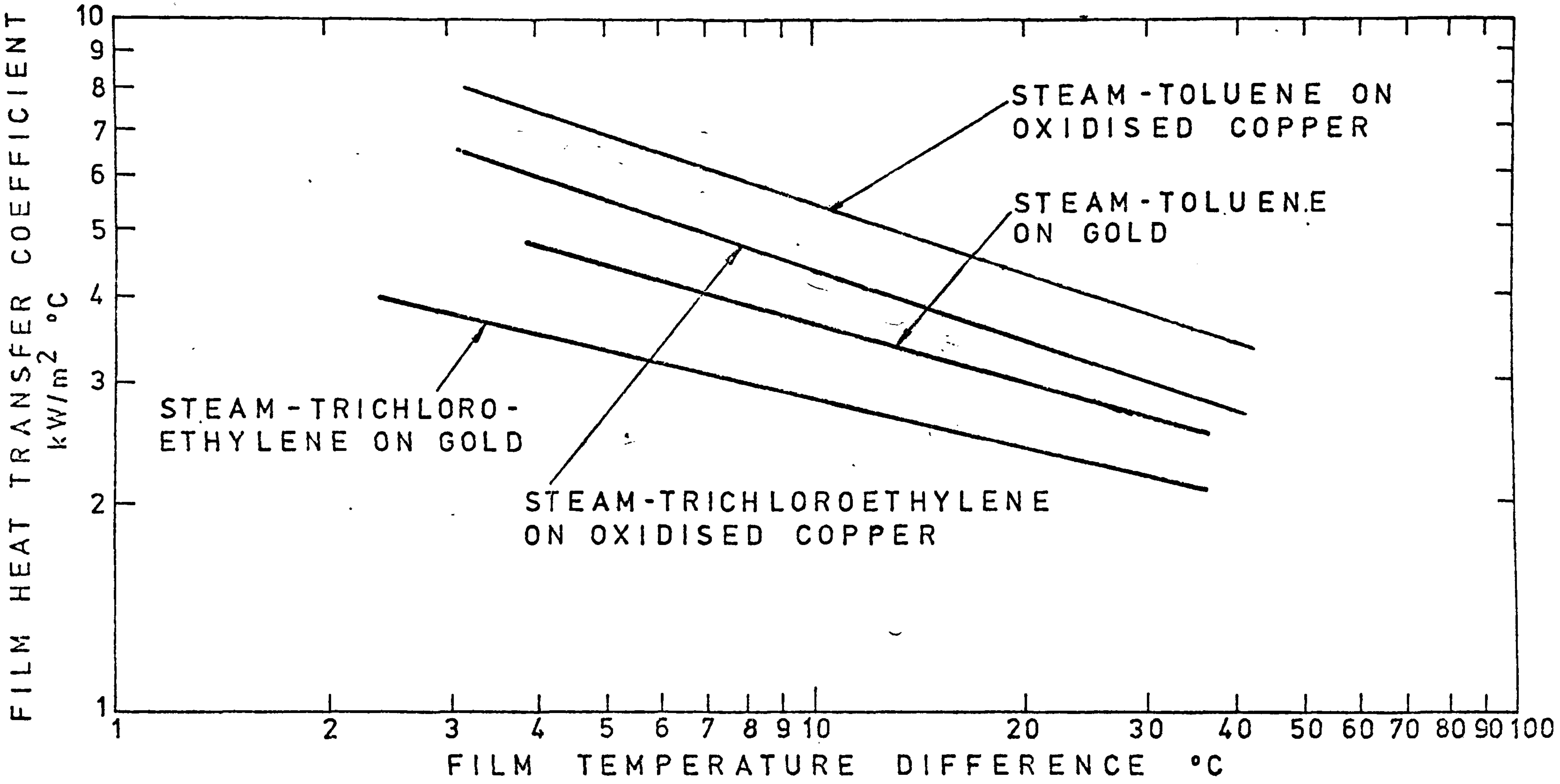

FIG. 6·4. COMPARISON OF THE TRENDS IN THE DATA FOR THE IMMISCIBLE SYSTEMS USED IN THIS STUDY

The flow mechanism for the water-toluene mixtures on the gold surface was similar to that described for the water-trichloroethylene mixtures. In this case however, there were far fewer standing water drops and many more tiny water drops flowing on or in the organic film. Also when the large drops drained from the surface they formed rivulets which persisted for considerable periods of time, whereas for the water-trichloroethylene case, when the drops drained they left only a trail of smaller drops in their wake.

The change in mechanism when condensing on oxidised copper or gold plated surfaces could be caused by a change in wetting properties. For instance pure steam condensing on a clean oxidised copper surface will spread over the surface(i.e. the contact angle is zero)), whereas pure water on a gold surface forms drops with a finite contact angle (approximately 70°).

In the condensation case, on the oxidised copper surface, both liquids spread over the surface and in competing for the surface form adjacent films. Further vapour condenses on these films, the organic vapour on the organic film and the steam on the water film.

For the gold surface the vapour initially condenses to form an organic film (organics spread on gold) and standing water drops. It would then seem that further water vapour can only condense on those water drops which protrude through the organic film. Eventually these drops would become large enough to roll from the surface under the action of gravity. When the drops roll they sweep a track through the organic film and other drops in their path, a track of small drops being left in the drop wake for the steam-trichloroethylene case and a small rivulet of water in the steam-toluene case.

In the steam-toluene case the water rivulet eventually drains and the organic film flows in over the surface, small water drops being left as "islands" in the organic film.

For the steam-trichloroethylene case it may be speculated that the following processes occur simultaneously:

1. The drops in the trail grow by condensation
2. The existing organic film spreads over the bare surface
3. Steam and organic vapour condense on the bare area, the organic forming a film and the water drops.

Thus again we have an organic film plus standing water drops. The above processes continue indefinitely.

Although the above interpretations explain the major features of the observed mechanisms, there are still several unresolved questions regarding the origin of the small organic droplets and the origin of the water droplets.

Bernhardt et al (1970) speculated that the small organic drops (present in or on the water films and drops) were nucleating on microscopic dust particles on the surface of the water, also the moving water drops could have the same origin.

A further possibility is that the organic is condensing directly onto the water drops. If this were the case it is possible that a nucleation barrier exists to the condensation of the organic on the water. Also any water condensing directly onto the organic film might also have to overcome a nucleation barrier. A more detailed discussion on nucleation phenomenon is given in section 6.5.

Another mechanism by which droplets can be produced in the respective films and drops is as follows. Although it is often stated that the organic-water mixtures used in immiscible condensation studies are totally immiscible this is rarely the case. Figures 6.5 and 6.6 show the solubility curves for water-toluene and water-trichloroethylene mixtures.

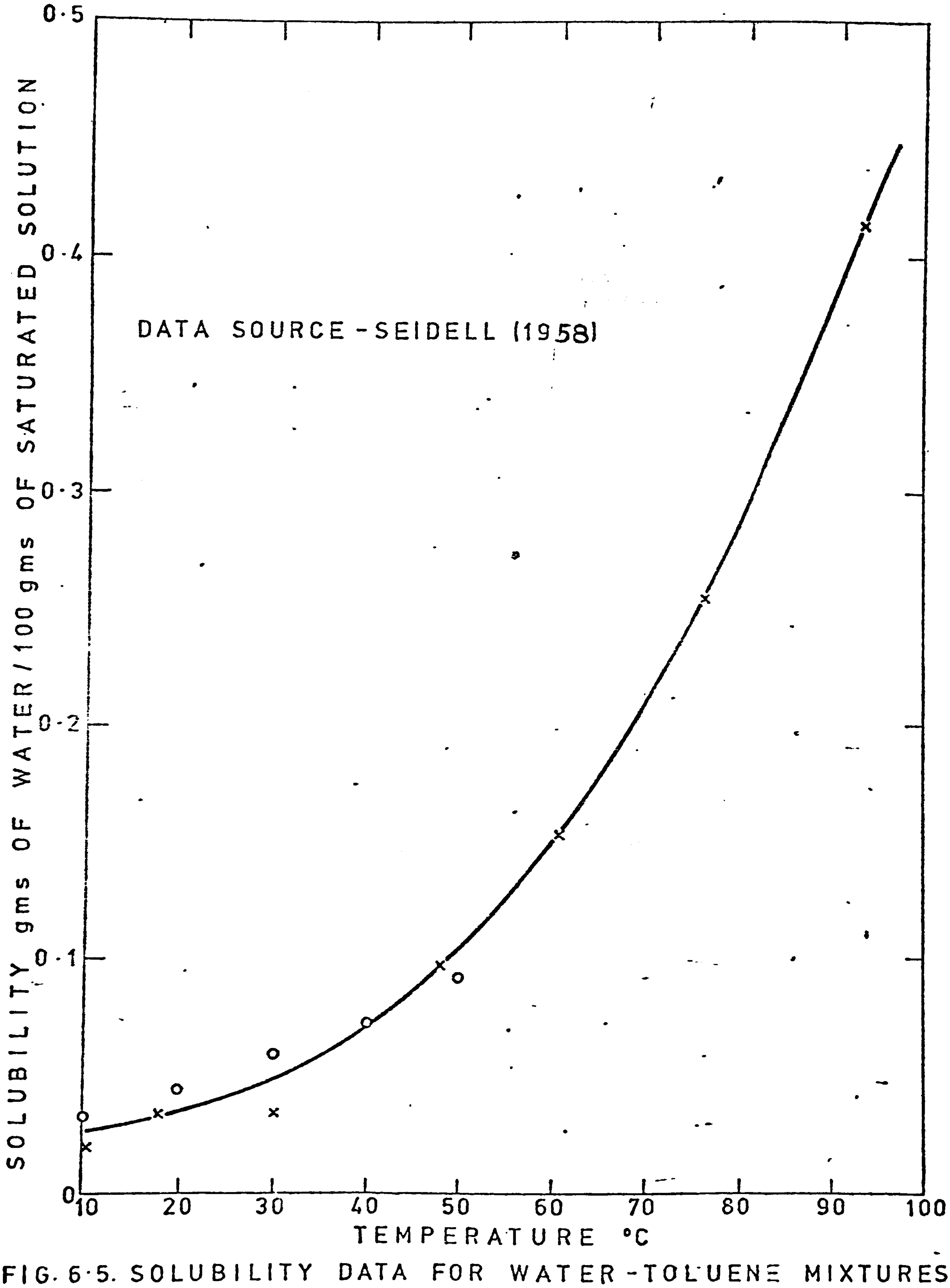

FIG. 6·5. SOLUBILITY DATA FOR WATER-TOLUENE MIXTURES

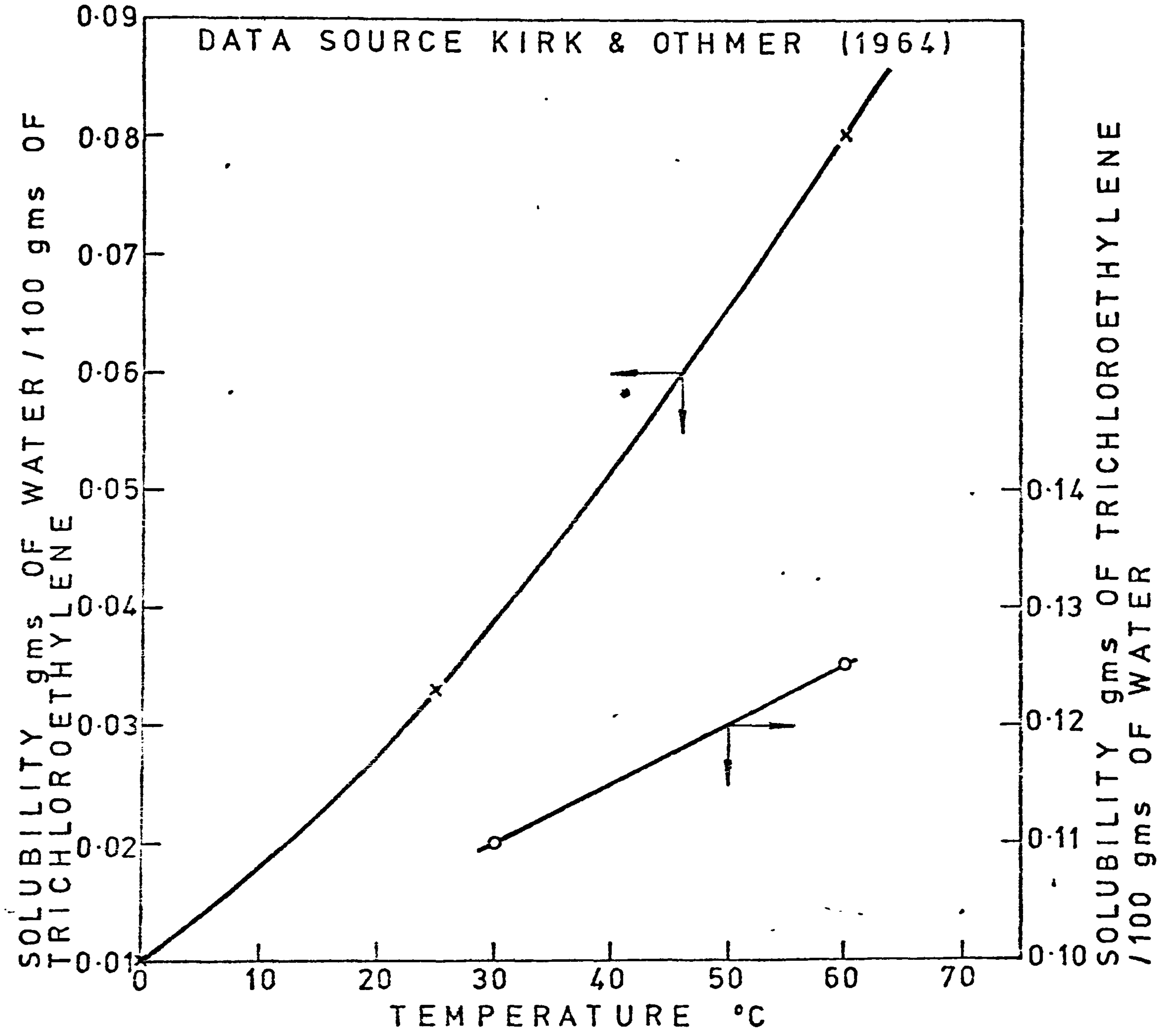

FIG. 6·6. SOLUBILITY DATA FOR WATER - TRICHLOROETHYLENE MIXTURES

Using the water-toluene curve as an example, it can be seen that if the temperature difference across the film were say $10^{\circ}C$, then for an eutectic mixture ($T_i \simeq 84.3^{\circ}C$) the solubility of water in toluene falls from 0.32 gm/(100 gms of saturated solution) at the interface, to 0.24 gm/(100 gms of saturated solution) at the wall. Thus 0.077 gms/(100 gms of saturated solution) of water will have precipitated, in the form of droplets, in the organic film.

Those droplets coming out of solution at the wall adhere to the wall, further droplets coalescing with these existing droplets until eventually the drops are large enough to protrude through the condensate film. They can then grow by direct condensation of steam.

The droplets that come out of solution within the film can either fall to the wall or rise to the film surface. If the drops are denser than the film they fall to the tube wall, and if they are less dense they rise to the surface of the film. The above description is for the top half of the tube, for the bottom half of the tube the droplet motion is reversed.

Although the above description was given for water drops in the organic film, the same processes apply for organic droplets coming out of solution in the water film (or drops).

Another puzzling aspect of the mechanism is that the organic phase forms drops on the water phase. The spreading concept of Harkins and Feldman (see section 2.4.1.1 chapter 2) offers an explanation in that essentially, if the appropriate spreading coefficient is negative then the organic will not spread on the water drops (or films).

The difficulty in using this concept is in deciding what surface tension values to use in equation (2.11). If the pure component values are used then the spreading coefficient will be positive and the organic should spread. However, since the two liquid phases are mutually saturated, the correct surface tension values are those of the mutually saturated phases and in this case it has been stated by Adamson (1967) that the spreading coefficient is usually close to zero or negative. Thus it is unlikely that the organic will spread on the water film (or drops).

It can be seen from the above discussion that the detailed mechanism of the condensation process is extremely complex.

6.3.3 Discussion of the heat transfer data

The data for the steam-toluene and steam-trichloroethylene mixtures condensing on the gold plated and oxidised copper tubes are shown in figures 6.7 and 6.8 respectively. It can be seen that the heat transfer coefficient obtained on the oxidised copper surface is greater than that obtained on the gold surface. However, it does seem that at high temperature differences the data are in closer agreement.

Since two fundamentally different mechanisms were observed on the two surfaces used it is not surprising that the heat transfer coefficients differ. From the data, it is obvious that the channelling mechanism gives higher heat transfer coefficients than the standing drop type mechanism. This confirms for horizontal tubes what Hazelton and Baker (1944) found for vertical tubes.

Figure 6.9 shows the comparison between the steam-toluene data for the present study with existing data. The data compared lie in the composition range 78-85% by weight toluene in the condensate, this is in comparison to the eutectic composition of approximately 80%.

From figure 6.9 it is apparent that the authors data on the oxidised copper surface is above most of the earlier data, whereas the authors data on the gold plated surface is in reasonable agreement with other data.

The condensation mechanism observed in the present study was of a channelling type and a standing drop type for the oxidised copper and gold

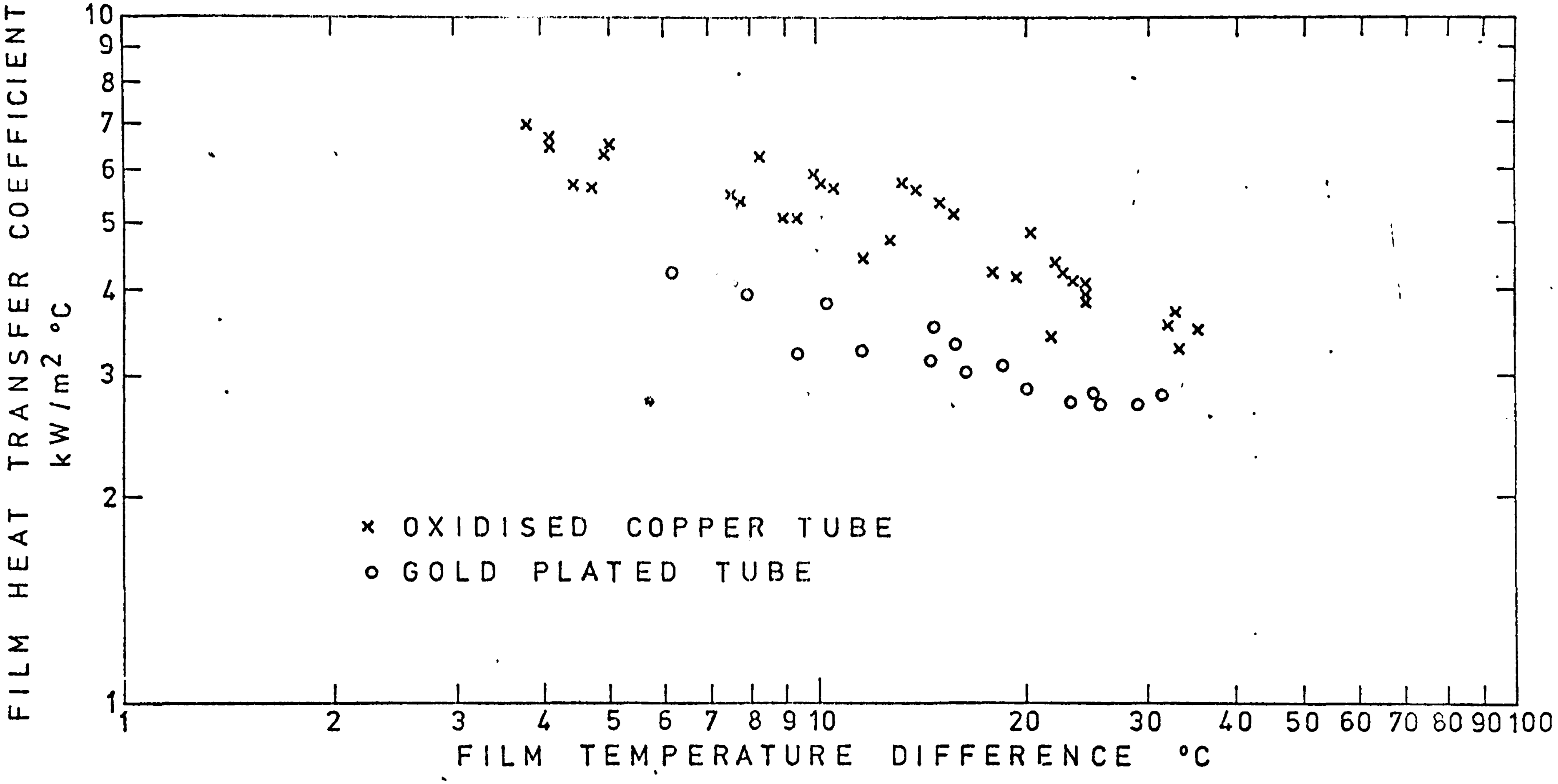

FIG. 6·7. COMPARISON OF THE STEAM-TOLUENE DATA FROM THIS STUDY

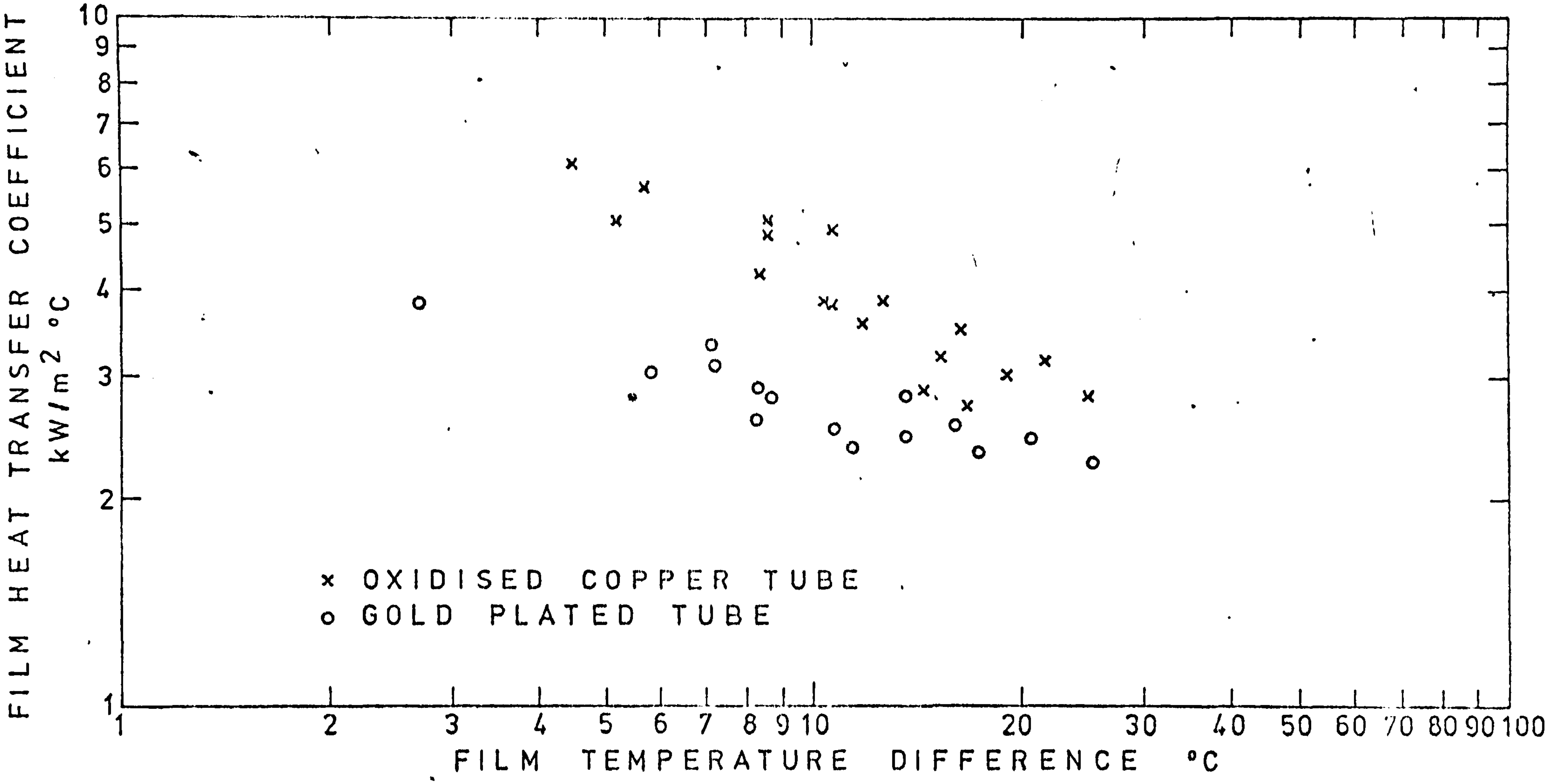

FIG. 6·8. COMPARISON OF THE STEAM-TRICHLOROETHYLENE DATA FROM THIS STUDY

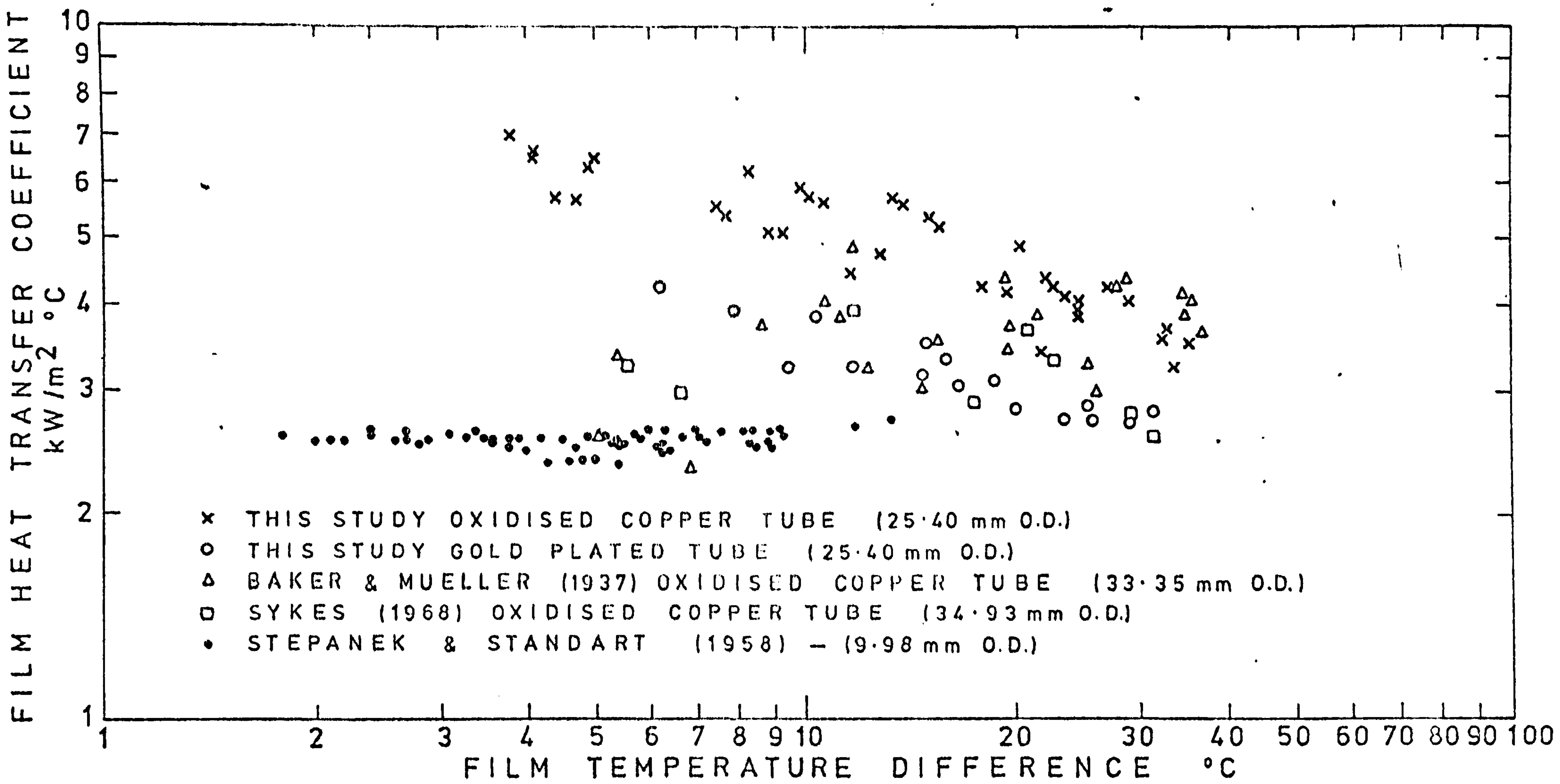

FIG. 6·9. COMPARISON OF EXISTING STEAM-TOLUENE DATA IN THE COMPOSITION RANGE 78-85% BY WEIGHT TOLUENE IN THE CONDENSATE plated tubes respectively. The mechanism described by Baker and Mueller (1937) appears to be of the standing drop type, whereas that by Sykes (1968) a mixture of the channelling and standing drop types. Thus it would be expected tnat Baker and Mueller and Syke's data should either agree with the gold plated data or lie between the gold plated and oxidised copper tube data of this study. And indeed for the majority of the data this is the case (see Figure 6.9).

It is interesting to note, however, that the surfaces used by the other two sets of workers were reported as being oxidised copper. Thus it seems that the mechanism obtained on apparently similar tubes can be different. In the vertical plate study by the author (described in appendix F) it was observed that a mixed channelling-standing drop mechanism was obtained. However, in this particular study on the vertical plate no precautions were taken to ensure that filmwise condensation of steam was obtained on the copper surface before use.

From the above discussion it would seem that the properties of an oxidised copper surface vary with time (this has previously been suggested by Ponter and Diah (1974)). Initially a standing drop type mechanism being obtained, this apparently transforming to a channelling type flow after some unknown period of time. A way of confirming the above suggestions would be to continuously condense a given vapour mixture on an initially polished copper tube, any variation of mechanism, with time, could then be observed.

An interesting feature of figure 6.9 is the position of the Stepanek and Standart (1958) data. Because this data is much lower than other data sets, several authors (Bernhardt et al (1970), Ponter and Diah (1974)) have suggested that the discrepancy is caused by the presence of incondensable gases. However, Standart (1973) has stated that a vent line had been included in the apparatus and that incondensable gases had been purged from the system. A possible reason for this data set being so far below the others is due to the effect of tube diameter. As was started in chapter 2 several workers have observed that for small diameter tubes the heat transfer coefficient increases as the tube diameter increases, and since the tube diameter of Stepanek and Standart (1958) was only 9.93 mm in comparison to the more common diameters of between 25.40 and 34.93 mm it is possible that this effect might explain the discrepancy.

Figure 6.10 shows the comparison between the authors steam-trichloroethylene data and Baker and Muellers (1937) data. As can be seen the agreement is not good the Baker and Muller (1937) data being considerably higher at high temperature differences than this study. The reasons for the discrepancy is not known, but presumably surface properties are of importance.

6.4 Comparison between theory and data

The comparison between the authors channelling data and the channelling model (equation (5.7)) is shown in figures 6.11 and 6.12. The theoretical line was calculated for the eutectic mixture in each case. From the figures it can be seen that the model underpredicts quite considerably.

In order to predict the data satisfactorily an enhancement factor of 1.5 is introduced. Thus equation (5.7) becomes

$$h = 1.5\ (h_{N1}/(1+Y) + h_{N2}\,Y/(1+Y)) \qquad (6.1)$$

The agreement between equation (6.1) and the data (see figures 6.11 and 6.12) is to within ± 20.0%.

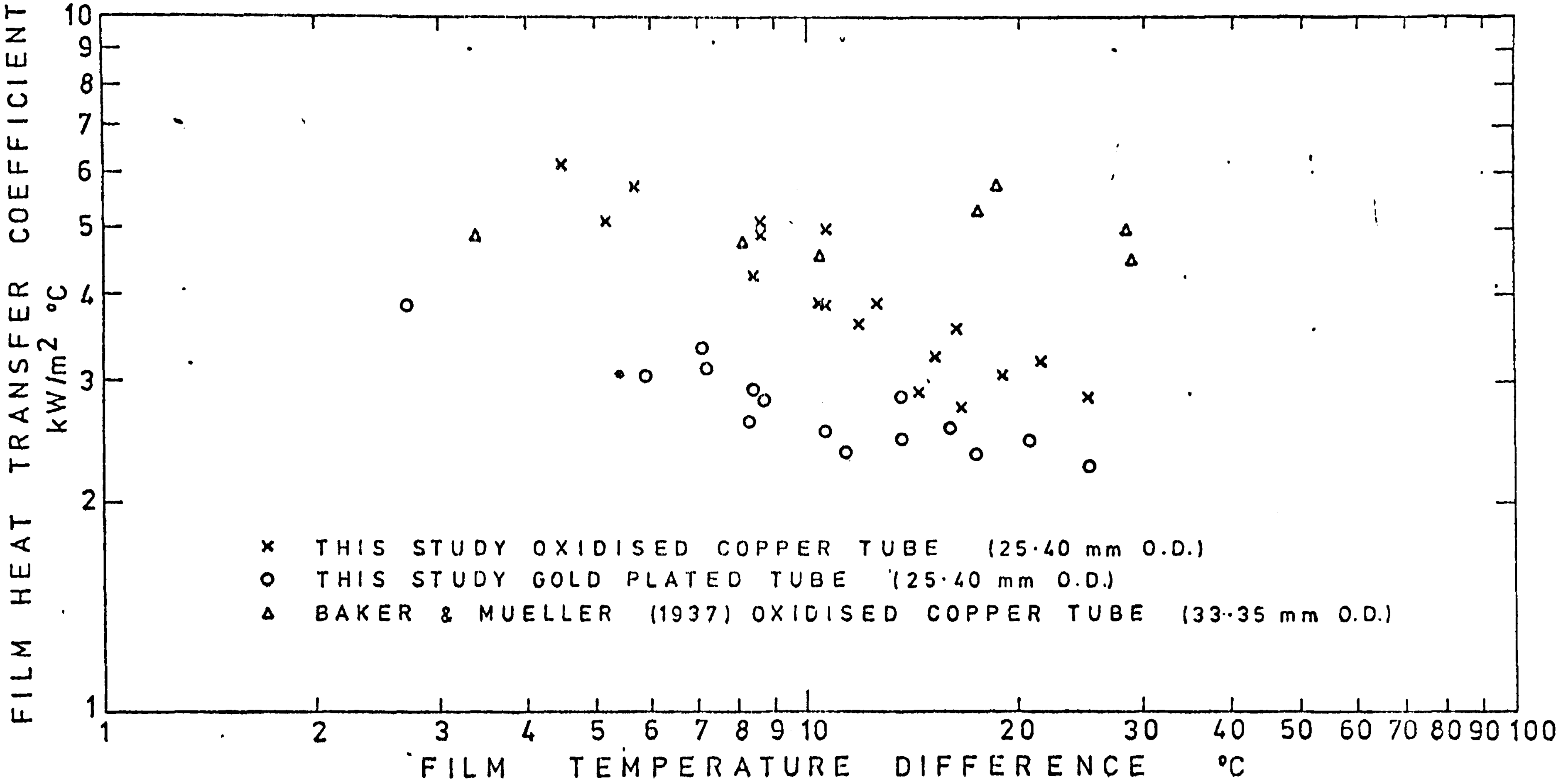

FIG. 6·10. COMPARISON OF EXISTING STEAM-TRICHLOROETHYLENE DATA IN THE COMPOSITION RANGE 92-95% BY WEIGHT TRICHLOROETHYLENE IN THE CONDENSATE

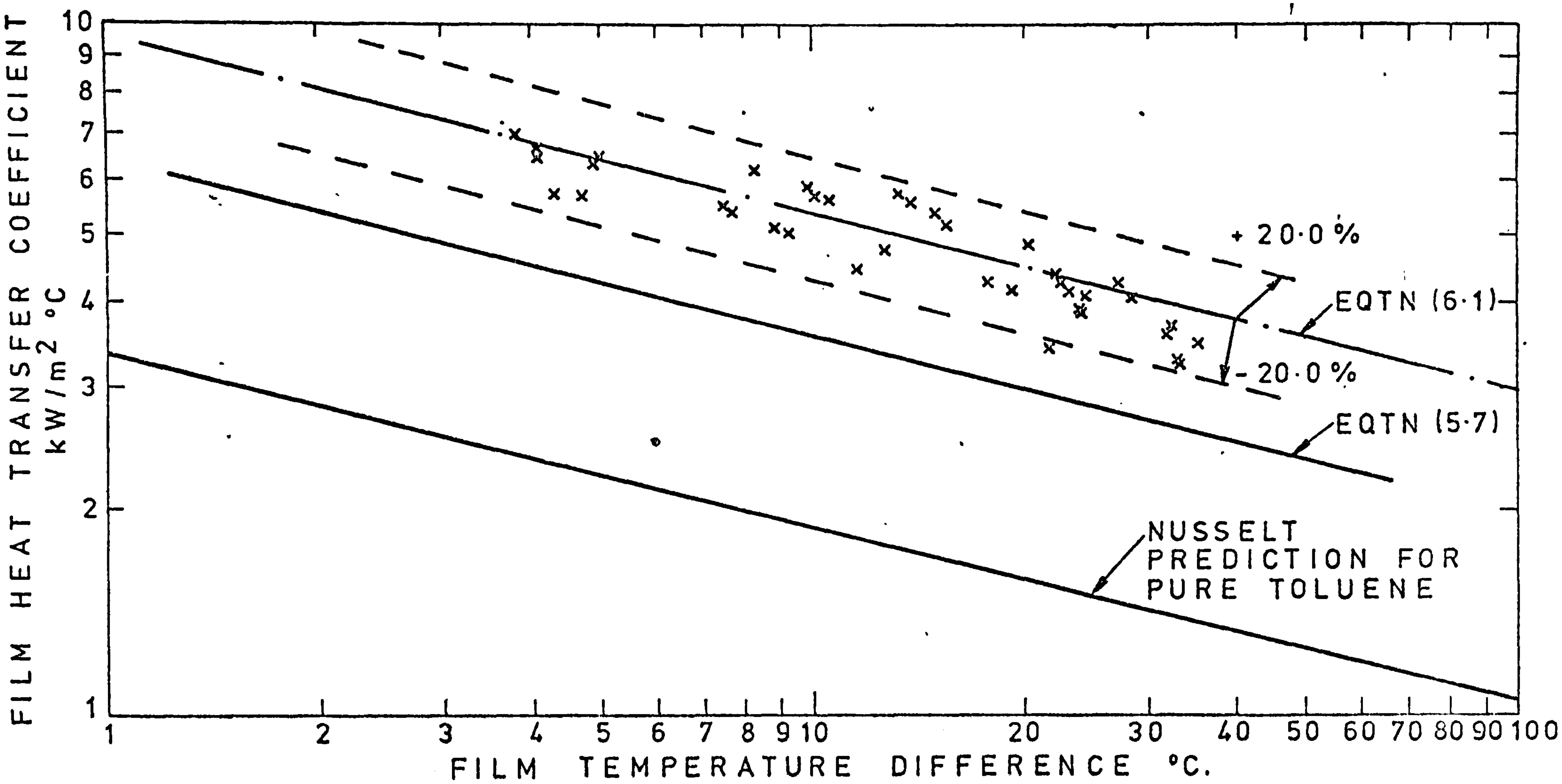

FIG. 6·11. COMPARISON BETWEEN THE CHANNELLING MODEL AND THE DATA FOR THE CONDENSATION OF STEAM-TOLUENE MIXTURES ON AN OXIDISED COPPER TUBE

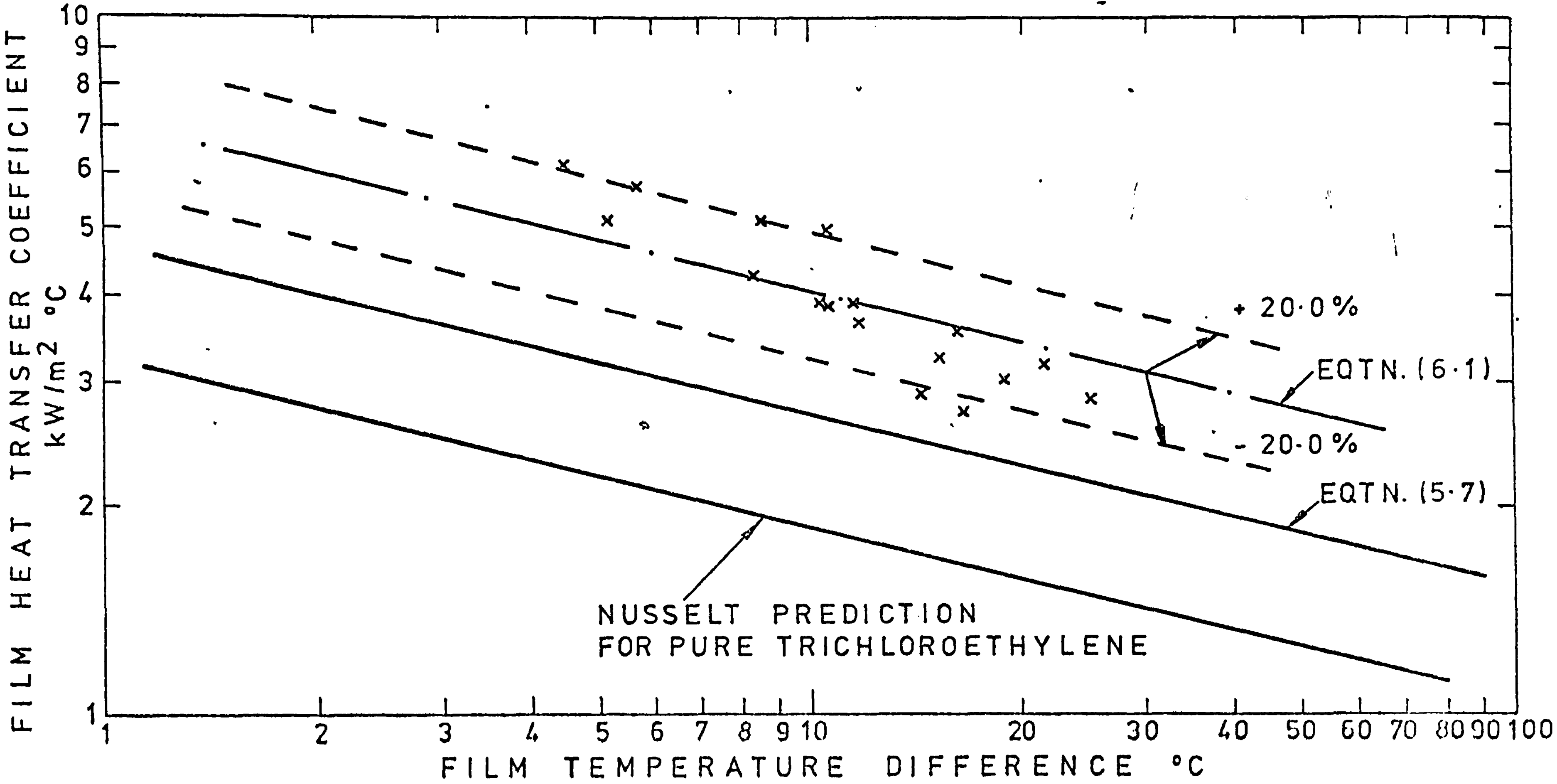

FIG. 6·12. COMPARISON BETWEEN THE CHANNELLING MODEL AND THE DATA FOR THE CONDENSATION OF STEAM-TRICHLOROETHYLENE MIXTURES ON AN OXIDISED COPPER TUBE

The discrepancy between the original theory (equation (5.7)) and the data may be due to the effects of the water drops protruding through the organic film and/or the small organic drops in the water films. Either of these could cause disturbances in the films, thus promoting better mixing and hence increased rates of heat transfer. If so, higher heat transfer coefficients than those predicted by the simple laminar film model would be expected.

It can be shown (chapter 5 section 5.2) that the Bernhardt correlation (equation (2.18)) is a special case of equation (5.7) and that for most organic-water mixtures the differences between the two equations is less than ± 5.0%. It is thus apparent that equation (5.7) will fit the same data as the Bernhardt correlation. Bernhardt et al (1972) have stated that their correlation fits the majority of the existing data to within an average error of ± 15.0%.

Since much of the earlier data has not been of a purely channelling type it would seem that the excellent predictions of the Bernhardt correlation have been somewhat fortutious.

In particular if the Bernhardt correlation is compared in detail with existing steam-toluene data it can be seen (figure 6.13) that the correlation does not predict the data to anywhere near ± 15.0%, overpredicting some data by factors of two or more and underpredicting a considerable amount of data by up to 50%.

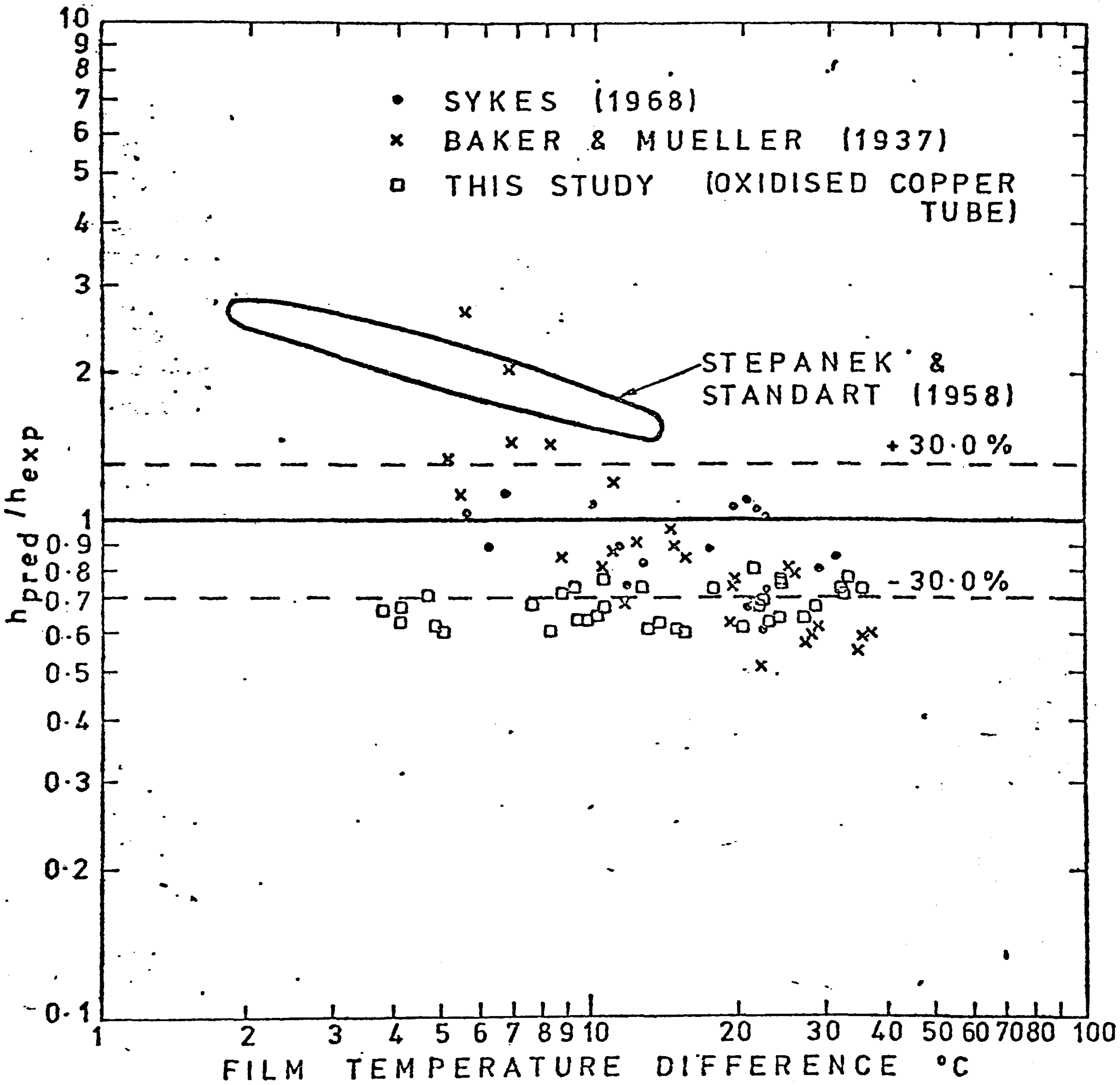

FIG. 6·13. COMPARISON OF BERNHARDTS EQUATION AGAINST EXPERIMENTAL DATA FOR STEAM TOLUENE MIXTURES

The standing drop model is compared with the authors data obtained on the gold plated tube in figures 6.14 and 6.15. It can be seen from figure 6.14 that for the steam toluene case the fractional area occupied by the drops has to be greater than 0.5 to give reasonable predictions. However, from observation of the cine films it is evident that the fractional area occupied by the drops is much less than the required 0.5.

From the description of the flow mechanism (for the steam-toluene case) given earlier (section 6.3) it is perhaps not so suprising that the standing drop model is not adequate (for the steam-toluene case), since rivulets of water are also present on the tube and at any instant of time occupy a finite fraction of the tube surface. Thus to predict the steam-toluene data adequately a combination of the channelling and standing drop models is probably more appropriate. To combine the two models, detailed measurements of the fractional area occupied by the drops and the water rivulets is required.

Figure 6.15 shows that the fractional area occupied by the drops in the steam-trichloroethylene case has to be in the range 0.6 to 0.3 to give reasonable predictions. And indeed the experimental observations confirm that such values are realistic. In this case the flow mechanism was purely of the standing drop type, so agreement between the model and observations would be expected.

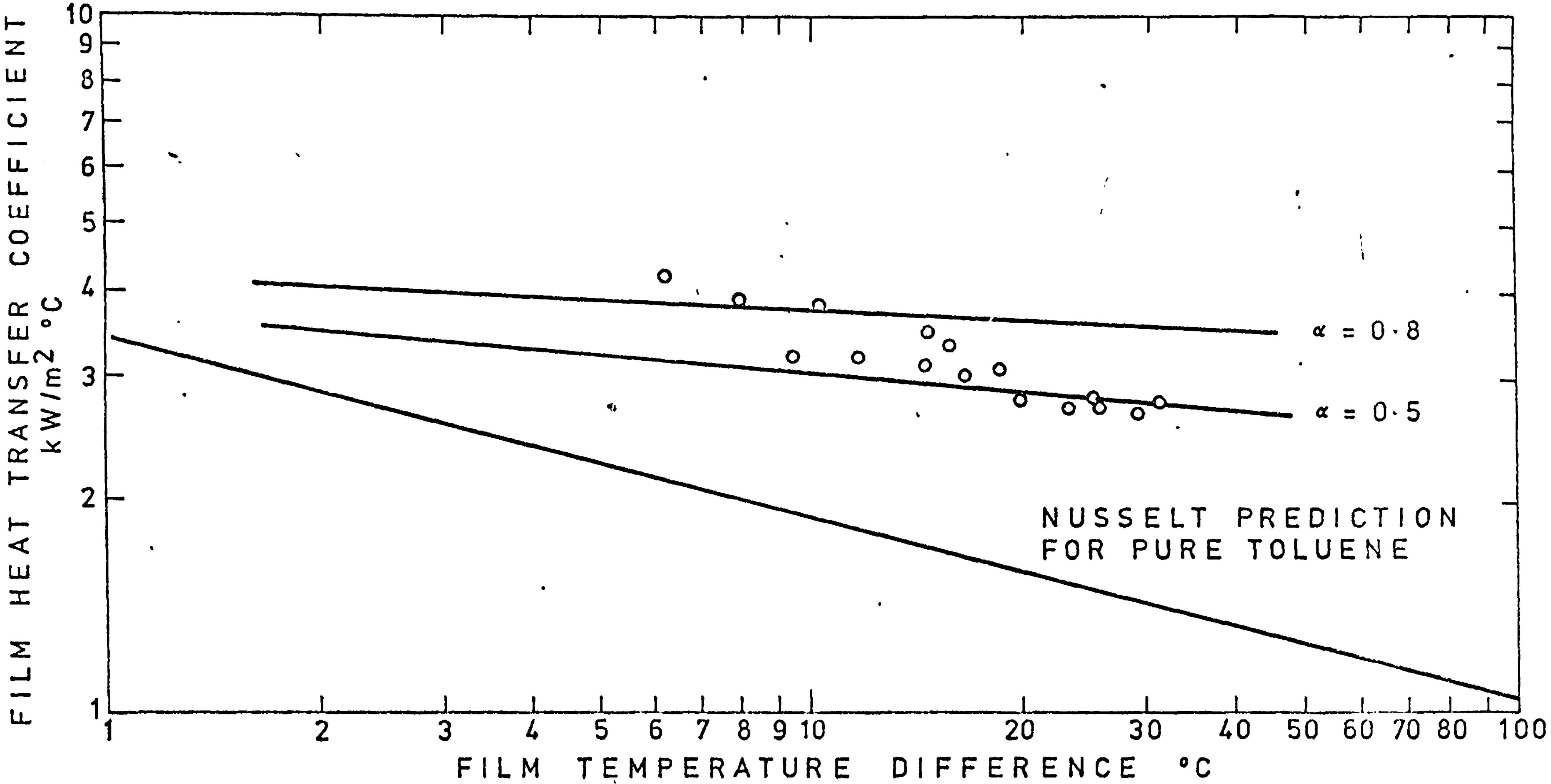

FIG. 6·14. COMPARISON BETWEEN THE STANDING DROP MODEL AND THE STEAM-TOLUENE DATA OBTAINED ON THE GOLD PLATED TUBE

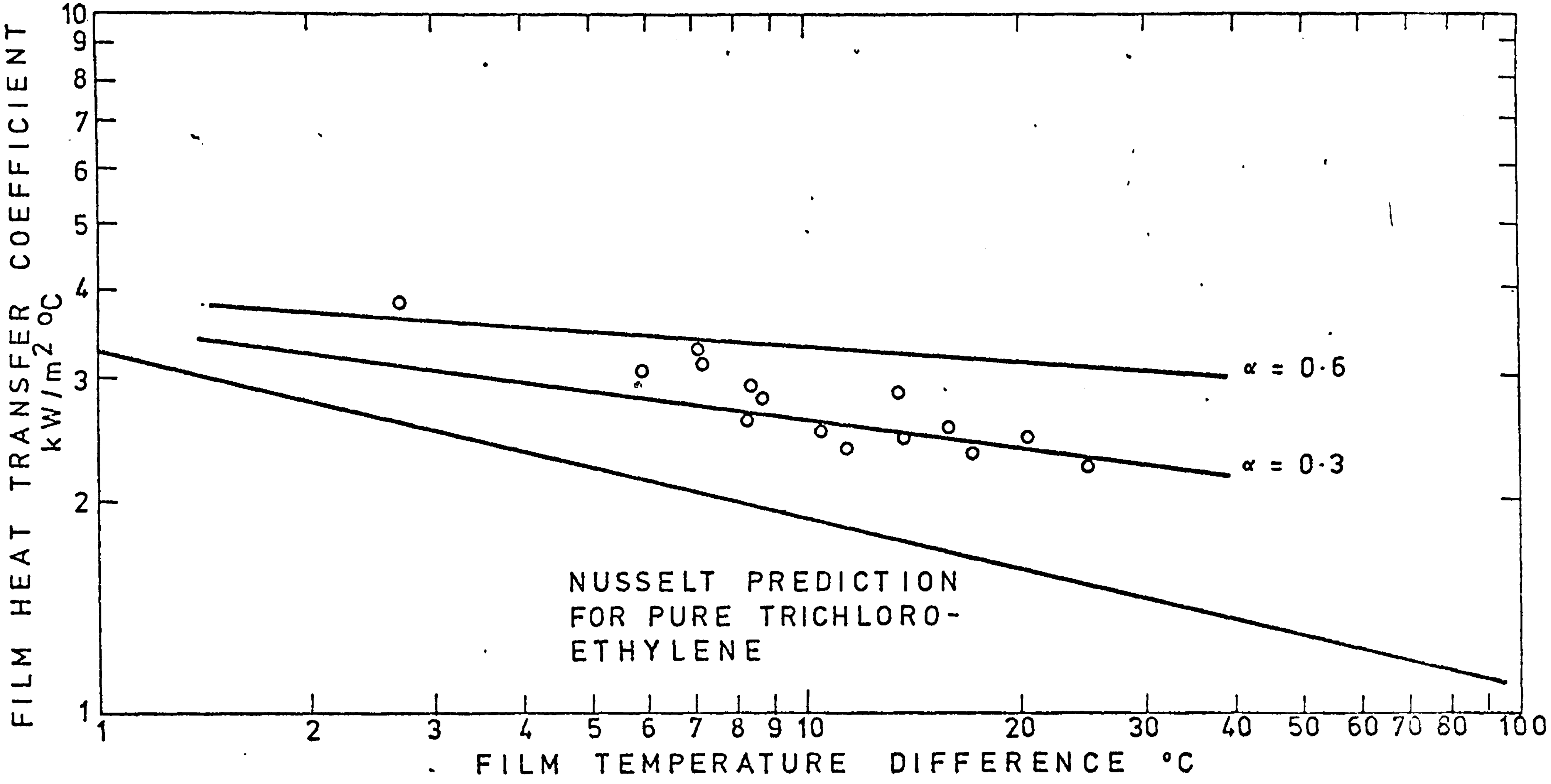

FIG. 6·15. COMPARISON BETWEEN THE STANDING DROP MODEL AND THE STEAM-TRICHLOROETHYLENE DATA OBTAINED ON THE GOLD PLATED COPPER TUBE

Several of the details of the flow mechanism have been ignored in the standing drop model such as the effect of the small organic droplets on the standing water drops. Presumably these organic drops will be a further resistance to heat transfer. Another important feature of the mechanism which has been ignored is the effect of the continual sweeping of the surface by the rolling drops. This would cause an increase in the heat transfer rate, since the organic film is continually being disturbed.

To support the standing drop model further work is needed to determine contact angles, dropsize distributions and the dropsize range which is effective in transferring heat to the wall. Also some suitable combination of the channelling and standing drop **model** might provide the basis for predicting any data set. A further requirement in trying to predict heat transfer coefficients for immiscible systems is a method of predicting the condensation mechanism for any combination of organic-water mixtures on any specified surface.

6.5 Nucleation barriers

As was stated earlier, the possibility exists that the organic vapour might be condensing on the water or the steam on the organic film. If this is the case then it is likely that a certain degree of vapour subcooling is required before condensation can take place, that is a nucleation barrier exists.

Consider the case of the organic vapour condensing on a water film (or drop). The free energy required to form a drop which is part of a spherical segment is (see appendix D),

$$\Delta G\ (r^*) = \frac{4}{3}\ \pi\ \sigma\ r^{*2}\ (2 + \cos\theta)\ (1 - \cos\theta)^2/4 \quad (6.2)$$

For organic vapour condensing on the water phase the contact angle is likely to be zero or near zero, thus the free energy to form a drop is zero or small. That is there is a very small or no barrier to the condensation of organic vapour on water.

However, when water is condensing on the organic it is known that water will not spread on organic films, that is a finite contact angle exists. Thus a free energy barrier will exist for this case. Unfortunately there seems to be no data available on the value of the required contact angles. In appendix D it can be seen from figure (D2) that for a steam-toluene mixture the degree of subcooling required is a strong function of contact angle. Thus with a contact angle of 30° the degree of subcooling required is approximately 2.7°C whereas for a 60° angle the degree of subcooling is approximately 9.2°C.

A few qualitative experiments were performed to try and test if nucleation barriers are of importance. To do this the following procedure was adopted. The cooling water (passing through the tube) temperature was raised to a level high enough to give very small film temperature differences ($\simeq 1.0^{\circ}C$ based on $T_e - T_w$). The condensation mechanism was then observed. If any significant barrier to the nucleation of one phase existed a change in appearance of the condensate would be expected. In fact it was found that for all the tests conducted the mechanisms were identical to those described earlier (section 6.3). Thus indicating that if nucleation barriers were present the vapour subcooling required to overcome the added resistance is less than 1.0°C.

An interesting speculation is that nucleation barriers are not important in "immiscible liquid condensation" because there are other mechanisms by which the steam may condense (for example the solubility effect described earlier in section 6.3) which offer less resistance than the nucleation path.

6.6 Conclusions and Recommendations

1. Nusselts equation is adequate for predicting film heat transfer coefficients for the condensation of pure steam, toluene and trichloroethylene.

2. The film heat transfer coefficients for the condensation of vapours of immiscible liquids decreases as the film temperature difference increases.

3. For binary immiscible systems two distinct fundamental modes of condensation were observed, namely a channelling mode and a standing drop mode.

4. The mode of condensation depended on the condenser surface being used, the channelling mode being observed on an oxidised copper surface and the standing drop mode on a gold plated surface.

5. For a given binary immiscible system the film heat transfer coefficients for the channelling mode are greater than those for the standing drop mode. That is the oxidised copper surface gives greater rates of heat transfer than the gold plated surface for a given mixture.

6. The origin of the small water and organic droplets, which have previously been attributed to nucleation effects, may be due to solubility phenomenon.

7. Models for predicting film heat transfer coefficients for both the channelling and standing drop modes have been presented.

Recommendations

1. The detailed condensation mechanism requires further study. Dropsize distributions, contact angles and detailed surface effects being of particular importance.

2. The detailed effects of tube diameter on the condensation process needs further study.

3. Other aspects which have so far received little or no attention in connection with the condensation of vapours of immiscible liquids are:

(a) Vapour shear effects

(b) Condensation on tube banks

(c) Intube condensation

(d) Work on mixtures of more than two components

(e) Effects of incondensable gases

All of the above are of particular importance in industrial applications.

References

AKERS, W.W. and TURNER, M.M. (1962). Condensation of vapours of immiscible liquids, A.I.Ch.E.J., No.5, pp.587-589.

ADAMSON, A.W. (1967) Physical chemistry of surfaces, John Wiley and Sons, second edition.

ARNOLD, E. (1970). U.K. Steam tables in S.I. units, Published for the U.K. committee on the properties of steam by Arnold Ltd. London.

BAKER, E.M. and MUELLER, A.C. (1937). Condensation of vapours on a horizontal tube Parts I and II. Ind. Eng. Chem. 29, No.9, pp.1065-1072 and also Trans. Am. Inst. Chem. Eng. 33, pp.539-558.

BAKER, E.M. and TSAO, U. (1940). Heat transfer coefficients for condensation of vapours of water and nonmiscible organic liquids, Ind. Eng. Chem. 32, No.8 pp.1115-1121 and also Trans. Am. Inst. Ch. E. 36, pp. 517-539.

BARNEA, E. and MIZRAHI, J. (1972). Heat transfer coefficient in the condensation of a hydrocarbon-steam mixture, Trans. Inst. Ch. E. 50, pp. 286-288.

BERNHARDT, S.H. (1970). Condensation of immiscible mixtures, Ph. D. Thesis, Univ. of Illinois.

BERNHARDT, S.H., SHERIDAN, J.J. and WESTWATER, J.W. (1972). Condensation of immiscible mixtures, A.I. Ch. E. Symp. Ser., Vol.68, No.118, pp.21-37.

BOYES, A.P. and PONTER, A.B. (1972). Condensation of immiscible binary systems, Chem. Proc. Eng. Heat Transfer Survey, pp.26-30.

BROMLEY, L.A. (1952). Effect of heat capacity of condensate, Ind. Eng. Chem. 44, pp. 2966-2969.

BUTTERWORTH, D. (1972). Personal communication.

COLLIER, J.G. (1972). Convective boiling and condensation, McGraw Hill Book Co., pp. 301-303.

CHADDOCK, J.B. (1957). Film condensation of vapour in a horizontal tube, Ref. Eng., 90-95, pp. 36-41.

CHEN, M.M. (1961). An analytical study of laminar film condensation: Part I - flat plates, Part II - single and multiple horizontal tubes, J. Heat Transf. Ser. C. 83, pp. 48-60.

CHUN, K.R. and SEBAN, R.A. (1971). Heat transfer to evaporating liquid films, J. Heat Transf. Ser. C. 93, pp.391-396.

COOPER, A.H., HALL MORRISON, E. and HENDERSON, H.E., Heat transfer of condensing organic vapours, Ind. Eng. Chem. 34, No.1, pp.79-84.

EDWARDS, D.A., BONILLA, C.F. and CICHELLI, M.T. (1948). Condensation of water, styrene and butadiene vapours, Ind. Eng. Chem. 40, No.6, pp.1105-1112.

Engineering Science Data Unit (E.S.D.U.) (1966). Approximate data on the viscosity of some common liquids, Item No. 66024.

E.S.D.U. (1968). Thermal conductivity of liquid halogenated aliphatic hydrocarbons including commercial refrigerants, Item No. 68024.

E.S.D.U. (1974). Thermal conductivity of liquid aromatic hydrocarbons, Item No. 74007.

E.S.D.U. (1974). Thermodynamic properties of toluene, Item No. 74024.

FATICA, N. and KATZ, D.L. (1949). Dropwise condensation, Chem. Eng. Progress, 45, pp.661-674.

FUJII, T., UEHARA, H. and ODA, K. (1972). Filmwise condensation on a surface with uniform heat flux and body force convection, Heat Transf. Jap. Res. Vol.1, No.4, pp.76-83.

GALLANT, R.W. (1970). Physical properties of hydrocarbons, Vols. 1 and 2, Gulf publishing Co.

HAZELTON, R.H. and BAKER, E.M. (1944). Condensation of vapours of immiscible liquids, Trans. Am. Inst. Ch. E. 40, pp. 1-28.

HILL, P.G., WHITING, H. and DEMETRI, E.P. (1963). Condensation of metal vapours during rapid expansion, J. Heat Transf. 83, pp. 303-317.

International Critical Tables (1930).

JAMIESON, D.T., IRVING, J.B. and TUDHOPE, J.S. (1974). The thermal conductivity of liquids a data survey to 1973, H.M.S.O. publication.

JEFFREYS, J.O. (1936). Cornell Univ., Exp. St., Bulletin No. 21.

KAPITZA, P.L. (1948). Wave motion in the flow of thin layers of viscous liquid, Zh. Eksp. Theoret. Fiz. 18, pp. 7-18.

KAWASKI, J., HAYAKAWA, T. AND FUJITA, S. (1972). Condensation of binary mixtures of vapours of immiscible liquids, Heat Transf. Jap. Res. Vol.1, No.1, pp. 33-38.

KIRK AND OTHMER (1964). Encylopedia of Chem. Tech., Sec. Ed.

KIRKBRIDE, C.G. (1933). Heat transmission by condensing pure and mixed substanced on horizontal tubes, Ind. Eng. Chem. 25, No.12, pp.1324-1330.

KOH, J.C.Y., SPARROW, E.M. and HARTNETT, J.P. (1961). The two phase boundary layer in laminar film condensation, Int. J.H.M. Transf. 2, pp. 69-82.

LABUNTSOV, D.A. (1957). Effect of temperature dependence of physical parameters of condensate on heat transfer in film condensation of steam, Teploenergetika, 4, pp.49-51.

MARSCHALL, E. and HICKMAN, R.S. (1973). Laminar gravity flow film condensation of binary vapour mixtures of immiscible liquids, J. of Heat Transf. Ser. C. 95, pp. 1-5.

MERTE, H. (1973). Condensation heat transfer, Adv. in Heat Transf., 9, pp.181-272.

MCADAMS, W.H. (1954). Heat Transmission, McGraw-Hill.

MIRKOVITCH, V.V. and MISSEN, R.W. (1961). Non filmwise condensation of binary vapours of miscible liquids, Can. J. Chem. Eng. 39, pp. 86-87.

NUSSELT, W. (1916). Surface condensation of water vapour, Parts I and II. Z. Ver. D. Ing. 60 No.27, pp. 541-546 and No. 28, pp.569-575.

PATTERSON, W.C. et al. (1937). The condensation of steam and heptane on vertical tubes. Trans. Am. Inst. Chem. Eng. 33, pp.216-241.

PATTON, E.L. and FEAGAN, R.A. (1941). Heat transfer coefficients for the condensation of mixed vapours of turpentine and water on a single horizontal tube, Ind. Eng. Chem. 33, No. 10, pp. 1237-1239.

PONTER, A.B. and DIAH, I.G. (1974). Condensation of vapours of immiscible binary liquids on orizontal copper and polytetrafluoroethylene coated copper tubes, Warme und Stoffubertragung, 7, No.2, pp.94-106.

POOTS, G. and MILES, R.G. (1967). Effects of variable physical properties on laminar film condensation of saturated steam on a vertical flat plate. Int. J. H. M. Transf. 10, pp. 1677-1692.

ROHSENOW, W.M. (1956). Heat transfer and temperature distribution in laminar film condensation, Trans, A.S.M.E. 78, pp. 1645-1648.

SALOV, V.S. and DANILOV, O.L. (1975). The condensation of binary mixtures of the vapours of immiscible liquids on non isothermal surfaces, Int. Chem. Eng., Vol.15, No.1, pp.39-43.

SEIDELL, A. (1958). Solubilities-Inorganic and Metal Organic Compounds, Vol.1, 4th edition. American Chem. Soc.

SPARROW, E.M. and GREGG, J.L. (1959). A boundary layer treatment of laminar film condensation, J. Heat Transf. Ser. C. 81, pp. 13-18.

STANDART, G. (1973). Personal communication

STEPANEK, J. and STANDART, G. (1958). Heat transfer during the condensation of mixtures of vapours of immiscible liquids. Collection Czech. Chem. Commun. 23, pp. 995-1011.

SYKES, J.A. (1968) Condensation heat transfer for vapours of immiscible liquids outside horizontal copper tubes. Ph. D. Thesis. Univ. of Maryland.

SYKES, J.A. and MARCHELLO, J.M. (1970) Condensation of immiscible liquids on horizontal tubes, Ind. Eng. Chem. Proc. Des. Dev. Vol.9, No.1, pp.63-71.

TANAKA, H. (1975). Measurement of dropsize distributions during transient dropwise condensation, J. Heat Transf. Ser. C. Vol.97, No.3, pp. 341-346.

TIMMERMANS, J. (1950). Physics chemical constants of pure organic compounds, Elsevier publishing Co.,

TIMMERMANS, J. (1965). Physico-chemical constants of pure organic compounds, Vol.2, Elsevier publishing Co.

TOBIAS, M. and STOPPEL, A.E. (1954). Condensation of vapours of water and immiscible organic liquids, Ind. Eng. Chem. 46, pp. 1450-1455.

TURNBULL, D. (1950). Kinetics of heterogeneous nucleation, The J. of Chem. Phys. Vol.18, No.2, pp. 198-203.

van der WALT, J. and KROGER, D.G. (1971) Heat transfer during film condensation of saturated and superheated freon-12, Prog. H.M. Transf. Vol.6, pp.75-98.

van ES, J.P. and HEERTJES. P.M. (1962). The condensation of vapour of a binary mixture. Brit. Chem. Eng. Vol. 7, No. 8, pp. 580-586.

VOSKRESENSKIY, K.D. (1948). Calculation of heat transfer in film condensation allowing for the temperature dependence of the physical properties of the condensate, USSR, Acad. Sci. OTK. (As reported by Poots and Miles (1967)).

YUSOFOVA, V.D. and NEIKOUCHT, N.N. (1970). An investigation of the process of condensation of gasoline vapour in the presence of water vapour, Int. Chem. Eng. 10, No.3, pp.422

Nomenclature

			Units S.I.
a	-	Weight fraction of component 1 or A	-
A	-	Total heat transfer area	m^2
A_F	-	Area occupied by the film	m^2
A_i	-	Area occupied by component i	m^2
b	-	Weight fraction of component 2 or B	
B	-	Length term in Nusselts equation	m
		$B = D_o$ for horizontal tubes	
		$B = L$ for vertical tubes	
c	-	Constant in equation (5.13)	-
C	-	Constant in Nusselts equation:	-
		$C = 0.728$ for horizontal tubes	-
		$C = 0.943$ for vertical tubes	-
C_1, C_2, C_3 and C_4 are constants in equations (2.29) and (2.30).			-
C_{Pi}	-	Specific heat of component i	J/kg °C
C_{pav}	-	Weight average of the pure component specific heats.	J/kg °C
d	-	Base diameter of a drop	m
d_{max}	-	Maximum drop diameter	m
d_{min}	-	Minimum drop diameter	m
D_o	-	Outside tube diameter	m
F	-	Cooling water flowrate	kg/s
g	-	Gravitational acceleration	m/s^2
Ga	-	Gallileo number - $(D_o^3 \rho^2 g/\mu^2)$	-
h	-	Heat transfer coefficient of immiscible condensates	W/m^2K
h_c	-	Chen film coefficient (equation (2.4) and (2.5))	W/m^2K
h_d	-	Heat transfer coefficient through a drop of base diameter d and contact angle θ	W/m^2K

h_e	Heat transfer coefficient for eutectic mixtures.	W/m^2K
h_H	Heat transfer coefficient defined by equation (2.21).	W/m^2K
h_L	Labuntsov film coefficient (equation (2.7)).	W/m^2K
h_{NF}	Nusselts coefficient for the film.	W/m^2K
h_{Ni}	Nusselts coefficient for component i.	W/m^2K
h_R	Rohsenow film coefficient (equation (2.3)).	W/m^2K
h_T	Heat transfer coefficient for a horizontal tube with a non isothermal wall.	W/m^2K
h_w	Chun and Seban film coefficient (equation (2.10)).	W/m^2K
I	Nucleation rate.	s^{-1}
k	Boltzmans constant in equations (D1) and (D5).	J/K
k_i	Thermal conductivity of component i.	W/m^oC
k_{av}	Weight average of the pure component thermal conductivities.	W/m^oC
k'_{av}	Volume average of the pure component thermal conductivities.	W/m^oC
k_s	Thermal conductivity evaluated at T_s	W/m^oC
k_w	Thermal conductivity evaluated at T_w.	W/m^oC
Ku	Kutateladze number - $(\lambda/C_p \Delta T_f)$.	-
l	Length of the copper block.	m
L	Length of a vertical surface	m
$\dot{m}_i$	Mass rate of condensation of component i.	kg/s
M_i	Molecular weight of component i.	-
n	Constant in equation (5.13)	-
n	= 0.67R in equation (2.25).	-
N	Probability density of time averaged dropsize distributions.	m^{-3}
N_o	Avogadros number (6.023×10^{26}).	kg mole^{-1}
Nu	Nusselt number - $(h\, D_o/k)$.	-
Oh	Ohnesorge number - $(\mu^2/\rho\, g\, D_o \sigma)^{\frac{1}{2}}$.	-

p^o	-	Vapour pressure	N/m^2
P	-	Constant in equation (2.2):	-
		P = 1.47 for horizontal tubes.	-
		P = 1.51 for vertical tubes.	-
P_c	-	Critical pressure	N/m^2
P_g	-	Vapour pressure over the drop at a system temperature Tg	-
Pr	-	Prandtl number - $(C_p \mu/k)$	-
P_∞	-	Vapour pressure for a planar interface at Tg.	N/m^2
q	-	Heat flux through the copper block	W/m^2
q_c	-	Condensate heat flux.	W/m^2
Q	-	Total heat load	W
Q_d	-	Is the heat transferred through a drop of base diameter d.	W
Q'_d	-	Is the heat transferred through all drops of base diameter d.	W
Q_D	-	Is the heat transferred through all drops in a given diameter range.	W
Q_F	-	Is the heat transferred through the film.	W
Q_i	-	Is the heat load for component i.	W
r^*	-	Critical nuclei radius	m
R	-	Gas constant (8.314×10^3)	J/kg mole K
S	-	As defined by equation (E17).	-
S_{BA}	-	Spreading coefficient for B on A.	N/m
t	-	Temperature in °C.	°C
t_{IN}	-	Cooling water inlet temperature.	°C
T_B	-	Boiling point	°C
T_c	-	Critical temperature	°C
T_e	-	Eutectoid temperature	°C
Tg	-	Temperature of the vapour system	K
T_K	-	Temperature in K.	K
T_s	-	Saturation temperature	°C
$T_{v(IN)}$		Vapour inlet temperature	°C
T_w	-	Tube surface temperature	°C
U	-	Velocity in the x direction	m/s

v_i -	Volume fraction of component i in the condensate	-
V_i -	Volume of liquid i on the tube surface.	m^3
V -	Total liquid volume on the tube surface.	m^3
x -	Is the height at which T_w is being calculated in equation (2.30).	m
x -	P_g/P	-
Y -	As defined by equation (5.8)	-

Greek Symbols

α -	Fractional area occupied by the drops	-
α' -	Angle of inclination of the surface	o
β -	Constant in equation (2.27) - = 0.063	$^oC^{-1}$
Γ -	Mass flowrate per unit width of condensate film	kg/sm
δ_i -	Film thickness of component i	m
$\Delta G(r^*)$	Free energy required to form a drop of radius r*.	J
Δr -	1-r	
Δt -	Cooling water temperature difference	^oC
ΔT_f -	Film temperature difference	^oC
$\Delta\rho$ -	$\rho_2 - \rho_1$	kg/m^3
$\Delta\sigma$ -	$\sigma_2 - \sigma_1$	N/m
ε -	$C_p \Delta T_f/\lambda$	-
θ -	Contact angle	o
θ_A -	Advancing contact angle	o
θ_R -	Receding contact angle	o
λ_{av} -	Weight average of the pure component latent heats of vapourisation	J/kg
λ_H -	$(a\lambda_1 + b\lambda_2)/a$	J/kg
λ_i -	Latent heat of vapourisation of component i.	J/kg
μ_{av} -	Weight average of the pure component viscosities	Ns/m^2
μ'_{av} -	Volumetric average of the pure component viscosities	Ns/m^2

μ_i	-	Viscosity of component i.	Ns/m^2
μ_s	-	Viscosity evaluated at the saturation temperature.	Ns/m^2
μ_w	-	Viscosity evaluated at the wall temperature.	Ns/m^2
ρ_{av}	-	Weight average of the pure component densities.	kg/m^3
ρ'_{av}	-	Volumetric average of the pure component densities	Kg/m^3
ρ_i	-	Density of component i.	kg/m^3
σ_i	-	Surface tension of component i.	N/m
σ_{ij}	-	Interfacial tension between liquids i and j.	N/m
ϕ	-	Angle at which T_w is calculated in equation (E1).	o
ϕ_L	-	Labunstov physical property correction factor.	-
ω	-	$C_1/\Delta \bar{T}_f$.	-

Subscripts

i - A or 1 and B or 2

A or 1 - Property of the wall wetting phase.

B or 2 - Property of the other phase.

Note

An overbar denotes a mean value.

APPENDIX A

PHYSICAL PROPERTIES

The following tables contain the correlations for calculating the physical properties used in this study.

Table 1 contains correlations for water, these were tested against data obtained from Arnold (1970), the agreement between the data and correlations is better than ±2.0% within the temperature ranges quoted.

Table 2 contains correlations for toluene, these were tested against data from various sources but mainly ESDU Items 74024 (1974), Item 74007 (1974), Item 66024 (1966), Jamieson et al (1973), International Critical Tables (1930) and Timmermans (1950) and (1965). The agreement between the data and correlations is better than ±5.0% within the given temperature ranges.

Table 3 contains correlations for trichloroethylene, these were again tested against various data sources, i.e. E.S.D.U. Item 68024 (1968), Item 66024 (1966), Jamieson et al. (1973), Sykes (1968), Gallant (1970) and Kirk and Othmer (1964). The agreement between the data and correlations is better than ±5.0% (except for the liquid heat capacity which is within ±10%) within the given temperature ranges.

TABLE A 1

PHYSICAL PROPERTY CORRELATIONS FOR WATER

Physical property	Equation
Liquid density ρ in kg/m^3	$\rho = 1000.0 - 45.05 \left[\frac{t - 5.0}{100.0}\right]^{1.65}$ Temperature range 0–220°C.
Liquid viscosity μ in Ns/m^2	$\mu = \exp\left[\frac{587.04}{136.1 + t} - 10.6604\right]$ Temperature range 0–320°C
Liquid thermal conductivity k in W/m K	$k = 0.688 - 7.76 \times 10^{-6} \quad (t - 125.0)^2$ Temperature range 0–140°C
Surface tension σ in N/m	$\sigma = 0.1082 - 7.2629 \times 10^{-5} T_K - 1.6052 \times 10^{-7} T_K^{\ 2}$ Temperature range 0–200°C
Liquid specific heat capacity C_p in J/kg K	$C_p = 4179.0 + 1.232 \times 10^{-2} \quad (t - 40.0)^2$ Temperature range 0–200°C
Latent heat of vapourisation λ in J/kg	$\lambda = 2.5018 \times 10^6 - 2368.0\ t - 5.254 \times 10^{-5} t^4$ Temperature range 0–240°C
Vapour pressure p^o in N/m^2	$\ln p^o = 23.2189 - 3842.56/(228.6 + t)$ Temperature range 0–240°C
Critical temperature T_c in °C	374.15 (647.30 K)
Critical pressure P_c in N/m^2	2.212×10^7 (218.3 atm)

TABLE A1 continued

Physical property	Equation
Normal boiling point (1 atm) T_B in °C	100.00 (373.15 K)
Molecular weight	18.0153

Where t is the temperature in °C and T_K the temperature in K.

Note

The equations above were obtained from Butterworth (1972).

TABLE A 2

PHYSICAL PROPERTY CORRELATIONS FOR TOLUENE

Physical property	Equation
Liquid density ρ in kg/m^3	$\rho = 885.45 - 0.92248\,t - 3.055 \times 10^{-6} t^3$ Temperature range 0–130°C
Liquid viscosity μ in Ns/m^2	$\mu = 1.540 \times 10^{-5} \exp(1067.8/T_K)$ Temperature range 0–110°C
Liquid thermal conductivity k in W/mK	$k = 0.1368 - 0.000273t$ Temperature range 0–120°C
Surface tension σ in N/m	$\sigma = 28.43 \times 10^{-3} \left[\frac{1 - \frac{T_K}{591.8}}{0.5046}\right]^{1.23}$ Temperature range 0–120°C
Liquid specific heat capacity C_p in J/kg K	$C_p = 1599.8 + 3.8895\,t$ Temperature range 0–300°C
Latent heat of vapourisation λ in J/kg	$\lambda = 3.634 \times 10^5 \left[\frac{1 - \frac{T_K}{591.8}}{0.3536}\right]^{0.37}$ Temperature range 0–260°C
Vapour pressure p^o in N/m^2	$\ln p^o = 20.90352 - \frac{3094.55}{219.377 + t}$ Temperature range 0–260°C
Critical temperature T_c in °C	318.7 (591.8K)

TABLE A2 continued

Physical property	Equation
Critical pressure P_c in N/m^2	4.104 x 10^6 (40.5 atm)
Normal boiling point T_B in °C	110.63 (383.73 K)
Molecular weight	92.1418

Where t is the temperature in °C and T_K the temperature is K.

TABLE A3

PHYSICAL PROPERTY CORRELATIONS FOR TRICHLOROETHYLENE

Physical property	Equation
Liquid density ρ in kg/m^3	$\rho = 1496.27 - 1.6493\ t$ Temperature range 0-100°C
Liquid viscosity μ in Ns/m^2	$\mu = 3.5714 \times 10^{-5} \exp(814.7/T_K)$ Temperature range 0-80°C
Liquid thermal conductivity k in W/m °C	$k = 0.1275 - 0.000398\ t$ Temperature range 0-80°C
Surface tension σ in N/m	$\sigma = 0.0312 - 1.075 \times 10^{-4} t - 1.25 \times 10^{-7} t^2$ Temperature range 0-100°C *
Liquid specific heat capacity C_p in J/kg K	$C_p = 946.2 + 0.8374t$ Temperature range 0-100°C *
Latent heat of vapourisation λ in J/kg	$\lambda = 2.4702 \times 10^5 \left[\frac{1 - \frac{T_K}{544.15}}{0.3878}\right]^{0.38}$ Temperature range 0-100°C
Vapour pressure p^o in N/m^2	$\ln p^o = 21.07558 - \frac{3027.9}{230.0 + t}$ Temperature range 0-100°C
Critical temperature T_c in °C	271.0 (544.15 K)
Critical pressure P_c in N/m^2	5.016×10^6 (49.5 atm)

TABLE A3 continued

Physical property	Equation
Normal boiling point T_B in °C	87.0 (360.15 K)
Molecular weight	131.3893

Where t is the temperature in °C and T_K the temperature in K.

This equation was derived from the data of Gallant (1970) and has not been compared against an independent data set.

APPENDIX B

Tabulated Results

The results obtained in the present study are tabulated in the following seven tables. From the error analysis (see appendix G) a heat balance error limit of ± 15% was set. That is any experimental run giving a heat balance error greater than ± 15% was rejected. Approximately 5% of the total number of runs were rejected.

Table B1 requires further explanation here it can be seen that the heat balances are all negative and of order -20%. This large error was due to the initial method of measuring the condensate flowrate, a consistently low reading being obtained. Since the condensate flowrate was not a reliable method of determining the heat transferred in this particular case the heat transferred to the cooling water was used to determine the heat transfer coefficients.

The error in measuring the condensate flowrate was corrected for all subsequent runs and for these the condensate flowrate was used to determine the heat transfer coefficients.

APPENDIX B

TABLE B 1

DATA for the CONDENSATION of PURE STEAM

Run No.	T_s °C	T_w °C	ΔT_F °C	q_c W/m^2	t_{IN} °C	Δt °C	F kg/s	Ht. Bal. %	h W/m^2°C	h_L W/m^2°C
1	100.0	86.0	14.0	148400	71.3	3.10	0.8360	- 17.2	10600	11378
2	100.0	86.9	13.1	152615	72.3	3.22	0.8240	- 15.3	11650	11595
3	100.0	89.2	10.8	129319	74.4	3.29	0.6818	- 19.9	11974	12205
4	100.0	90.1	9.9	123027	76.2	3.22	0.6671	- 17.4	12427	12470
5	100.0	89.7	10.3	128812	78.9	2.38	0.9434	- 16.5	12506	12345
6	100.3	90.3	10.0	119880	79.9	2.20	0.9491	- 15.2	11988	12449
7	101.2	92.1	9.1	105851	82.5	2.04	0.9069	- 12.3	11632	12784
8	99.9	91.6	8.3	101434	82.4	1.95	0.9113	- 13.0	12221	13041
9	100.0	83.7	16.3	180441	66.1	3.19	0.9866	- 25.5	11070	10927
10	100.0	85.3	14.7	174239	68.9	3.09	0.9818	- 24.7	11853	11233
11	99.5	84.8	14.7	183089	68.4	3.02	1.0523	- 18.9	12455	11233
12	99.7	86.5	13.2	170438	71.4	2.84	1.0405	- 19.5	12912	11563
13	100.0	89.9	10.1	137178	77.7	2.38	1.0026	- 18.5	13582	12416
14	99.8	90.4	9.5	132031	78.5	2.30	0.9926	- 21.2	13898	12621
15	99.8	91.5	8.3	102198	82.4	1.82	0.9817	- 18.5	12313	13037
16	99.8	92.4	7.4	93492	84.1	1.69	0.9614	- 18.7	12634	13465
17	99.8	92.0	7.8	96361	84.1	1.61	1.0443	- 20.2	12354	13267
18	99.7	92.6	7.1	89041	85.2	1.51	1.0194	- 19.6	12541	13608
19	99.7	93.9	5.8	77546	87.3	1.38	0.9764	- 20.6	13370	14327
20	99.7	84.0	5.7	78478	87.5	1.39	0.9746	- 24.4	13768	14415

TABLE B 2

DATA for the CONDENSATION of TOLUENE

Run No.	T_s °C	T_W °C	ΔT_F °C	q_c W/m^2	t_{IN} °C	Δt °C	F kg/s	Ht. Bal. %	h W/m^2 °C	h_L W/m^2 °C
1	110.7	100.4	10.3	21,349.8	98.8	0.45	0.7310	- 9.4	2072.8	1834.4
2	110.2	77.3	32.9	48,159.0	73.6	0.87	0.8478	- 12.2	1463.8	1384.6
3	110.2	79.2	31.0	45,570.0	75.4	0.89	0.8148	- 8.4	1470.0	1404.6
4	110.2	81.0	29.2	41,656.7	77.1	0.83	0.8376	- 3.7	1426.6	1425.0
5	110.2	84.7	25.5	39,191.0	81.1	0.79	0.8411	- 3.0	1536.9	1472.1
6	109.8	90.6	19.2	31,791.4	87.1	0.83	0.7604	14.6	1655.8	1578.0
7	109.8	93.0	16.8	29,198.4	89.9	0.75	0.7232	7.2	1738.0	1630.2
8	110.0	94.7	15.3	26,441.5	91.5	0.79	0.6500	11.4	1728.2	1667.9
9	110.2	84.7	25.5	42,498.3	81.1	0.78	0.8180	- 14.1	1666.6	1472.8
10	110.7	93.9	16.8	31,063.2	90.9	0.68	0.6930	- 12.6	1849.0	1627.8
11	110.3	96.6	13.7	27,190.4	94.0	0.61	0.7431	- 3.0	1984.7	1712.0
12	111.2	99.7	11.5	23,947.6	97.1	0.66	0.6774	6.9	2082.4	1783.7
13	110.6	99.1	11.5	24,091.4	96.4	0.67	0.6633	6.0	2094.9	1785.6
14	111.2	81.0	30.3	46,683.2	74.5	1.42	0.5576	- 2.1	1540.7	1411.6
15	111.1	87.0	24.6	41,903.6	81.7	1.27	0.5494	- 1.8	1703.4	1492.6
16	110.9	89.9	21.0	35,962.5	85.0	1.20	0.5342	2.7	1712.5	1541.9
17	111.0	95.8	15.2	30,089.9	92.4	0.94	0.5293	- 5.2	1979.6	1668.2
18	110.3	78.6	31.7	47,432.7	73.1	1.15	0.6793	- 5.7	1496.3	1397.1
19	110.3	84.2	26.1	41,840.9	79.5	1.01	0.6728	- 6.8	1603.1	1464.6
20	110.4	92.4	18.0	33,197.4	88.4	0.95	0.5407	- 9.9	1844.3	1602.7
21	110.2	93.7	16.5	30,345.2	90.0	0.92	0.5368	- 6.8	1839.1	1637.6

TABLE B 3

DATA for the CONDENSATION of PURE TRICHLOROETHYLENE

Run No.	T_s °C	T_w °C	ΔT_F °C	q_c W/m^2	t_{IN} °C	Δt °C	F kg/s	Ht. Bal. %	h W/m^2 °C	h_L W/m^2 °C
1.	87.0	83.2	3.8	9,166	81.8	0.34	0.4974	6.5	2412	2279
2	87.0	82.0	5.0	10,660	80.2	0.42	0.4818	9.0	2132	2137
3	87.0	80.2	6.8	13,416	77.9	0.51	0.4862	6.4	1973	1977
4	86.6	66.8	19.8	25,720	62.7	0.83	0.5412	0.1	1299	1533
5	86.6	68.0	18.6	24,087	64.0	0.79	0.5412	2.1	1295	1556
6	86.6	70.2	16.4	22,681	66.6	0.77	0.5445	5.8	1383	1603
7	86.7	71.0	15.7	21,792	67.4	0.76	0.5406	7.5	1388	1619
8	86.8	73.0	13.8	21,349	69.8	0.68	0.5404	- 0.2	1547	1672
9	86.9	75.3	11.6	17,864	72.3	0.65	0.5436	13.1	1540	1741
10	86.8	77.3	9.5	15,020	74.9	0.54	0.5396	10.8	1581	1825
11	86.9	78.9	8.0	13,952	76.5	0.54	0.5021	11.9	1744	1907
12	86.3	80.3	6.0	12,078	78.9	0.44	0.5118	- 5.0	2013	2041
13	87.0	82.9	4.1	9,597	81.5	0.34	0.4975	2.2	2365	2245

TABLE B 4

DATA for the CONDENSATION OF STEAM-TOLUENE MIXTURES ON AN OXIDISED COPPER TUBE

Run No.	T_s °C	T_w °C	ΔT_F °C	q_c W/m^2	t_{IN} °C	Δt °C	F kg/s	Ht. Bal. %	COND. COMP. % W/W Toluene	h W/m^2 °C	h_L (Tol.) W/m^2 °C	T_V (IN) °C
1	84.4	62.7	21.7	73,938	54.0	1.54	0.8727	4.7	83.0	3407	1545	97.9
2	84.5	60.1	24.4	95,187	50.1	1.75	0.9595	1.5	78.8	3901	1499	96.4
3	84.6	60.2	24.4	94,201	49.6	1.74	0.9304	- 1.3	79.5	3861	1499	96.3
4	84.9	65.5	19.4	81,061	56.8	1.45	0.9579	- 1.6	80.1	4178	1590	96.8
5	84.5	62.0	22.5	95,627	51.7	1.67	0.9834	- 1.7	80.1	4250	1530	98.3
6	84.5	62.5	22.0	95,960	52.5	1.60	0.9832	- 5.5	80.1	4362	1539	98.9
7	83.6	65.7	17.9	76,517	57.8	1.44	0.9722	5.9	79.8	4275	1625	97.5
8	83.7	70.9	12.8	60,490	65.5	1.04	0.9684	- 3.5	78.7	4726	1770	99.9
9	84.4	75.0	9.3	46,956	71.2	0.73	0.9630	- 12.4	79.5	5049	1917	98.5
10	84.1	75.2	8.9	45,221	71.6	0.70	0.9608	- 12.9	81.5	5081	1943	100.6
11	84.1	76.4	7.7	41,241	72.8	0.82	0.7459	- 13.5	81.9	5356	2014	98.7
12	83.9	72.2	11.7	52,007	65.8	1.45	0.6496	4.8	80.7	4445	1810	98.3
13	84.2	73.6	10.6	59,392	64.6	1.94	0.5078	- 4.9	78.2	5603	1862	99.3
14	84.3	74.1	10.2	58,069	65.2	1.90	0.5038	- 5.1	78.5	5693	1884	99.7
15	84.3	70.4	13.9	77,437	61.7	1.56	0.8226	- 5.0	78.0	5571	1732	99.2
16	84.4	71.1	13.3	75,783	62.6	1.56	0.8319	- 1.5	78.3	5698	1753	99.3
17	84.3	68.8	15.5	80,135	59.7	1.64	0.8848	4.2	80.8	5170	1685	100.0
18	84.3	69.3	15.0	80,100	60.3	1.62	0.8758	1.7	80.0	5340	1700	100.4
19	84.3	59.8	24.5	99,274	48.0	1.94	0.9093	2.4	83.3	4052	1497	99.2
20	84.3	60.9	23.4	96,080	49.5	1.92	0.9112	4.8	84.7	4106	1516	100.8
21	84.4	64.2	20.2	97,546	52.5	2.02	0.9040	7.3	80.4	4829	1573	100.8
22	84.4	49.1	35.3	123,479	33.4	2.32	1.0111	9.3	80.9	3498	1359	95.2
23	84.3	51.5	32.8	120,999	36.3	2.33	1.0106	12.1	80.6	3689	1386	96.7
24	84.2	50.9	33.3	108,525	37.6	2.10	0.9941	10.6	82.4	3259	1380	98.8
25	83.9	51.8	32.1	114,982	37.4	2.25	0.9801	10.1	80.8	3582	1394	96.4
26	83.8	54.9	28.9	117,623	40.5	2.30	0.9796	9.8	80.6	4070	1433	97.3
27	84.0	56.9	27.1	114,498	42.7	2.27	0.9791	11.7	81.2	4225	1458	97.4
28	84.7	74.8	9.9	57,915	67.3	1.51	0.6695	0.0	78.9	5850	1887	99.4
29	84.7	76.4	8.3	50,597	69.9	1.31	0.6690	- 0.6	80.7	6096	1971	99.9
30	84.8	77.3	7.5	41,273	71.9	1.14	0.6333	0.3	82.3	5503	2021	102.7
31	84.8	79.9	4.9	30,899	76.3	0.79	0.6324	- 7.4	84.0	6306	2249	104.7
32	84.8	79.8	5.0	32,280	76.1	0.85	0.6324	- 4.9	83.8	6456	2236	104.5
33	84.7	80.0	4.7	26,461	76.6	0.82	0.6293	10.7	83.3	5630	2272	103.1
34	84.7	80.6	4.1	27,081	77.5	0.77	0.6352	3.6	82.6	6605	2357	102.9
35	84.9	80.8	4.1	26,613	77.7	0.79	0.6229	7.1	81.0	6491	2359	101.9
36	85.0	81.2	3.8	26,463	78.3	0.77	0.6258	4.3	79.8	6964	2401	102.4
37	84.6	80.2	4.4	24,996	77.2	0.65	0.6679	0.6	84.2	5681	2322	101.7

TABLE B 5

DATA for the CONDENSATION OF STEAM-TOLUENE MIXTURES ON A HORIZONTAL GOLD PLATED COPPER TUBE

Run No.	T_s °C	T_W °C	ΔT_F °C	q_c W/m^2	t_{IN} °C	Δ t °C	F kg/s	Ht.Bal. %	Cond. comp. % Tol.	h W/m^2 °C	$h_{L(Tol)}$ W/m^2 °C	$T_{v(IN)}$ °C
1	84.4	53.3	31.1	87,515	41.0	1.88	0.8379	3.74	79.1	2814	1406	98.2
2	84.4	55.4	29.0	78,677	44.0	1.76	0.8374	7.93	80.8	2713	1432	99.4
3	84.6	58.7	25.9	71,510	49.1	1.62	0.8596	11.63	83.5	2761	1475	100.6
4	84.6	61.4	23.2	63,800	52.7	1.46	0.8615	13.26	84.0	2750	1518	101.3
5	84.6	64.6	20.0	57,280	57.5	1.23	0.8606	6.49	84.7	2864	1579	102.4
6	84.7	66.0	18.7	58,007	57.8	1.41	0.8015	11.86	83.3	3102	1605	98.3
7	84.7	68.2	16.5	50,672	61.4	1.23	0.8008	11.88	84.1	3071	1659	100.7
8	84.9	70.3	14.6	46,311	64.4	1.10	0.7953	8.39	83.9	3172	1711	101.1
9	84.9	69.0	15.9	53,297	61.5	1.50	0.6851	10.64	83.2	3352	1674	99.2
10	85.1	73.4	11.7	38,470	68.4	1.10	0.6096	12.99	84.5	3288	1809	101.7
11	84.9	75.4	9.5	31,008	71.3	0.88	0.6862	11.77	86.0	3264	1908	102.5
12	84.6	74.4	10.2	39,505	59.2	1.13	0.6085	0.52	81.5	3873	1875	99.7
13	84.6	76.7	7.9	31,131	72.2	0.97	0.5238	12.03	83.0	3947	1998	100.8
14	84.8	78.6	6.2	26,537	75.1	0.77	0.6295	4.63	82.3	4280	2123	101.3
15	84.6	59.3	25.3	72,434	43.5	1.87	0.7760	15.16	83.7	2863	1484	100.0
16	84.8	69.9	14.9	52,970	53.3	1.19	0.7819	1.12	83.0	3555	1701	100.5

Table B 6

DATA for the CONDENSATION OF STEAM-TRICHLOROETHYLENE MIXTURES ON A HORIZONTAL OXIDISED COPPER TUBE

Run No.	T_s ^oC	T_W ^oC	ΔT_f ^oC	q_c W/m^2	t_{IN} ^oC	Δt ^oC	F kg/s	Ht.Bal. %	Cond.comp. % $^w/_w$ Trich	h $W/m^2 {}^oC$	$h_{L(Trich)}$ $W/m^2 {}^oC$	$T_{v(IN)}$ ^oC
1	74.9	69.7	5.2	26,572	65.5	0.88	0.5302	1.5	92.5	5110	2182	90.0
2	73.5	59.0	14.5	41,862	52.3	1.24	0.6482	10.8	94.0	2887	1703	74.2
3	73.5	63.1	10.4	40,248	58.2	0.88	0.9024	13.1	93.9	3870	1844	74.2
4	73.5	65.1	8.4	35.624	60.6	0.80	0.9083	-	93.7	4241	1942	74.7
5	73.3	62.6	10.7	52,762	54.4	1.47	0.6718	7.7	93.1	4931	1832	83.2
6	73.3	64.7	8.6	43,757	57.8	1.28	0.6713	12.4	92.9	5088	1931	83.3
7	73.4	48.3	25.1	70,682	37.3	1.80	0.6827	- 0.1	93.4	2816	1492	75.4
8	73.4	51.7	21.7	69,115	41.2	1.74	0.6822	- 1.4	93.0	3185	1545	76.1
9	73.2	58.0	15.2	49,155	49.7	1.60	0.5501	3.2	94.0	3232	1683	73.8
10	73.1	61.2	11.9	43,087	54.0	1.44	0.5460	4.5	93.8	3609	1785	73.9
11	73.2	62.5	10.7	41,161	55.1	1.45	0.5459	10.3	93.7	3585	1834	74.2
12	73.5	64.9	8.6	41,902	57.9	1.41	0.5492	6.2	93.8	4884	1932	76.0
13	73.6	67.9	5.7	32,692	62.6	1.06	0.5450	1.5	94.1	5705	2133	75.1
14	73.6	69.1	4.5	27,447	64.3	0.94	0.5375	5.5	95.0	6127	2266	74.8
15	73.6	56.9	16.7	45,539	48.6	1.52	0.5502	5.5	93.9	2720	1643	74.7
16	73.6	61.0	12.6	48,969	51.7	1.81	0.5204	10.6	93.5	3880	1759	77.3
17	73.6	54.6	19.0	57,539	44.6	1.77	0.5475	- 3.0	93.5	3024	1593	78.1
18	73.5	57.3	16.2	57,316	47.8	1.79	0.6371	14.3	93.4	3538	1658	89.7

TABLE B 7

DATA for the CONDENSATION OF STEAM-TRICHLOROETHYLENE MIXTURES ON A HORIZONTAL GOLD PLATED COPPER TUBE

Run No.	T_s °C	T_w °C	ΔT_f °C	q_c W/m^2	t_{IN} °C	Δt °C	F kg/s	Ht. Bal. %	Cond. Comp. % $^w/_w$ Trich	h W/m^2°C	h_L W/m^2°C	$T_{v(IN)}$ °C
1	73.3	56.1	17.2	40,403	49.9	1.20	0.6015	2.87	93.8	2349	1633	88.6
2	73.5	59.8	13.7	34,099	54.6	0.97	0.6009	- 1.49	92.5	2489	1714	89.9
3	73.4	63.1	10.6	26,670	59.2	0.78	0.5970	0.43	94.2	2516	1860	90.6
4	73.8	71.1	2.7	10.457	69.1	0.38	0.5296	11.69	94.5	3873	2606	91.9
5	74.4	67.3	7.1	23,629	63.2	0.85	0.4594	- 5.06	94.5	3328	2044	91.0
6	73.8	65.4	8.4	24,427	60.3	1.06	0.4597	14.53	93.9	2908	1967	91.1
7	73.9	65.2	8.7	24,272	60.3	0.85	0.5493	11.01	92.8	2803	1952	90.1
8	74.1	66.9	7.2	22,414	62.6	0.74	0.5490	4.81	94.0	3126	2044	90.7
9	73.8	67.9	5.9	28,074	64.1	0.65	0.5452	12.22	94.3	3048	2142	90.9
10	73.9	62.5	11.4	27,222	57.2	1.00	0.5363	12.87	94.0	2382	1825	88.6
11	73.8	67.5	8.3	21,762	61.5	0.76	0.5347	6.76	93.7	2616	1972	90.8
12	73.7	48.6	25.1	57,173	40.4	1.54	0.6891	6.96	92.9	2275	1513	91.4
13	73.5	52.9	20.6	50,931	45.0	1.46	0.6798	11.96	94.5	2470	1586	92.5
14	73.7	57.7	16.0	41,124	51.3	1.22	0.6730	14.46	93.6	2572	1684	92.5
15	73.2	59.6	13.6	38,896	52.6	1.12	0.6483	7.34	94.6	2860	1753	91.5

APPENDIX C

Determination of n for use in the standing drop model

From Chapter 5 section 5.2 the heat transfer coefficient predicted by the model is,

$$h = k\ f\ (\Theta)\ \alpha\ \left(\frac{n+3}{n+2}\right) \left[\frac{d_{max}^{n+2} - d_{min}^{n+2}}{d_{max}^{n+3} - d_{min}^{n+3}}\right] + (1-\alpha)\ h_{NF} \quad \text{(C1)}$$

The maximum drop diameter is 3.0 mm while the minimum diameter is $2\delta \sin\Theta$ $(1 - \cos\Theta)$, with $\Theta = 70^{\circ}$.

$$d_{min} = 2.8561\,\delta \quad \text{(C2)}$$

The fractional area occupied by the drops, within the expected dropsize range, has been reported by Bernhardt (1970) to be from 0.64 to 0.16, the time average, however, seems to be between 0.6 and 0.4. The value of $\alpha = 0.5$, therefore seems to be a reasonable guess with which to start calculating n for the steam-toluene and steam-trichloroethylene data obtained on the gold plated tube. This initial guess could be refined later but the uncertainties in the theory and the data do not make such refinements worthwhile at this stage.

From Fatica and Katz (1949), $f(\Theta)$, = 10.2 when $\Theta = 70^{\circ}$. To further simply the calculations only one value of k, the drop (water) thermal conductivity will be used, that is $k = 0.655$ W/m°C (at 60°C). Substituting the above values into (C1) gives,

$$h = 3.3405 \left[\frac{n+3}{n+2}\right] \left\{\frac{(3 \times 10^{-3})^{n+2} - d_{min}^{n+2}}{(3 \times 10^{-3})^{n+3} - d_{min}^{n+3}}\right\} + 0.5\ h_{NF} \quad \text{(C3)}$$

The mean value of the minimum drop diameter for the steam-toluene and steam trichloroethylene data obtained on the gold plated tube (calculated using (C2)) is 1.785 x 10^{-4}m. The value of n to give the best fit to the data is - 1.0 hence (C1) becomes in general,

$$h = 13.362\, \alpha \left[\frac{3 \times 10^{-3} - d_{min}}{9 \times 10^{-6} - d^2_{min}} \right] + (1 - \alpha) h_{NF} \qquad (C4)$$

APPENDIX D

Nucleation barriers in immiscible liquid condensation

When condensing vapour mixtures of immiscible liquids the possibility exists that if a continuous film of one liquid covers the surface the other condensing liquid may not be able to condense until a resistance to nucleation has been overcome.

The homogeneous rate of nucleation of a liquid from a supersaturated vapour is given by Hill et al (1963) as

$$I = \left(\frac{P_g}{k\,T_g}\right)^2 \left(\frac{M}{N_o\,\rho_L}\right) \left(\frac{2\,\sigma\,N_o}{\pi\,M}\right)^{\frac{1}{2}} \quad \exp\left(\frac{-\Delta G(r^*)}{k\,T_g}\right), \qquad \text{(D1)}$$

where I is the rate of nucleation, k Boltzmans constant, T_g the vapour temperature, M the molecular weight, N_o Avogadro's number, ρ_L the liquid drop density, σ the liquid drop surface tension and $\Delta G(r^*)$ the free energy required for nucleation of a drop. For the homogeneous case $\Delta G(r^*)$ is given by (Hill et al (1963)) as,

$$\Delta\ G(r^*) = \frac{4}{3}\,\sigma\,\pi\ {r^*}^2. \qquad \text{(D2)}$$

Turnbull (1950) has shown that the free energy required to form a drop which is part of a spherical segment is

$$\Delta\ G(r^*) = \frac{4}{3}\pi\sigma {r^*}^2 (2 + \cos\Theta)\ (1 - \cos\Theta)^2/4 \qquad \text{(D3)}$$

$$= \Delta G\ (r^*)_H C,$$

where $\Delta G(r^*)_H$ is defined by equation (D2), Θ is the contact angle and $C = (2 + \cos\Theta)\ (1 - \cos\Theta)^2/4$. Combining (D1) and (D3) gives

$$I = \left(\frac{P_g}{k\ T_g}\right)^2 \left(\frac{M}{N_o\ \rho_L}\right) \left(\frac{2\sigma N_o}{\pi\ M}\right)^{\frac{1}{2}} \exp\left(\frac{-4\,\sigma\pi {r^*}^2 C}{3\ k\ T_g}\right) \qquad \text{(D4)}$$

but $r^* = 2\sigma M/\rho_L RT\ln(P_g/P_\infty)$ (Hill et al. (1963) hence

$$I = \left(\frac{P_g}{kT_g}\right)^2 \left(\frac{M}{N_o\,\rho_L}\right) \left(\frac{2\sigma N_o}{\pi M}\right)^{\frac{1}{2}} \exp\left(\frac{-16\,\sigma^3 \pi C M^2}{3kT_g^3\,\rho_L^2\,R^2(\ln P_g/P_\infty)^2}\right) \qquad \text{(D5)}$$

The part of equation (D5) which is of greatest interest for this study is C, that is $(2 + \cos\theta)(1 - \cos\theta)^2/4$. Consider a steam-toluene mixture, if the water forms the continuous phase we must consider the condensation of toluene onto the water surface. The contact angle for most organics on water is zero or near zero, from equation (D3) this means the free energy for the formation of a drop is zero, or near zero, that is there is no barrier to the condensation of toluene onto the water surface.

However, if we consider steam condensing on toluene it is known that water will not spread on a toluene surface (or indeed on most organic liquids with which it is immiscible) and hence a finite contact angle between the two liquids will exist. Since there are no data available for the required contact angles a range of contact angles will be considered.

Consider the system shown in figure D1. The problem is to calculate T_i such that the nucleation rate I is significant. In this case P_g is the partial pressure of the steam in the gas phase (and thus also its vapour pressure), thus for a eutectic vapour mixture.

$P_g = 760\,(1-0.444) = 423$ mm Hg $\simeq 5.6394 \times 10^5$ dynes/cm^2, $T_g = 84.34^{\circ}C = 357.49K$,
$\rho_L = 0.9692$ gm/cm^3
(at $84.34^{\circ}C$), $N_o = 6.023 \times 10^{26}$ g mole^{-1} (Avogadros constant),

$k = 1.3808 \times 10^{-16}$ ergs/K (Boltzmans constant), $\sigma = 61.7$ ergs/cm^2

(at $84.34^{\circ}C$), $R = 8.314 \times 10^7$ ergs/g moleK(Gas constant) and $M = 18.0$.

From equation (D5), $I = 1.4591 \times 10^{26} \exp\left[-31.13\ C/(\ln x)^2\right]$, Collier (1972) gives a rate of 10^{11}–10^{16} as being a significant nucleation rate, thus taking the smaller value

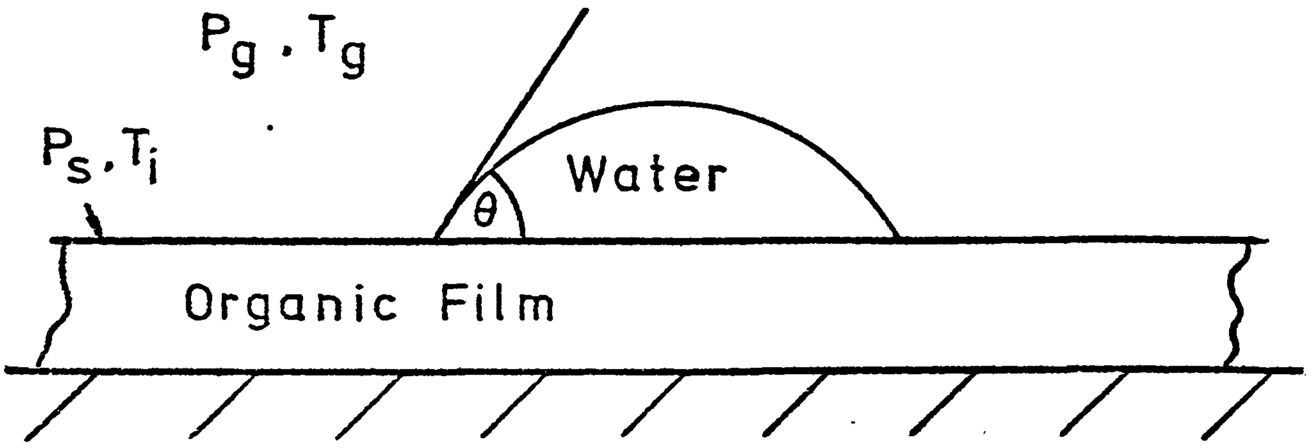

FIG. D1. THE NUCLEATION OF A WATER DROP ON AN ORGANIC FILM.

$\exp\left[-31.13\ C / (\ln x)^2\right] = 10^{11}/1.4591 \times 10^{26} = 6.8535 \times 10^{-16}$,

therefore $\ln x = (0.8915\ C)^{\frac{1}{2}}$. (D6)

As an example let $\theta = 30^{\circ}$ then $C = 0.01285$ and hence $x = 1.1130 = P_g/P_{\infty}$, thus $P_{\infty} = 380.0$ mm Hg that is the vapour pressure of the steam at the interface must be 380.0 mm Hg and $T_i = 81.60^{\circ}C$ before any significant nucleation takes place. The degree of vapour subcooling required is hence (84.34 – 81.60) 2.70°C. Figure D2 gives the degree of subcooling required for various values of the contact angle.

The major errors in this analysis are probably in the calculation of the pre-exponential term in equation (D5). However, since a variation of two to three orders of magnitude has little effect it is not as important as the exponential term.

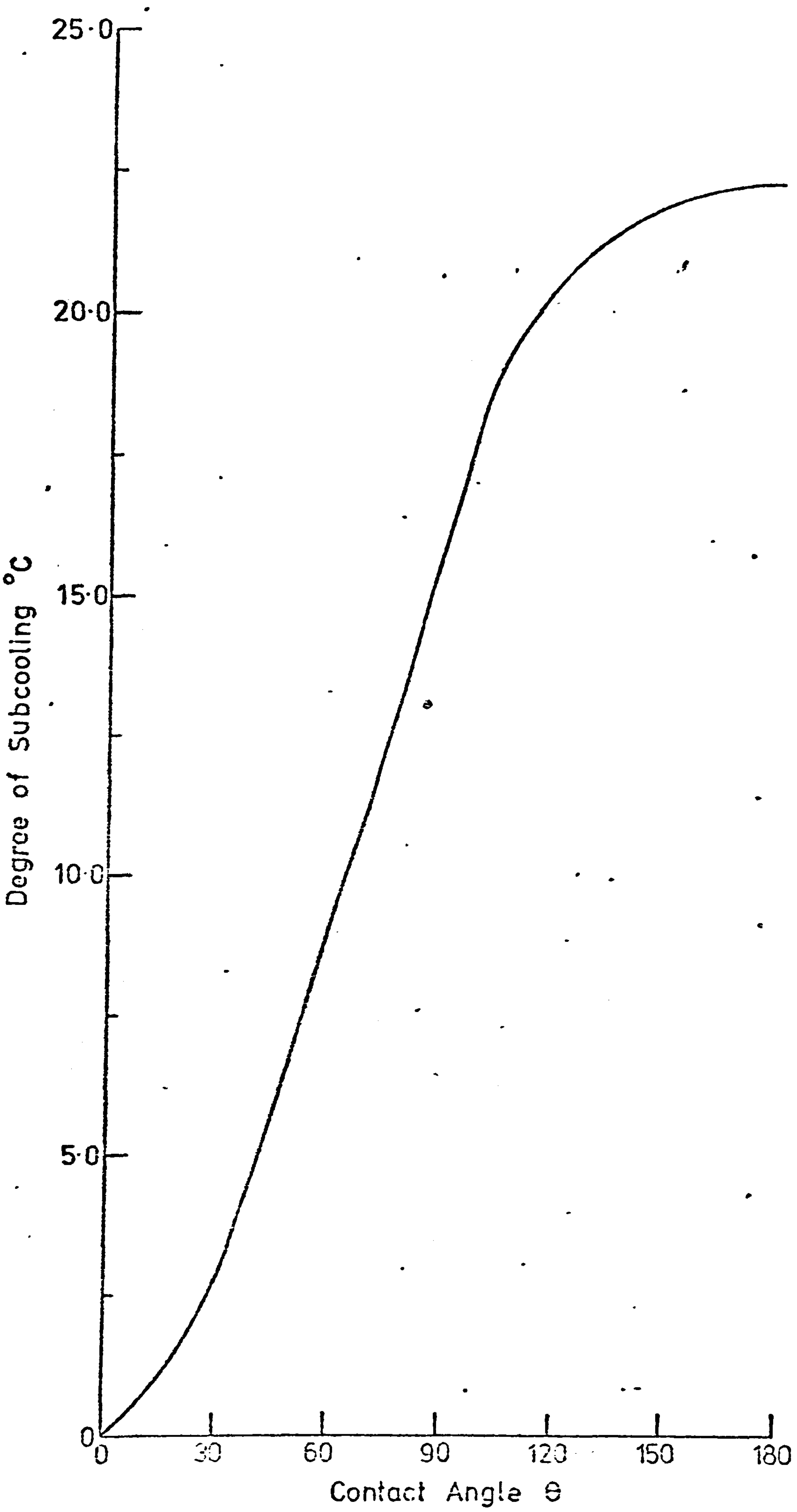

FIG.D2. PLOT OF CONTACT ANGLE Vs DEGREE OF SUBCOOLING FOR THE CONDENSATION OF A STEAM - TOLUENE EUTECTIC MIXTURE

APPENDIX E

The Effect of Variable Wall Temperature on the Laminar Film Condensation of a Pure Vapour

The following analysis is an attempt to determine the effects of a non-uniform wall temperature on the condensing film heat transfer coefficient. The assumptions made are those of Nusselt (see chapter 2 section 2.2), here however, the wall temperature is not constant but is given by the empirical correlation of Salov and Danilov (1975) i.e.,

$$T_w = \bar{T}_w + C_1 \cos \phi \qquad \text{(E1)}$$

where $\bar{T}_w$ is the mean wall temperature, T_w the actual wall temperature at angle ϕ and C_1 a constant.

From equation (E1) it is apparent that

$$\Delta T_f = \Delta \bar{T}_f \left(1 - \frac{C_1}{\Delta \bar{T}_f} \cos \phi\right) \qquad \text{(E2)}$$

putting $\omega = C_1 / \Delta \bar{T}_f$ then equation (E2) becomes

$$\Delta T_f = \Delta \bar{T}_f \left(1 - \omega \cos \phi \right). \qquad \text{(E3)}$$

The coordinate system used is shown in figure E1. The momentum equation can be reduced to

$$(\rho_L - \rho_V) g \sin \phi = - \mu_L \frac{d^2 u}{dy^2} \qquad \text{(E4)}$$

Integrating equation (E4) gives

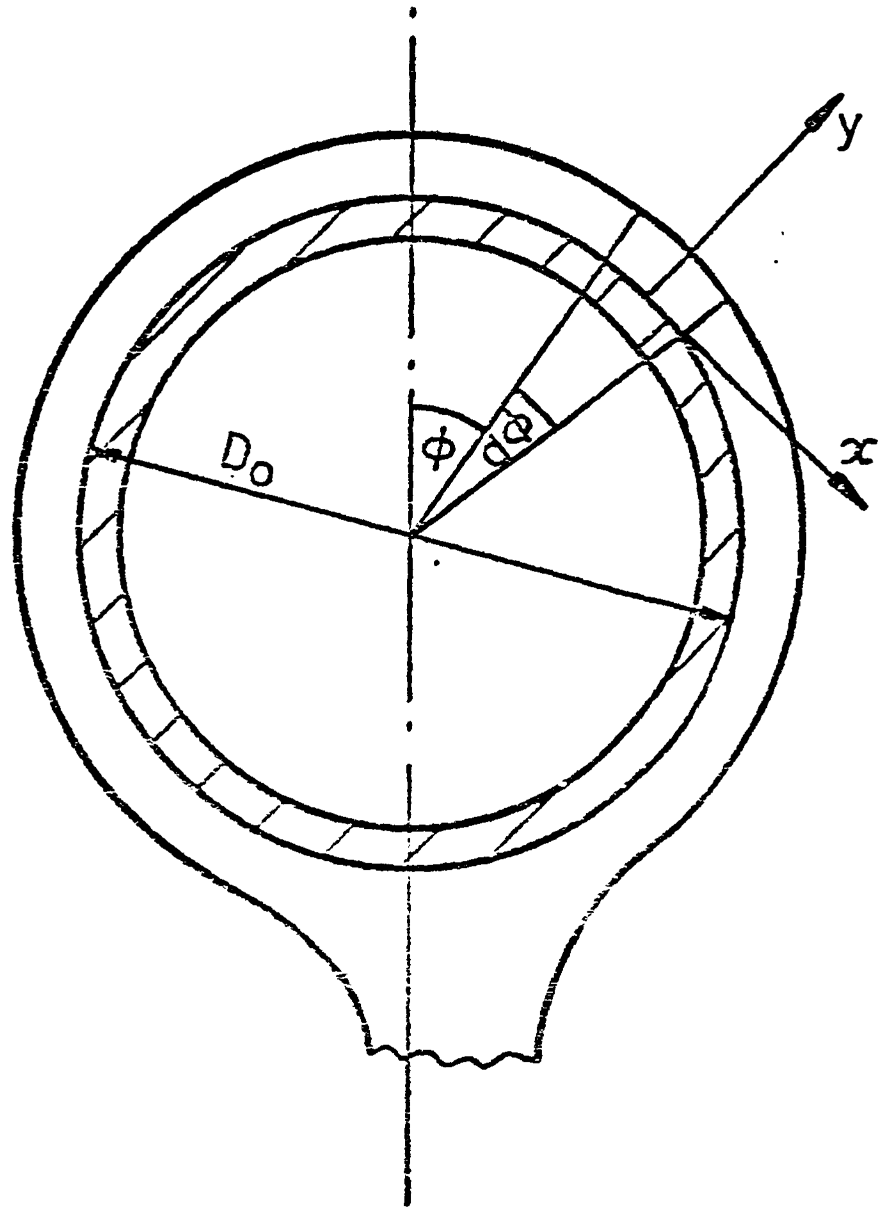

FIG. E1. CO-ORDINATE SYSTEM FOR THE NON ISOTHERMAL WALL ANALYSIS.

$$u = \frac{(\rho_L - \rho_V) g \sin \emptyset}{\mu_L} (y\delta - y^2/2) \qquad \text{(E5)}$$

where u is the velocity in the x direction. The mean film velocity is hence

$$\bar{u} = \frac{(\rho_L - \rho_V) g \sin \emptyset \, \delta^2}{3\mu} . \qquad \text{(E6)}$$

The heat transferred through unit width of the element d $\emptyset$ is given by

$$q = \lambda \, d\Gamma = \frac{k}{\delta} \Delta T_f \left(\frac{D_o}{2}\right) d\emptyset \qquad \text{(E7)}$$

where Γ the condensate flowrate per unit width of film is

$$\Gamma = \rho_L \, \bar{u} \, \delta . \qquad \text{(E8)}$$

Combining equations (E6), (E8) and differentiating we get

$$d\Gamma = \frac{\rho_L(\rho_L - \rho_V) g}{3\mu} \, d(\delta^3 \sin \emptyset) . \qquad \text{(E9)}$$

Thus from equations (E3), (E7) and (E9) it is apparent that,

$$(1 - \omega \cos \emptyset) \, d\emptyset = K \{ \delta^4 \cos \emptyset \, d\emptyset + 3 \, \delta^3 \sin \emptyset \, d\delta \} \qquad \text{(E10)}$$

where $K = 2(\rho_L (\rho_L - \rho_V) g \; \lambda/(3kD_o \, \Delta\bar{T}_f \, \mu)$. .

Putting $K\,\delta^4 = Z$ and noting that $3\delta^3\,d\delta = \frac{3}{4}d\,(\delta^4)$ equation (E10) reduces to

$$\frac{dZ}{d\phi} + \frac{4\,Z\cot\phi}{3} + \frac{4\,\omega\cot\phi}{3} - \frac{4\,\mathrm{cosec}\,\phi}{3} = 0, \qquad \text{(E11)}$$

the solution of this ordinary differential equation being,

$$Z = \frac{4}{3\sin^{4/3}\phi}\int_0^{\phi}\left(\sin^{\frac{1}{3}}\phi - \omega\sin^{\frac{1}{3}}\phi\cos\phi\right)d\phi. \qquad \text{(E12)}$$

The condensate film heat transfer coefficient is hence given by

$$h = \frac{k}{\delta} = k\,K^{\frac{1}{4}}/Z^{\frac{1}{4}} \qquad \text{(E13)}$$

However, it is not the absolute value which is of interest but the value relative to the Nusselt prediction for an isothermal wall. Following the derivation above with the isothermal wall assumption gives

$$h_N = k\,K^{\frac{1}{4}}/Z_N^{\frac{1}{4}} \qquad \text{(E14)}$$

where

$$Z_N = \frac{4}{3\sin^{4/3}\phi}\int_0^{\phi}\sin^{\frac{1}{3}}\phi\,d\phi. \qquad \text{(E15)}$$

Combining (E12),E(13), (E14) and (E15) gives

$$\frac{h_T}{h_N} = \left[1 - \frac{\omega}{S}\right]^{-\frac{1}{4}}, \qquad \text{(E16)}$$

where

$$S = \frac{4}{3 \sin^{4/3} \phi} \int_0^{\phi} \sin^{\frac{1}{3}} \phi \, d\phi. \qquad \text{(E17)}$$

The ratio of the mean heat transfer coefficient is given by

$$\bar{h}_T/\bar{h}_N = \int_0^{\pi} h \, d\phi \Big/ \int_0^{\pi} h_N \, d\phi. \qquad \text{(E18)}$$

Using equations (E12), (E13), (E14), (E15), and (E18) it can be shown that,

$$\bar{h}_T/\bar{h}_N = \int_0^{\pi} d\phi/[s-\omega]^{\frac{1}{4}} \Big/ \left\{ \int_0^{\pi} d\phi/S^{\frac{1}{4}} \right\}. \qquad \text{(E19)}$$

Since Chaddock (1957) has shown that

$\frac{1}{\pi} \int_0^{\pi} d\phi/S^{\frac{1}{4}} = 0.805542$, equation (E17) becomes

$$\bar{h}_T/\bar{h}_N = 0.39515 \int_0^{\pi} d\phi \Big/ [s-\omega]^{\frac{1}{4}}. \qquad \text{(E20)}$$

Equations (E16) and (E20) have been solved numerically (using Simpsons rule) for various values of ω and ϕ, the results are shown in figures E2 and E3 respectively.

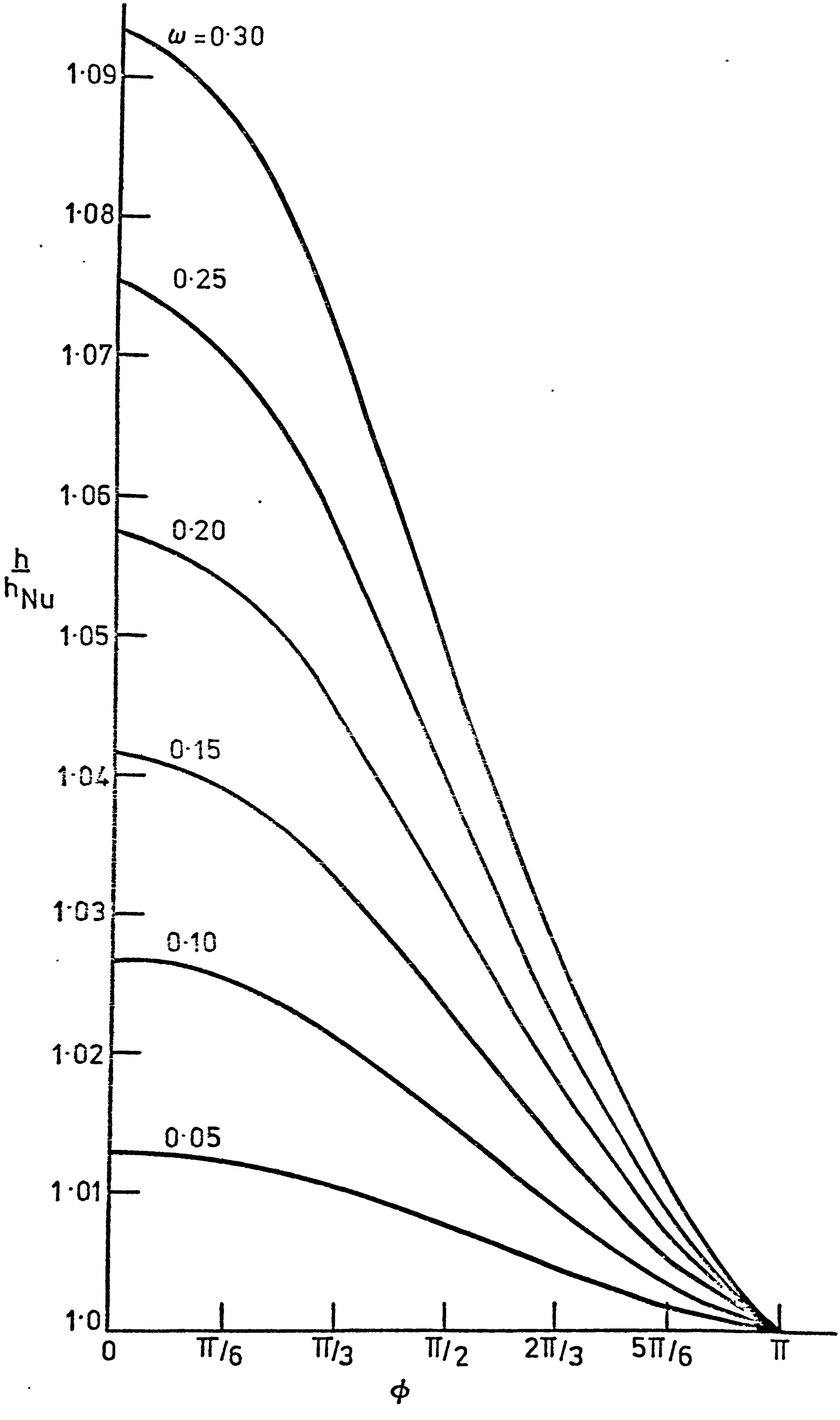

FIG. E2. PLOT OF h/h_{Nu} vs ϕ.

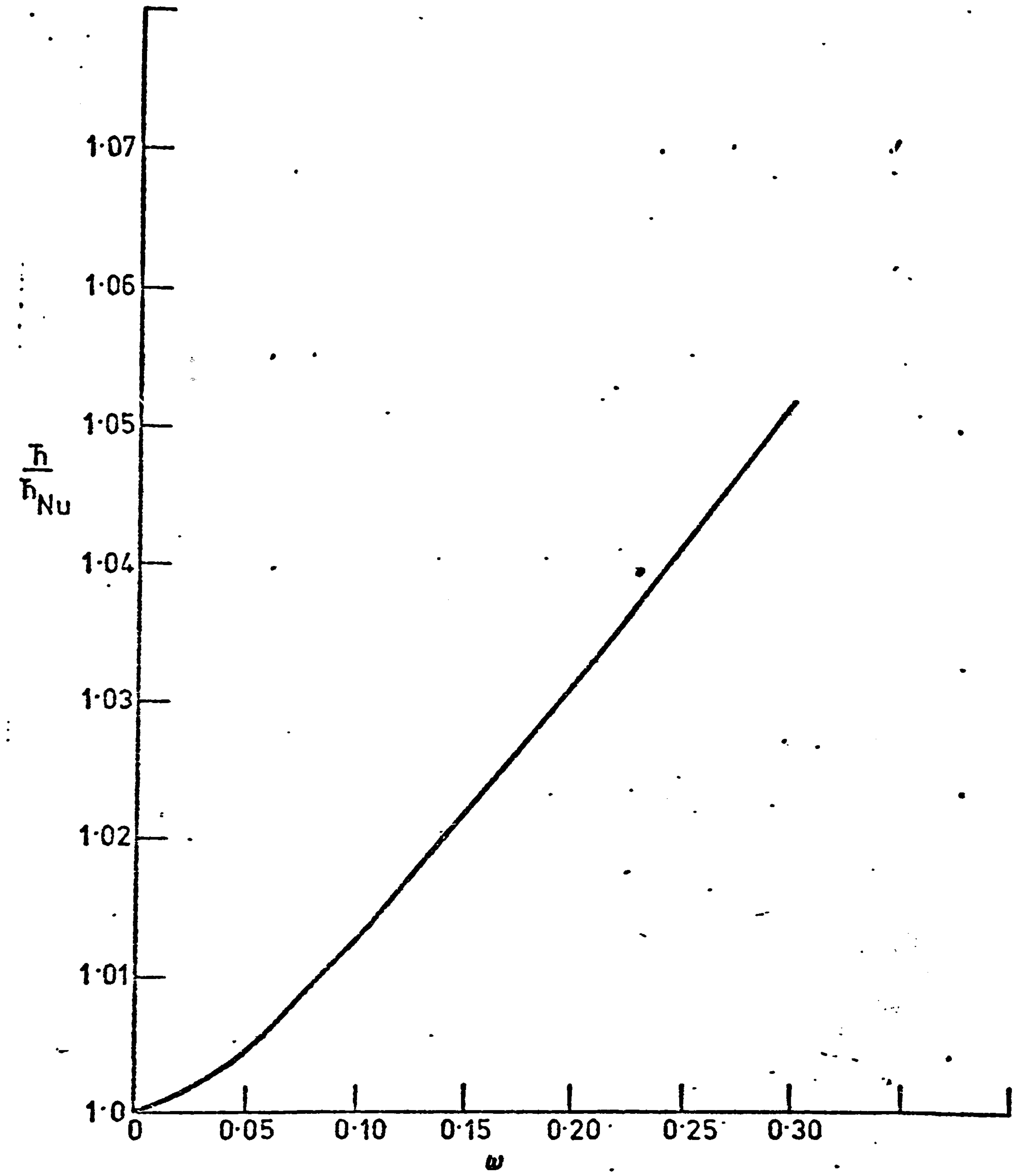

FIG. E3. PLOT OF $\bar{h}/\bar{h}_{Nu}$ vs ω.

APPENDIX F

Experiments on a Vertical Copper Surface

F.1 Introduction

At a late stage in the present study it was decided that useful information on detailed mechanisms might be obtained using a simple flat plate. The apparatus described below was designed primarily to give such information and at the same time provide quantitative heat transfer data.

Unfortunately it was only possible to do relatively few commissioning runs with this apparatus, however, some limited qualitative and quantitative data have been obtained.

The following sections will give a brief description of the apparatus and the results obtained.

F.2 Apparatus and procedure

The flow diagram of the apparatus is shown in Figure F1; essentially the equipment consists of the following items:

1. Boiler - this is simply a 2 litre round bottomed flask fitted with a reflux condenser. The heat required to boil the liquids is provided by a 2.0 KW isomantle.

2. Test section - this is described in more detail in section F.2.1.

3. Total condenser - this consists of a coil made up of several turns of 7 mm bore glass tube in a 75 mm i.d. glass jacket. The cooling water passes through the coil.

4. Constant temperature bath - this provides the cooling water, at a controlled temperature, to the test section.

5. Collection vessels - these are simple conical flasks fitted with ground glass stoppers to accomodate the glass feed lines.

The lines connecting the various items of equipment are of glass and P.T.F.E. tubing, all of the vapour lines being lagged with fibreglass.

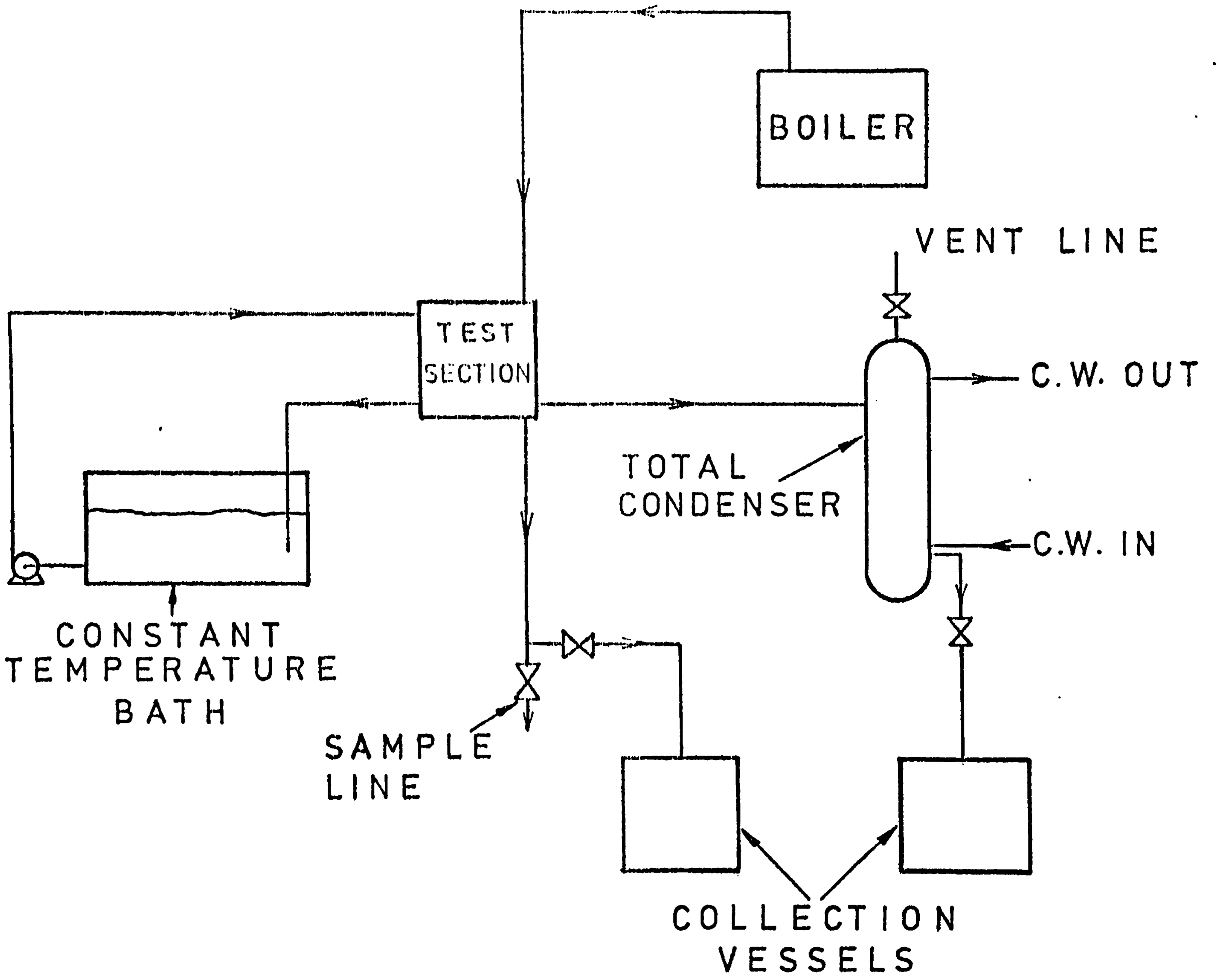

FIG. F1. BLOCK DIAGRAM OF THE FLAT PLATE RIG

F.2.1 Test section

Figure F2 shows the details of the test section, essentially this is a 90 x 60 x 50 mm copper block. The temperature distribution through the block is measured by four equally spaced stainless steel sheathed copper constantan thermocouples. The measuring junctions are located on the centre line of the block as shown in figure F2. These thermocouples were calibrated in situ by simply placing the block in a constant temperature bath, the bath temperature being determined by a National Physical Laboratory tested mercury in glass thermometer. The accuracy of calibration was estimated to be within $\pm$ 0.1°C.

The condensing surface can be viewed through a glass window, when running the apparatus this window was kept clear of condensate by shining a 500 W lamp onto the glass, this also provided the illumination for viewing.

F.2.2 Procedure

Before taking any readings the condensing face of the copper block was polished with successively finer grades of emery paper until uniformly smooth. It was then washed with acetone and distilled water before installation.

To commission the apparatus the procedure was as follows:

1. The cooling water supplies to the reflux and total condenser were turned on.

2. The cooling water to the block was turned on and the temperature adjusted to the required value.

3. The mixture to be boiled was placed in the boiler and the iso-mantle switched on.

4. When the system had reached steady state, that is when the block thermocuples were giving a constant output the following readings were taken:

1. Thermocouple readings

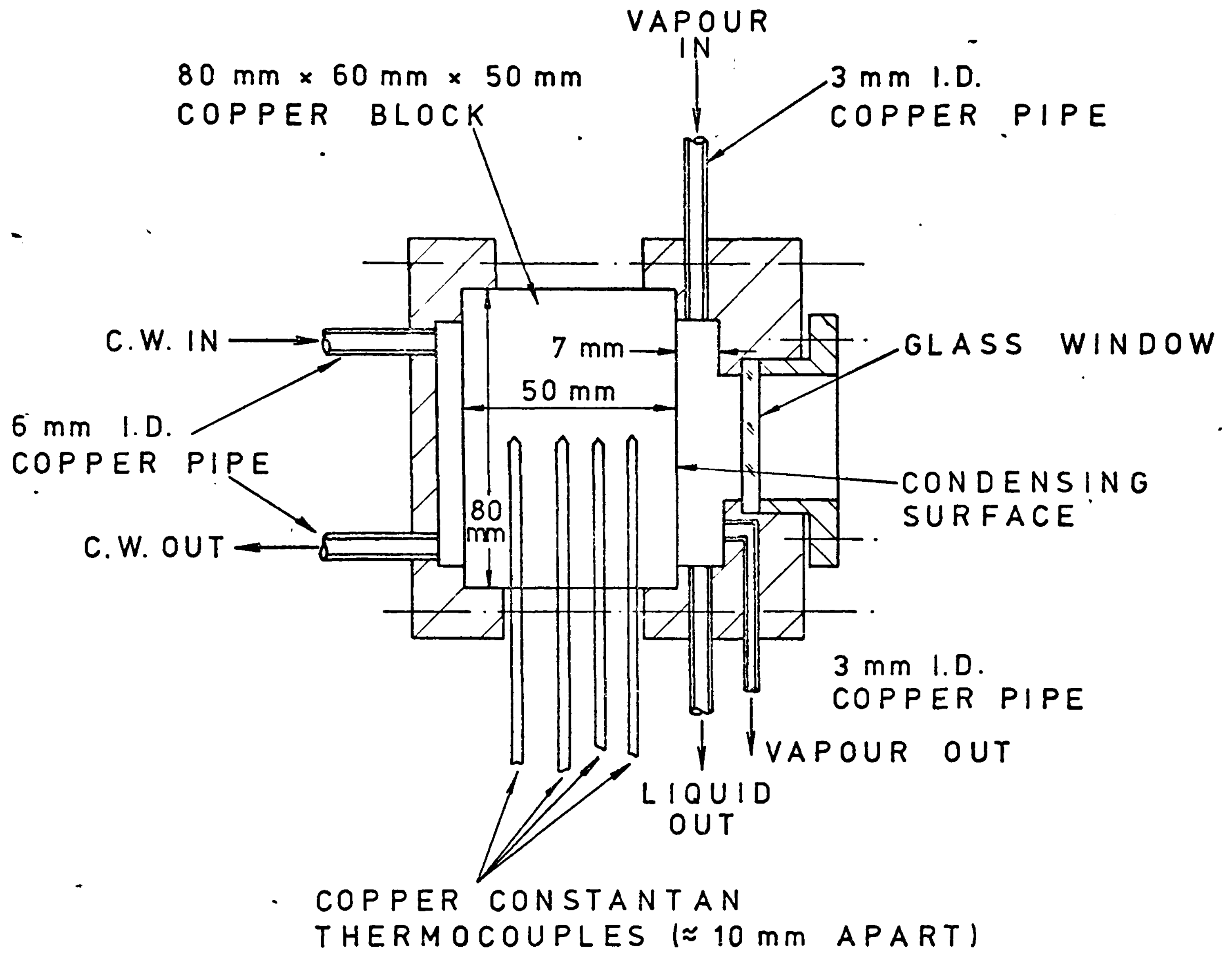

FIG. F2. DETAILS OF THE TEST SECTION

2. Condensate flowrates

3. Atmospheric pressure.

Note

The build up of incondensable gases was prevented by periodically venting from the total condenser.

F.3 Results and Discussions

The heat transfer results were calculated as follows. The heat flux through the block was determined from the measured temperature profile using

$$q = k \ \Delta T_1/1 \qquad \text{(F1)}$$

where k is the thermal conductivity of copper (389 W/m°C), 1 and ΔT_1 are the distance and temperature difference between the thermocouple nearest the condensing surface and the one furthest away. The condensing film heat transfer coefficient is then given by

$$h = q/ \ \Delta T_f \qquad \text{(F2)}$$

where $\Delta T_f = T_s - T_w$. T_s the saturation temperature was determined from the atmospheric pressure reading and T_w the wall temperature by extrapolation of the measured temperature profile.

The results obtained are shown in Fig. F3, as can be seen except for three points the heat transfer coefficient does not vary appreciably with temperature difference.

The condensation mechanism was as follows: the toluene formed a film while the water formed both standing drops and rivulets. The area occupied by the drops was substantially greater than the area occupied by the water rivulets Fig. F4. Small drops were observed on both the standing water drops and water rivulets.

F.3.1 Discussion

Since originally the reported results were only meant to be of a

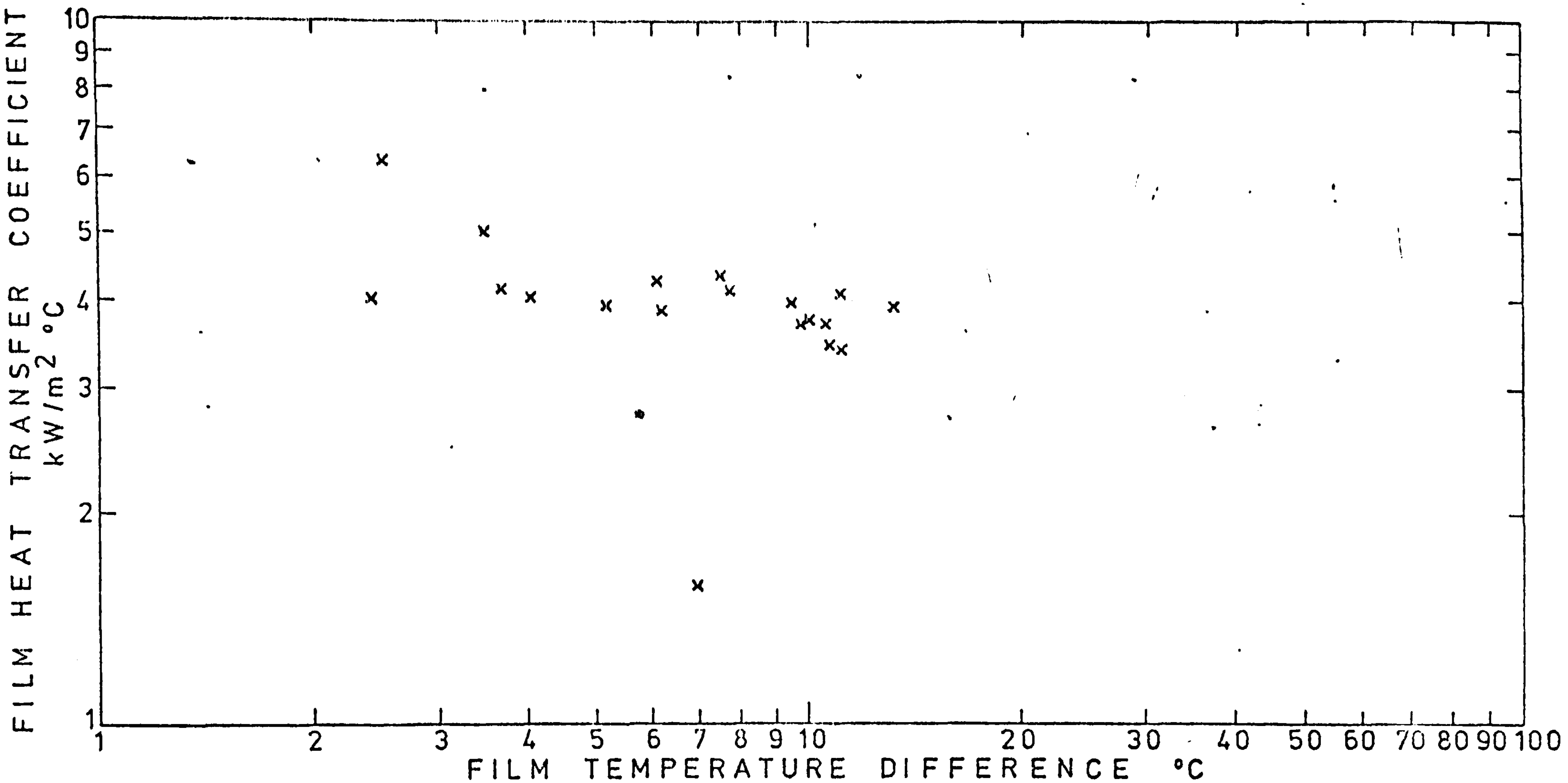

FIG. F3. FILM HEAT TRANSFER COEFFICIENTS FOR THE CONDENSATION OF STEAM-TOLUENE MIXTURES ON A VERTICAL COPPER BLOCK

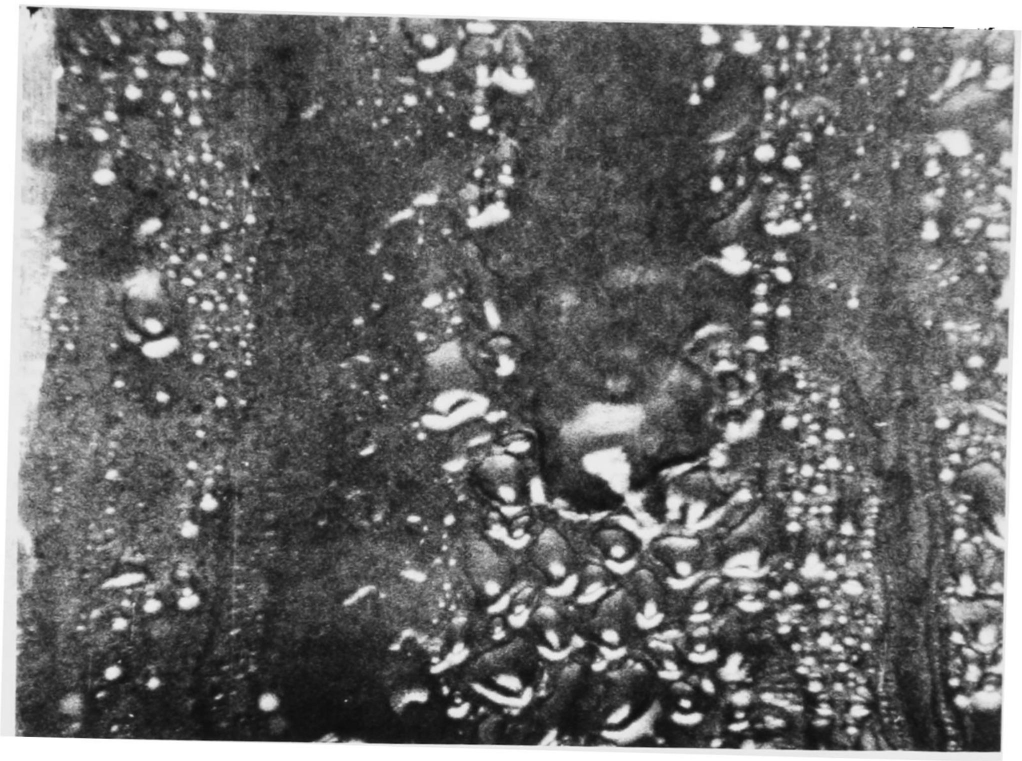

FIG. F4 FLOW PATTERN for the CONDENSATION of STEAM-TOLUENE MIXTURES ON A VERTICAL FLAT PLATE preliminary nature (i.e. commissioning runs) no attempt was made to uniformly oxidise the copper condensing surface before use. Thus the results have been obtained on a surface which has been changing with time.

APPENDIX G

Error Analysis

The film heat transfer coefficient is calculated using the equation

$$h = m\lambda / A\Delta T_f, \qquad (G1)$$

where m is the rate of condensation, λ the latent heat of vapourisation. A the heat transfer area and ΔT_f the film temperature difference.

The estimated accuracy in measuring the above variables is as follows:

Error on A – this value is assumed correct

Error on λ – the physical property correlations are accurate to ± 1.0%.

Error on m – the measurement of the rate of condensation is estimated to be accurate to within ± 4.0%.

Error on ΔT_f – this variable is determined from the expression $\Delta T_f = T_s - T_w$ and since T_s and T_w can be measured to within ± 0.2°C the error in calculating ΔT_f is ± 0.4°C. Thus if $\Delta T_f = 4.0°C$ the error is ± 10.0% but if $\Delta T_f = 40.0°C$ the error is only ± 1.0%.

From the above estimates of the errors it is apparent that the minimum theoretical errors in measuring the heat transfer coefficient are ± 15.0% if $\Delta T_f = 4.0°C$ and ± 6.0% when $T_f = 40.0°C$. Thus for the present measurements an accuracy limit of ± 15.0% on the heat transfer data is reasonable.

The heat balance is determined from the expression $Q_{CW}/Q_{COND}-1$, where Q_{CW} is the heat gained by the cooling water and Q_{COND} is the heat lost by the condensate.

The condensate heat load is determined from $Q_{COND} = m\lambda$. Thus from the above discussion it is apparent that Q_{COND} can be measured to within ± 5.0%.

The cooling water heat load is obtained from $Q_{CW} = F\,\Delta t\,C_p$. The estimated maximum error in measuring F is ± 5.0%, while the correlation for determining C_p is better than ± 1.0%. Again the error in determining Δt depends on its magnitude, the absolute error is ± 0.05°C.

From the above it is apparent that if $\Delta t = 0.5°C$ the error in measuring the heat balance will be ± 21.0% (5.0 + 1.0 + 10.0 - 5.0) and if $\Delta t = 1.0°C$ the error is ± 17.0% (5.0 + 1.0 x 5.0 + 5.0).

The limit set on the rejection of data because of bad heat balances was set at ± 15.0%, and the cooling water temperature difference maintained above 0.5°C were possible.