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1. Introduction

In practice, heat is transferred to a boiling liquid from the heated walls or the tube bundles in a boiler. The heat source in both cases may be a fluid, electricity or nuclear energy.

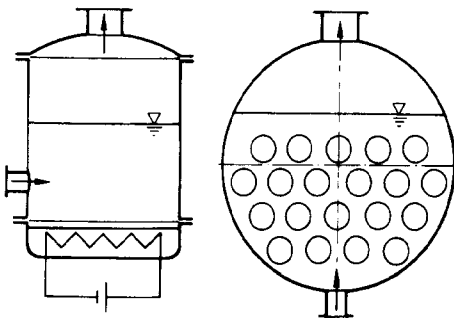


Fig. 1. Evaporator in which the heated surfaces are flat walls, tube bundles, or assemblies of metal rods

If the temperature T_w of an element immersed in a liquid is raised by the application of heat to a value above the saturation temperature T_{sat} of the liquid, the relationship between the heat flux $\dot{q} = \dot{Q}/A$ and the temperature difference $\Delta T = T_w - T_{sat}$ will be similar to that shown in Figs. 2a and 2b [1; 2]. Fig. 2a applies for a moderate reduced pressure $p^* = p/p_c$ (p_c = pressure at the critical point); and Fig. 2b, for a reduced pressure close to the critical point. The parameter in both diagrams is the surface coefficient of heat transfer α , which is defined by

$$\dot{q} = \alpha \Delta T. \quad (1)$$

It can be seen from Fig. 2a that, if the temperature difference or superheat is small, the increase in heat flux \dot{q} is somewhat more than directly proportional to the rise in temperature ΔT (dash-dotted line; free convection without bubble formation). At a given temperature difference, which depends on the saturation pressure, vapour bubbles commence to form at the heated surface (point A in Fig. 2a), and \dot{q} increases with ΔT to a

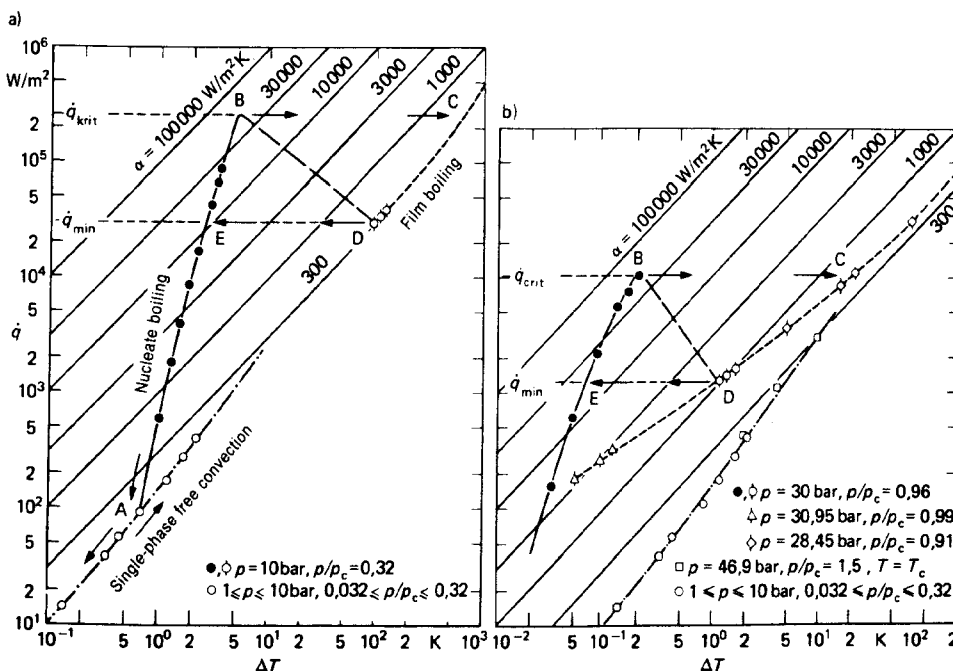


Fig. 2a and 2b. Heat flux \dot{q} as a function of the temperature difference ΔT in nucleate and film boiling and single-phase free-convection boiling. The boiling liquid is Refrigerant R 115 (C_2F_3Cl), and the heated surface is a horizontal copper tube [2].

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much greater extent than it does in free convection without bubble formation (full-line curves with solid dots in Figs 2a und 2b).

The pronounced increase in \dot{q} with ΔT is characteristic of heat transfer during nucleate boiling. Vapour bubbles are formed at gas or vapour filled cavities on technically rough heating surfaces. They reduce local overheating of the liquid; and when they are detached from the wall, cooler liquid flows in to take their place in the overheated boundary layer [3–6]. If the superheat is slight, only a few of the cavities participate in nucleation; but, as it increases, a greater number of sites in the heated surface are activated. This can be explained qualitatively by the fact that the liquid surrounding a growing bubble nucleus must be overheated to effect a rise in pressure, which is approximately given by

$$\Delta p \sim \frac{\sigma}{r} \quad (2).$$

Afterwards, as ΔT becomes greater, more and more cavities with a smaller radius of curvature r at the liquid interface are activated. As a consequence, the heated surface is gradually covered by a growing number of active nucleation sites.

In analogy, the number of active nucleation sites also increases with a rise in the saturation pressure, because the surface tension σ then decreases and the slope of the vapour pressure curve $(dp/dT)_{\text{sat}}$ becomes greater. An example is given by the full-line curves with solid dots in both diagrams shown in Fig. 2, which presents numerical values that are typical for many organic liquids. It can thus be seen that, if the pressure is raised from 32% to 96% of the critical value, the temperature difference required to effect a given heat flux is reduced to approximately one-tenth of its original value.

Nucleate boiling allows heat to be transferred at comparatively high rates by applying small temperature gradients. However, there is a limit to the extent that the heat flux can thus be increased. The temperature difference ΔT at which this maximum heat flux \dot{q}_{crit} (point B in Fig. 2) occurs depends on the saturation pressure. In this state of very intense nucleate boiling, a coherent film of vapour commences to form on parts of the heated surface. The further behaviour of the system consisting of the boiling liquid and the heated wall depends on the nature of the heat source, as is described below.

(a) Heating by a fluid

If the value of ΔT associated with \dot{q}_{crit} is continuously increased by raising the inlet temperature of the heating fluid, more points will exist at which a coherent, insulating film of vapour can be formed for some length of time. As a result, the rate of heat transfer will be reduced, although the average temperature (with respect to time and location) of the heated wall has risen. This range of partial film boiling extends to point D, which is referred to as the minimum heat flux \dot{q}_{min} or the Leidenfrost point. At this point, the heated surface

is completely covered by a film of vapour, and total film boiling commences. The heat flux thus increases again – at first slowly and then at a rate that is more than directly proportional to ΔT . This implies that the heat transfer coefficient also rises, mainly as a result of the growing turbulence in the film of vapour and the gradual increase in heat transfer by radiation.

(b) Heating by electricity or nuclear energy

In this form of heating, the heat flux \dot{q} applied to the heated surface is largely independent of the wall temperature and thus of ΔT . If its value is higher than \dot{q}_{crit} , it can no longer be completely absorbed by the boiling liquid, and the wall temperature rapidly increases. A film of vapour will then cover the heated surface, and the temperature difference will rise to a value corresponding to point C in Figs. 2a and 2b. In each diagram, C represents the first stable operating point for the transition to film boiling at a surface to which a heat flux is applied and at both the pressures shown in the diagrams, i.e. 10 bar and 30 bar. The overheating that thus ensues, and depends on the saturation pressure and the nature of the liquid may lead to destruction or burnout of the heated wall. A reduction in heat flux to a value less than that at point C gives rise to stable operating points in the range of total film boiling (dashed line between C and D). If the heat flux falls below the minimum value \dot{q}_{min} for film boiling, a transition takes place in the \overline{DE} direction into the range of nucleate boiling. It is analogous to that in the \overline{BC} direction.

Details on these relationships are given in fundamental studies by Stephan [7; 8] and in other publications [9–13; 159–161].

It is worth noting that the heat transfer coefficient and the maximum heat flux in nucleate boiling and the minimum heat flux in film boiling depend considerably on pressure, whereas heat transfer in total film boiling and in free convection without nucleation is only slightly influenced by the pressure (and thus the temperature) in the saturated state. The only exception arises in the vicinity of the critical point, owing to the wide range within which some properties vary, even with slight changes in temperature and pressure.

2. Convective boiling

Transient or local convective boiling without bubble formation at the heated surfaces may occur in evaporators that run at low heat fluxes or at small temperature gradients, e.g. in refrigeration. The case does not differ from that of heat transfer in single-phase free convection, as is evident from a comparison of results obtained for a single-phase liquid in the supercritical state with those presented for a boiling liquid in Fig. 2b. Thus the following applies for laminar free convection:

$$\text{Nu} = 0.60(\text{Gr Pr})^{1/4} \quad (3).$$

And the following applies for the turbulent boundary layer at the heated surface:

$$Nu = 0.15 (Gr Pr)^{1/3} \quad (4)$$

The Prandtl number in each case is about 2–100. The characteristic dimension to insert in the Nusselt Nu and Grashof Gr numbers is the *Krischer* length of flow path [14] – or simply the diameter if the body concerned is a horizontal cylinder. If the product of the Grashof and Prandtl numbers is 2×10^7 , the two relationships yield identical values for the Nusselt number. In other words, the transition from turbulent to laminar flow in the boundary layer probably lies in the $10^7 < Gr Pr < 10^8$ range. This agrees with the shape of the well-known mean *Krischer* curve [14] shown in Fig. 3. The slope of the curve at $Gr Pr = 10^6$ is $1/4$ for laminar flow; and at $Gr Pr = 10^9$, it is $1/3$ for turbulent flow¹.

The relationships concerned are illustrated in Figs. 4a and 4b. The liquids that have been chosen as examples for plotting the curves are various refrigerants (halocarbons). They represent a group that, up to now, has been investigated in greatest detail in studies on pool boiling. Fig. 4a is a log-log plot of measured values [2] for the heat transfer coefficient α against the heat flux \dot{q} . The temperature difference ΔT has been included as a parameter in order to give an idea of its relationship to \dot{q} . The measured values that have been plotted are valid for pressures between 0.3 % and 30 % of the critical value. The data for Refrigerant 113

($C_2F_3Cl_3$) reveal that the saturation pressure can be varied by a factor of 20 without transgressing the range of experimental scattering. A fact for which allowance must be made in interpreting the diagram is that the results plotted embrace heating by plain horizontal tubes and by copper tubes with external fins of various geometries (fins of 1.5–3.5 mm height and at 1.3–3.8 mm spacing). The heat transfer coefficient α and the heat flux \dot{q} for the finned tubes have been expressed in terms of the area of their total outer surfaces.

The values measured for each refrigerant on plain and finned tubes can be interpolated to form a common linear characteristic. Consequently, heat transfer by free convection without nucleation depends primarily on the outer surfaces of the tubes, and the other characteristics of the fin geometry are of subordinate importance.

It is evident from Fig. 4b that the results can be represented by the relationships between the Nusselt, Grashof and Prandtl numbers expressed by Eqns (3) and (4) and that the range of scattering involved is $\pm 20\%$. In addition to the measurements on halocarbon refrigerants, the diagram includes the best straight lines drawn through points determined in experiments performed by *Jakob* and *Linke* [16] on vertical and horizontal surfaces with water and tetrachloromethane at atmospheric pressure. It can be seen that these linear relationships lie within the range of scattering for halocarbon refrigerants.

On the whole, therefore, Eqns (3) and (4) permit the determination of heat transfer during pool boiling of liquids *without nucleation*. They are also valid for single-phase free convection for any given fluid in evaporators of conventional geometry within the range of Prandtl numbers quoted [15].

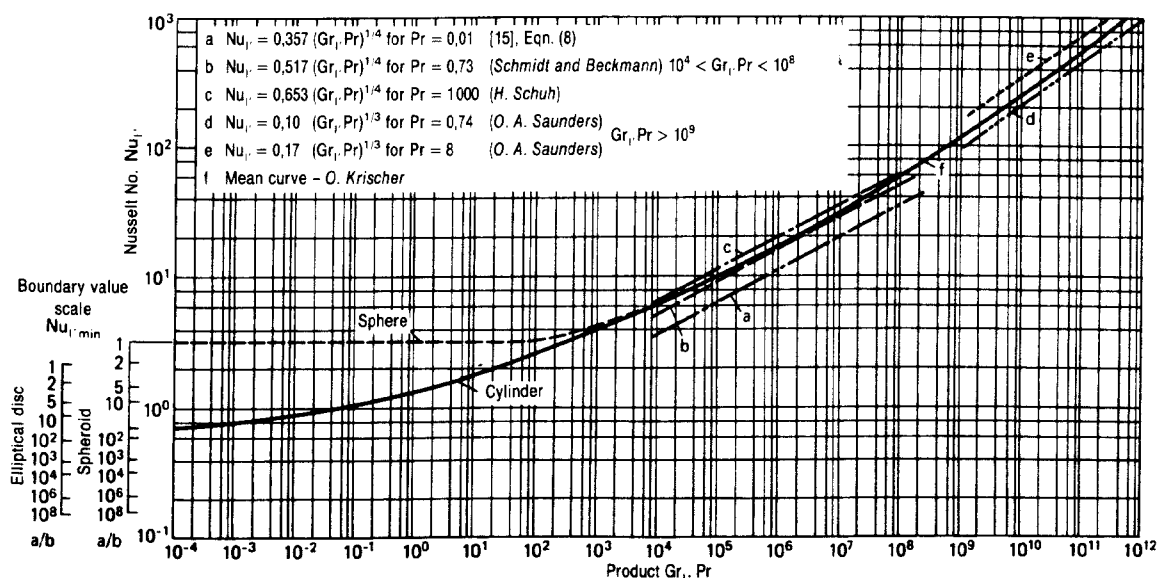


Fig. 3. Experimental and theoretical results in free-convection heat transfer around bodies of various geometries and Prandtl numbers [15]

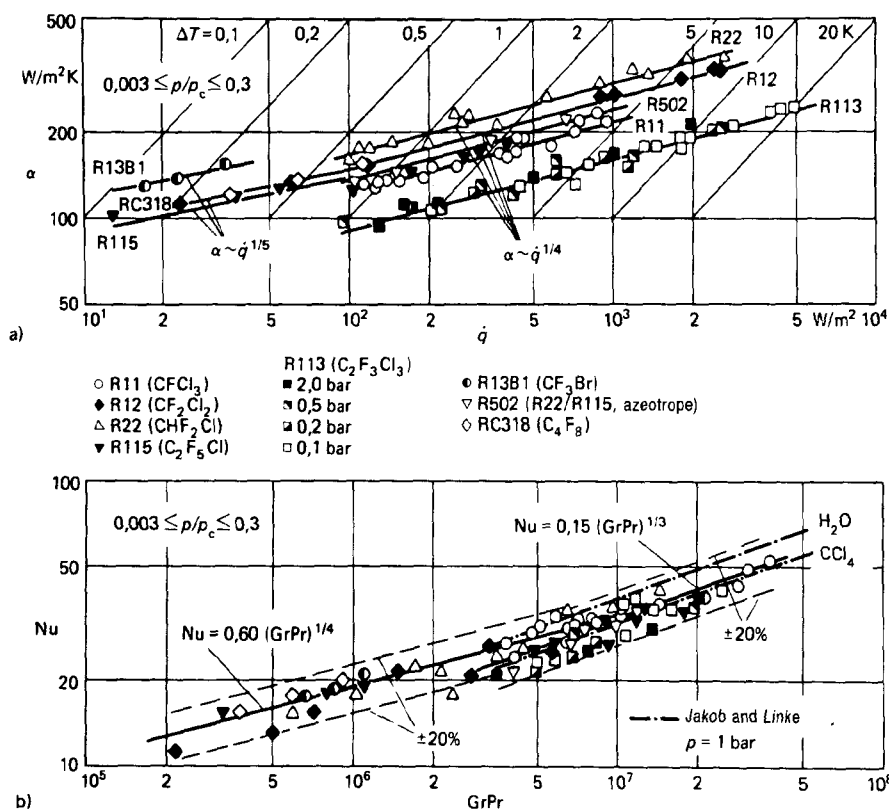


Fig. 4.

Pool boiling heat transfer without bubble formation from plain and finned horizontal tubes to various halocarbon refrigerants at pressures between 0.3% and 30% of the value at the critical point [2]

(a) heat transfer coefficient as a function of the heat flux with the superheat ΔT of the tube surfaces as parameter;

(b) Nusselt number as a function of the product of the Grashof and Prandtl numbers. The error limits of $\pm 20\%$ are entered as dashed lines. Dot-dashed lines for water and carbon tetrachloride according to Jakob and Linke [16].

3. Nucleate boiling

3.1 Nucleate boiling of pure substances

Despite the large number of treatises that have appeared on the subject in the literature [17–19; 159], no coherent theory yet exists that would allow heat transfer coefficients during nucleate pool boiling to be predicted to within the accuracy required in engineering.

Thus, in the light of the current state of the art, only empirical or semiempirical correlations can be applied in practice. Their development has proved that it is expedient to devote separate treatment to the main groups of variables, i.e. the properties of the liquids, the nature of the heated surfaces, and the operating parameters (viz. the heat flux \dot{q} and the pressure p). By this means, a relationship can be established for a reduced heat transfer coefficient α/α_0 that is analogous to an equation for α that is frequently resorted to in the literature, i.e.

$$\frac{\alpha}{\alpha_0} = C_w F(p^*) \left(\frac{\dot{q}}{\dot{q}_0} \right)^n \quad (5)$$

The right-hand side of this equation states the *relative* effect of the properties of the heated surface (which are embraced by the term C_w), the saturation pressure and the heat flux in relation to any given state freely selected as a reference. The properties of the liquid at the reference pressure p_0^* are contained in the reference value α_0 ; in other words, α_0 must be determined by experiment or calculated independently of Eqn (5).

Reference values chosen for the calculation presented below are an average heat flux of $\dot{q} = 20\,000$ W/m² and the reduced pressure $p_0^* = p_0/p_c = 0.1$. This pressure lies within the range of values, viz. 2–6 bar, that has been the subject of frequent experimental studies on most of the organic liquids that are of interest in engineering (cf. pp. Dc 1–Dc 5). The reference adopted for the heated surface is a metal with an arithmetic mean roughness height, as defined in ISO 4287/1:1984, of $R_{ao} = 0.4$ μ m (cf. Fig. 12). The reference state with the relevant heat transfer coefficient α_0 is illustrated in Fig. 5, in which the example chosen is Refrigerant R 12.

Effect of heat flux and saturation pressure

The term $(\dot{q}/\dot{q}_0)^n$ in Eqn (5) expresses the relationship between the heat transfer coefficient α and the heat flux \dot{q} and is referred to a given pressure (Fig. 5). The exact form of the relationship depends on the size distribution of the active nucleation sites, i.e. on the roughness pattern of the heated surface [21–24; 130; 132; 162]. It can be approximately represented by the exponential term in Eqn (5), but the index n , and thus the slope of the curves in Fig. 5, decrease steadily with a rise in pressure, i.e.

$$n = n(p^*) \quad (6)$$

This is evident from the upper parts of Figs. 6–12, which were plotted from experimental data for many organic and cryogenic liquids and water. In the large

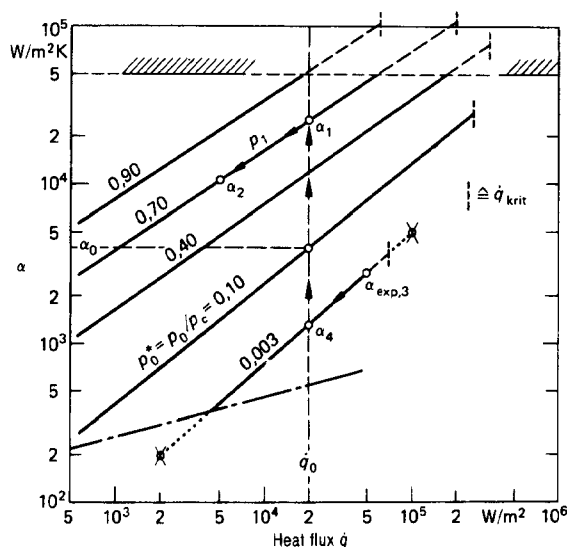


Fig. 5. Conversion of measured or calculated heat transfer coefficients into the corresponding values at other pressures and heat fluxes taking Refrigerant R 12 as an example.

Dash-and-dotted line: free convection without nucleation; vertical dashed lines: maximum heat flux \dot{q}_{crit} in nucleate boiling. The arrows indicate how an experimentally determined value $\alpha_{exp,3}$ for the heat transfer coefficient can be converted into the corresponding value α_2 at a different pressure and at a different heat flux.

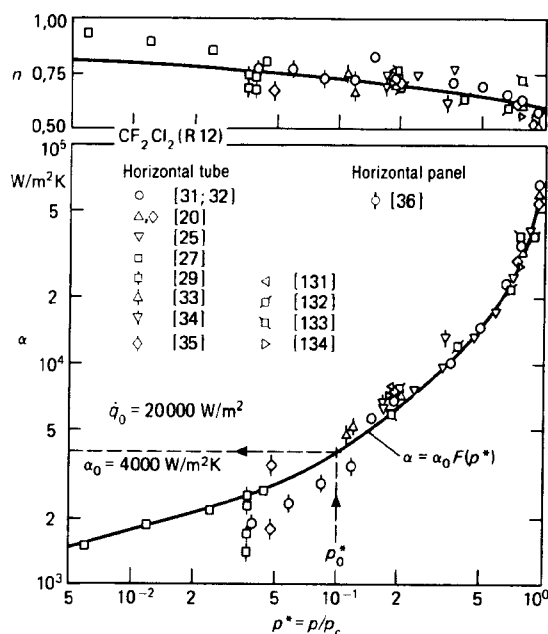


Fig. 6. Heat transfer coefficient α and the index n in the term containing the heat flux as functions of the reduced pressure p^* for Refrigerant R 12. Example for the determination of $\alpha_{0,exp}$ as listed in Table 1, from measured values.

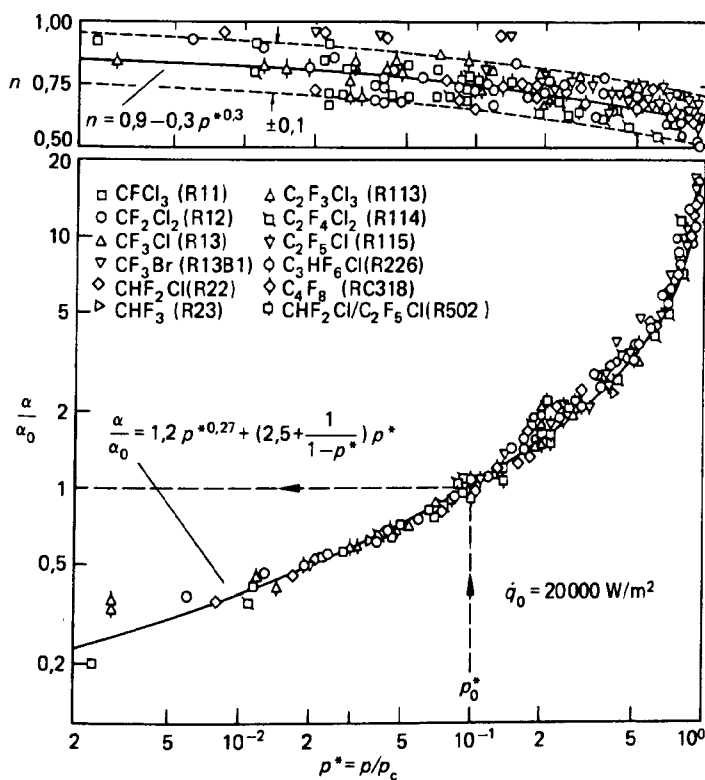


Fig. 7. Relative relationship to pressure of the heat transfer coefficient in the nucleate boiling of halocarbons and the relationship to pressure of the index n in the term containing the heat flux [3; 20; 25; 27; 31–41; 131–135]. The curve through the points in the upper diagram fits Eqn (7a); and the curve in the lower diagram, Eqn (8a).

majority of cases, the heated surfaces in the experiments were horizontal tubes². The question on whether

2) The experimental data chosen to plot the curves shown in Figs. 6–11 were derived from the literature. Only measurements at pressures of up to 90% of the relevant critical pressure were taken into consideration, because scattering increases considerably at higher pressures and nucleate boiling in this range is of no engineering significance at present.

it is permissible to apply the results of measurements on thin wires is dealt with in the discussion on Fig. 13.

The curves obtained over a wide range of pressures for the fluids investigated, with the exception of water and helium, fit the following equation:

$$n = 0.9 - 0.3 p^{*0.3} \quad (7a),$$

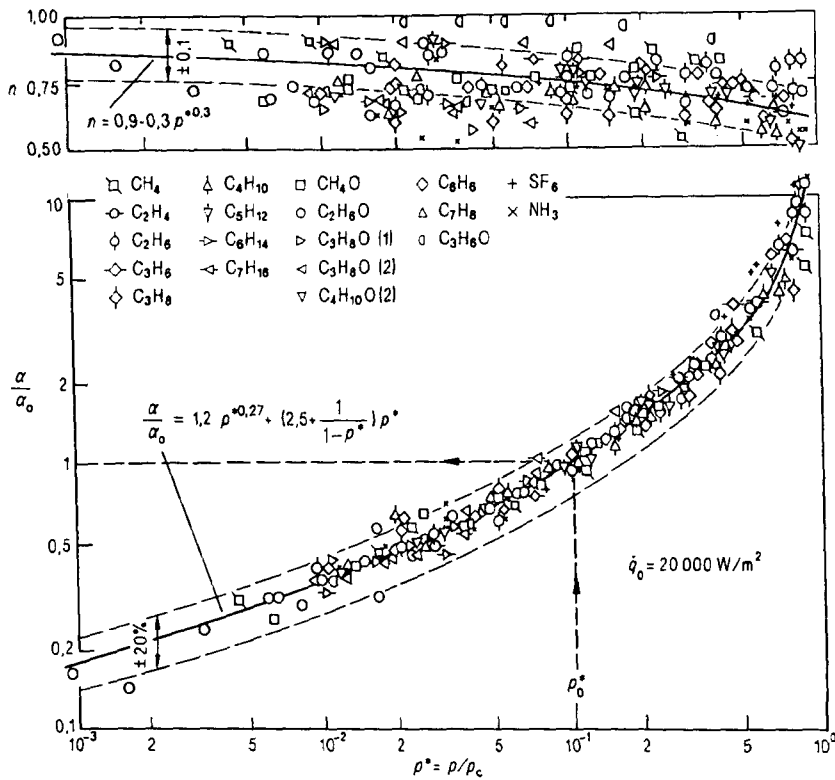


Fig. 8. Relative relationship to pressure of the heat transfer coefficient in the nucleate boiling of organic liquids, ammonia, and sulfur hexafluoride and the relationship to pressure of the index n in the term containing the heat flux [27; 28; 42–61; 134; 165; 166]. The curve through the points in the upper diagram conforms to Eqn (7a); and the curve in the lower diagram, to Eqn (8a).

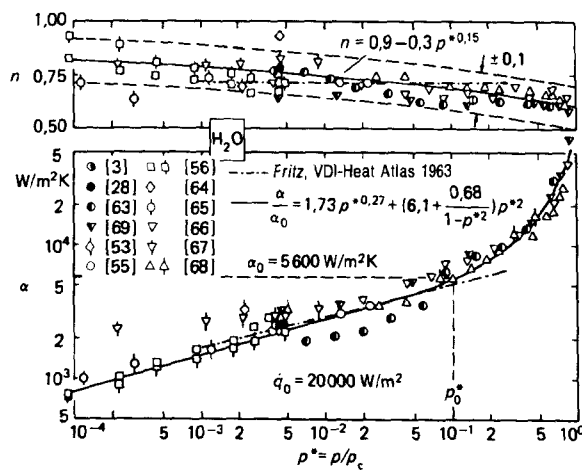


Fig. 9. Heat transfer coefficient α and the index n in the term containing the heat flux as functions of the reduced pressure p^* for water [3; 28; 53, 55–57; 63–69]. The curve through the points in the upper diagram fits Eqn (7b); and the curve in the lower diagram, Eqn (8b).

where $n = 0.75$ if $p^* = p_0^* = 0.1$, and $n = 0.62$ if $p^* = 0.8$.

Eqn (7a) was derived from measurements on halocarbon refrigerants (cf. Fig. 7), because these substances constituted the most frequently studied group in the past. It has also been verified by fresh measurements on nitrogen and other cryogenic liquids [131; 163–165]. Recent measurements on hydrocarbons (and halocarbon refrigerants) heated by horizontal tubes rubbed with emery led, on an average, to

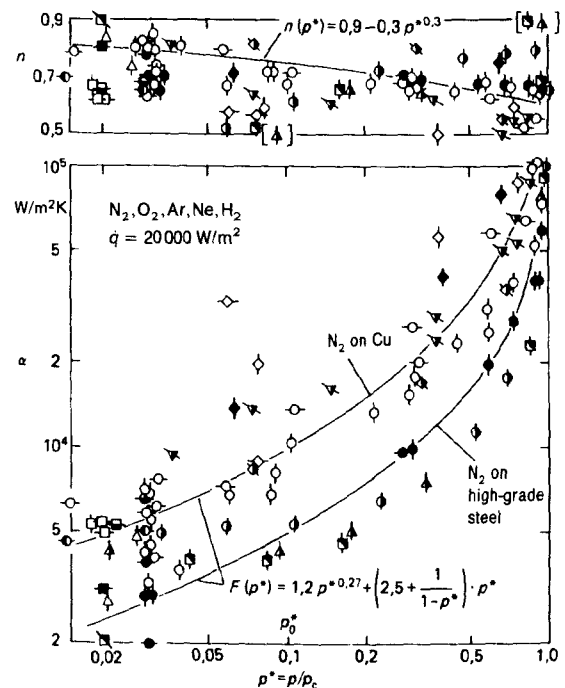


Fig. 10. Heat transfer coefficient α and the index n in the term containing the heat flux as functions of the reduced pressure p^* for cryogenic liquids [44; 47; 48; 50; 70–74; 131; 136; 140–151]. The curve through the points in the upper diagram fits Eqn (7a); and the curve in the lower diagram, Eqn (8a).

N ₂	O ₂	Ar	Ne	H ₂	Horizontal tube	—
Cu	\square	\triangle	∇	\diamond	Horizontal plate	
St	\bullet	\square	\triangle	∇	Wire	\
Pt	\circ	\square	\triangle	∇		
Al	\bullet					
Au	\circ	\square				

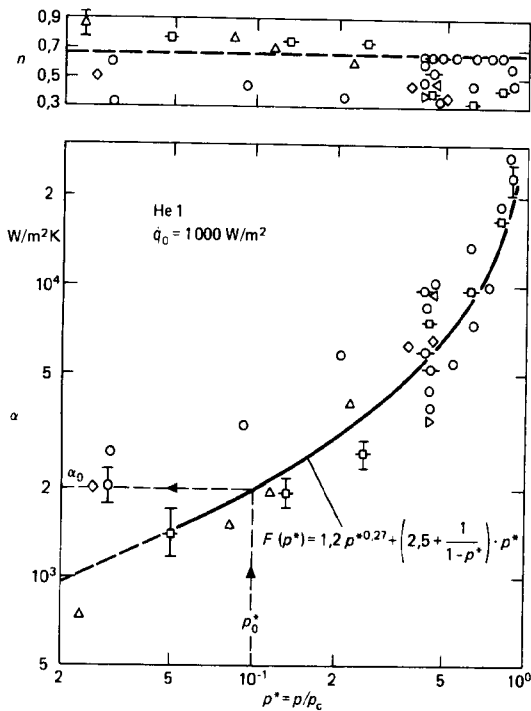


Fig. 11. Heat transfer coefficient α and the index n in the term for the heat flux as functions of the reference pressure p^* for helium [47; 74–77; 152–158]. The curve through the points in the lower diagram fits Eqn (8a).

Tube	Plate
Cu ○	○
Pt ◇	◇
St □	□
Al	△
	△ (Al 9.6)
	▷ (Al 9.5)

numerical values of n that were somewhat higher than those obtained from Eqn (7a) [134; 165–167]. No average relationship between the index n and p^* has yet been determined for helium beyond the limits of experimental scattering (cf. Fig. 11).

The $n(p^*)$ relationship for water is less pronounced, i.e.

$$n = 0.9 - 0.3 p^{*0.15} \quad (7b)$$

where $n = 0.69$ if $p^* = p_0^* = 0.1$ and $n = 0.61$ if $p^* = 0.8$.

Although Eqn (7b) has been derived from more recent measurements, it also embraces a previous suggestion by Fritz (cf. upper part of Fig. 9).

If the values determined for α exceed ca. $50\,000 \text{ W/m}^2 \text{ K}$, it can usually be assumed that the system is operating close to the maximum heat flux in nucleate boiling (Fig. 5). Within this operating range, the figure calculated from Eqn (7a) or (7b) for the exponent n may be too high, and the heat transfer coefficient obtained by extrapolation of the linear characteristics in Fig. 5 to \dot{q}_{crit} may well be too large. If an evaporator is to be designed to operate in this range, it would be advisable to verify the calculations by experiment.

If a known value of α at the one pressure has to be converted into the corresponding value at another pressure, the relative relationship of α to pressure, which is represented by the function $F(p^*)$ in Eqn (5), must be given. It is evident from Figs. 6–11 that F can be obtained empirically as a function of the reduced pressure p^* . Thus, if $\dot{q} = 20\,000 \text{ W/m}^2$, further development of correlations in the literature [25–27] allow the following to be derived from Figs. 6, 7, 8, 10 and 11:

$$F(p^*) = 1.2 p^{*0.27} + \left(2.5 + \frac{1}{1 - p^*}\right) p^* \quad (8a).$$

A slightly different relationship can be obtained for water from Fig. 9, i.e.

$$F(p^*) = 1.73 p^{*0.27} + \left(6.1 + \frac{0.68}{1 - p^*}\right) p^{*2} \quad (8b).$$

In each case, the first term in the sum governs the relationship at low pressures, and the sudden increase in α when the critical pressure is approached is largely described by the last term. It is evident from the lower diagram in Fig. 9 that the absolute value for α and the relative pressure relationship, as determined for water by the Fritz method, practically agree with those obtained by the new proposed method within the average pressure range previously investigated.

The upper limit for the validity of Eqn (8) is given by $p^* = 0.9$, because measured values for $p^* > 0.9$ are scanty and are also unreliable owing to the slight overheating of the surfaces at these pressures. Another reason is that pressures of $p^* > 0.9$ are of little interest in practice. The lower limit recommended is an absolute pressure of $p = 0.1$ bar, because nucleation at even lower pressures differs drastically from that at higher pressures; for instance, bubbles with diameters larger than 1 cm at the instant of detachment are often formed.

Effect of the properties of the heated surface

The quantitative effect exerted by the heated wall and the texture of its surface on the heat transfer coefficient over a wide range of pressures in nucleate boiling is still largely unknown (cf. [132; 134; 162]). Consequently a method of general validity for the determination of the variable C_w in Eqn (5) cannot be submitted. A relationship that was found by Stephan [3] in 1963 is

$$\alpha \sim R_p^{0.133} \quad (9a).$$

However, it can be applied only if the pressure at the boiling point is close to atmospheric and the heated surfaces concerned have been machined in the same manner, e.g. by drawing or turning.

The roughness parameter R_p , as defined in the former German standard DIN 4762/1 : 1960, was chosen by Stephan, because it can be related to the volume of vapour trapped in the roughness cavities of the heating surface. This relationship is not implied in the new definition of R_p in ISO 4287/1 : 1984. The relationship between R_p in the former definition and the arithmetic mean roughness height R_a , as defined in ISO 4287/1 : 1984, is shown in Fig. 12, which was plotted from the results of comparison tests performed on metal

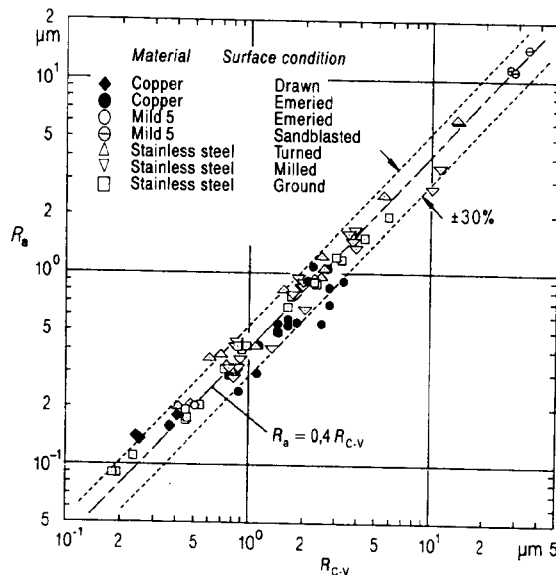


Fig. 12. Relationship between the former parameter R_p , as defined in DIN 4762/1:1960 to the arithmetic mean roughness height R_a , as defined in ISO 4287/1:1984, for metal surfaces finished by various means [168]

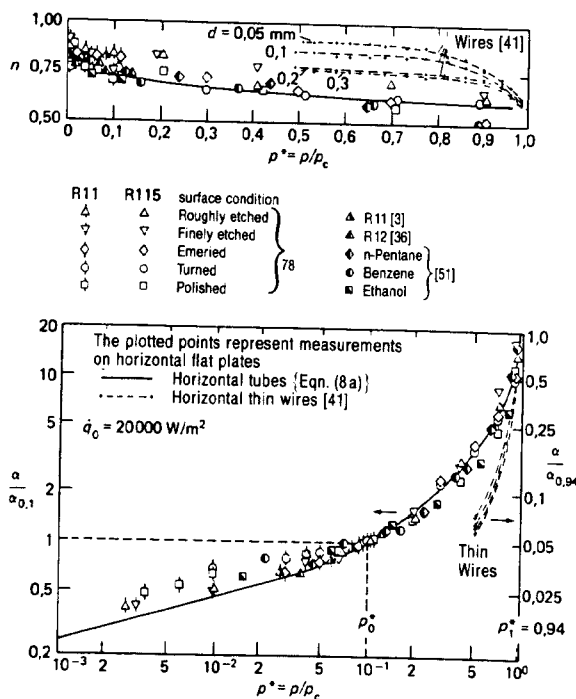


Fig. 13. Relative relationship to pressure of the heat transfer coefficient in nucleate boiling and the relationship to pressure of the index n in the term containing the heat flux for various organic liquids on horizontal tubes, plates and wires.

Ordinate on the left: tubes and plates

Ordinate on the right: wires

surfaces that had been finished in various ways. It can be represented roughly by the following equation:

$$R_a = 0.4 R_p \quad (9b)$$

The term C_w in Eqn (5) can thus be obtained from Eqns (9a) and (9b), i.e.

$$C_w = \left(\frac{R_a}{R_{a0}} \right)^{0.133} \quad (9c)$$

As was previously stated, the reference value to be applied in the calculation method is

$$R_{a0} = 0.4 \mu\text{m}.$$

Reference value α_0

The influence exerted by the properties of the liquid at the reference pressure³ is expressed in Eqn (5) by the collective term α_0 . Thus allowance can be made for the effects mentioned above by rearranging Eqn (5) to give

$$\frac{\alpha}{\alpha_0} = \left[\frac{\dot{q}}{\dot{q}_0} \right]^{n(p^*)} \cdot F(p^*) \left[\frac{R_a}{R_{a0}} \right]^{0.133} \quad (10)$$

Resort should be taken to any experimental data available on the liquid/wall system concerned or at least on the liquid in question. Empirical values for α_0 at $p_0^* = 0.1$ and $\dot{q}_0 = 20000 \text{ W/m}^2$ are listed in Table 1 for a number of liquids of industrial significance. In order to ensure a reliable estimate, the figures quoted for some of the liquids are somewhat lower than the experimental mean values.

The references quoted in the captions to Figs. 6–11 and from some other sources were resorted to for the evaluation. In most cases, the heated surfaces were single horizontal copper tubes. The average figure determined in recent measurements for the arithmetic roughness height is $R_a = 0.4 \mu\text{m}$ and has thus been adopted as the reference value R_{a0} in Eqn (10) and Table 1. The results obtained in former works embraced by this study have also been referred to $R_{a0} = 0.4 \mu\text{m}$, although most of them did not give any quantitative information on roughness⁴.

If there are no experimental data on the heat transfer coefficient for the fluid concerned, an equation developed by Stephan and Preußer [28] is suggested for the determination of the reference value α_0 , i.e.

$$\text{Nu} = 0.1 \left[\frac{\dot{q}_0 d_0}{\lambda' T_s} \right]^{0.674} \left[\frac{q''}{q'} \right]^{0.156} \left[\frac{\Delta h_v d_0^2}{a'^2} \right]^{0.371} \cdot \left[\frac{a'^2 q'}{\sigma d_0} \right]^{0.350} \left[\frac{\eta' c_p'}{\lambda'} \right]^{-0.16} \quad (11)$$

where

- ' property of the boiling liquid,
- " property of the saturated vapour,
- λ thermal conductivity,
- ρ density
- a diffusivity,
- σ surface tension,
- η dynamic viscosity,
- c_p specific heat capacity,
- Δh_v specific enthalpy of vaporization.

3) At other pressures, the effect of the relationship between the properties of the liquid and the temperature and pressure is included in the term $F(p^*)$ in Eqn (8).

4) On the whole, the adoption of this reference value will yield an estimate that is on the safe side, because the heated surfaces in experimental equipment are mostly new and smooth, with the consequence that their mean roughness height ought to be less than $0.4 \mu\text{m}$. If the numerical values in Table 1 are referred to a figure of $R_a = 1 \mu\text{m}$, as is suggested in Chapter Hbb, a further factor of safety would be introduced; according to Eqn (9c), this would entail another 13% on the safe side.

Table 1. Heat transfer coefficients $\alpha_{0, \text{calc}}$ and $\alpha_{0, \text{expl}}$ for various fluids at $p_0^* = (p/p_c)_0 = 0.1$ and $\dot{q} = 20\,000 \text{ W/m}^2$ and $R_{a0} = 0.4 \mu\text{m}$ (heated surfaces relevant to $\alpha_{0, \text{expl}}$, i.e. predominantly single horizontal copper tubes)

Substance	Formula	p_c bar	$\alpha_{0, \text{calc}}$ from Eqs. (8) & (11) $\text{W/m}^2 \text{ K}$	$\alpha_{0, \text{expl}}$ $\text{W/m}^2 \text{ K}$
Methane	CH_4	46.0	8060	7000
Ethane	C_2H_6	48.8	5210	4500
Propane	C_3H_8	42.4	4000	4000
Butane	C_4H_{10}	38.0	3300	3600
n-Pentane	C_5H_{12}	33.7	3070	3400
i-Pentane	C_5H_{12}	33.3	2940	2500
Hexane	C_6H_{14}	29.7	2840	3300
Heptane	C_7H_{16}	27.3	2420	3200 ^a
Cyclohexane	C_6H_{12}	40.8	2420	—
Benzene	C_6H_6	48.9	2730	2000–3500 ^b
Toluene	C_7H_8	41.1	2910	2200–3100 ^b
Diphenyl	$\text{C}_{12}\text{H}_{10}$	38.5	2030	2100 ^a
Methanol	CH_3O	81.0	2770	3000–6500 ^b
Ethanol	$\text{C}_2\text{H}_5\text{O}$	63.8	3690	4400
n-Propanol	$\text{C}_3\text{H}_7\text{O}$	51.7	3170	3800 ^a
i-Propanol	$\text{C}_3\text{H}_7\text{O}$	47.6	2920	3000
n-Butanol	$\text{C}_4\text{H}_9\text{O}$	49.6	2750	2600 ^a
i-Butanol	$\text{C}_4\text{H}_9\text{O}$	43.0	2940	4500 ^a
Acetone	$\text{C}_3\text{H}_6\text{O}$	47.0	3270	3200–4700 ^b
	R 11	44.0	2690	2800
	R 12	41.6	3290	4000
	R 13	38.6	3910	3900
	R 13 B1	39.8	3380	3500
	R 22	49.9	3930	3900
	R 23	48.7	4870	4400 ^a
	R 113	34.1	2180	2650
	R 114	32.6	2460	3800
	R 115	31.3	2890	4200
	R 123	36.7	2600	—
	R 134a	40.6	3500	4500 ^a
	R 152a	45.2	4000	—
	R 226	30.6	—	3700
	R 227	29.3	—	3800 ^a
	RC 318	28.0	2710	4200
	R 502	40.8	2900	3300
Chloromethane	CH_3Cl	66.8	4790	4400
Tetrachloromethane	CCl_4	45.6	2320	1900–4800 ^b
Tetrafluoromethane	CF_4	37.4	4500	4750 ^a
Water	H_2O	220.64	6400	5600
Ammonia	NH_3	113.0	8090	7000
Carbon dioxide	CO_2	73.8	4170 ^c	5100 ^a
Sulfur hexafluoride	SF_6	37.6	2700 ^c	3700
Oxygen	O_2	50.5	6930	9500 ^a on Cu 7200 ^a on Pt
Nitrogen	N_2	34.0	7360	10 000 on Cu 7000 on Pt 5000 on VA
Argon	Ar	49.0	6500	8200 ^a on Cu 6700 ^a on Pt
Neon	Ne	26.5	15 000	20 000 ^a on Cu
Hydrogen	H_2	13.0	20 000	24 000 ^a on Cu
Helium	He	2.28	1 990 ^{d, e}	2000 ^c

a) Very few experimental data available

b) Too much scattering of experimental data to allow a measured value to be determined

c) Eqn (11) solved for the pressure at the triple point, because $p^* = 0.03$ is less than the pressure at the triple point

d) Eqn (11) solved for $p^* = 0.3$ ($p_{\text{He}} = 0.68 \text{ bar}$)

e) The figures are valid for $\dot{q}_0 = 1000 \text{ W/m}^2$, because nucleate boiling of helium no longer applies at $\dot{q}_0 = 20\,000 \text{ W/m}^2$

The characteristic dimension for the determination of the Nusselt number $Nu = \alpha d_0 / \lambda'$ is the bubble diameter at departure d_0 and is given by

$$d_0 = 0.0149 \beta \left(\frac{2\sigma}{g(q' - q'')} \right)^{0.5} \quad (12).$$

The numerical values for the contact angle β are 45° for water, 1° for cryogenic liquids, and 35° for other fluids.

Eqn (11) has been based on a set of analogous correlations that apply to several groups of substances and were derived by Stephan and Abdelsalam [19]. Since it has been specifically adapted for values measured at pressures close to atmospheric, it is suitable for the determination of α at the reduced pressure $p^* = 0.03$, which corresponds to a pressure of 1–2 bar at the boiling point of many organic liquids (cf. column headed p_c in Table 1).

However, Eqn (11) does not adequately represent the dependence of α on pressure (cf. [28]), and is therefore not used for the direct determination of the reference value α_0 at $p_0^* = 0.1$. In this case, α_0 is obtained in two steps:

- (a) from Eqn (11) for $p^* = 0.03$;
- (b) conversion from $p^* = 0.03$ into the value at $p_0^* = 0.1$ by means of Eqn (8).

The heat transfer coefficients calculated from Eqns (11) and (8) for very many of the fluids listed in Table 1 agree well with the values determined by experiment. In the cases where larger differences occur, the calculated values would usually allow equipment to be designed on the safe side. The biggest differences (ca. 14%) in the opposite direction apply to water and ammonia, but the great differences that were observed in earlier measurements on methane and ethane have not been confirmed by the most recent measurements [165; 166].

Thus the comparison of experimental with calculated values in Table 1 also gives an impression of the reliability of Eqn (11), in cases where no empirical data on α exist for the substance concerned.

For the group of halocarbon refrigerants, similar agreement between experimental and calculated values is obtained by a mathematical method developed by Slipcevic [29]; and an equation devised by Hirschberg [30] yields results that, on an average, differ slightly less than those obtained by Eqns (11) and (8) from the experimental values.

If measured data are available but some doubt exists on their convertibility into values that are suitable for the application in question, preference should be given to the mathematical method represented by Eqns (11) and (8) for the determination of α_0 . Examples of where this arises are experimental layouts in which the evaporator cannot be thermostatically controlled with sufficient accuracy, measurements for which the heater elements are thin wires, and cases in which boiling occurs at the faces of cylindrical rods of moderate thickness.

The results of measurements on horizontal tubes ($d \geq 8$ mm), flat plates ($D \geq 80$ mm) and thin wires ($0.05 \leq d \leq 0.3$ mm) are compared in Fig. 13. They are presented in the forms of the relative relationship between α and pressure and the relationship between the pressure and the index n in the relationship $\alpha \sim q^n$. It can be seen that the scatter of values obtained for the horizontal tubes and the plates fall within a common range, which corresponds to those represented in Figs. 6–11; but that the $\alpha(p^*)$ and $n(p^*)$ functions for thin wires are significantly different and involve an additional parameter, i.e. the diameter of the wires. However, results obtained by Shi [169] in more recent experiments on various halocarbons heated over a horizontal platinum wire of 67 mm length and 0.1 mm diameter are more in line with the relative relationship between α and pressure that is described by Eqn (8a). In addition, the relationship that they reveal between n and the pressure is only slightly more pronounced than that derived from Eqn (7a), if moderate to high heat fluxes are applied and the pressures are not too low, i.e. $p^* \geq 0.1$.

Effect of industrial-scale evaporators

Examples of parameters, other than those embraced by Eqns (10) and (11), that may affect the performance of industrial-scale free-convection evaporators are

- the geometry of the individual heating elements (e.g. finned tubes and tubes with surfaces to augment boiling) and the entire layout of the evaporator (tube bundles; horizontal and inclined flat walls, etc);
- additives and adulterants in the boiling liquid (e.g. wetting agents, oils, or dissolved substances that are deposited on the heated surfaces);
- the operation of the plant (e.g. the admission of sub-cooled liquid and intermittent operation with on-off control).

Allowance for these effects can be made by appending appropriate corrections to the heat transfer coefficient α_{10} calculated from Eqn (10) and by introducing an average heat transfer coefficient $\bar{\alpha}$ for the entire evaporator, i.e.

$$\frac{\bar{\alpha}}{\alpha_{10}} = C_G C_{Ad} C_{Op} \quad (13)$$

where C_G is the correction for the geometry; C_{Ad} for the adulterants in the boiling liquid; and C_{Op} for the particular mode of operation.

The design of large shell-and-tube evaporators is hampered by the lack of reliable information in the literature. One reason for the scantiness of reliable data is the factor of uncertainty involved in scaling up the results obtained in research setups. Another reason is that the three parameters for which corrections are allowed usually occur simultaneously in large pilot plants and cannot be separated. Hence, the results gained on individual items or equipment or their assemblies cannot be presented, even to a limited extent, in a generalized form. In the sections that follow,

the nature and magnitude of the factors in Eqn (13) will be discussed in the form of hints for drawing up estimates⁵.

Effect of geometry

As is evident from Fig. 13, there are no differences from the aspect of the geometry between the values of α for individual horizontal tubes and those for horizontal flat walls⁶. Roughly the same applies to vertical heated surfaces, because an improvement in heat transfer brought about by additional convection at the lower end entails an impairment at the upper end due to the restriction imposed by the rising cloud of bubbles on the liquid inflow. In some cases, the flow velocity at the lower end of vertical heated surfaces or through horizontal tube bundles may greatly exceed the upward velocity of the free-flowing bubbles. Under these circumstances, heat transfer must be determined by the method described in Chapter Hb for forced-circulation evaporators.

Finned tubes and tubes with artificial nucleation sites

Within the range of initial nucleate boiling, horizontal evaporator tubes with external fins transfer heat more efficiently than plain tubes, particularly if the heat flux is low. This is illustrated in Fig. 14 for the range of pressures extending from 0.3% to 30% of the value at the critical point. In almost the entire range of heat flux concerned, the finned tube has a higher heat transfer coefficient, expressed in terms of the external area, than the plain tube⁷. The increase in α can be ascribed to the great roughness at the crests of the fins and the additional convection caused by the bubbles that rise along their flanks.

If the performance of finned tubes is described in the form of Eqns (5)–(8), the value obtained for the index n will be less than that for a plain tube, and the effect of pressure in the term $F(p^*)$ will be weaker. A preliminary estimate for the effect of the heat flux in finned tubes can be obtained from the following equation:

$$n_f(p^*) = n(p^*) - 0.1 h/t_f \quad (14)$$

where n_f and n are the indices in the terms for the heat flux ratio {in Eqn (5)} for the finned and plain tubes

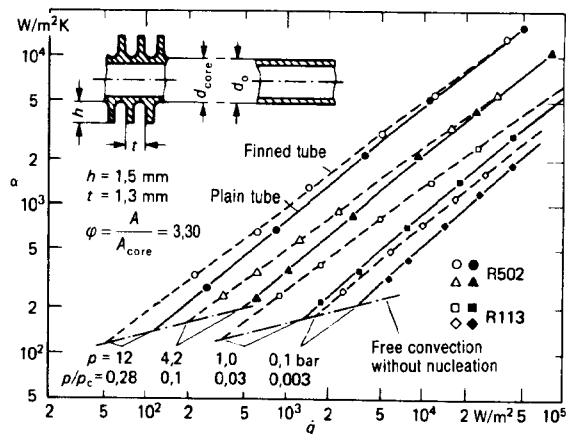


Fig. 14. α as a function of \dot{q} for a finned and a plain tube of the same core diameter. The parameter is the saturation pressure p or the reduced pressure p^* . Both α and \dot{q} are expressed in terms of the total outer area of the respective tube.

respectively and $n(p^*)$ can be obtained from Eqn (7); h is the height of the fins; and t_f is the free spacing between fins. Approximate allowance can be made for the reduced effect of pressure by substituting the term $F(p^*/\sqrt{\phi})$ for $F(p^*)$ in Eqn (8), i.e.

$$F_i = F(p^*/\sqrt{\phi}) \quad (15)$$

where ϕ is the ratio of the surface area of the finned tube to that of a plain tube of the same core diameter.

If no experimental data are available for verification, Eqns (14) and (15) should not be used except for copper tubes of conventional dimensions and unless the pressure is within the $0.03 \leq p^* \leq 0.3$ range.

Allowance for the roughness of finned tubes should not be in the form of Eqn (9), because the great dissimilarities in the surface structure of the fin crests and the other parts of the tubes entail that the conditions are completely different to those encountered for plain tubes. However, a great allowance has already been introduced by the modifications made in the relationship between the heat transfer coefficient, the heat flux, and the pressure.

A fact that can be exploited in estimating the reference value α_{of} for finned tubes is that the heat transfer coefficients for both finned and smooth copper tubes are approximately the same at a heat flux of about 10^5 W/m^2 and a reduced pressure of $p^* = 0.03$. This is evident from Fig. 14. Thus the first step would be to calculate the heat transfer coefficient that applies for both tubes at $p^* = 0.03$ and $\dot{q} = 10^5 \text{ W/m}^2$ from the relevant figures for α_o , $F(p^*)$ and $n(p^*)$. The value for α_o can be obtained from Table 1 or Eqn (11); for $F(p^*)$, from Eqn (8); and for $n(p^*)$, from Eqn (7). The reference value α_{of} can then be derived by a backward calculation to $\dot{q} = 20000 \text{ W/m}^2$, as determined from Eqn (14), and $p^* = 0.1$, as determined from Eqn (15) (cf. Example 2 and Fig. 20).

5) Priority in the near future must be devoted to the improvement of Eqns (10) and (11) [79], and the factors in Eqn (13) should be verified by experiment in each case for the individual items of equipment or assemblies.

6) On the one hand, this applies primarily to the relationship of the heat transfer coefficient to the heat flux and the pressure as shown in Fig. 13. On the other hand, differences may occur in the absolute values of α for horizontal tubes and those for flat plates. One reason for this is that flow development and edge effects are more difficult to eliminate in horizontal plates than in horizontal tubes.

7) In Fig. 14 and in all further general considerations on finned tubes, the heat transfer coefficient defined by Eqn (1) was not derived from the temperature at the actual surface of the tubes but from that at the base of the fins. It is thus referred to as the "apparent heat transfer coefficient".

The method suggested here yields results that agree well with measured values. Equally good agreement is achieved by a mathematical method devised by Slipcevic [137] (see also [138]).

Evaporator tubes with artificial nucleation sites have been the subject of experiment for some time, particularly for boiling halocarbon refrigerants. It can be seen from Figs. 15 and 16 that they can often increase the heat transfer coefficient to a value much higher than that attained by conventional finned tubes. Examples of the means adopted to promote bubble formation are the flattening of finned tubes between rolls and the sintering of porous metal layers onto the tubes. Impurities in the pores may drastically reduce the heat

transfer coefficient in the course of time, but there is a lack of systematic long-term studies on the subject.

Horizontal tube bundles

If the heat flux is low or moderate, the average heat transfer coefficient in nucleate boiling at the outer surfaces of horizontal tubes in a bundle is higher than that for a single tube. One reason for this is that the liquid inlet is designed to cope with the requirements for the entire bundle of tubes, with the result that the flow rate over the lowermost bank is higher than that over a single tube. Another reason is the rising cloud of bubbles within the bundle.

A number of similarly derived proposals have appeared in the literature for the mathematical treatment of the contributions thus made towards heat transfer [37; 80; 81]. In the discussion on Eqn (13), attention was drawn to the problems involved in the interpretation of experimental results and to their application in the design of tube-bundle evaporators. In view of these remarks, there would be no justification for a complicated mathematical outlay; and, for this reason, the following simple method, analogous to those described elsewhere [37; 80], is recommended.

First of all, the average heat transfer coefficient for the tube bundle $\bar{\alpha}$ is calculated from the coefficient α_u for the undermost row and the relative improvement $\bar{\alpha}/\alpha_u$ within the bundle. Thus

$$\bar{\alpha} = \alpha_u \frac{\bar{\alpha}}{\alpha_u} \quad (16)$$

α_u is calculated by adding the heat transfer coefficient for the one single tube in nucleate boiling $\alpha_{\text{one, nb}}$ and that in natural convection without nucleation $\alpha_{\text{one, conv}}$, i.e.

$$\alpha_u = \alpha_{\text{one, nb}} + \alpha_{\text{one, conv}} \quad (17)$$

where f depends on the size of the tube bundle and the anticipated inlet flow velocity. The lower limit is $f = 0.5$; and the upper, $f = 1$.

Evaluation of data extracted from the literature gives the following approximate expression for $\bar{\alpha}/\alpha_u$:

$$\frac{\bar{\alpha}}{\alpha_u} = 1 + \left(2 + \frac{\dot{q} \varphi}{1000 \text{ W/m}^2} \right)^{-1} \quad (18)$$

where \dot{q} is the heat flux, expressed in terms of the total outer surface area of the tubes, and φ is the area ratio ($\varphi = 1$ for plain tubes).

Eqn (18) applies in the $1000 \text{ W/m}^2 \leq \dot{q} \varphi \leq 20000 \text{ W/m}^2$ range and for near-atmospheric pressure, i.e. about 0.5–10 bar. The evaluation of Eqns (16–18) for bundles of plain and finned tubes is illustrated in Fig. 17 and, in connection with Example 2, in Fig. 20.

A condition that has been assumed in the derivation of these equations is that $\dot{q} = \text{constant}$ for the entire tube bundle. It is certainly not satisfied if the bundles are heated by a liquid. In this case, the heat flux for the individual segments or rows in the bundle must be estimated from the difference between the temperature of the heating medium and that of the boiling liquid at

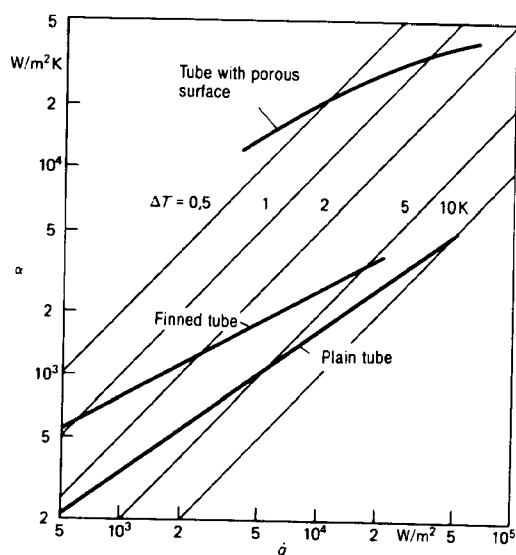


Fig. 15. Heat transfer coefficient α as a function of the heat flux \dot{q} for various types of tubes in the nucleate boiling of Refrigerant R12 at a pressure of 3.1 bar [82].

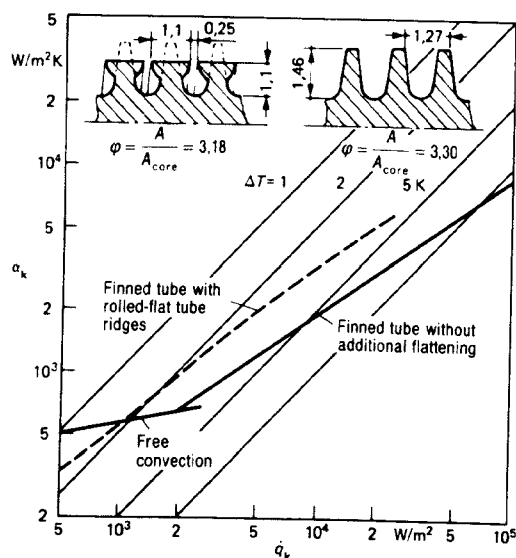


Fig. 16. Heat transfer coefficient α as a function of the heat flux \dot{q} in nucleate boiling of Refrigerant 11 at a pressure of 1.3 bar for two tubes with fins of the same basic structure [87], but one of the tubes was rolled to flatten the ridges of the fins.

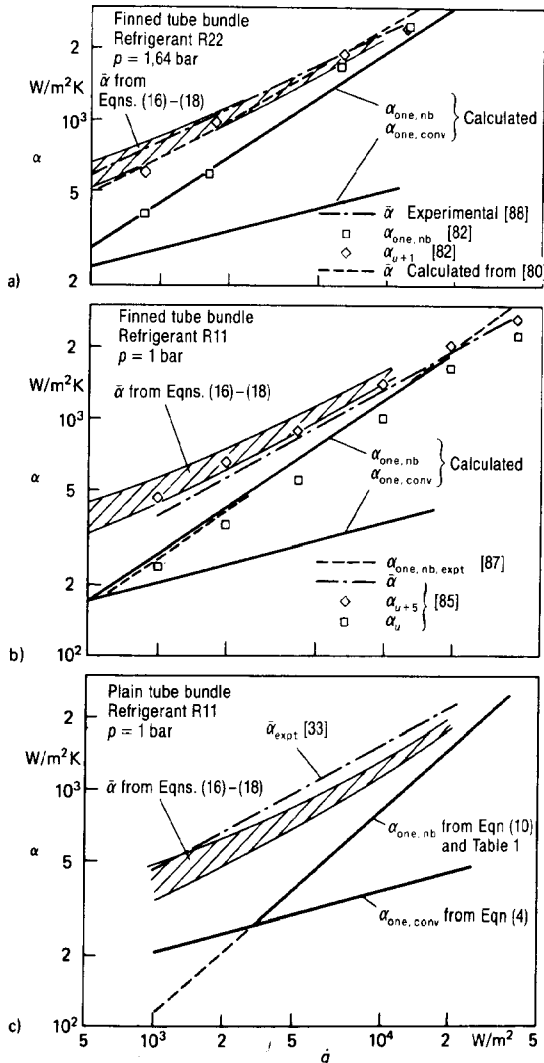


Fig. 17. Calculated and measured values of the heat transfer coefficient as functions of the heat flux in nucleate boiling in tube bundles.

a and b, finned-tube bundles; c, plain-tube bundles. The term α_{u-5} denotes the coefficient for the fifth bank of tubes as counted from the undermost (see specimen calculation in Example 2 and Fig. 20).

the inlet to the tube bundle⁸. In addition, the flow velocity within the bundle should be of the same order of magnitude as the free upward velocity of the individual rising bubbles. If it is much higher, the mathematical methods for two-phase flow should be resorted to [98–102].

Effect of impurities

As a rule, adulteration of the boiling liquid by another impairs heat transfer (cf. Section 3.2).

The only impurities that can increase the transfer of heat above that of the pure boiling liquid if they are added in low concentrations are surfactants, but the improvement is usually slight [35; 36; 82–84]. The

8) Examples of heat transfer coefficients determined experimentally in a row of tubes within a bundle if the heat flux varies at the lower rows are given in the literature [38; 85; 86].

increase can be ascribed to the decrease in surface tension and the associated reduction in the energy required to produce viable bubbles. Typical results obtained on adding oil to boiling refrigerants on flat plates [36; 84] and at finned tubes [82] are shown in Fig. 18. In some cases, an increase in α is evident.

Impurities in the form of dissolved or suspended additives that are deposited on the heated surface usually give rise to heat transfer coefficients that are lower than those for nucleate boiling of the pure liquid.

Apart from the presentation of results of individual measurements, no method yet exists for estimating the effect of impurities on the heat transfer coefficient in nucleate boiling.

Effect of evaporator operation

All the experimental results that have been described up to now in Section 3.1 and the mathematical methods developed from them apply to the case in which the potential nucleation sites are fully activated at the wall superheat and the liquid is in the saturated state associated with the reigning pressure at some distance from the heated surface. If the evaporator does not allow these conditions to be adhered to, the heat transfer coefficients in the range of initial nucleate boiling are lower than those that would be expected from Eqns (5)–(12).

On-off controllers or gradual increments in heat flux increase the likelihood that nucleation is not completely activated. As a consequence, an operating point is reached between fully activated nucleation and free convection without bubble formation, as is illustrated by the hatched zone in Fig. 19.

A corresponding effect is brought about by the entry of subcooled liquid. In cases of this nature, therefore, Eqn (3) or Eqn (4) should be resorted to instead of Eqn (5) in order to allow a safeguard, even if the heat flux or superheat is somewhat higher than that corresponding to the commencement of nucleate boiling.

Example 1

Calculate the length of a single cylindrical rod of 10 mm outer diameter required to transfer 1 kW for the evaporation of water at $p = 100$ bar. The superheat is $\Delta T = 3$ K.

$$\dot{Q} = A \dot{q} = \pi D L \alpha \Delta T.$$

$$\text{The desired length is } L = \dot{Q} / \pi D \alpha \Delta T.$$

The heat transfer coefficient is calculated in five stages.

(a) According to Table 1, the reference value is

$$\alpha_{0, \text{exp}} = 5600 \text{ W/m}^2 \text{ K at } \dot{q}_0 = 20\,000 \text{ W/m}^2 \text{ and } p_0^* = 0.1.$$

A value of ca. 14% higher, i.e. 6400 W/m² K, is obtained from Eqns (11) and (8).

(b) Conversion into the value at $p_1 = 100$ bar

$$\text{Since } p_c = 220.64 \text{ bar (Chapter Dc), } p_1^* = p_1/p_c = 0.453.$$

$$\text{Eqn (8b) then gives } \alpha_1/\alpha_0 = F(p_1^*) = 2.816.$$

$$\alpha_1 = 2.816 \alpha_0 = 15\,768 \text{ W/m}^2 \text{ K at } p_1 = 100 \text{ bar and}$$

$$\dot{q}_1 = 20\,000 \text{ W/m}^2.$$

(c) $\alpha(\dot{q})$ at p_1

$$\text{Eqn (7b) gives } n(p_1^*) = 0.634.$$

$$\text{Thus } \alpha \sim \dot{q}^{0.634} \text{ at } p_1$$

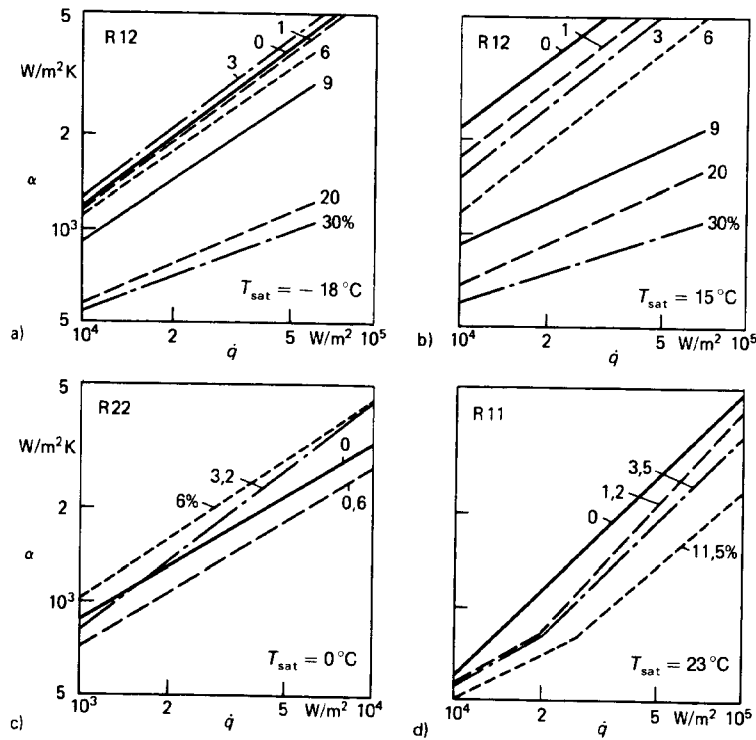


Fig. 18.

Heat transfer coefficient α as a function of the heat flux \dot{q} for nucleate boiling of refrigerant-oil mixtures.

a and b: R 12 on a horizontal plate [36]. c: R 22 on a horizontal finned tube [82]. d: R 11 on a horizontal plate [84].

The parameter is the mass fraction of oil in %.

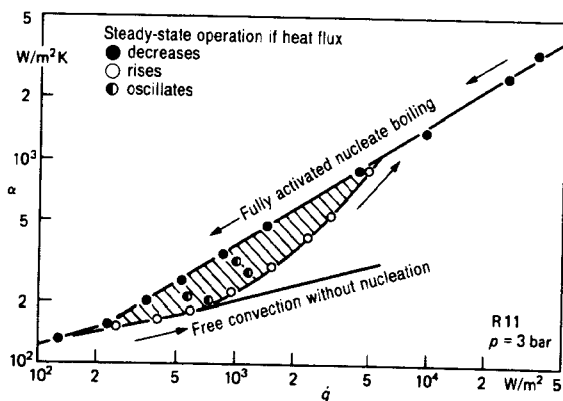


Fig. 19. Effect of the mode of operation on the heat transfer coefficient at the initiation of nucleate boiling evaporation: Refrigerant R 11 on a finned tube at $p = 3.0$ bar [38]. Similar results have been obtained elsewhere [83; 86].

- (d) $\alpha(\Delta T)$ at p_1
 $\alpha \sim \dot{q}^{0.634} = (\alpha \Delta T)^{0.634}$ according to Eqn (1).
 Therefore, $\alpha \sim T^{1.73}$

- (e) α_2 at $\Delta T_2 = 3$ K and $p_2 = p_1$.

$$\frac{\alpha_2}{\alpha_1} = \left(\frac{\Delta T_2}{\Delta T_1} \right)^{1.73} = 4.442; \Delta T_1 = \frac{\dot{q}_1}{\alpha_1} = 1.267 \text{ K}$$

$$\alpha_2 = 4.442 \alpha_1 = 70\,042 \text{ W/m}^2 \text{ K and } \dot{q}_2 = 210\,125 \text{ W/m}^2.$$

Since $\alpha_2 > 50\,000 \text{ W/m}^2 \text{ K}$, it should be checked how close the heat flux \dot{q} is to the maximum in nucleate boiling (Fig. 5). It follows from Eqn (26) in Section 4 that $\dot{q}_{crit} > 10^6 \text{ W/m}^2 \gg \dot{q}_2$. (The values of the maximum heat flux that can be obtained for water are much higher than those for most organic liquids).

The length of rod required is thus

$$L = \frac{1000}{0.01 \times 210300 \pi} \text{ m} = 15.1 \text{ cm}$$

Example 2

An evaporator has to be designed in which Refrigerant R 22 boils at 0°C . The tubes have fins of $h = 1.5$ mm height spaced at $t_f = t - b = 0.95$ mm. The area ratio is $\varphi = 3.18$. Determine $\alpha(\dot{q})$ for the tube bundle.

The calculation proceeds in the following four steps.

- (a) α_0 for a single plain tube

According to Table 1, $\alpha_{0,exp} = 3900 \text{ W/m}^2 \text{ K}$ if $p_0^* = 0.1$ and $\dot{q}_0 = 20\,000 \text{ W/m}^2$. A value of $3930 \text{ W/m}^2 \text{ K} \approx 3900 \text{ W/m}^2 \text{ K}$ is obtained from Eqns (11) and (8).

- (b) α_{of} for a single finned tube

It is assumed that the heat transfer coefficient for a finned tube is identical to that for a plain tube if $p^* = 0.03$ and $\dot{q} \approx 10^5 \text{ W/m}^2 \text{ K}$, i.e. that $\alpha_{if} = \alpha_1$ for $\dot{q}_1 = 100\,000 \text{ W/m}^2 \text{ K}$.

Conversion from $\alpha_{0,exp}$ at $p_0^* = 0.1$ for the plain tube to α_2 at $p_2^* = 0.03$ when $\dot{q}_2 = \dot{q}_0 = 20\,000 \text{ W/m}^2 \text{ K}$

$$\alpha_2/\alpha_0 = F(p^* = 0.03) = 0.572 \text{ according to Eqn (8a). Hence } \alpha_2 = 2230 \text{ W/m}^2 \text{ K.}$$

Conversion from α_2 to α_1 at $\dot{q}_1 = 10^5 \text{ W/m}^2$ when $p_1^* = p_2^* = 0.03$

$$\frac{\alpha_1}{\alpha_2} = \left(\frac{100\,000}{20\,000} \right)^n = 3.595; \alpha_1 = 8017 \text{ W/m}^2 \text{ K}$$

where $n(p^* = 0.03) = 0.9 - 0.3 p^{*0.3} = 0.795$ according to Eqn (7a).

Proceeding backwards from $\alpha_1 = \alpha_{if}$ to α_{2f} for a finned tube when $p^* = 0.03 = \text{constant}$

$$\frac{\alpha_{2f}}{\alpha_{1f}} = \left(\frac{20\,000}{100\,000} \right)^{n_f} = 0.3587; \alpha_{2f} = 2876 \text{ W/m}^2 \text{ K}$$

where $n_f = n(p^* = 0.03) - 0.1 h/t_f = 0.637$ according to Eqn (14).

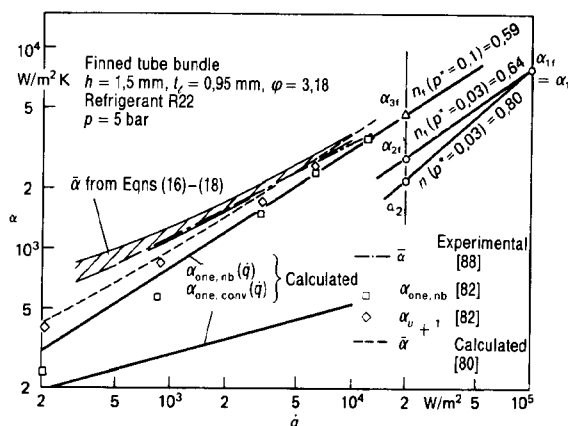


Fig. 20. α as a function of \dot{q} , as determined from Eqns (4)–(18) (see Example 2). Comparison with experimental data [82; 88] and with the figures obtained by a mathematical method proposed by Slipcevic [80]. The term α_{u+1} denotes the heat transfer coefficient for the bank of tubes immediately above the undermost.

Conversion into α_{of} at the reference pressure $p_0^* = 0.1$ when $\dot{q}_0 = 20000 \text{ W/m}^2 = \text{constant}$

$$\alpha_{of} = \frac{F_t(p_0^*/\sqrt{\phi})}{F_t(p_2^*/\sqrt{\phi})} = \frac{0.7508}{0.4574} = 1.641.$$

Hence $\alpha_{of} = 4720 \text{ W/m}^2 \text{ K}$ at $p_0^* = 0.1$ and $\dot{q}_0 = 20000 \text{ W/m}^2$.

- (c) Conversion into the value at the pressure p_3 of R 22 at a boiling temperature of 0°C

The saturation pressure of R 22 is $p_3 = 4.98$ at 0°C . According to Chapter Dc, $p_c = 49.9$ bar. Hence if $p_3 = 4.98$ bar, $p_3/p_c = p_3^* = 0.1$.

Since $p_3^* = p_0^*$, $\alpha_{3f} = \alpha_{of}$ and no further calculation is required. {If $p_3^* \neq p_0^*$, the conversion would have to proceed to the end as in step (b) above.}

The following figures thus apply for a single tube at p_3^* :

$$\alpha_{one, nb} = C_{nb} \dot{q}^{0.592} = 13.42 \dot{q}^{0.592}$$

for $n_{3f} = n(p_3^* = 0.1) - 0.1 h/t_f = 0.592$ according to Eqns (7 a) and (14) and for $C_{nb} = \alpha_{3f}/20000^{0.592} = 13.42$.

- (d) Determination of $\alpha(\dot{q})$ for the tube bundle from Eqns (16)–(18)

$$\bar{\alpha} = \alpha_u \frac{\bar{\alpha}}{\alpha_u} = (\alpha_{one, nb} + f \alpha_{one, conv}) \left[1 + \left(2 + \frac{\dot{q} \phi}{1000 \text{ W/m}^2} \right)^{-1} \right],$$

$$\bar{\alpha} = (13.42 \dot{q}^{0.592} + 51.77 f \dot{q}^{0.25}) \left[1 + \left(2 + \frac{\dot{q} \phi}{1000 \text{ W/m}^2} \right)^{-1} \right]$$

where $\alpha_{one, conv} = C_{conv} \dot{q}^{0.25} = 51.77 \dot{q}^{0.25}$, as determined from Eqn (4) on page Ha 3 and from the properties listed in Chapter D. If the upper and lower limits for the estimate are taken to be $f = 1$ and $f = 0.5$ respectively, the hatched zone in Fig. 20 will be obtained. The unit to which the numerical values for α pertain throughout the entire calculation is $\text{W/m}^2 \text{ K}$; and for \dot{q} , W/m^2 .

3.2 Nucleate boiling of binary and multicomponent mixtures

The heat transfer coefficient α in nucleate boiling of binary mixtures is less than the molar average α_m , as defined by Eqn (19 a), of the heat transfer coefficients α_1 and α_2 for the individual components:

$$\alpha_m = \bar{x}_1 \alpha_1 + \bar{x}_2 \alpha_2 \quad (19 a)$$

where \bar{x}_1 and \bar{x}_2 are the molar fractions of Components 1 and 2.

This relationship was first established by Bonilla and Perry [89] and has since been confirmed in many other studies that have been reviewed [96].

When a binary mixture boils, the more volatile component is concentrated within the bubbles of vapour; and the less volatile, in the superheated interface around the bubbles. As a consequence, the saturation temperature of the liquid in the vicinity of the heated wall $T_{sat W}$ is elevated. Thus the motive temperature difference is actually $T_W - T_{sat W}$, but what is normally taken to determine α is the difference between the temperature of the heated wall and the saturation temperature of the undisturbed liquid at the relevant pressure, i.e.

$$\alpha = \frac{\dot{q}}{\Delta T} = \frac{\dot{q}}{T_W - T_{sat}(p_{sat})} \quad (19 b).$$

Accordingly, the heat transfer coefficient ought not to fall below the molar average, as indicated by

Eqn (19 a), if the composition of the growing vapour bubbles is to be the same as that of the undisturbed liquid, e.g. if the mixture were azeotropic. This premise has been confirmed – at least qualitatively – by measurements on systems of a corresponding nature [39; 53; 55; 90; 91; 97] (Fig 23). If one of the components is a surfactant and is present in low concentrations, the impairment of heat transfer may be compensated or even outweighed (cf. Fig. 18). In the literature, the improvement is attributed to the reduction in the energy required to produce viable bubbles. This group of substances will not be further discussed here.

It has been demonstrated that the energy required to produce viable bubbles in all other mixtures is greater than that for the pure components [92; 93]. The attendant reduction in the number of bubbles formed at the heated surface entails a decrease in the heat transfer coefficient.

If the pressure is close to atmospheric, α can be determined by a method proposed by Stephan and Körner [92], in which the superheat ΔT required to transfer the heat flux \dot{q} into a mixture is regarded as a real magnitude that can be broken down into an ideal ΔT_{id} and an excess ΔT^E component, i.e.

$$\Delta T = \frac{\dot{q}}{\alpha} = \Delta T_{id} + \Delta T^E \quad (20).$$

In analogy to Eqn (19 a), the ideal component for a binary mixture can be defined by

$$\Delta T_{id} = \bar{x}_1 \Delta T_1 + \bar{x}_2 \Delta T_2 \quad (21)$$

where ΔT_1 and ΔT_2 are the temperature differences required to transfer \dot{q} to the pure substances 1 and 2 respectively⁹.

If Fig. 21 is compared to Fig. 22, it can be seen that the excess component can be expressed qualitatively as a function of the difference between the concentration of the more volatile substance in the vapour and its concentration in the liquid, i.e. $\bar{y}_1 - \bar{x}_1$. It is thus evident that the differences between the concentration on

⁹ The heat transfer coefficient defined by $\alpha_{id} = \dot{q}/\Delta T_{id}$ is not identical to the α_m defined by Eqn (19 a). If, however, ΔT_2 does not differ very much from ΔT_1 , the difference between α_m and α_{id} is insignificant.

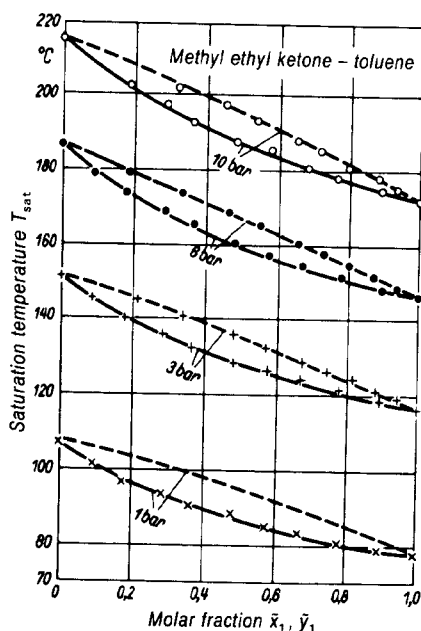


Fig. 21. Boiling diagram for a mixture of methyl ethyl ketone and toluene at pressures between 1 and 10 bar [55]

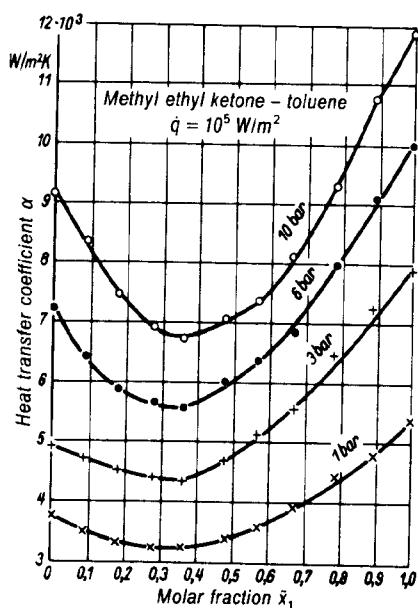


Fig. 22. Relationship between the heat transfer coefficient and the composition of a mixture of methyl ethyl ketone and toluene at a heat flux of $\dot{q} = 10^5 \text{ W/m}^2$ and pressures between 1 and 10 bar

the boiling point curve and that on the dew point curve and between the heat transfer coefficient and the molar average are greatest at medium values for the concentration. The correlation suggested is

$$\Delta T^E / \Delta T_{id} = K_{12} (\bar{y}_1 - \bar{x}_1) \quad (22)$$

The constant of proportionality K_{12} is composed of a pressure function and a magnitude A_0 that is roughly independent of pressure and concentration [92], i.e.

$$K_{12} = A_0 \left(0.88 + 0.12 \frac{p}{\text{bar}} \right) \quad (23)$$

Values for A_0 are listed in Table 2. In principle, they can be determined by a single measurement at

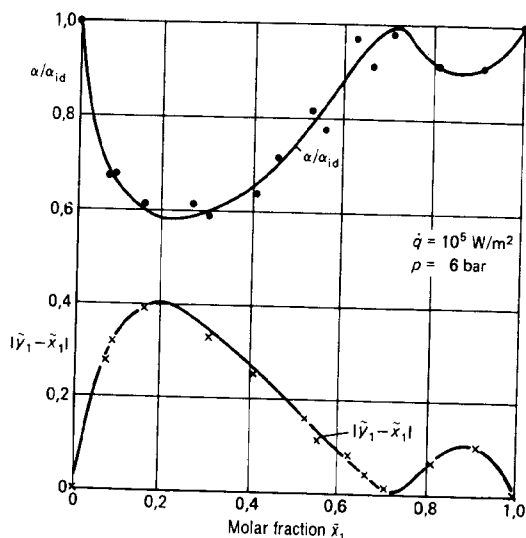


Fig. 23. Ratio α/α_{id} and the concentration difference $\bar{y}_1 - \bar{x}_1$ as functions of the composition of a methanol/benzene mixture at $\dot{q} = 10^5 \text{ W/m}^2$ and $p = 6 \text{ bar}$ [97]

Table 2. Experimental values for the term A_0 in Eqn (23) [93]

System	A_0
Acetone-ethanol	0.75
Acetone-butanol	1.18
Acetone-methanol	1.19
Acetone-water	1.40 on copper 0.81 on nickel
Ethanol-benzene	0.42
Ethanol-cyclohexane	1.31
Ethanol-water	1.21 on copper 0.71 on nickel
Benzene-toluene	1.44
Heptane-methylcyclohexane	1.95
Isopropanol-water	2.04
Methanol-ethanol	1.39
Methanol-benzene	1.08
Methanol-amy alcohol	0.80
Methanol-water	0.56
Methyl ethyl ketone-toluene	1.32
Methyl ethyl ketone-water	1.21
Propanol-water	3.29
Water-ethylene glycol	1.47
Water-glycerol	1.50
Water-pyridine	3.56

any given concentration and any near-atmospheric pressure [92–95]. Eqns (20)–(22) can be combined to yield

$$\frac{\alpha}{\alpha_{id}} = \frac{1}{1 + K_{12} (\bar{y}_1 - \bar{x}_1)} \quad (24)$$

where

$$\frac{1}{\alpha_{id}} = \frac{\bar{x}_1}{\alpha_1} + \frac{\bar{x}_2}{\alpha_2} \quad (21a)$$

The ratio α/α_{id} and the difference in concentration $\bar{y}_1 - \bar{x}_1$ for a methanol/benzene mixture at $p = 6 \text{ bar}$ and $\dot{q} = 10^5 \text{ W/m}^2$ are shown as functions of \bar{x}_1 in Fig. 23.

The mathematical method represented by Eqns (20)–(24) is restricted to comparatively high heat fluxes of about 10^5 W/m^2 and pressures close to atmospheric. If it is applied over a wide range of operating conditions, large errors may arise, because it entails a linear relationship between the pressure and the proportionality constant K_{12} and a heat flux effect that is independent of concentration.

Stephan and Preußer [28; 93–95] extended the method represented by Eqns (20)–(24) to embrace nucleate boiling of multicomponent mixtures at low pressures. In this case, the value of α for the mixtures is determined by inserting the properties of the mixtures concerned in Eqns (11) and (12) and multiplying by the following expression for the specific mixture:

$$\left(1 + \sum_{i=1}^{n-1} (\tilde{y}_i - \tilde{x}_i) \left(\frac{\partial \tilde{y}_i}{\partial \tilde{x}_i} \right)_{\tilde{x}_j, p} \right)^{-0.0733} \quad (25)$$

where $\tilde{y}_i - \tilde{x}_i$ is the difference between the concentration of the vapour and that of the liquid for Component i , and $(\partial \tilde{y}_i / \partial \tilde{x}_i)_{\tilde{x}_j, p}$ is the slope of the $\tilde{y}_i(\tilde{x}_i)$ curve in the \tilde{x}_i, \tilde{y}_i diagram for constant pressure and constant concentration of the other components.

The expanded version of Eqn (11) yielded satisfactory results when it was checked against previous figures obtained in measurements on binary mixtures and figures for two ternary systems, viz. acetone-methanol-water and methanol-ethanol-water. Until further experimental data are available, the correlation should be resorted to only for first approximations, particularly if the heat transfer coefficient has to be determined for mixtures with more than two components.

A mixture of sulfur hexafluoride and Refrigerant R 13 B1 has been taken as an example in Fig. 24 to illustrate $\alpha(q, p^*, \tilde{x}_1)$ relationships at elevated reduced pressures over a wide range of heat fluxes [61]; p^* was obtained from the experimental value for the critical pressure for each of the five mixtures.

It can be derived from the diagram in the centre that the relationship between the index n in the relationship $\alpha \sim \dot{q}^n$ and the reduced saturation pressure in mixtures is similar to that for the individual components. However, the concentration also exerts a significant effect. A comparison of the two lower diagrams clearly reveals that narrow limits are imposed in establishing an analogy between the relationship to concentration of the ratio α/α_{id} and that of the difference $\tilde{y}_1 - \tilde{x}_1$. In the first place, the α/α_{id} minima occur at significantly lower SF_6 concentrations than those corresponding to the $\tilde{y}_1 - \tilde{x}_1$ maxima. In the second place, the impairment of heat transfer in mixtures becomes continuously more pronounced until the critical point is approached, whereas the difference in concentration becomes less. It would appear that the heat transport becomes less than that for the pure substances in the vicinity of the heated surfaces. This is because less bubbles are detached, although the number formed per unit area of the heated surface increases. Another reason is that the decrease in the velocity at which the bubbles rise detracts from the intensity

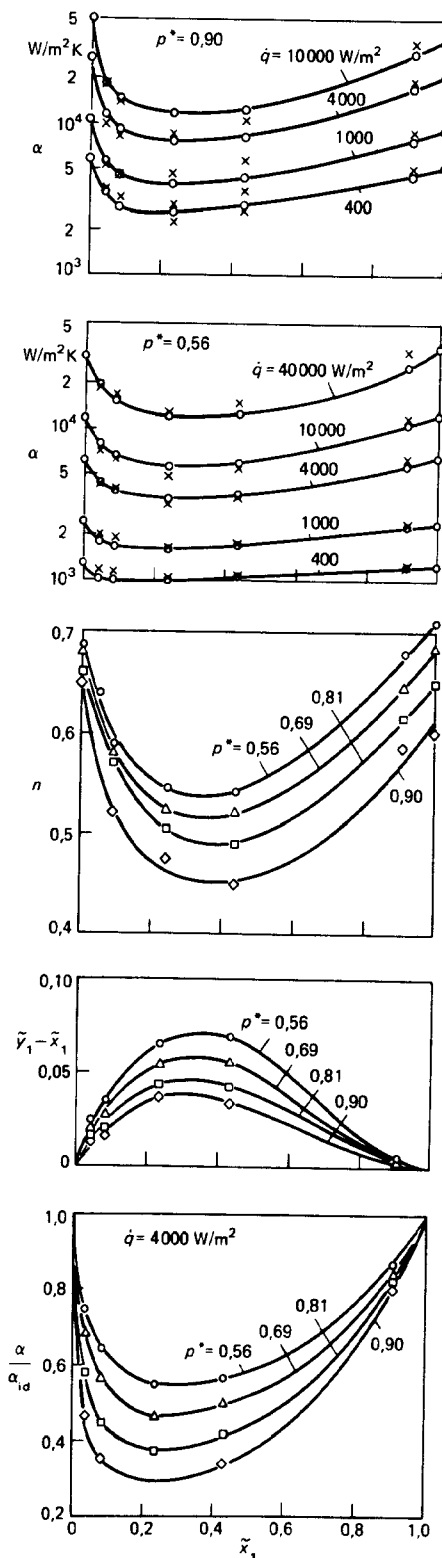


Fig. 24. Boiling of binary mixtures at elevated pressure. The example taken is sulfur hexafluoride/Refrigerant R 13 B1. The two upper diagrams show the heat transfer coefficient at two reduced pressures as a function of the molar concentration of the more volatile component \tilde{x}_1 with the heat flux as parameter. The three lower diagrams show the exponent n in the relationship $\alpha \sim \dot{q}^n$, the concentration difference $\tilde{y}_1 - \tilde{x}_1$, and the ratio α/α_{id} as functions of \tilde{x}_1 with the pressure as parameter. According to Bier et al [61], but with slightly altered values of $n(p^*)$. Additional values calculated from Eqn (26) with $B_0/B_1 = 10^4 \text{ s/m}$ for $p^* = \text{constant}$ have been entered (x) in the two upper diagrams.

of mixing in the boundary layer. Yet another fact that emerges from Fig. 24 is that method entailed by Eqns (22) and (23) is much less complicated than a mathematical approach that embraces all the effects mentioned. The lack of experimental data for a large number of mixtures over a wide range of temperatures and pressures has held back the development of the appropriate mathematical methods, but a promising new line of approach has been suggested by Schlünder [128] (cf. [139]).

This method takes into account the diffusion that takes place in the layer of liquid close to the wall. Some simplifications are introduced, including the assumption that no fundamental difference in the evaporation mechanism exists between that on the bubbles of the pure substances and that on those of binary mixtures. The following equation has thus been derived:

$$\frac{\alpha_{id}}{\alpha} = 1 + \frac{\alpha_{id}}{\dot{q}} \left(\frac{\partial T_{sat}}{\partial \tilde{x}_1} \right) (\tilde{y}_1 - \tilde{x}_1) \cdot \left(1 - \exp \left(- B_0 \frac{\dot{q}}{\rho_f \beta_f \Delta h_v} \right) \right) \quad (26)$$

where β_f is the mass transfer coefficient and B_0 is a parameter that Schlünder has approximately equated to unity but can be adapted to accommodate experimental results¹⁰. Values calculated from Eqn (26) have been plotted in Fig. 24. They reproduce fairly faithfully the deterioration in heat transfer that was determined experimentally on the SF₆/R 13 B1 system even at high pressures (cf. the crosses entered in the two upper diagrams shown in Fig. 24). This is despite the fact that only the one fixed numerical value of the matching parameter ($B_0/\beta_f = 10^4$ s/m) was taken for both pressures and all the values of heat flux. If a function of \dot{q} and p^* were to be substituted for B_0/β_f , the measured values would be more faithfully reproduced (see [61] for other systems).

Eqn (26) has been formally applied to multicomponent systems and further simplified by the following substitution [170]:

$$\frac{\partial T_{sat}}{\partial \tilde{x}_1} (\tilde{y}_1 - \tilde{x}_1) = \sum_{i=1}^{i=n-1} (T_{sat,n} - T_{sat,i}) (\tilde{y}_i - \tilde{x}_i) \quad (26a)$$

For binary mixtures, T_1 and T_2 can be inserted in Eqn (26a) for the saturation temperatures of the pure components, i.e. $T_{sat,n} = T_2$ and $T_{sat,i} = T_1$. If they are determined at the same reference pressure, the values given in Table 3 and Fig. 25 can be obtained for the term B_0/β_f in Eqn (26) for binary systems with differently spaced boiling points [171; 173]. The result generally obtained at the highest pressure, i.e. $B_0 \approx 1$, obviously confirms the assumption that was made in deriving the model for Eqn (26) and the numerical value that was inserted in Eqn (26) for β_f , i.e. $\beta_f = 2 \times 10^{-4}$ s/m.

10) Only the order of magnitude, viz. $(1-5) \cdot 10^{-4}$ m/s, is known for β_f , as has been deduced from absorption measurements [139]. Hence the quotient B_0/β_f can also be regarded as a fitting parameter.

Table 3. Spacing of boiling points of various systems and numerical values for the ratio B_0/β_f adopted as a fitting parameter in Eqn (26) for $\dot{q} = 20\,000$ W/m² and for three reduced pressures [171; 173]

System	Spacing of boiling points K	B_0/β_f (10 ⁴ s/m) at		
		$p^* = 0.1$	0.5	0.9
R 22/R 115 ^a	2.8	≥ 5	2.8	~ 0.5
SF ₆ /R 13B1	6.0 ^b	^c	^c	~ 0.5
SF ₆ /R 22	23.0 ^b	1.2	0.8	~ 0.5
SF ₆ /R 12	34.0 ^b	0.7	0.65	~ 0.5
R 23/R 115	44.0	0.7	0.7	^d
R 22/R 114	44.4	~ 0.5	~ 0.5	~ 0.5

a) Forms an azeotrope.

b) Sulfur hexafluoride is in the solid phase.

c) Eqn (26) or (26a) cannot be adapted to accommodate $\dot{q} > 10\,000$ W/m² (cf. [171]).

d) No experimental data are available for $\dot{q} = 20\,000$ W/m².

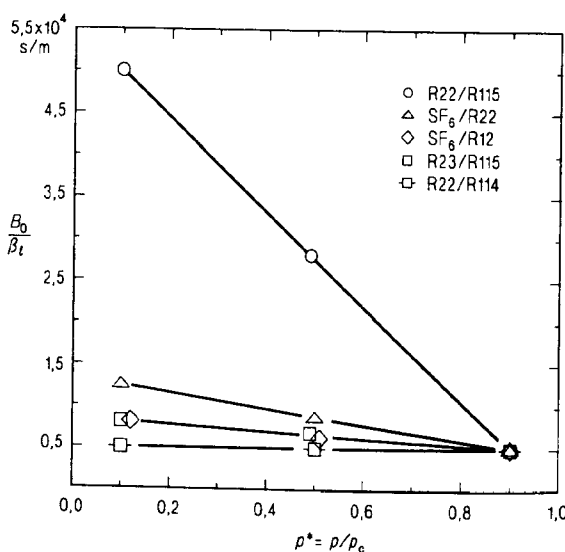


Fig. 25. Relationship between the constant of proportionality B_0/β_f in Eqns (26) and (26a) at $\dot{q} = 20\,000$ W/m² and the reduced pressure for the systems listed in Table 3

If the boiling points are widely spaced (> 30 K), the value of B_0/β_f for moderate and high fluxes remains approximately the same, even in the direction of low pressures. However, it increases if the reference boiling points are closely spaced (cf. Fig. 25), or if they are widely spaced but the heat flux is small [171; 173]. The p , T , x phase diagram for systems with closely spaced ($< \text{ca. } 10$ K) reference boiling points, and particularly for those with azeotropes, is a major factor in Eqn (26) and must therefore be accurately known [135; 174]. A mathematical method that was developed for pure substances [23; 24; 132] and applied to binary mixtures by Bier and Bayer [171; 175] is just as good as Eqn (26) in reproducing the experimental results obtained on the systems listed in Table 3 [173].

4. Maximum heat flux during nucleate boiling

According to Kutateladze [103] and Zuber et al. [104–106], the maximum heat flux \dot{q}_{crit} in nucleate boiling (point B in Fig. 2) can be calculated from

$$\dot{q}_{\text{crit}} = K_1 \Delta h_v \rho_v^{0.5} (\sigma (\rho_l - \rho_v) g)^{0.25} \quad (27)$$

where Δh_v is the enthalpy of vaporization, σ is the surface tension, and ρ_v and ρ_l are the densities of the vapour and the liquid respectively.

From a theoretical study on a flat plate for which some simplifications were introduced, Zuber derived a numerical value of 0.13 for the factor K_1 . By bringing the calculations into line with measured values, Kutateladze obtained numerical values of $K_1 = 0.13$ to $K_1 = 0.16$.

Eqn (27) is valid for saturated and slightly subcooled liquids and for flat plates or plain tubes as heating elements. Values for water derived from it have been plotted in Fig. 26. The difference in density between the vapour and the liquid $\rho_l - \rho_v$ is approximately related to the surface tension σ by $\sigma \sim (\rho_l - \rho_v)^{1/4}$ and is thus actually represented in Eqn (27) to the power of 1.25.

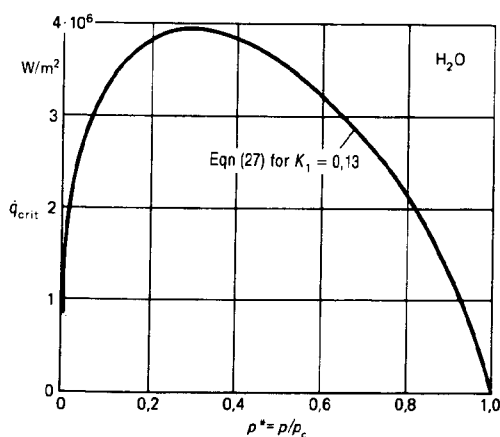


Fig. 26. Maximum heat flux in the nucleate boiling of water as determined from Eqn (27)

The only other properties to be determined for the solution of the equation are the enthalpy of vaporization and the vapour density.

Many equations have been submitted in the literature for the calculation of the maximum heat flux in nucleate boiling. Most of them contain the terms $\Delta h_v \rho_v^{0.5}$ and $(\rho_l - \rho_v)^m$ in which the exponent m is given by $1.25 < m < 1.5$ [103–110].

Of all the factors that have been investigated up to now for their effect on \dot{q}_{crit} , the pressure is the one to which most attention has been devoted. Studies by Borishansky [125] and Mostinsky [126] indicate that the interpolation of existing experimental data can give rise to the following function, which is analogous to Eqn (8) for the relationship between the pressure and the heat transfer coefficient in nucleate boiling:

$$\dot{q}_{\text{crit}}/\dot{q}_{\text{crit},0} = 2.8 p^{*0.4} (1 - p^*) \quad (28)$$

where $\dot{q}_{\text{crit},0}$ is the value for the maximum heat flux at a reference pressure $p_0^* = 0.1$.

Values of \dot{q}_{crit} calculated from Eqn (28) as functions of pressure are compared with experimental values in Fig. 27. It can be seen that the function lies between the centre and the lower limit of the range of experimental scattering. The curve of \dot{q}_{crit} for water, as determined from Eqn (27), has been included in the diagram in the form of a dashed line. In the $10^{-3} \leq p^* \leq 0.6$ range of pressure, it deviates by less than 5% from the values determined from Eqn (28), but at $p^* = 0.9$, the difference becomes +50%. In other words, the absolute values of \dot{q}_{crit} determined from Eqn (27) are considerably higher than very many of the experimental values in the range of high reduced saturation pressures. This applies particularly to organic liquids [12; 34; 41; 112; 113]. For this reason, it is suggested that the relationship of \dot{q}_{crit} to pressure be calculated from Eqn (28) and that the reference value $\dot{q}_{\text{crit},0}$ be determined from experimental data obtained at pressures close to p_0^* or from Eqn (27) for p_0^* .

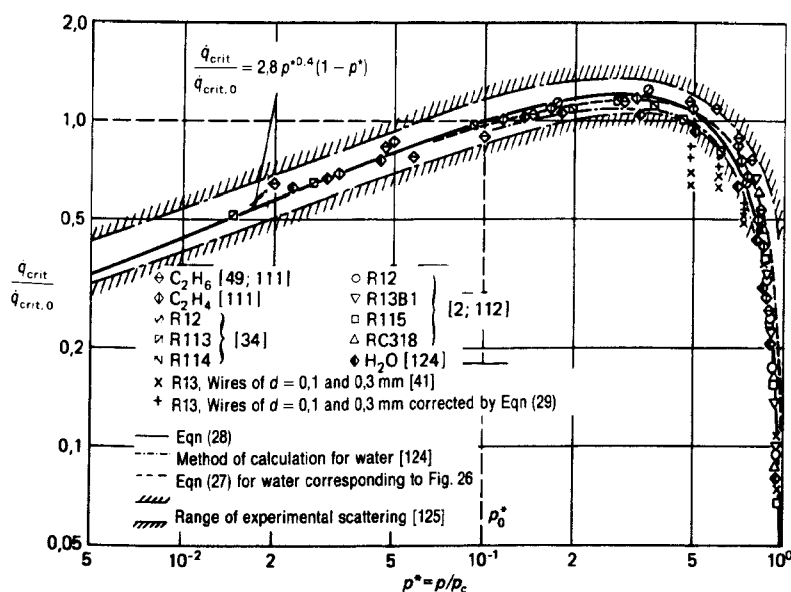


Fig. 27.

Relationship between the relative pressure and \dot{q}_{crit} as determined from Eqns (27) and (28) and from the results of experiments in which the values measured have been referred to $p^* = 0.1$. If this pressure does not lie within the range of measurements, the values determined must be referred to the lowest pressure in the experiments [34] or to $p^* \approx 0.9$ [2; 41].

Labuntsov [114] demonstrated that, at very low reference pressures, the relative relationship to pressure is noticeably less pronounced than would be presumed from Eqns (27) and (28). According to the results of his measurements on water and ethanol at heated surfaces of various geometries and properties, the relationship is similar to that for liquids or fused metals, which can be described by $\dot{q}_{\text{crit}} \sim p^{*m}$ where $0.15 < m < 0.20$. Consequently, the exponent 0.4 in Eqn (28) is somewhat too high for very low reduced pressures. An indication to this effect¹¹ is given on the left edge of Fig. 27.

If the numerical value for the characteristic dimension of the heated surface L from which the maximum heat flux is obtained is of the order of $\{\sigma/(\rho_l - \rho_v)g\}^{0.5}$, the correction proposed by Lienhard and Dhir [110] is

$$\dot{q}_{\text{crit}, L} = K_2 \dot{q}_{\text{crit}, 27} \quad (29)$$

where

$$K_2 = 1.19(L)^{-0.25}, \quad \text{and} \quad L = L/(\sigma/(\rho_l - \rho_v)g)^{0.5} \quad (29a)$$

and $\dot{q}_{\text{crit}, 27}$ is the maximum heat flux as determined from Eqn (27).

Eqn (29) is valid for the following range:

$$0.1 \leq L \leq 2 \quad (29b).$$

The correction factor does not apply if $L > 2$, in which case $K_2 = 1$. Hence examples of the upper limits for the correction of Eqn (27) by Eqn (29) are

$L_{\text{max}} = 5 \text{ mm}$ for water at $p = 1 \text{ bar}$ ($p^* = 0.0045$) and $L_{\text{max}} \approx 1 \text{ mm}$ for Refrigerant R 12 at $p = 20 \text{ bar}$ ($p^* = 0.5$).

The dimensions represented by L are the radius for spheres and horizontal cylinders and the height for fins. The effect of the correction factor K_2 in Eqn (29) is indicated in Fig. 27 by the shift in the results of measurements on thin wires [41] at $p^* = \text{const.}$ (a shift from points marked by a cross to points marked by a plus sign). It can be seen that the correction reduces the systematic deviations in the results obtained for thin wires.

Many other factors that affect the maximum heat flux in nucleate boiling are not embraced by Eqns (27)–(29). Examples are the roughness of the heated surface and potential differences in liquid flow [115], the type of heating [34], and impurities and the wetting of the heated surface [113]. The results of studies on these effects are sometimes contradictory, with the consequence that they cannot be expressed in mathematical terms.

On the whole, therefore, all that can be derived from Eqns (27)–(29) is a more or less fair approximation for the maximum heat flux in nucleate boiling.

Typical values measured for the maximum heat flux \dot{q}_{crit} in nucleate boiling of *binary mixtures* have been

11) For this reason, it could be recommended that the pressure range be divided into two [127], e.g.
 $(\dot{q}_{\text{crit}}/\dot{q}_{\text{crit},1}) = A_1 p^{*0.4}(1 - p^*)$ for $p^* \geq 0.1$, and
 $(\dot{q}_{\text{crit}}/\dot{q}_{\text{crit},2}) = A_2 (p^{*0.2} + p^{*0.5})$ for $p^* \leq 0.1$.

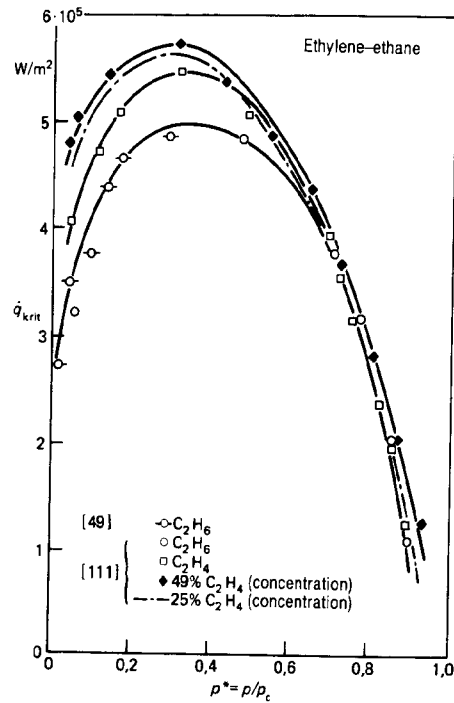


Fig. 28. Maximum heat flux as a function of the reference pressure for ethane, ethylene, and two of their mixtures [111]

plotted in Fig. 28. They lie between or above the corresponding values for the pure substances to an extent that depends on the properties of the mixture and the pressure [111]. In analogy to the deterioration in heat transfer, the highest values of \dot{q}_{crit} frequently occur at intermediate concentrations (cf. the curve through the points marked \bullet in Fig. 28)¹². The same phenomena as those cited in Section 3.2 as being responsible for the deterioration in heat transfer (i.e. the added resistance to heat transport caused by diffusion, the increase in the energy required for bubble formation, and lower bubble densities) retard the coalescence of bubbles to form a film and thus provide a qualitative explanation.

Eqn (27) can be taken to calculate \dot{q}_{crit} for binary mixtures. The reservations are the same as those for the individual components, and the values to be inserted for the properties are those of the mixture. The same recommendation has been submitted by Stephan and Preußer [28], whose measured values for \dot{q}_{crit} at near-atmospheric pressure always lay between those for the pure components. If the values of \dot{q}_{crit} for the mixture should happen to be higher, Eqn (27) yields an estimate that is more likely to lie on the safe side.

5. Film boiling

Stable, free or forced convection film boiling with due allowance for the effects of radiation has been investigated in great theoretical and experimental detail by

12) A trend that can be recognized in Fig. 28 is that the relative difference becomes particularly great at high reduced pressures. It also follows from the results of the experiments on boiling mixtures plotted in Fig. 24 [129].

Bromley [116; 117]. The result of his studies on free convection is the following implicit relationship between the heat transfer coefficient in film boiling, the coefficient α_{cond} for heat transport purely by conduction, and the coefficient α_{rad} purely for radiation through the film of vapour [116]:

$$\alpha = \alpha_{\text{cond}}(\alpha_{\text{cond}}/\alpha)^{1/3} + \alpha_{\text{rad}} \quad (30).$$

His rough explicit equation for the $0 < \alpha_{\text{rad}} < 10 \alpha_{\text{cond}}$ range, which is of interest in engineering, is

$$\alpha = \alpha_{\text{cond}} + \alpha_{\text{rad}} \left[\frac{3}{4} + \frac{1}{4}(1 + 2.62 \alpha_{\text{cond}}/\alpha_{\text{rad}})^{-1} \right] \quad (30a).$$

A more exact calculation by *Roetzel* [118] led to an equation that yields somewhat higher values for the heat transfer coefficient than those obtained from Eqn (30a), viz.

$$\alpha = \alpha_{\text{cond}} + \alpha_{\text{rad}} \left[\frac{4}{5} + \frac{1}{5}(1 + 3 \alpha_{\text{cond}}/\alpha_{\text{rad}})^{-1} \right] \quad (30b).$$

The difference is 6.7% if $\alpha_{\text{rad}} \ll \alpha_{\text{cond}}$; and a bare 1%, if $\alpha_{\text{rad}} = 10 \alpha_{\text{cond}}$.

The coefficient of heat transfer for the radiation component α_{rad} is obtained from Eqn (1) by substituting \dot{q}_{rad} for \dot{q} and from the correlation $\dot{q}_{\text{rad}} = C_{12}(T_w^4 - T_1^4)$ given in Chapter Ka. The transport of heat by conduction through the vapour film is given by

$$\alpha_{\text{cond}} = K_f(L\Delta T)^{-1/4} \left[\frac{\lambda_v^3 \rho_v \Delta h \Delta \rho g}{\eta_v} \right]^{1/4} \quad (31)$$

where

L is the characteristic dimension for the heater element ($L = d$ for a horizontal tube and $L = H$ for a vertical element).
 λ_v is the thermal conductivity; ρ_v , the density, and η_v , the dynamic viscosity of the vapour – referred in each case to the mean temperature of the vapour film $T_m = 0.5(T_w + T_f)$.

$\Delta \rho = \rho_l - \rho_v$ is the difference in density and

$\Delta h = h_v - h_l$ is the difference in enthalpy between the vapour and the liquid.

The numerical value for the empirical constant K_f for a horizontal heater element is 0.62 ± 0.04 [116]; and for a vertical wall, 0.8 [118].

The correlation $\alpha \sim (L\Delta T)^{-1/4}$ in Eqn (31) has been confirmed in the high reduced pressure range by measurements at low values of ΔT ($\alpha_{\text{rad}} \rightarrow 0$) on heater elements of 0.1 and 8 mm diameter [2; 119]. The relationship of the heat transfer coefficient to pressure in film boiling is described by the various properties embraced by the final collective term in Eqn (31). Various pointers indicate that it is very slight, e.g. the exponent is small, opposing trends cancel out, and differences occur over a wider range of temperatures. Exceptions arise for heater elements of very small diameter (Fig. 2 in [2]) and in measurements on carbon dioxide from the triple to the critical point [120].

An equation derived by *Pitschmann* and *Grigull* [121] allows the coefficient of heat transfer in film boiling on thin wires to be calculated. It is an extension of *Bromley's* theory and has been well confirmed by experiment [41; 113; 120].

A mathematical method that has been proposed for film boiling of binary mixtures on vertical heater

surfaces [122] embraces the effect of diffusion on the temperature at the surface of the film and on the heat transfer coefficient (cf. [123]).

The lower limit in stable film boiling (point D in Fig. 2) is of subordinate interest in engineering. It has been described in a number of mathematical studies that have been reviewed in the literature [41; 113] but give rise to results that differ widely from one another and from measured values [112; 113]. It can be derived from measurements at elevated reduced pressures ($p^* \geq 0.6$) that the ratio of the maximum heat flux \dot{q}_{crit} in nucleate boiling to the minimum heat flux \dot{q}_{min} in film boiling is roughly independent of pressure and the properties of the liquid and the vapour [34; 45; 112]. Thus

$$\dot{q}_{\text{crit}}/\dot{q}_{\text{min}} \approx 8 \pm 1.5 \quad (32).$$

Together with Eqns (27) and (28), this correlation allows a first approximation for \dot{q}_{min} .

6. References

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