

UNIVERSITY OF THE WITWATERSRAND, JOHANNESBURG

SCHOOL OF MECHANICAL ENGINEERING

LATENT HEAT TRANSPORT AND MICROLAYER EVAPORATION
IN NUCLEATE BOILING

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Doctor of Philosophy

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(ii)

DECLARATION BY CANDIDATE.

I, Harald Hans Jawurek hereby declare that this thesis is my own work and that the material presented herein has not been submitted for any degree at any other university.

H. H. Jawurek

ABSTRACT

Part 1 of this work provides a broad overview and, where possible, a quantitative assessment of the complex physical processes which together constitute the mechanism of nucleate boiling heat transfer. It is shown that under a wide range of conditions the primary surface-to-liquid heat flows within an area of bubble influence are so redistributed as to manifest themselves predominantly as latent heat transport, that is, as vaporisation into attached bubbles.

This and related findings are applied in the derivation of a new pool boiling heat transfer correlation. The correlation allows the prediction of boiling curves (q/A versus ΔT) at any pressure provided that one boiling curve for the same surface-liquid combination is available as a reference.

Part 2 deals in greater detail with one of the component processes of latent heat transport, namely microlayer evaporation. A literature review reveals the need for synchronised records of microlayer geometry versus time and of normal bubble growth and departure. An apparatus developed to provide such records is described. High-speed ciné interference photography from beneath and through a transparent heating surface provided details of microlayer geometry and an image-reflection system synchronised these records with the bubble profile views. Results are given for methanol and ethanol boiling at sub-atmospheric pressures and at various heat fluxes and bulk subcoolings. In all cases it is found that microlayers were of sub-micron thickness, that microlayer thinning was restricted to the inner layer edge (with the thickness elsewhere remaining constant or increasing with time) and that the contribution of this *visible* evaporation to the total vapour flow into bubbles was negligible. The observation of thickening towards the outer microlayer edge, however, demonstrates that a liquid replenishment flow occurred simultaneously with the evaporation process.

The effects of such replenishment are investigated by means of a numerical analysis of microlayer heat transfer. The analysis indicates that the *net* evaporation from the continuously replenished microlayers was a significant contributor to the total bubble growth. The optical results of the present study are thus reconciled with the deductions of published surface-thermometry studies.

The forces acting on the observed bubbles are analysed and are found to be compatible with a replenishment mechanism.

Microlayer interferometry studies founded on published portions of the present study are discussed and controversial results analysed.

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NOMENCLATURE

Nomenclature separate from or additional to that given below is defined in Appendices 2, 18 and 19. Primary units are shown against the symbols; multiples or submultiples of the units may appear in the text.

a	constant; Chapter 6: interval characterising Gaussian distribution ($2a = d\lambda/\lambda_0^2$)
A	heat transfer area, m^2
A_i	area of bubble influence, m^2
A_{NC}/A_T	fraction of total area supporting natural convection
b	constant
c	constant; Appendix 18: heat capacity, $kJ/kg\ K$
C_1, C_2, C_3	constants
C, C_L	heat capacity (of liquid), $kJ/kg\ K$
C_c, C_{sf}	constants
d	thickness; microlayer thickness, m
d_0	initial microlayer thickness, m
d_c	diameter of cylindrical increment of bubble volume, m
d_{max}	maximum microlayer thickness, m
d_s	base diameter of spherical cap increment of bubble volume, m
d'	optical thickness ($= nd$), m
d''	microlayer thickness given by 2nd maximum of reflectance, m
d^*	displacement thickness of laminar boundary layer, m
D_c	diameter of bubble base contact area, m

D_{cmax}	maximum value of D_c , m
D_d	equivalent spherical bubble diameter at departure, m
$\bar{D}_d$	mean value of D_d , m
D_i	diameter of area of bubble influence, m
D_{max}	maximum diameter of bubble, m
$\bar{D}_{max}$	mean value of D_{max} , m
D_s	equivalent spherical diameter, m
D_w	bubble diameter at maximum bubble width (Fig. 7.7), m
$\exp()$	$e()$
E	linear enlargement (size on print/life size)
f	bubble departure frequency, s^{-1}
$\bar{f}$	mean value of f , s^{-1}
$\bar{V}_d$	mean volumetric vapour flow per bubble source, m^3/s
$f()$	function
g	gravitational acceleration, m/s^2
g_e	gravitational acceleration on earth, m/s^2
G_d	superficial mass velocity of vapour departing from surface, $kg/s\ m^2$
Gr	Grashof number = $L^3 \rho^2 \beta g \Delta T / \mu^2$
h	heat transfer coefficient, $W/m^2\ K$
h_c	height of cylindrical increment of bubble volume, m
h_s	height of spherical cap increment of bubble volume, m
H	height of bubble, m
i	counter: 1, 2, 3 ...
I	intensity of illumination, $J/s\ m^2$; Appendix 7: electric current, A

Ja	Jakob number = $\rho_L C_L \Delta T_{\text{sat}} / \rho_V \lambda$
Ja*	modified Jakob number = $\rho_L C_L T_{\text{sat}} / \rho_V \lambda$
k	thermal conductivity, W/m K
K, K*	constants
K_D, K_H, K_V	constants
$\ln$	natural logarithm
log	logarithm to base 10
L	characteristic length, m; Appendix 8 : length of fin, m
m	Chapter 2: constant characterising surface-liquid combination; Chapter 6: order of interference; Appendix 8: function defined by equation A8.2
M	molecular mass, kg/kmol
n	refractive index (of glass); constant or number of bubble sources present on area A
n_o	refractive index of vapour
n_1	refractive index of liquid (microlayer)
n_2	refractive index of SnO_2 film
n_f	refractive index of thin film
N	number of active nucleation sites per unit heater area, m^{-2}
N'	number of bubble sources (including coalescence effects) per unit heater area, m^{-2}
N_o	Chapter 2: constant; Chapter 3: number of potentially active nucleation cavities of mouth radius r_{max} per unit heater area, m^{-2}
N_{inst}	instantaneous value of N, m^{-2}

N_R	dimensionless heat flux defined by equation 2.43
Nu	Nusselt number = hL/k
p	function defined by equation 6.21
P	pressure, N/m^2
P_{crit}	critical pressure, N/m^2
P_g	buoyancy (gravity) force per unit area, N/m^2
P_i	inertia force per unit area, N/m^2
P_s	surface tension force per unit area, N/m^2
P_v	viscous force per unit area, N/m^2
P_s^0	vapour pressure corresponding to T_s , N/m^2
P_{sat}^0	vapour pressure corresponding to T_{sat} , N/m^2
ΔP	pressure difference (between vapour in bubble and liquid at infinity), N/m^2
Pr	Prandtl number = C_p/k
q	rate of heat transfer, W
q_B	mean q due to evaporation into rising bubbles, W
q_{BWE}	mean q due to evaporation at the walls of attached bubbles, W
q_C	mean q by convection from thermal layer into liquid bulk, W
q_{DENs}	mean q due to recondensation of attached bubbles, W
q_L	mean q from heating surface into adjacent liquid, W
q_{LH}	mean q due to evaporation at all surfaces of attached bubbles (latent heat transport), W
q_{LHvis}	portion of q_{LH} due to visible vapour detachment from surface, W
q_{ML}	mean q due to evaporation of microlayers, W

q_{PSE}	mean q due to free evaporation at pool surface, W
q_T	mean total rate of heat transfer from heater into adjacent fluid phases, W
q_V	mean q from heating surface directly to vapour within attached bubbles, W
q^i	mean rate of heat transfer per area of bubble influence (due to a particular heat transfer process), W
q_{BB}^i	hypothetical q^i by conduction over area actually occupied by bubble base, W
q_{BC}^i	q^i from thermal layer to bulk due to bubble-induced convection, W
q_{BWE}^i	q^i due to evaporation from side walls and dome of attached bubbles, W
$q_{CONDENS}^i$	q^i due to recondensation of attached bubbles, W
q_L^i	q^i from heating surface into adjacent liquid, W
q_{LA}^i	q^i from heater into liquid annulus surrounding bubble, W
q_{LH}^i	q^i due to evaporation at all surfaces of attached bubbles, W
q_{LHvis}^i	portion of q_{LH}^i due to visible detachment of vapour, W
q_{LW}^i	q^i from heater to adjacent liquid during waiting period, W
q_{LWA}^i	$= q_{LW}^i + q_{LA}^i$
q_{ML}^i	q^i due to microlayer evaporation, W
q_{ML}^{i*}	q^i due to apparent microlayer evaporation, W
q_T^i	q^i (total) from heater to adjacent fluid phases, W
q_V^i	q^i from heater directly to vapour within attached bubbles, W

q^*	internal heat generation per unit time and volume, W/m^3
q/A	heat flux, W/m^2
$(q/A)_{BC}$	heat flux, referred to total heater area, due to bubble-induced convection ($= Nq_{BC}^i$), W/m^2
$(q/A)_c$	critical (burnout) heat flux, W/m^2
$(q/A)_{LH}$	heat flux, referred to total heater area, due to latent heat transport ($= Nq_{LH}^i$), W/m^2
$(q/A)_{LHvis}$	heat flux, referred to total heater area, due to visible latent heat transport ($= Nq_{LHvis}^i$), W/m^2
$(q/A)_{ML}$	heat flux, referred to total heater area, due to microlayer evaporation ($= Nq_{ML}^i$), W/m^2
$(q/A)_{NB}$	heat flux, referred to total heater area, due to nucleate boiling ($= Nq_T^i$), W/m^2
$(q/A)_{NC}$	natural convection heat flux, W/m^2
$(q/A)_T$	total heat flux, W/m^2
$(q/A)_{T-NC}$	$= (q/A)_T - (q/A)_{NC}$, W/m^2
$(q/A)_{ML}^2$	mean local heat flux due to microlayer evapora- tion at thermocouple site, W/m^2
$(q/A)_{NB}^i$	modified $(q/A)_{NB}$ defined by equation 2.60, W/m^2
$(q/A)^i$	mean heat flux over area of bubble influence, W/m^2
$(q/A)_{LH}^{V_{max}}$	modified latent heat flux defined by equation 2.2, W/m^2
Q	energy, heat, J
$(Q/A)_{TD}$	heat conducted from unit heater area during temperature drop, J/m^2
ΔQ	heat transferred in time t , J
ΔQ_i	heat transferred from one area of bubble influence in time t , J

$(\Delta Q/A)_i$	heat transferred from unit area within area of bubble influence in time t , J/m^2
r	Chapters 2, 3, Appendix 4: mouth radius of potentially active nucleation cavity, m; Chapters 4 to 9, Appendix 14: bubble base radius, m
r_{max}	Chapters 2, 3, Appendix 4: mouth radius of largest potentially active cavity, m; Chapters 4 to 9: maximum bubble base radius, m
$r_{d''}$	bubble base radius corresponding to d'' , m
$r_{d''max}$	maximum value of $r_{d''}$, m
r_s	mouth radius of nucleation cavity for which N is unity, m
r^*	mouth radius of active nucleation cavity, m
r_{min}^*	mouth radius of smallest active cavity, m
r_{max}^*	mouth radius of largest active cavity, m
R	bubble radius; Chapter 4: radius of hemispherical bubble; Chapter 9: equivalent spherical bubble radius, m
R	universal gas constant = 8,314 kJ/kmol K
R_c	critical bubble radius, m
R_d	radius of bubble at departure, m
R_i	radius of area of bubble influence, m
Re	Reynolds number = $vL\rho/\mu$
Re_b	bubble Reynolds number
Re_x	Reynolds number for flow in x direction over flat plate
s	distance from $x = 0$ to evaporating microlayer face, m
t	time, s

t_g	time of bubble growth to reach radius R , s
t_w	waiting period (time between departure of one bubble and nucleation of next bubble), s
$t_{w(1)}$	primary waiting period (see Figure 8.1), s
$t_{w(2)}$	secondary waiting period (see Figure 8.1), s
t^*	time of growth to reach maximum bubble base diameter, s
T	temperature, $^{\circ}\text{C}$
T	absolute temperature, K
T_b	bulk liquid temperature, $^{\circ}\text{C}$
T_{in}	initial temperature of heater, $^{\circ}\text{C}$
T_o	temperature at root of fin, $^{\circ}\text{C}$
T_r	reduced temperature ($= T/T_{\text{critical}}$)
T_s	temperature of evaporating surface, $^{\circ}\text{C}$
T_s	absolute temperature of evaporating surface, K
T_{sat}	saturation temperature of liquid, $^{\circ}\text{C}$
T_{sat}	absolute saturation temperature of liquid, K
T_w	heater wall temperature, $^{\circ}\text{C}$
$T_{w(\text{mean})}$	mean heater wall temperature, $^{\circ}\text{C}$
T_{∞}	temperature of liquid at infinity ($= T_b$), $^{\circ}\text{C}$
ΔT	temperature difference, K
ΔT_{sat}	wall superheat ($= T_{w(\text{mean})} - T_{\text{sat}}$), K
ΔT_{sub}	bulk subcooling ($= T_{\text{sat}} - T_b$), K
v	velocity, m/s
V	bubble volume, m^3
V_d	volume of bubble at departure, m^3
V_e	volume of vapour released (evaporated) into bubble up to point of departure, m^3

$V_{\max}$	maximum volume of bubble, m^3
V_v^m	vapour volume resulting from complete evaporation of one stagnant microlayer, m^3
V_{LE}^*	apparent volume of microlayer liquid evaporated, m^3
V_{VE}^*	volume of vapour due to apparent microlayer evaporation, m^3
w	mass rate of evaporation per unit area, $kg/s\ m^2$
W_B	power dissipated over boiling surface, W
W_M	measured power input to heater, W
x	Chapter 2: perpendicular distance from heater wall into liquid; Chapter 4: distance along flat plate; Chapter 9: distance in direction of conduction (from $x = 0$ within glass); Appendix 7: distance (from root) along length of fin, m
$\bar{x}$	optical path difference ($= 2d'$), m
y_0	half of fin thickness, m

GREEK SYMBOLS

α	thermal diffusivity ($= k/\rho C$), m^2/s
β	coefficient of cubic expansion, K^{-1} ; Chapter 2: bubble contact angle (through liquid), rd
δ	thickness of thermal layer, m
δ_c	thickness of thermal layer when $R = R_c$, m
δ_d	thickness of thermal layer when $R = R_d$, m
ϵ_h	eddy diffusivity of heat transfer, m^2/s
ϵ_m	eddy diffusivity of momentum transfer, m^2/s
η	evaporation coefficient
κ	contrast of interference fringes

(xxx)

κ_c	contrast with respect to imperfect collimation
κ_m	contrast with respect to imperfect monochromaticity
κ'	overall contrast
λ	latent heat of vaporisation, kJ/kg; Chapter 6: wavelength of light in medium of refractive index of unity, m
λ_o	wavelength of light at peak of energy distribu- tion, m
λ'	wavelength of light in medium of refractive index n ($= \lambda/n$), m
$d\lambda$	bandwidth of energy distribution (light source), m
Λ	function defined by equation 3.29
ρ	density, kg/m ³ ; Chapter 6: reflectance ($= I_{\text{reflected}}/I_{\text{incident}}$)
ρ_o	reflectance of lower surface of heater
ρ_1, ρ_2	reflectance of light and dark fringes
ρ_3	reflectance of bubble surroundings
ρ'	overall reflectance
σ	surface tension, J/m ² ; Chapter 6: reciprocal wavelength ($= 1/\lambda$), m ⁻¹
σ_o	Chapter 6: reciprocal peak wavelength ($= 1/\lambda_o$), m ⁻¹
σ'	$\sigma - \sigma_o$, m ⁻¹
τ	optical transmittance
τ_{max}	peak transmittance (of filter)
ϕ	function defined by equation 2.45
ψ	function defined by equation 3.27; Chapter 6: angle subtended by pinhole at collimator, rd

SUBSCRIPTS AND SUPERSSCRIPTS

1, 2, 3	subscripted physical properties refer to regions 1, 2, 3
i	area of bubble influence
inst	instantaneous
L	liquid
T	total
V	vapour
-	(bar over symbol) arithmetic mean value
∞	infinite or 'semi-infinite' medium

CHAPTER 1INTRODUCTION1.1 BACKGROUND AND HISTORY OF THESIS

The work reported in this thesis was started in 1963, when the author, supported by a bursary from the South African Atomic Energy Board, joined the boiling research programme of the Department (now School) of Mechanical Engineering at this University.

The programme had been initiated by Dr C J Rallis in 1956, with the general aim of taking a fundamental, but broad and common-sense look at the mechanism of nucleate pool boiling heat transfer. Experimental studies had been conducted, using modest apparatus, on the heat transfer rates and on the bubble frequencies, volumes and concentrations during nucleate pool boiling at atmospheric pressure from immersed wire heaters. The most important results of this work were published by Rallis, Greenland and Kok [1]^{*} in 1961. It was shown that in saturated boiling the latent heat transport, that is, the heat flow due to evaporation into bubbles up to their point of departure, was a major contributor to the total boiling heat transfer. This finding was completely contrary to the views generally accepted at the time.

It was decided that the author would continue this work and would extend measurements to subatmospheric and to slightly elevated pressures. To this end a more elaborate

^{*}References are, for ease of location, listed at the end of the thesis, after the appendices.

apparatus was developed, with E Bayer, who had since joined the team, taking the lead in its construction.

Simultaneously, a further set of data, previously recorded and roughly analysed by Rallis and two undergraduate students, S Hyman and D J Carliell, was re-analysed by the author and used as a vehicle for theoretical work. The results were presented by Rallis and Jawurek in 1964, both in the journal literature [2] and, in modified form, at the Third United Nations Conference on the Peaceful Uses of Atomic Energy [3]. The studies, firstly, confirmed the previous latent heat transport findings, secondly, clarified certain aspects of the interrelation of bubble frequency and departure volume, and thirdly, presented new ideas on nucleation and surface characterisation. This work is submitted as a part of this thesis. Since some aspects of it have been confirmed by subsequent workers, and others have been extended in this thesis, it is presented in review form at appropriate points in the text. The original study in its entirety is given in an appendix.

Several years later this work was taken up again and its chief conclusions applied to the problem of correlating and predicting heat transfer rates in nucleate pool boiling. The derivation and testing of the correlation is reported fully in this thesis. An abbreviated version was presented at the Johannesburg Heat Transfer Conference of 1974 [4].

During the construction and testing of the new apparatus mentioned earlier, the author became increasingly interested in one of the several heat transfer processes which contribute to latent heat transport, namely microlayer evaporation. The microlayer phenomenon, that is, the formation of thin liquid films beneath the bubble base and their evaporation into the bubble interior, had been quite recently discovered and, at the time, constituted one

of the least understood aspects of boiling. It was decided that the author would conduct an optical study of microlayer thickness, and that Bayer would continue the latent heat transport measurements alone.

In 1966 the author joined the Engineering Division of the South African Atomic Energy Board at Pelindaba, and it was there that the microlayer study was conducted as one of the contributory projects of the 'Pelinduna' Reactor Concept [5]. The microlayer thickness measurements required fairly advanced optical apparatus, high-speed ciné facilities and film-processing services. These were all built up from scratch at Pelindaba. In this and the subsequent experimental work the author was assisted by D J Maritz, a part-time engineering student. Experimental runs were completed early in 1968. At that stage the Pelinduna Concept had just been abandoned and the author assumed other full-time duties. The analysis of the microlayer data, the derivation of the heat transfer correlation mentioned earlier and the writing up of this thesis had thus to be conducted in spare time.

1.2 STRUCTURE AND OUTLINE OF THESIS

This thesis falls naturally into two parts. Part 1, occupying Chapters 2 and 3, takes an overall view of the mechanism of nucleate boiling and established that the group of heat transfer processes known collectively as latent heat transport, firstly, is a chief contributor to the total boiling heat transfer, and secondly, forms a useful basis for the correlation of such heat transfer. With the importance of latent heat transport thus established, Part 2, occupying the remaining chapters, takes a detailed view of one of the component processes of latent heat transport, namely microlayer evaporation.

Chapter 2 attempts, by way of an interpretive review of published work (including the author's work presented in Appendix 2), to provide a broad outline of the mechanism of nucleate pool boiling heat transfer. The various primary surface-to-liquid heat transfer processes within an area of bubble influence are identified and are shown to manifest themselves partly as latent heat transport and partly as bubble-induced convection. Of these the latent heat transport is shown to predominate under a wide range of conditions, and the remaining heat transfer, due to the intermittent bubble-induced convection, is shown to approximate to that of undisturbed natural convection.

These findings are applied in Chapter 3 to the problem of correlating and predicting heat transfer rates during saturated pool boiling. A new correlation results, which allows the prediction of boiling curves, q/A versus ΔT_{sat} , at any pressure, provided that at least one boiling curve for the same surface is available as a reference.

Chapter 4 provides a review of previous microlayer research. This points to a need for synchronised records, at various conditions of normal boiling, of: (1) microlayer shape and thickness versus time, and (2) overall bubble growth and detachment. Unambiguous records of microlayer geometry should, it is concluded, be obtainable by high-speed cine interference photography from beneath and through a transparent heating surface.

Chapter 5 describes the boiling apparatus which was built up to permit such measurements. A number of uncommon techniques employed in the preparation of the heating surface are given in some detail.

Chapter 6 provides, in its opening sections, an analysis of the optics of interference by amplitude division at a microlayer in contact with the heating surface used. The

analysis firstly determines the ranges of liquid refractive index, and thus the test liquids, that give fringes of good visibility, and secondly, dictates certain requirements of the optical apparatus. The remainder of the chapter describes and analyses the complete optical apparatus.

Chapter 7 describes experimental procedure and data processing.

Chapter 8 presents the experimental results. Microlayers are shown to be of sub-micron thickness and the contribution of visible microlayer evaporation (thinning) is found to be negligible. Both findings are apparently at variance with the deductions of previous microthermometry-based studies. Evidence is, however, presented of a microlayer liquid replenishment occurring simultaneously with the evaporation process.

These findings are analysed and discussed in Chapter 9. It is shown that in the presence of such liquid replenishment the net microlayer evaporation can be a significant contributor to latent heat transport. Thus the findings of the present study are reconciled with those of the microthermometry studies. The forces acting on bubbles are evaluated and buoyancy is identified as the force tending to lift the bubble continuously off its microlayer and thus to cause the liquid inflow.

Chapter 10 presents a summary and the conclusions of the study and offers suggestions for future work.

1.3 UNITS

SI units are used throughout this thesis. The treatment follows that recommended by the Bureau of Standards [6], with the following additional features:

All pressures are expressed in newtons per square metre (N/m^2). The name pascal (Pa) has recently been recommended for this unit. Since the pressure data in this work have already been changed once from the previously recommended bar (10^5 N/m^2), and the newton per square metre is legitimate within the SI and recalls the meaning of pressure as force per unit area, it is retained in this thesis.

Two units of temperature, the kelvin (K) and the degree Celsius ($^{\circ}\text{C}$) are legitimate within the SI. In the present work temperature levels (points on a temperature scale) are expressed in the familiar degrees Celsius, and temperature differences (driving potentials) in kelvin. The use of separate units for temperature and temperature difference was already advocated by Jakob [7] for purposes of clarifying heat transfer work.

7.

PART 1

LATENT HEAT TRANSPORT

CHAPTER 2THE MECHANISM OF SATURATED NUCLEATE POOLBOILING: AN INTERPRETIVE REVIEW2.1 INTRODUCTION2.1.1 Aims and Scope

The primary aim of this review is to present a coherent description and, where possible, a quantitative assessment of the physical processes which together constitute the mechanism of nucleate pool boiling.

The customary enumeratory 'literature survey' is unlikely to supply such an outline. In its unprocessed state the literature on boiling offers several controversies, conflicting mechanistic concepts and even cases of apparently contradictory experimental results. Much of this conflict, however, appears to be traceable to loose nomenclature, to the extrapolation of experimental conclusions to conditions unrelated to those of the original studies, and to the adherence to preconceived notions. A careful reassessment, an ordering and a conciliatory overall view of the literature would thus appear to be called for. Such treatment will be attempted and will, it is hoped, allow a unified, if rough, outline of the boiling mechanism to emerge.

In view of the complexity of the boiling processes, treatment will be restricted mainly to the simplest mode of nucleate boiling, namely saturated pool boiling in the presence of isolated bubbles. However, several mechanistic ideas, applicable to this restricted field, arose out of studies

into subcooled boiling, forced-convection boiling and high-flux boiling with bubble coalescence; thus some coverage of these cases will be necessary.

The review will generally place more stress on experimental studies than on theory. Nevertheless, since the development of mechanistic concepts was closely coupled with attempts at correlating boiling heat transfer, certain major correlations will be examined in some detail.

The second aim of this chapter is to assess the group of heat transfer processes known collectively as 'latent heat transport'. The assessment is intended, firstly, to correct widely held misconceptions concerning latent heat transport, and secondly, to provide the quantitative background for the derivation (in Chapter 3) of a new boiling heat transfer correlation.

The third aim of this review is to submit in the correct perspective a study on latent heat transport published by Rallis and the author in 1964. Presentation within the review should allow the work to be judged more objectively than in an artificially contrived separate chapter.

The final aim of this review is to direct attention at one particular process contributing to the total boiling heat transfer, namely microlayer vaporisation. The review thus aims to introduce the subject of Part 2 of this thesis.

2.1.2 Presentation and Method

The presentation of the review will be semi-historical, that is, the material to be covered will be broken down into a number of main groups, each reflecting a broad line of mechanistic thinking or of experimental approach; the groups will be treated consecutively and within each group developments will be followed through in roughly historical

order. The breakdown and classification are intended as presentation aids only and will thus be neither rigorous nor detailed. Excessive separation of the material would in any event jeopardise one of the main aims of the review, namely the comparison and reconciliation of results arising from widely different approaches.

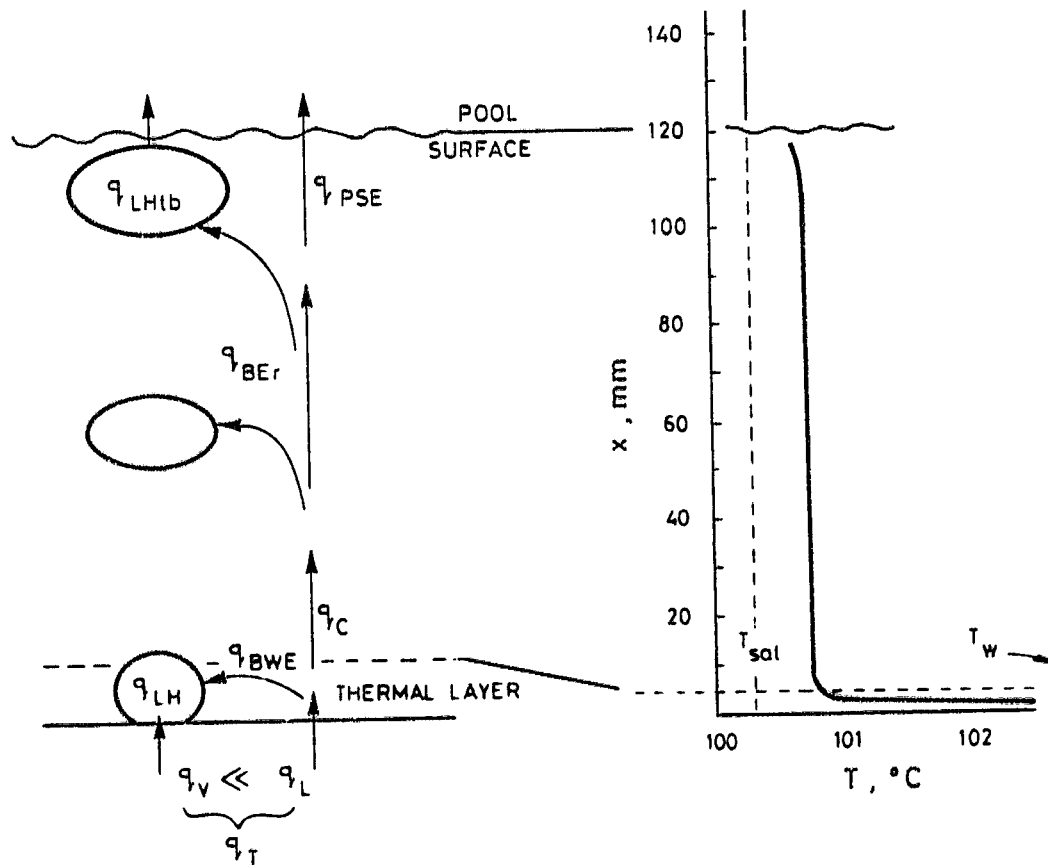
In reviewing experimental work particular attention will be directed at the conditions and the range of parameters covered in each study. This will lead to a rough assessment of the range of validity of each conclusion. In the reviewing of proposed mechanisms and correlations a careful check will be maintained on the compatibility of the theory with the properly assessed experimental conclusions.

All terms and symbols will be clearly defined or, where published ambiguities or contradictions exist, redefined.

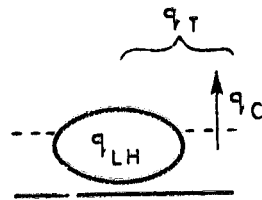
The re-examined mechanistic conclusions will be incorporated into a simple scheme of equations and sketches which will be added to and amended as the review progresses. The final version of this scheme will, it is hoped, contain the required outline of the boiling mechanism.

2.2 INSIGNIFICANT LATENT HEAT TRANSPORT. MICROLEVECTION

The earliest detailed studies of the mechanism of nucleate boiling were conducted by Jakob and his co-workers in the 1930's [1]. These studies indicated that in saturated pool boiling the transfer of energy at the heating surface and its subsequent distribution in the fluid phases takes place as summarised in Figure 2.1 and equation 2.1 below. The quantities q are mean rates of heat transfer from the entire heater area, each identifying a particular transfer path or process. They are interrelated as follows:



(a) DISTRIBUTION OF HEAT FLOW

(c) TEMPERATURE DISTRIBUTION
(water, $(q/A)_T = 43 \text{ kW/m}^2$)

(b) BUBBLE DEPARTURE

FIG. 2.1 DISTRIBUTION OF HEAT FLOW
AND TEMPERATURE IN SAT-
URATED POOL BOILING.
(according to Jakob [7])

$$q_T = q_L + q_V (=negl.) \quad (2.1a)$$

$$= q_{BWE} + q_C \quad (2.1b)$$

$$= q_{LH} + q_C \quad (2.1c)$$

$$= q_{LH} + q_{BER} + q_{PSE} \quad (2.1d)$$

$$= q_{LHtb} + q_{PSE} \quad (2.1e)$$

where:

q_T (T = TOTAL) is the total heat transferred per unit time from the entire heater area into the fluid phases

q_V (V = VAPOUR) is the heat transferred from the heating surface directly into the vapour within bubbles; it is considered negligibly small compared with q_T

q_L (L = LIQUID) is the heat transfer from the heating surface into the liquid where it establishes a thin thermal layer (see Fig. 2.1c); it is essentially equal to q_T

q_{BWE} (BWE = BUBBLE WALL EVAPORATION) is that portion of q_L which is transferred from the bubble surroundings (mainly the thermal layer) to the walls of attached bubbles, there resulting in vapour evolution

- q_{LH} (LH = LATENT HEAT) is the heat flow associated with vapour evolution at all surfaces of attached bubbles; it is commonly called 'latent heat transport'; according to the present description it is equal to q_{BWE} ; in saturated boiling it may be measured from the mass flow rate of vapour visibly detaching from the heater (see Fig. 2.1b)
- q_C (C = CONVECTION) is that portion of q_L not involved in evaporation into attached bubbles; it is transported from the thermal layer into the liquid bulk by convection and is responsible for the bulk superheat (see Fig. 2.1c)
- q_{BER} (BER = BUBBLE EVAPORATION, rising) is that portion of q_C which results in evaporation into rising bubbles
- q_{PSE} (PSE = POOL SURFACE EVAPORATION) is the remaining portion of q_C which results in free evaporation from the pool surface
- q_{LHtb} (LHtb = LATENT HEAT, terminal bubbles) is the heat flow associated with bubbles bursting at the pool surface.

Jakob and his co-workers conducted extensive photographic studies of bubble dynamics in low-flux saturated boiling. These showed that the volumes of terminal bubbles at the pool surface far exceeded those of bubbles detaching from the heater (i.e., in the terminology of equation 2.1: $q_{LHtb} \gg q_{LH}$). This implied that q_{BER} far exceeded q_{LH} and thus that q_C far exceeded q_{LH} . In other words, the heat transport from the thermal layer to the bulk was dominated by convection rather than by evaporation into attached bubbles (i.e. latent heat transport). The stirring action of bubbles, Jakob argued, provided convective

velocities sufficient to account for the increased heat transfer coefficients in boiling.

It must be stressed that Jakob's observations were limited to low heat fluxes, that is, to surfaces sparsely populated by bubbles. Under such conditions the heat transfer over the entire surface must of necessity be dominated by convection. This limitation was overlooked by most subsequent workers. They thus extrapolated Jakob's findings to all conditions of boiling and attributed generality to the notions of insignificant latent heat transport and of the predominance of the bubble-induced convection mechanism.

Similar mechanistic conclusions were reached in two influential studies of subcooled boiling.

In the first study, that of Gunther and Kreith of 1949 [8], water was boiled at such levels of subcooling that bubbles grew and collapsed without detaching from the heater. High-speed cine records provided data on bubble frequency, f , maximum bubble volume, $V_{\max}$, and the number of bubble sources per unit area, N . From these, in an effort to evaluate the latent heat transport contribution, the following heat flux was determined

$$(q/A)_{\text{LH}}^{\text{V}_{\max}} = \rho_V \lambda N f V_{\max} \quad (2.2)$$

Under typical conditions (e.g. $(q/A)_T = 10^6 \text{ W/m}^2$, $\Delta T_{\text{sub}} = 96 \text{ K}$) this flux accounted for a mere 2 per cent of the total heat flux. The heat flux due to latent heat transport, that is, due to vapour evolution at the walls of attached bubbles, is given by

$$(q/A)_{\text{LH}} = \rho_V \lambda N f V_e \quad (2.3)$$

where V_e is the volume of vapour released into a typical attached bubble. Clearly, if $(q/A)_{\text{LH}}^{\text{V}_{\max}}$ is to measure latent

heat transport $(q/A)_{LH}$, then V_{max} , the maximum visible bubble volume, must be equal to V_c . In subcooled boiling these volumes can be equal only if bubble growth and recondensation (collapse) proceed strictly sequentially. Gunther and Kreith's bubbles, however, protruded well into the subcooled liquid bulk (see Fig. 2.2), thus, in principle, permitting simultaneous evaporation at the bubble base and condensation towards the apex. The authors considered the possibility of such mass transfer, but argued that their bubbles were surrounded by essentially stagnant liquid films and that under such conditions the mass transfer mechanism could be neglected. Thus they concluded that $(q/A)_{LH}^{V_{max}}$ was an adequate measure of $(q/A)_{LH}$ and hence that latent heat transport was a negligible contributor to the heat transport from the thermal layer to the bulk. This transport they attributed to liquid 'microconvection' excited by the rapid growth and collapse movement of bubbles.

In a second and similar subcooled-boiling study Rohsenow and Clark in 1951 [9] presented an analysis of McAdams' [10] high-speed cine records of forced-convection boiling. Under the conditions of these tests bubbles detached, grew for a short distance from the heater and then collapsed. Using several approximations, Rohsenow and Clark showed that the latent heat flux due to the formation of bubbles of maximum volume was negligible in comparison with the total flux. Thus the latent heat flux due to visible detachment of bubbles (of departure volume V_d), that is, the following heat flux:

$$(q/A)_{LHvis} = \rho_V \lambda N f V_d \quad (2.1)$$

was also negligible. Rohsenow and Clark apparently took this quantity to be a measure of the total latent heat transport as given by equation 2.3. They thus reached the same mechanistic conclusion as Gunther and Kreith. The possibility of simultaneous evaporation into and condensation from attached bubbles, although reiterated by Zmola [11], received no thorough consideration.

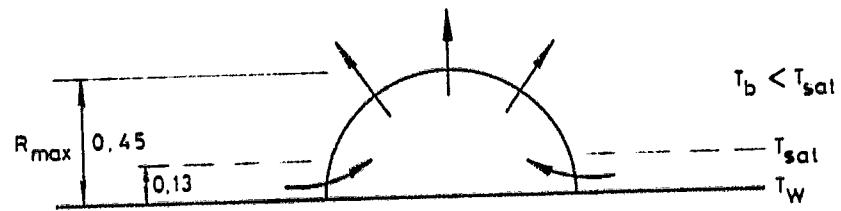


FIG 2.2 MASS TRANSFER THROUGH ATTACHED BUBBLES IN SUBCOOLED BOILING.
 (Dimensions, in mm, refer to Gunther and Kreith's experiment [8])

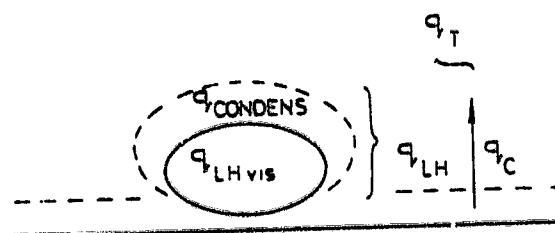


FIG.2.3 BUBBLE DETACHMENT IN SUBCOOLED BOILING

In the presence of such mass transfer through attached bubbles the heat transport from the thermal layer to the bulk takes place as summarised in Figure 2.3 and equation 2.5 below (cf Fig. 2.2 and equation 2.1):

$$q_T = q_{LH} + q_C \quad (2.5a)$$

$$= q_{LHvis} + q_{CONDENS} + q_C \quad (2.5b)$$

where:

$q_{CONDENS}$ (CONDENS = CONDENSATION) is the mean rate of heat transfer due to recondensation of attached bubbles, and

q_{LHvis} (LHvis = LATENT HEAT, visible) is the remaining portion of q_{LH} which detaches visibly from the surface as vapour.

Rohsenow and Clark's conclusion concerning the predominance of the microconvection mechanism is thus based on the assumption that the term $q_{CONDENS}$ is negligible.

In 1956 Snyder [12] and Edwards [13] presented detailed postulates on the mechanism of mass transfer through attached bubbles. According to these the term $q_{CONDENS}$ is not only far from negligible, but, at high heat fluxes, the chief contributor to the total heat transfer in subcooled forced-convection boiling.

Dealing first with evaporation, Snyder and Edwards proposed that this takes place largely from the surface of a thin liquid film existing between the bubble base and the heating surface. Such a 'microlayer', they argued, is formed owing

to the rapid initial growth of the bubble and the non-slip condition at the solid-liquid interface. The distribution of heat flow near the surface, with microlayers present, is summarised in Figure 2.4 and equation 2.6 below:

$$q_T = q_L + q_V (= \text{negl.}) \quad (2.6a)$$

$$= q_{ML} + q_{BWE} + q_C \quad (2.6b)$$

$$q_{ML} + q_{LH} + q_C \quad (2.6c)$$

where:

q_{ML} (ML = MICROLAYER) is the mean rate of heat transfer due to evaporation from the microlayers.

In Snyder and Edwards' view $q_{ML} \gg q_{BWE}$ and hence $q_{ML} \approx q_{LH}$.

Next, Snyder demonstrated the flow of latent heat from microlayers through bubbles. He applied Plesset's equation [14] for the steady one-dimensional vapour flow between surfaces at different temperatures to a hypothetical, square and flat-topped bubble. On the assumption of an evaporation coefficient (see [15, 16]) of unity, he showed that a temperature difference of a mere 2 K between the microlayer surface and the bubble top is sufficient to maintain a latent heat flux of the order of 10^7 W/m² through the bubble. In forced-convection subcooled boiling, with the bubble apex extending well into the turbulent core, such a temperature difference should be maintained with ease. Thus, he argued, at high heat fluxes the latent heat flow evolved at the microlayers and transferred through bubbles could exceed by orders of

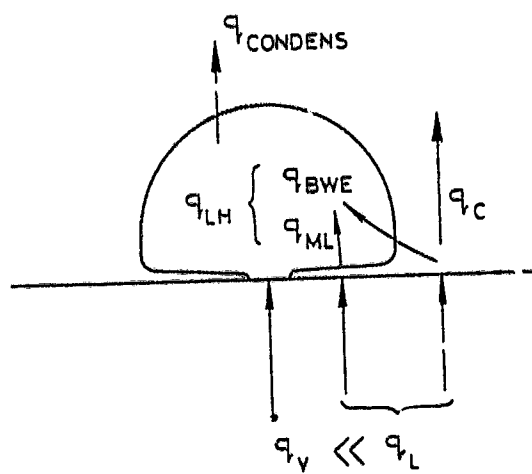


FIG. 2.4 BREAKDOWN OF HEAT FLOW; MICROLAYER PRESENT.

magnitude the visible latent heat flow due to bubbles of maximum volume.

No evidence for the existence of microlayers was available in 1956 and thus Snyder's proposals were largely ignored. Even in the absence of both microlayers and forced convection, however, it would seem conceivable that a significant latent heat flow should occur between the lower side walls of bubbles and their apexes (see q_{BWE} and $q_{CONDENS}$ in Fig. 2.4).

It may be seen then, that the evidence supporting the concept of negligible latent heat transport was questionable for the case of subcooled boiling and incomplete for the case of saturated boiling. Nevertheless, the concept gained ground and by extrapolation, coupled with loose semantics, was widely applied to all cases of boiling. The resultant concept of bubble-induced convection as the predominant boiling heat transfer mechanism thus became the general basis for further work.

2.2.1 Early Heat Transfer Correlations Based on Microconvection

Bubble-induced convection, in its earliest interpretation, was seen as an agitation process inducing turbulence in the vicinity of the heating surface, similar to that existing in single-phase forced convection. This interpretation led to the derivation of a number of pool boiling heat transfer correlations having the general form

$$Nu = f(Re, Pr) \quad (2.7)$$

but with the Nusselt and Reynolds groups formulated in terms of bubble parameters. The choice of different bubble dimensions

and velocities and the use of different expressions for these quantities resulted in a wide variety of correlations (see reviews [17, 18, 19]). Among these the correlations of Rohsenow [20], Forster and Zuber [21] and of Kutateladze and Mishenco (see [22]) have been of lasting popularity.

For a particular pressure and for a particular liquid all equations of this type may be reduced to the form

$$(q/A) = a \Delta T_{\text{sat}}^b \quad (2.8)$$

where a is a constant characterising the surface-liquid combination and b is a constant under all conditions.

The derivation of correlations and the disadvantages inherent in equation 2.8 are illustrated by the equation of Rohsenow [20]. The dimensionless groups in equation 2.7 were defined as follows:

$$\text{Nu} = \{(q/A)D_d\} / (\Delta T_{\text{sat}} k_L) \quad (2.9)$$

$$\text{Re} = G_d D_d / \mu_L \quad (2.10)$$

where G_d , the superficial mass velocity of vapour departure from the surface, was assumed proportional to $(q/A)\lambda$, and D_d , the mean diameter of bubbles at departure was given by the equation of Fritz [23]

$$D_d = \text{const.} \beta \{\sigma/g (\rho_L - \rho_V)\}^{1/2} \quad (2.11)$$

By fitting the dimensionless groups to pool boiling data [24, 25] covering a wide range of pressures Rohsenow established his correlation as follows:

$$\text{Re Pr}_L / \text{Nu} = C_{sf} \text{Re}^{0.33} \text{Pr}_L^{1.7} \quad (2.12)$$

that is

$$\frac{C_L \Delta T_{sat}}{\lambda} = C_{sf} \left[\frac{(q/A)}{\mu_L \lambda} \left\{ \frac{\sigma}{g(\rho_V - \rho_L)} \right\}^{\frac{1}{2}} \right]^{0.33} \left[\frac{C_L \mu_L}{k_L} \right]^{0.7} \quad (2.13)$$

or (cf equation 2.8)

$$(q/A) = 1/C_{sf} \cdot f(\text{physical properties}) \cdot \Delta T_{sat}^3 \quad (2.14)$$

where the constant C_{sf} (incorporating the bubble contact angle θ) was dependent on the surface-liquid combination. The exponent of ΔT_{sat} (constant b in equation 2.8) was invariable at 3 (other correlations of the same type gave values of 2 to 4).

Except for minor deviations at subatmospheric pressure [26] Rohsenow's equation satisfactorily correlated the effect of pressure on pool boiling heat transfer, thus meeting one of its primary aims. The correlation of surface effects, however, was found to be severely limited by the invariable exponent of ΔT_{sat} . Experimentally the exponent could be varied between 3 and 25 on one and the same heater merely by changing its surface finish, that is, its nucleation properties [27 - 33]. Adjustment of the constant C_{sf} was thus recognised as insufficient for surface characterisation. Clearly all correlations reducible to the form of equation 2.8 were similarly limited.

This finding, together with experimental studies into the interrelation between heat flux and nucleation site concentration [28, 34, 35] led to the derivation of modified microconvection correlations containing nucleation terms. Kurihara and Myers [28] for example, using dimensionless groups as defined by Rohsenow [20], arrived at the following improved correlation:

$$(q/A) = \text{const.} (\rho_V/\mu_L)^{0,33} \text{Pr}_L^{-0,89} N^{0,33} \Delta T_{\text{sat}} \quad (2.15)$$

which for a particular fluid at a particular pressure may be reduced to

$$(q/A) = a \Delta T_{\text{sat}}^b N^c \quad (2.16)$$

Equation 2.15 and others reducible to the form of equation 2.16 (see also Section 2.4) were tested by Hall and Hatton [31] against their own data on several liquids boiling from heaters of varying surface finish. Correlation was not really satisfactory. The dependence of flux on nucleation site concentration (i.e. the 'constant' c in equation 2.16) was itself found dependent on surface finish.

Evidently, the forced-convection analogy produced no entirely satisfactory heat transfer correlations. Its search for adequate expressions describing bubble nucleation, growth and detachment did, however, encourage a large number of fundamental bubble studies of lasting importance.

2.3 FURTHER PROPOSALS ON BUBBLE-INDUCED CONVECTION

Below are summarised a number of further theoretical proposals on the nature of bubble-induced convection. All are based on the assumption that such convection is the sole mode of heat transport from the thermal layer to the liquid bulk, the latent heat transport contribution being negligible. The proposals range from correlation-orientated analogies, similar in approach to the forced-convection analogy, to detailed postulates on the liquid flow in the vicinity of single bubbles.

According to the early natural-convection analogy of Nishikawa and Yamagata [36] the stirring action of bubbles in pool boiling may be interpreted as an enhancement of

natural convection. A heat transfer correlation was thus developed having the general form

$$\text{Nu} = \text{const.} (\text{Gr.Pr})^n \quad (2.17)$$

where bubble parameters were used to evaluate the dimensionless groups. The final correlation is reducible to the form of equation 2.8 and subject to the limitations already discussed.

Zuber [37] proposed a more detailed natural-convection analogy for the specific case of pool boiling from a horizontal surface in the region of isolated bubbles. With such boiling, he argued, the liquid circulation in the pool is essentially the same as that existing above a horizontal surface in turbulent natural convection. He showed that the equations of Malkus and of Thomas and Townsend (see [37]) for the heat transfer coefficient in single-phase turbulent natural convection can be made to apply in isolated-bubble pool boiling provided that a mean fluid density is defined that takes account of the vapour hold-up. Owing to the inclusion of a nucleation site concentration term the final correlation obtained is reducible (for a particular liquid at a particular pressure) to equation 2.16.

Zuber considered his mechanism to be applicable only in the region of isolated bubbles. At higher heat fluxes, with the onset of bubble coalescence, he proposed that the convective transport ceases and that latent heat transport becomes operative.

A correlation-approach roughly related to that of Zuber's natural convection analogy is embodied in Tien's stagnation flow model of pool boiling [38]. According to this model columns of rising bubbles induce a flow pattern above the heating surface similar to that in steady axisymmetrical stagnation flow, but in the reverse direction (see Fig. 2.5). Existing stagnation flow solutions were modified to derive a

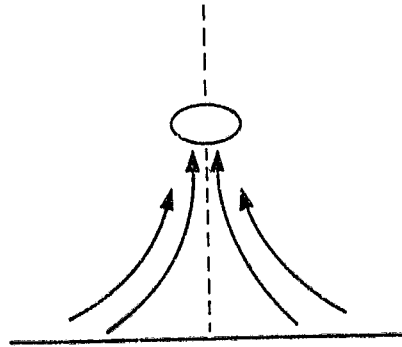


FIG. 2.5 TIEN'S STEADY STAGNATION FLOW MODEL [38]

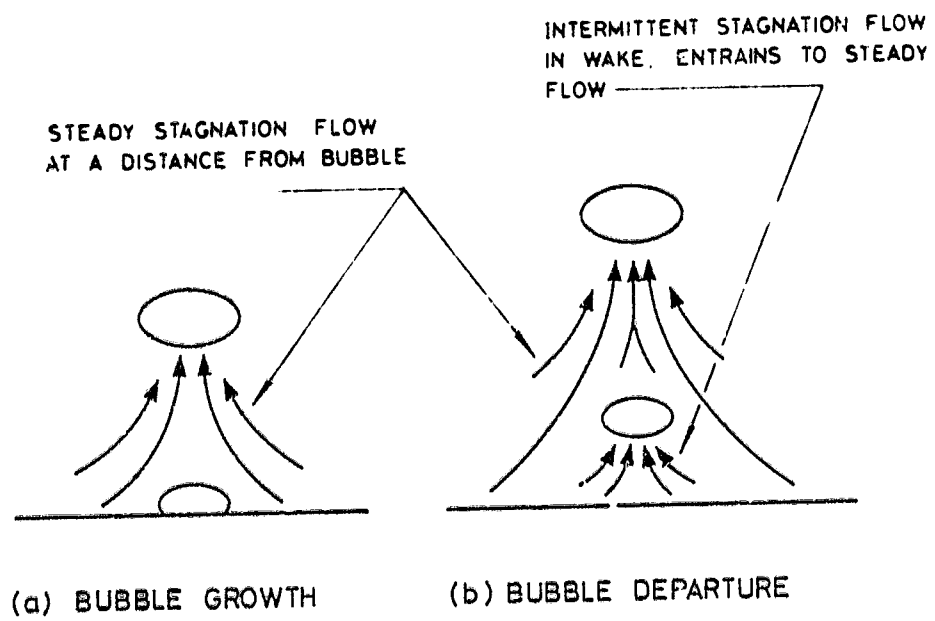


FIG. 2.6 BOEHM AND LIENHARD'S MODIFIED STAGNATION FLOW
MODEL [40]

heat transfer correlation reducible to the form of equation 2.16. The model ignored all intermittent flow effects and was later retracted by Tien himself [39].

In Boehm and Lienhard's modified model [40] local intermittent stagnation flows, due to bubble departure, are superimposed on Tien's steady stagnation flow (see Fig. 2.6). The transient stagnation flows, it was argued, are entrained rapidly to the steady flow and can be approximated by transient conduction cycles. The two mechanisms were assumed to contribute equally to the total boiling heat transfer. The resulting interrelation of heat flux, temperature difference and nucleation site concentration was of the form of equation 2.16.

Finally we discuss the 'vapour-liquid exchange' mechanism originally proposed by Forster and Greif in 1959 [41] and, under the new name of 'bulk-convection' mechanism, analysed in detail by Han and Griffith in 1965 [42]. This mechanism describes a bubble as acting as a liquid pump: the bubble of negligible latent heat content pushes up a portion of the thermal layer into the cooler bulk; after bubble detachment or collapse an intruding slug of cooler liquid makes contact with the surface, is heated by transient conduction, the next bubble then nucleates and the cycle is repeated.

Comments on the merits of the mechanisms discussed above will be given at various points later in this chapter.

The widely accepted bulk convection mechanism will now be treated in more detail. First, however, it is convenient to introduce the concept of area of bubble influence.

2.3.1 Area of Bubble Influence

The concept of 'area of bubble influence' may be stated as

follows: A bubble, in growing and departing from a nucleation site, has associated with it an 'area of influence', that is, a limited portion of the heating surface the heat transfer from which is influenced by the action of the bubble; outside the areas of influence the heating surface is totally unaffected by bubble action and, in the case of pool boiling, supports natural convection.

The first experimental evidence of bubble influence as being limited in range appears to have been given by Hsu and Graham in their schlieren-photography study of 1961 [43]. Their photographs, although difficult to interpret, do show that a bubble in departing leaves behind it an area of reduced thermal layer thickness. The diameter, D_i , of this area (for their horizontal strip heater) appears to be roughly equal to two bubble departure diameter, D_d . Subsequent studies (see Section 2.4) obtained values of D_i ranging up to $3D_d$.

It should be noted that neither Hsu and Graham's study nor subsequent schlieren studies compared the thermal layer thickness outside the areas of influence with that occurring in pure (non-boiling) natural convection at the same temperature difference. Thus, while there is evidence of intense bubble influence in the immediate vicinity of a nucleation site, there is no evidence of the absence of long-range influence. The pool circulation set up by rising bubbles must to some extent intensify the convection over the 'non-boiling' portions of the surface. For these areas treatment along the lines of Zuber's natural convection analogy [37] would seem more appropriate.

Nevertheless, the concept of a limited area of bubble influence is extremely useful as a working hypothesis. It will be adopted in this chapter and will be partially justified in Section 2.6 and Chapter 3.

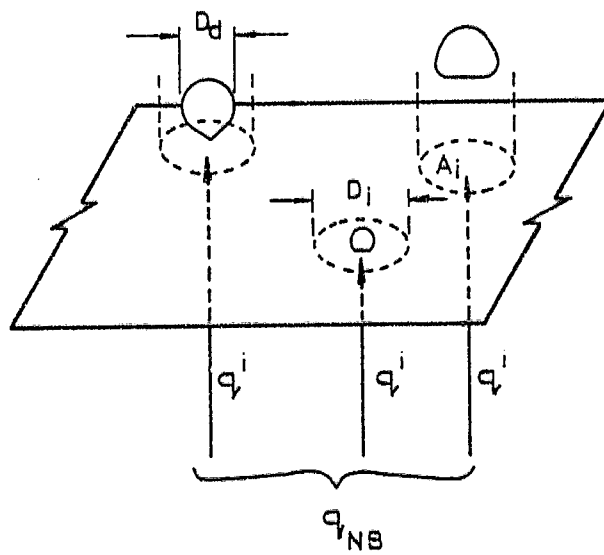


FIG. 2.7 AREA OF BUBBLE INFLUENCE, A_i

Thus the total heat flux from a surface supporting nucleate boiling may be separated as follows:

$$(q/A)_T = (q/A)_{NB} + (q/A)_{NC} (A_{NC}/A_T) \quad (2.18)$$

where $(q/A)_{NB}$ is the heat flux (referred to total area) due to nucleate boiling alone, that is, due to areas of influence and is given by

$$(q/A)_{NB} = Nq^i \quad (2.19)$$

where q^i is the mean rate of heat transfer per area of bubble influence (see Fig. 2.7); $(q/A)_{NC}$ is the heat flux due to natural convection when it alone is operative and (A_{NC}/A_T) is the area fraction available for such undisturbed natural convection.

2.3.2 The Bulk Convection Mechanism of Han and Griffith

Han and Griffith [42] have provided the most detailed analysis of heat transfer by bulk convection within an area of bubble influence. Since their work has found wide acceptance and since certain of its aspects will be required later in this chapter, the analysis is examined in some detail below.

Han and Griffith described the process of bulk convection from a horizontal surface as follows: At the instant of bubble departure (stage 1 in Fig. 2.8) the bubble of negligible latent heat content totally strips away the thermal layer over its entire area of influence, that is, a circular area of assumed diameter $2D_d$. At the same time ($t = 0$) liquid from the main bulk, at temperature T_∞ , rushes to the heating surface ($x = 0$), the temperature of which is invariable at T_w ; the thermal layer then reforms by pure conduction (stage 2); after time t_w (the waiting period)

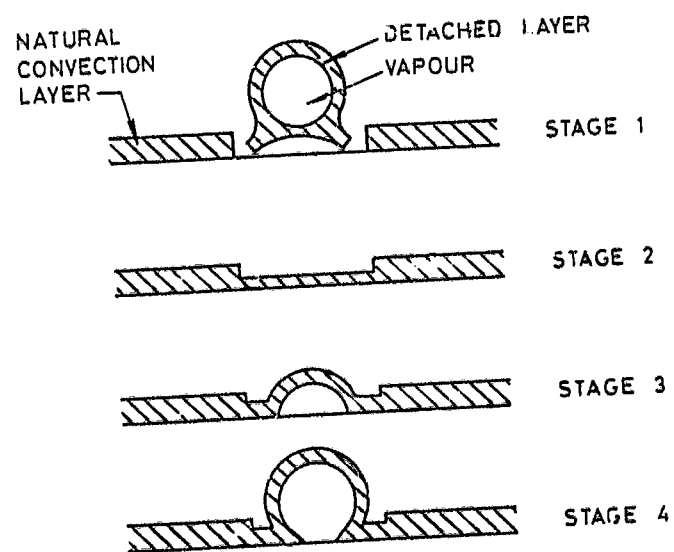


FIG. 2.8 BULK CONVECTION MODEL (Han and Griffith [42])

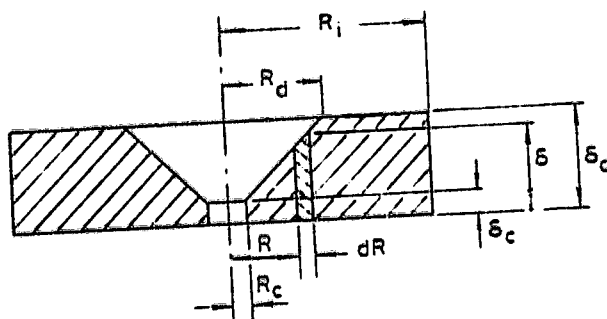


FIG. 2.9 SECTION THROUGH TRANSIENT CONDUCTION LAYER
IN AREA OF BUBBLE INFLUENCE (Han and Griffith [42])

the next bubble nucleates and, in growing, pushes up a section of the thermal layer (stage 3) until the bubble departure size is again reached (stage 4) and the thermal layer is again stripped away and dissipated in the bulk of the liquid.

Each heat transfer cycle from the area of influence into the liquid ($x > 0$) is thus subject to the transient conduction equation

$$\partial T / \partial t = \alpha (\partial^2 T / \partial x^2) \quad (2.20)$$

with initial conditions

$$T = T_w \text{ at } x = 0, t = 0 \quad (2.21a)$$

$$T = T_\infty \text{ at } x > 0, t = 0 \quad (2.21b)$$

and boundary conditions

$$T = T_w \text{ at } x = 0, t > 0 \quad (2.21c)$$

$$T = T_\infty \text{ at } x = \infty, t > 0 \quad (2.21d)$$

With the liquid treated as infinite in the x direction the standard solution of this problem is

$$(T - T_\infty) = (T_w - T_\infty) \operatorname{erfc} \{x / 2(\alpha t)^{1/2}\} \quad (2.22)$$

and

$$(\partial T / \partial x)_{x=0} = - (T_w - T_\infty) / (\pi \alpha t)^{1/2} \quad (2.23)$$

It was assumed that the temperature distribution throughout the thermal layer is linear and is given by equation 2.23. The thickness of this layer, δ , is thus

$$\delta = (\pi \alpha t)^{\frac{1}{2}} \quad (2.24)$$

with the liquid beyond δ uniform at temperature T_{∞} .

The heat transferred in time t from unit area within A_i into the thermal layer is

$$(\Delta Q/A)_i = \int_0^{\infty} (T - T_{\infty}) C_L \rho_L dx \quad (2.25)$$

which by equation 2.22, integration and substitution from equation 2.24 becomes

$$(\Delta Q/A)_i = 2\rho_L C_L (T_w - T_{\infty}) \delta / \pi \quad (2.26)$$

Figure 2.9, reproduced from Han and Griffith's paper, shows the thickness of the transient conduction layer about to be pushed up by the bubble, δ , as a function of bubble radius, R . Thus

$$\delta = \{\pi \alpha (t_w + t)\}^{\frac{1}{2}} \quad (2.27a)$$

$$\delta_c = \{\pi \alpha t_w\}^{\frac{1}{2}} \quad (2.27b)$$

$$\delta_d = \{\pi \alpha (t_w + t_g)\}^{\frac{1}{2}} \quad (2.27c)$$

where δ is the thickness for the general case (with t now measured from the end of the waiting period); δ_c is the thickness reached at the end of the waiting period (at which time a bubble of critical radius, R_c , nucleates); and δ_d is the thickness reached at the time of bubble departure (i.e. after the waiting period, t_w , and the bubble growth period, t_g).

The enthalpy contained in the transient layer for one area

of bubble influence thus follows from equations 2.26 and 2.27 as

$$\Delta Q_i = \int_{R_c}^{R_d} 2\rho_L C_L (T_w - T_\infty) (\delta/\pi) 2\pi R \cdot dR \\ + \pi (R_i^2 - R_d^2) 2\rho_L C_L (T_w - T_\infty) (\delta_d/\pi) \quad (2.28)$$

With $R_c \ll R_d$ and with δ assumed to vary linearly with R equation 2.29 reduces to the following:

$$\Delta Q_i = 2\rho_L C_L (T_w - T_\infty) \{R_i^2 \delta_d^2 - (R_d^2/3)(\delta_d - \delta_c)\} \quad (2.29)$$

At bubble departure this enthalpy is instantaneously mixed into the liquid bulk. The product $f\Delta Q_i$ thus gives the mean rate of heat transfer per area of bubble influence, q_i^1 , and hence the heat flux due to the nucleate boiling process (see equation 2.19) is

$$(q/A)_{NB} = 2\rho_L C_L (T_w - T_\infty) \sum^N \{f[R_i^2 \delta_d^2 - (R_d^2/3)(\delta_d - \delta_c)]\} \quad (2.30)$$

The total heat flux then follows from equation 2.18 as

$$(q/A)_T = (q/A)_{NB} + (1 - \sum^N R_i^2)(q/A)_{NC} \quad (2.31)$$

In the region of isolated bubbles Han and Griffith assumed $R_i = 2R_d$, as already stated; in the region of closely packed bubbles they set $R_i < 2R_d$, suitably corrected for overlap of areas of influence.

For purposes of testing equations 2.30 and 2.31 Han and Griffith performed a carefully controlled subcooled pool boiling run on water at atmospheric pressure in which certain parameters entering into the correlation were measured; the remaining parameters were evaluated from theory developed earlier in the paper. The agreement between measured and

calculated heat fluxes was good. This agreement was taken as evidence of the correctness of the proposed mechanism.

2.3.3 Thermal Layer Recovery by Convection

Stefan [44] has suggested an area-of-influence heat transfer mechanism which, although arrived at independently of Han and Griffith, may be considered an extension of their bulk convection mechanism. According to Stefan the recovery of the thermal layer takes place not by transient conduction but by a series of convection steps: Immediately after bubble departure heat transfer takes place by forced convection with the liquid flow approximately parallel to the heater and towards the nucleation site; this is followed by natural convection during the remainder of the waiting period and then, during the bubble growth period, by further forced convection parallel to the heater but with the flow direction now away from the nucleation site.

Stefan made these ideas the starting point for dimensional analysis and for the derivation of an empirical heat transfer correlation. His mechanism, although having a more realistic feel than that of Han and Griffith has not found as many adherents, possibly because of the absence of formal analysis.

One of Stefan's mechanistic steps, namely the forced convection parallel to the heater and towards the nucleation site, has received further attention from Ruckenstein [45] and from Heled et al [46]. In these studies, however, it was considered the sole contributor to the area-of-influence heat transfer.

2.3.4 The Bulk Convection Correlation of Mikic and Rohsenow

In the work of Mikic and Rohsenow [47] a slight simplification of Han and Griffith's model together with a novel method of

surface characterisation led to a heat transfer correlation having practical engineering value.

The simplification arose from the assumption that the bubble contact area is small compared with the total area of bubble influence. The transient conduction cycles could thus be treated as extending uniformly over the entire area of influence. From equation 2.23 which, with the liquid bulk at saturation temperature, becomes

$$(\partial T / \partial x)_{x=0} = - \Delta T_{\text{sat}} / (\pi \alpha t)^{1/2} \quad (2.32)$$

the instantaneous heat flux throughout A_i is

$$(q/A)_{\text{inst}}^i = k \Delta T_{\text{sat}} / (\pi \alpha t)^{1/2} \quad (2.33)$$

Since the thermal layer is replaced at the frequency of bubble departure, f , the mean heat flux over A_i is

$$(q/A)^i = f \int_0^{1/f} (k \Delta T_{\text{sat}}) / (\pi \alpha t)^{1/2} dt = 2 k \Delta T_{\text{sat}} f^{1/2} (\pi \alpha)^{-1/2} \quad (2.34)$$

Assuming A_i to be $(\pi/4)(2D_d)^2$, as before, and summing over all nucleation sites, N , Mikic and Rohsenow obtained the mean heat flux due to the nucleate boiling process:

$$\begin{aligned} (q/A)_{\text{NB}} &= \pi D_d^2 N (q/A)^i \\ &= 2\pi^{1/2} (k\rho C)_L^{1/2} f^{1/2} D_d^2 N \Delta T_{\text{sat}} \end{aligned} \quad (2.35)$$

Comparison of this equation with Han and Griffith's equation 2.30 shows the advantages of the simplified treatment.

The derivation of the complete correlation, although not strictly relevant to the discussion of the boiling mechanism is summarised below for purposes of later reference.

Equation 2.35 now required relationships for N , f and D_d . Concerning N , Mikic and Rohsenow applied the results of Brown (see [47]), who, using a novel experimental technique (not described), showed that the number of active nucleation cavities (per unit area) of cavity mouth radius larger than $r_{\min}^*$ may be approximated by

$$N = C_1 (r_s / r_{\min}^*)^m \quad (2.36)$$

where r_s , the radius for which N is unity, C_1 , a dimensional constant, and the exponent m , characterise a particular surface-liquid combination. For $r_{\min}^*$, the radius of the smallest active cavity, the relationship of Griffith and Wallis [48] was used:

$$r_{\min}^* = (2 T_{\text{sat}} \sigma) / (\lambda \rho_V \Delta T_{\text{sat}}) \quad (2.37)$$

giving

$$N = C_1 r_s^m (\lambda \rho_V / 2 T_{\text{sat}} \sigma)^m \Delta T_{\text{sat}}^m \quad (2.38)$$

Bubble departure diameter was obtained from the equation of Cole and Rohsenow [49]

$$D_d = C_2 \left[\sigma / \{g(\rho_L - \rho_V)\} \right]^{1/2} \left[(\rho_L C_L T_{\text{sat}}) / (\rho_V \lambda) \right]^{5/4} \quad (2.39)$$

in which C_2 is a liquid-dependent constant, and bubble frequency was obtained from the following relationship, derived by Cole [50]:

$$f D_d = C_3 \left[\sigma g(\rho_L - \rho_V) / \rho_L^2 \right]^{1/4} \quad (2.40)$$

in which C_3 is a liquid-independent constant.

Substitution of equations 2.38, 2.39 and 2.40 into

equation 2.35 led to the correlation

$$(q/A)_{NB} = \frac{C_1 C_2 C_3^{1/2} r_s^m}{\pi^{1/2} 2^{(m-1)}} \left[k\rho C \right]_L^{1/2} \left[\frac{\lambda \rho_V}{\sigma T_{sat}} \right]^m \left[\frac{\sigma g(\rho_L - \rho_V)}{\rho_L^2} \right]^{1/8} \\ \times \left[\frac{\sigma}{g(\rho_L - \rho_V)} \right]^{3/4} \left[\frac{\rho_L C_L T_{sat}}{\rho_V \lambda} \right]^{15/8} \cdot \Delta T_{sat}^{(m+1)} \quad (2.41)$$

This was rearranged to read

$$N_R = B (\phi \Delta T_{sat})^{(m+1)} \quad (2.42)$$

where

$$N_R = (q/A)_{NB} \left[\sigma / \{ g(\rho_L - \rho_V) \} \right]^{1/2} / \left[\lambda \rho_V \right] \quad (2.43)$$

is a dimensionless heat flux;

$$B = (r_s/2)^m (2/\pi^{1/2}) g^{-3/8} C_1 \cdot C_2^{5/3} \cdot C_3^{1/2} \quad (2.44)$$

is a constant depending on the surface-liquid combination,
and

$$\phi^{(m+1)} = \frac{k_L^{1/2} \rho_L^{17/8} C_L^{19/8} \lambda^{(m-23/8)} \rho_V^{(m-15/8)}}{\mu_L (\rho_L - \rho_V)^{9/8} \sigma^{(m-11/8)} T_{sat}^{(m-15/8)}} \quad (2.45)$$

is a group containing physical properties and the surface-liquid-dependent constant m .

The correlation was intended to allow the prediction of boiling fluxes, $(q/A)_{NB}$, versus ΔT_{sat} at all pressures,

provided that at least one set of data of $(q/A)_{NB}$ versus ΔT_{sat} for any one pressure and for the surface-liquid combination of interest, was available as a reference. The reference data were to provide the only unknowns in the correlation equations, namely the nucleation constants B and m .

This procedure was tested by Mikic and Rohsenow for Addoms' data on water (see [51]) and for Cichelli and Bonilla's data on organics [24], all covering a wide range of pressures. With the natural convection portion of the total flux neglected, that is, with $(q/A)_{NB}$ approximated by $(q/A)_T$, good correlation was obtained.

The present section has dealt with theoretical proposals on the mechanism of bubble-induced convection. All mechanisms were based on the assumption that such bubble convection is the predominant mode of heat transfer from the thermal layer to the liquid bulk. The validity of these ideas will now be examined by reviewing direct experimental studies on the bubble-induced convection processes.

2.4 EXPERIMENTAL STUDIES ON BUBBLE-INDUCED CONVECTION

Experimental studies on bubble convection have generally employed either optical techniques (schlieren and shadow-graph photography) or microthermometry in the vicinity of bubbles. The optical studies provided considerable detail on the flow of liquid at different temperatures in the area of bubble influence, but were, by their nature, uninformative on absolute temperature values. The thermometry studies, on the other hand, while generally more quantitative, gave less complete flow descriptions.

2.4.1 Optical Studies

The pioneer schlieren study of Hsu and Graham [43] has already been mentioned. The photographs (or at least the published prints) did not show much flow detail, but led to the valuable concept of an area of bubble influence.

Subsequent schlieren studies [52 - 57], elaborating on the area-of-influence flow, showed that the transport of the thermal layer into the bulk is entirely different in saturated and subcooled boiling. The two cases are discussed separately below.

(a) Saturated Boiling

Saturated boiling of water was studied by Béhar and Séméria [52, 53] using thin-wire heaters, and in considerable detail by Isshiki and Tamaki [56], using thin wires as well as rods of pencil lead as heaters.

These studies revealed that the bubble-induced flow of the thermal layer takes place as shown in Figure 2.10.

Stage 1 in Figure 2.10 shows the thermal layer and above it a decaying wake due to the previous bubble. In stage 2 a bubble has recently nucleated and is entirely enveloped by the thermal layer, while in stage 3 the bubble apex has grown beyond the layer. In stage 4 the bubble has detached and left behind it an area of partially destroyed thermal layer. A cone-shaped wake of superheated liquid (bubble tail) follows the bubble.* Stage 5 shows the partially

*The first observation of the bubble tail is due to Brauer [58] while its shape has apparently been predicted theoretically by Hara [59].

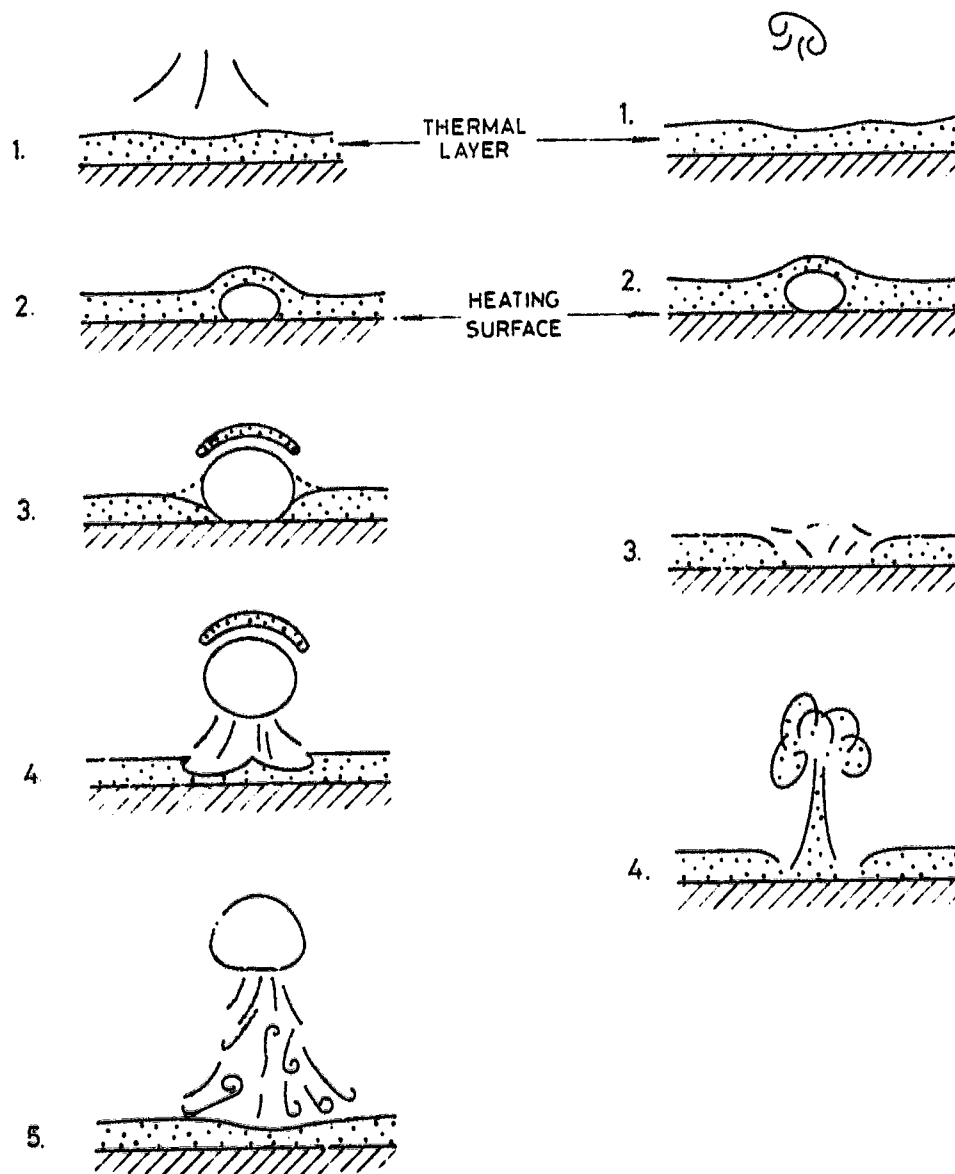


FIG. 2.10

SATURATED BOILING, FORMATION
AND DESTRUCTION OF THERMAL
LAYER

FIG. 2.11

HIGHLY SUBCOOLED BOILING,
BUBBLE COLLAPSE AND
EJECTION OF HOT LIQUID
SLUG.

recovered thermal layer and the eddies which in some cases occur in the wake.

In Béhar and Séméria's observations [52, 53] certain intermittent bubbles in stage 4 appear to push up a detached cap-shaped portion of the thermal layer. (The associated superheat is presumably dissipated partly into the liquid bulk and partly into bubble growth). This mechanism was not detected with bubbles of normal waiting periods, nor at pressures higher than atmospheric.

For saturated boiling then, the process of bubble-induced convection consists of the mixing into the liquid bulk of the superheated wake and, when present, of the superheated cap. Clearly this process, as well as the primary surface-to-liquid heat transfer, differs considerably from that postulated in the bulk-convection mechanism. The thermal layer is not stripped away entirely over the area of bubble influence, nor is it replaced by a liquid slug from the pool bulk. Inflow appears rather to occur parallel to the heating surface. It would thus appear that the bulk-convection models [42, 47] overestimate the effect of bubble convection.

(b) Highly Subcooled Boiling

In subcooled boiling bubble recondensation (collapse) takes place either after bubble detachment or, at very high subcooling, without detachment. Béhar and Séméria [52, 53] studied attached collapse, while Bähr [57] studied collapse within a distance of one bubble diameter from the surface. Both studies dealt with water boiling from thin wires and revealed essentially the same thermal layer flow; see Figure 2.11.

Stages 1 and 2 in Figure 2.11 are similar to the saturated case. The collapse (stage 3) on or near the surface is followed by the ejection of a slug of hot liquid (stage 4)

which apparently possesses a vortex structure [57]. While the slug is dissipated in the liquid bulk the thermal layer recovers. The cycle is then repeated.

The mechanism of slug ejection has apparently not been analysed. Bähr [57] has suggested that the condensation-induced inrush of liquid is asymmetrical and leads to a pressure pulse towards the surface which, on reflection, forms the ejected vortex.

By borrowing from recent findings on cavitation [60] a different and apparently novel explanation of slug ejection may here be suggested. A vapour cavity collapsing on or near a solid surface can become involuted so that a liquid jet is formed on the cavity apex; the jet pierces the vapour, strikes the surface, and is reflected as a slug. Appendix 1 gives a more detailed discussion of the process as well as some estimates of jet velocities under typical conditions of subcooling.

Neither these estimates nor the photographs of Séméria and Bähr allow a quantitative assessment of the contribution of slug ejection to the thermal layer transport or the total area-of-influence heat transfer. It seems conceivable, however, that the contribution is significant.

2.4.2 Microthermometry Studies

Several workers have studied bubble-induced convection by measuring local temperature fluctuations in the vicinity of bubbles [61 - 64]. Fast-response microthermocouples linked to oscilloscopes or high-speed recorders and synchronised high-speed ciné photography gave temperatures as a function of time and of distance from the moving bubble wall.

Bonnet et al [61] and Séméria and Flamand [62] conducted such tests on water boiling on horizontal plates; the pressure was atmospheric and the bulk both saturated and subcooled. Van Stralen and Sluyter [63] dealt with saturated boiling of water and of water-butadiene mixtures from 0,2 mm diameter platinum wires. Taken together, these three studies generally confirmed the findings of the schlieren observations. Thus the flow descriptions (including the incomplete destruction of the thermal layer) were found to be essentially in order. Two additional points were, however, revealed: (1) The superheated caps (see stages 3 and 4, Figure 2.10) were entirely absent, and (2) a portion of the thermal layer piled up near the bubble base, persisted until bubble growth was well advanced (see dotted line, stage 4, Figure 2.10) and was finally removed not by bubble-induced convection, but, according to van Stralen [63, 65], by evaporation into the bubble.

In addition, Séméria and Flamand, in a group of tests at mild subcooling (approximately 4 K), showed that at distances greater than the maximum thermal layer thickness the local temperature varied between a maximum of T_{sat} and a minimum which depended on the distance from the surface. Thus the entrained bubble tails were never superheated. This would again seem to suggest that the bulk convection models [42, 47] overestimate the contribution of bubble convection to the total area-of-influence heat transfer.

A direct attempt at the quantitative assessment of the bubble-convection contribution is found in the study of Jacobs and Shade [64]. Temperature probing, similar to that discussed above, together with high-speed schlieren photography was applied to subcooled boiling (at atmospheric pressure) of incompletely de-aerated, technical grade carbon tetrachloride. The boiling surface was an electrically heated, horizontal platinum ribbon. Tests were limited to

the transient period (1 to 2 s) following the energising of the heater.

It was found that the majority of the bubbles contained superheated vapour during the growth period and remained superheated until collapse was well advanced. Superheating ranged from 0 to 6 K and showed no pattern with heat flux or bulk subcooling. (In the studies discussed above [61 - 63] the vapour was always saturated.) Rising bubbles carried a thin laminar cap of superheated liquid on the upper surface and trailed a wake of hot liquid. Jacobs and Shade showed that the superheated caps were most likely due to heat transfer from the superheated interior of bubbles to the adjacent liquid, that is, the caps did not originate from the thermal layer of the heating surface and were not manifestations of bulk convection.

From an analysis (incompletely reported) of temperature readings in the bubble wake Jacobs and Shade estimated the sensible heat transport due to wake flow. Since neither nucleation site concentration nor maximum bubble volumes were reported this heat transfer cannot be related to the area-of-influence heat transfer. If it is assumed, however, that the entire heater area was influenced by bubble action, then for the three levels of subcooling studied, the percentage contribution of wake flow to the total area-of-influence heat transfer may be estimated as 13,6 per cent at $\Delta T_{\text{sub}} = 5,5 \text{ K}$; 18,2 per cent at $\Delta T_{\text{sub}} = 25 \text{ K}$ and 34 per cent at $\Delta T_{\text{sub}} = 50 \text{ K}$.

It would appear then that, at least at mild subcooling, bubble-induced convection is a significant contributor, but not the major contributor to the heat transport from the thermal layer into the bulk.

Finally, Jacobs and Shade estimated the contribution of 'latent heat' to the total heat transfer. The contribution was found to be negligible. The quantities estimated were,

however, the visible latent heat transport, q_{LHvis} , (see Section 2.2) and not the total latent heat transport.

2.4.3 Interim Conclusions

The schlieren and temperature-probing studies reviewed above were largely qualitative in nature and covered a limited range of test conditions. Nevertheless they suggest the following interim conclusions:

Highly Subcooled Boiling

(1) In highly subcooled boiling bubble-induced convection consists of the condensation-induced ejection of hot liquid slugs from the thermal layer into the liquid bulk.

(2) The contribution of this process to the total thermal-layer-to-bulk heat transport is unknown, but could conceivably be large.

Saturated and Near-saturated Boiling

(1) In saturated and near-saturated boiling bubble-induced convection consists essentially of the entrainment into the bubble wake of portions of the thermal layer and of the dissipation of this wake into the liquid bulk. Under certain conditions this process is augmented by the further lifting up of thermal layer fragments sitting capwise on the bubble apexes.

(2) Within each area of bubble influence such bubble-induced convection is apparently a significant, but not a major contributor to the total heat transport from the thermal layer into the liquid bulk.

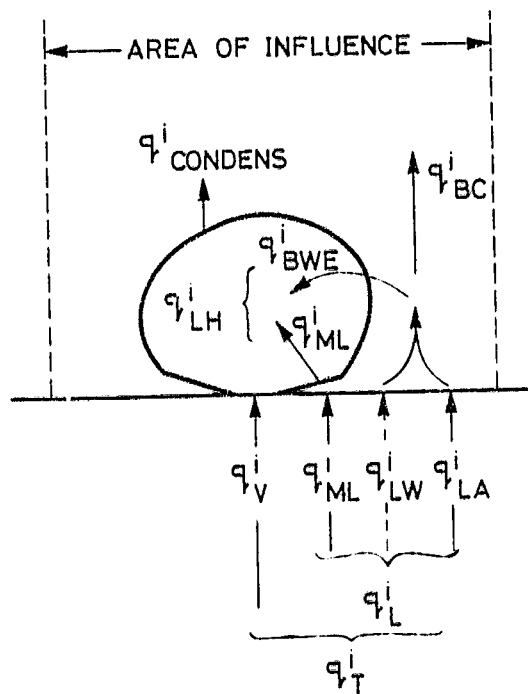
(3) The primary area-of-influence heat transfer (from the surface into the adjacent liquid) is convective. During the waiting period the convective velocities arise from the wake flow of the departed bubble and are directed radially inwards towards the nucleation site. During the bubble growth period the convective velocities (over the annular, unoccupied portion of the area of influence) are the resultants of the wake flow of the previous bubble and the source flow induced by the growth of the attached bubble.

It would appear then that neither the primary surface-to-liquid heat transfer nor the subsequent thermal-layer-to-bulk heat transfer take place as postulated in the bulk convection models. The success of heat transfer correlations based entirely on bulk convection thus requires explanation. Such an explanation will be attempted in Chapter 3.

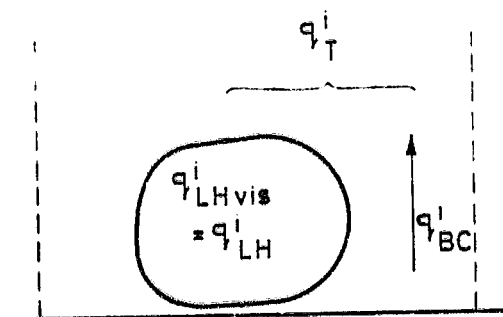
2.5 BREAKDOWN OF HEAT FLOW IN AREA OF BUBBLE INFLUENCE

The foregoing sections have shown that an area of bubble influence has associated with it a complex pattern of primary transfer processes at the heating surface and subsequent redistributions of energy in the fluid phases. A formal breakdown of these heat flows for both saturated and subcooled boiling is summarised in Figure 2.12 and equation 2.46 below. Each q^i is a mean rate of heat transfer (for one area of bubble influence., identifying a particular heat transfer process (observed or postulated), a related group of processes or a convenient fraction of these.

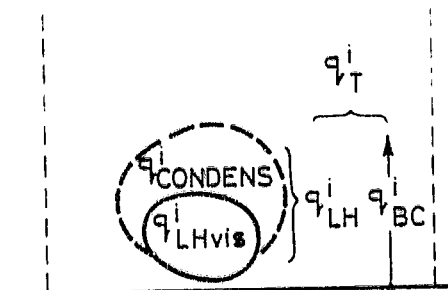
The heat flows are interrelated as follows:



(a) GENERAL CASE, SATURATED AND SUBCOOLED BOILING



(b) BUBBLE DEPARTURE, SATURATED BOILING



(c) BUBBLE DEPARTURE, SUBCOOLED BOILING

FIG. 2.12 BREAKDOWN OF HEAT FLOW IN AREA OF BUBBLE INFLUENCE

$$q_T^i = q_L^i + q_V^i (= \text{negl.}) \quad (2.46a)$$

$$= q_{LW}^i + q_{ML}^i + q_{LA}^i \quad (2.46b)$$

$$= q_{ML}^i + q_{IWA}^i \quad (2.46c)$$

$$= q_{ML}^i + q_{RWE}^i + q_{BC}^i \quad (2.46d)$$

$$= q_{IH}^i + q_{BC}^i \quad (2.46e)$$

$$= q_{IHvis}^i + q_{CONDENS}^i + q_{BC}^i \quad (2.46f)$$

where:

q_T^i (T = TOTAL) is the total heat transferred per unit time in the area of bubble influence; it may be resolved into q_V^i and q_L^i

q_V^i (V = VAPOUR) is the heat transferred from the heating surface directly to the vapour within the bubble; it is considered negligibly small compared with q_L^i

q_L^i (L = LIQUID) is the collective term for the heat transferred by various mechanisms from the heating surface into the liquid; it is essentially equal to q_T^i and may be resolved into the three components next listed

q_{LW}^i (LW = LIQUID, WAITING PERIOD) is the heat transfer during the waiting period (thermal layer recovery period)

- q_{ML}^i (ML = MICROLAYER) is the heat transferred to the liquid microlayer and causing microlayer vaporisation
- q_{LA}^i (LA = LIQUID, ANNULUS) is the heat transfer into the liquid annulus surrounding the attached bubble
- q_{LWA}^i (LWA = LIQUID, WAITING PERIOD and ANNULUS) is a collective term for the sum of q_{LW}^i and q_{LA}^i ; it manifests itself in the next two modes
- q_{BWE}^i (BWE = BUBBLE WALL EVAPORATION) is the heat transferred from the bubble surroundings to the walls of attached bubbles (microlayer excluded), there resulting in vapour evolution
- q_{BC}^i (BC = BUBBLE CONVECTION) is the portion of q_{LWA}^i not involved in evaporation into attached bubbles; it is transferred to the liquid bulk by convection induced by bubble movement
- q_{LH}^i (LH = LATENT HEAT) is the collective term for the heat flows associated with vapour evolution (latent heat transfer) at all surfaces of the attached bubble
- $q_{CONDENS}^i$ (CONDENS = CONDENSATION) is that portion of q_{LH}^i which, in subcooled boiling, recondenses from the attached bubble
- q_{LHvis}^i (LHvis = LATENT HEAT, visible) is the remaining portion of q_{LH}^i which detaches visibly from the surface as vapour.*

*The presentation of this breakdown derives in part from the work of Judd and Merte [66]; the quantities here identified are, however, different.

Clearly the chain of equation 2.46 could be continued by considering the further redistribution of heat occurring after bubble departure. In saturated boiling, for example, q_{BC}^i is redistributed between evaporation into rising bubbles and evaporation at the liquid surface at the top of the pool (see Section 2.1). These events, however, occur well away from the heating surface and are no longer associable with a particular area of bubble influence.

2.5.1 Summary of Foregoing Sections in Terms of New Nomenclature

The main points covered in the earlier sections of this chapter may now be summarised in terms of the newly-defined q^i nomenclature as follows:

The concept of negligible latent heat transport, $q_{LH}^i \ll q_T^i$, derives from early bubble observations in which not the ratio q_{LH}^i/q_T^i , but quantities obscurely related to it were estimated. In the case of Jakob's low-flux saturated boiling studies (see Section 2.2) the quantity in effect estimated, and found to be low, was $(q/A)_{LH}/(q/A)_T$, that is, $Nq_{LH}^i/(q/A)_T$. At low heat fluxes (low N) this quantity is of necessity low, irrespective of the ratio q_{LH}^i/q_T^i . In the subcooled boiling studies of Gunther and Kreith [8] and Rohsenow and Clark [9] not q_{LH}^i , but q_{LHvis}^i was estimated and found to be negligible. Neglect of the unknown term $q_{CONDENS}^i$, loose semantics and arbitrary extrapolation finally led to the application of the concept of negligible latent heat transport to all conditions of nucleate boiling.

With q_{LH}^i considered negligible, bubble-induced convection, q_{BC}^i , was accepted as the predominant thermal-layer-to-bulk heat flow, that is $q_{BC}^i \approx q_T^i$. A large number of mechanistic suggestions and heat transfer models for q_{BC}^i (and for the related primary transfer processes q_{LW}^i and q_{LA}^i) resulted.

In the most widely accepted of these, the bulk convection models, q_{LW}^i and q_{LA}^i were approximated by transient conduction into the liquid, while q_{BC}^i was ascribed to the periodic stripping-away of the entire transient conduction layers by departing bubbles. The resulting heat transfer correlation, were numerically successful.

Experimental studies on the thermal layer, however, indicated (at least for saturated and near-saturated boiling) that q_{BC}^i does not predominate the thermal-layer-to-bulk heat transfer and does not constitute a major fraction of q_T^i . The primary heat transfer processes q_{LW}^i and q_{LA}^i , on the other hand, were found to be convective and controlled by bubble movement.

Latent heat transport, q_{LH}^i , and its postulated microlayer contributor, q_{ML}^i , remained uninvestigated.

2.6 LATENT HEAT TRANSPORT

This section attempts, firstly, to review all available bubble measurements in saturated boiling which provide estimates of the area-of-influence latent heat transport contribution q_{LH}^i/q_T^i . The measurements cover a wide range of test conditions, heating surfaces and fluids, but are limited to atmospheric and sub-atmospheric pressures. All measurements conducted at terrestrial and reduced gravities point unanimously to a high latent heat contribution with q_{LH}^i/q_T^i invariably greater than 0,5 and in most cases greater than 0,7. At multi-g accelerations, however, the contribution appears to fall off. Further, at terrestrial gravity, the time-mean heat flow q_{BC}^i , due to the intermittent bubble-induced convection, is found to be of the same magnitude as the natural convection heat flow were it to operate undisturbed over the entire area of bubble influence.

Secondly, this section reviews indirect evidence, based on microlayer studies and bubble coalescence arguments, which corroborates the latent heat transport measurements.

Finally, concerning subcooled boiling, the section reviews indirect and theoretical estimates of the heat flow q_{CONDENS}^i , that is, of the portion of q_{LH}^i which recondenses from attached bubbles. The estimates suggest strongly, if not conclusively, that even in subcooled boiling $q_{\text{LH}}^i (= q_{\text{LH}}^i + q_{\text{CONDENS}}^i)$ is of comparable magnitude to q_T^i .

2.6.1 First Measurements of $(q/A)_{\text{LH}}$. Rallis et al

The first measurements of latent heat transport in saturated boiling appear to be those of Rallis, Greenland and Kok [1], presented in 1961.

Measurements were taken on water and ethanol boiling from a nickel wire of 0,51 mm diameter at an ambient pressure of 83 kN/m². The heat flux (referred to total area) due to latent heat flow into attached bubbles was expressed as follows:

$$(q/A)_{\text{LH}} = \rho_V \lambda \bar{f} \bar{V}_d N \quad (2.47)$$

Mean bubble frequency, $\bar{f}$, mean bubble departure volume, $\bar{V}_d$, and bubble source concentration, N , were obtained by multi-exposure flash photography. Vapour properties were evaluated at saturation. The latent heat flux $(q/A)_{\text{LH}}$ was shown to increase steadily with increasing heat flux, until in the vicinity of the critical flux, it approached or equalled the total flux; see Figures 2.13 and 2.14. At no stage was its contribution negligible.

Since the concept of an area of bubble influence had not yet been established, no estimates of the latent heat contribution q_{LH}^i/q_T^i were provided. However, it may be argued that the

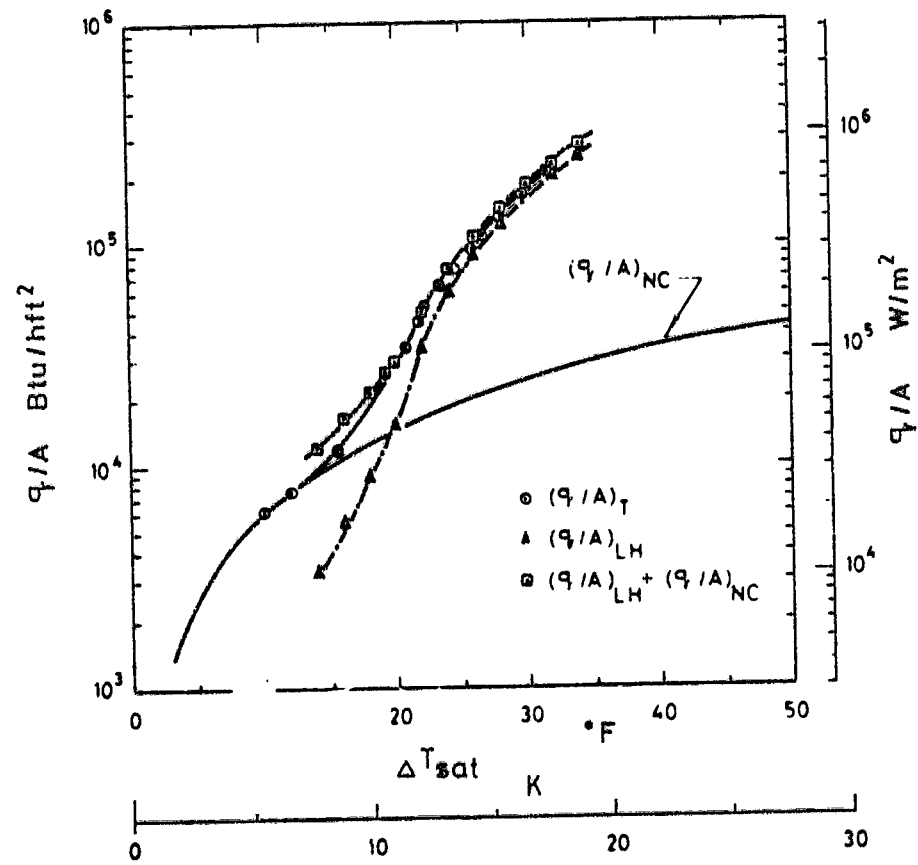


FIG. 2.13 LATENT HEAT TRANSPORT IN SATURATED BOILING
 RALLIS ET AL [1]
 Water, thin wire, 83 kN/m^2

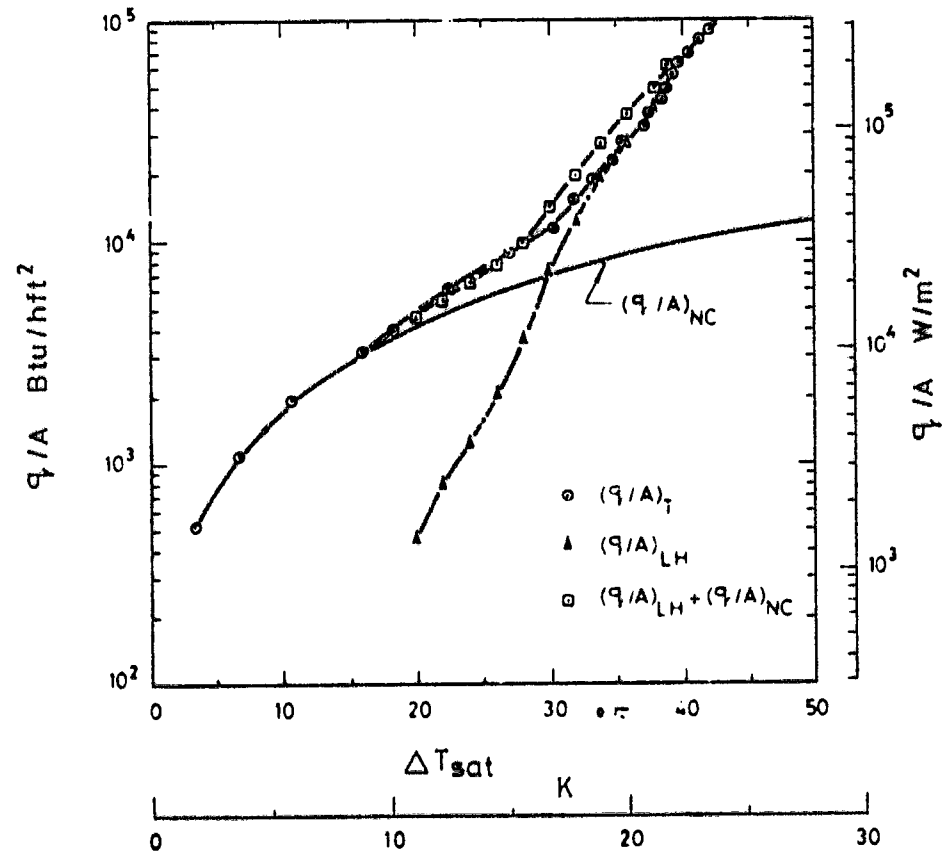


FIG. 2.14 LATENT HEAT TRANSPORT IN SATURATED BOILING
 RALLIS ET AL [1]
 Ethanol, thin wire, 83 kN/m^2

onset of lateral coalescence of attached bubbles implies the overlap of areas of bubble influence. Appendix 2, dealing with the subsequent study of Rallis and Jawurek [2], shows that for water boiling from a 0,51 mm diameter wire such coalescence sets in at a heat flux of approximately 195 kW/m². From this flux upwards then, the entire heater in the study of Rallis et al may be considered as occupied by areas of influence, with the flux ratio $(q/A)_{LH}/(q/A)_T$ consequently equal to the area-of-influence heat flow ratio q_{LH}^i/q_T^i . Figure 2.13 gives this ratio as 0,76 at 195 kW/m², and as 0,93 at the highest heat flux recorded. For the ethanol run the onset of coalescence is unknown; however, at the highest data points in Figure 2.14 $(q/A)_{LH}$ is identical with $(q/A)_T$.

It may be concluded therefore, that the latent heat transport contribution q_{LH}^i/q_T^i for water ranged from 0,76 to 0,93 at moderate to high heat fluxes, and for ethanol was equal to 1,0 at high fluxes.

The photographic technique employed in this study suffered from two inadequacies: Firstly, bubble volumes could not invariably be determined at the point of departure. Secondly, bubble frequencies could be measured only at low heat fluxes; high-flux values were obtained by extrapolation from an empirical equation fitted to low-flux data. Thus there is an element of doubt as to the accuracy of the latent heat flux figures. Their order-of-magnitude correctness, however, cannot be questioned.

2.6.2 First Evidence of Microlayer Effects. Moore and Mesler

Also in 1961, shortly after the publication of the findings of Rallis et al, Moore and Mesler [67] reported pool boiling runs in which a special microthermocouple [68]

provided local surface temperatures of the heater. The test liquid was water at saturation temperature and atmospheric pressure; the heater was a horizontal Nichrome ribbon. At heat fluxes of 426 to 637 kW/m² the thermocouple temperature (assumed to be the true surface temperature) was observed to drop intermittently by 11 to 17 K in 1 to 3 ms and to recover in 6 to 12 ms. Moore and Mesler could explain these fluctuations only by invoking the microlayer concept of Snyder [12] and Edwards [13]. In terms of this concept the heat required to evaporate the microlayer is extracted from the heating surface and causes the surface temperature to drop; after microlayer dry-out or bubble detachment the temperature recovers until the next bubble and microlayer are formed.

From the measured temperature records Moore and Mesler calculated $(Q/A)_{TD}$, the heat (per unit area) extracted from the surface during the temperature drops. The product of this and bubble frequency gave time-average fluxes due to microlayer evaporation at the thermocouple site, that is

$$(q/A)_{ML}^i = f (Q/A)_{TD} \quad (2.48)$$

The magnitude of $(q/A)_{ML}^i$ was 70 to 90 per cent of the overall heat flux (q/A) . Moore and Mesler argued that at their high heat fluxes the entire surface behaved like the thermocouple site. With the thermocouple sites having an active diameter of 0,13 mm this would imply that the entire heater area was covered by areas of bubble influence. If this argument is accepted and if the thermocouple temperature is accepted as the true heater surface temperature then the ratio q_{ML}^i/q_T^i is similarly of the order of 0,7 to 0,9.

The results of Rallis et al indicate that with a wire in high-flux boiling latent heat transfer q_{LH}^i is a major contribution to the total area-of-influence heat transfer q_T^i . The work of Moore and Mesler tends to confirm this

result for a flat ribbon heater and points to microlayer evaporation, q_{ML}^i , as a possible major contributor to the latent heat transfer.

2.6.3 Indirect Evidence of Significant Latent Heat Transport in Subcooled Boiling

During the years 1959 to 1962 the mechanism of mass transfer through attached bubbles in highly subcooled boiling received considerable attention from Bankoff and his co-workers [69 - 72]. Evidence presented in support of this mechanism was indirect and largely centred around the demonstration that conditions at the bubble apexes are non-stagnant and thus conducive to condensation, that is, to a significant contribution by $q_{CONDENS}^i$ to q_T^i . Firstly, it was pointed out [69, 70] that Ellison [73], using a graphite tracer, noted tangential velocity components on the surfaces of attached bubbles in subcooled pool boiling. Secondly, Bankoff and Mikesell [69] were able to correlate Gunther's data [74] on the bubble dynamics in highly subcooled forced-convection boiling, only on the assumption of turbulence at the bubble walls. Thirdly, Bankoff [70] noted, that in the same study of Gunther's [74] attached bubbles slid downstream at approximately 0.8 times the bulk velocity, thus indicating significant shear at the bubble surfaces.

Bankoff and Mikesell [69] in estimating the heat transfer from the condensing surfaces of Gunther's bubbles made use of empirical correlations for heat transfer from solid spheres in turbulent flow. They estimated that under the conditions of Gunther's experiment the condensation contribution $q_{CONDENS}$ and thus also the latent heat flux $(q/A)_{LH}$ might become significant.

In an attempt to assess this contribution more accurately Bankoff and Mason [71] measured heat transfer coefficients at the walls of steam bubbles condensing in a stream of cold

water. A metered flow of dry saturated steam was injected via a pinhole orifice through an unheated wall into a reservoir where a submerged water jet of controllable velocity and subcooling impinged head-on with the bubbles. Condensation rates were computed from high-speed ciné records of the bubble growth-collapse cycles. The heat transfer coefficients ranged from $7,4 \times 10^4$ to $1,82 \times 10^6$ W/m²K. The results of those tests where bubble size, shape and frequency approximated those in Gunther's study [74] were combined with Gunther's bubble population data to calculate the latent heat contribution $(q/A)_{LH}/(q/A)_T$. This ratio was always significant and at least an order of magnitude larger than the 1 to 2 per cent previously estimated [8, 9]. As the heat flux increased towards the critical flux the ratio approached unity. Thus at high heat fluxes the ratio q_{LH}^i/q_T^i also approached unity.

It should be noted, however, that Bankoff and Mason's experimental conditions differed significantly from those in forced convection boiling: firstly, the subcooled water flow was normal, rather than transverse to the direction of bubble growth, and, secondly, bubble growth took place on an unheated surface, that is, in the absence of a thermal boundary layer.

Both these departures from real conditions were corrected by Snyder and Robin in a more recent condensation study [75]. Their apparatus provided fully developed turbulent flow parallel to the surface and also the build-up of a thermal boundary layer by means of a special heating plate. The measured steam input to the bubble was independent of the heating of the plate. A high-speed ciné camera recorded bubble growth and collapse. The calculated bubble wall heat transfer coefficients were close to those obtained by Bankoff and Mason [71] and were virtually the same with and without a boundary layer. This finding again refutes the argument that bubbles in surface boiling are surrounded by

a film of hot, lifted-up boundary layer which inhibits condensation. The authors further showed that under conditions similar to those of Gunther's experiment [74] 10 to 100 times the vapour volume of a single bubble flowed through the bubble during its lifetime, that is $q_{LH}^i \gg q_{LHvis}^i$.

Robin and Snyder [76] next presented a mathematical model for the mass transfer through the artificially produced bubbles. Their treatment of condensation at the interface between the bubble vapour and the turbulent stream, and of the subsequent diffusion of latent heat into the turbulent core each involved a major unknown. In the condensation case the unknown was the controversial kinetic condensation coefficient [15, 77], and in the diffusion case, the eddy diffusivity of heat transfer, ϵ_h , at the bubble surface. The condensation coefficient was assumed equal to unity. The eddy diffusivity, ϵ_h , was established by trial and error during the numerical solution of the differential equations describing the model. The final ϵ_h value for each set of conditions was the one that gave agreement between the measured and the 'predicted' bubble growth and collapse. The values thus determined were virtually identical with the theoretical values of momentum eddy diffusivity, ϵ_m , at a distance of one bubble radius from the wall. The agreement between ϵ_h and ϵ_m was interpreted as strong support of the correctness of the model. Uncertainties concerning the kinetic condensation coefficient, see [77], however, detract somewhat from the strength of this argument.

In a further paper Robin and Snyder [78] extended their analysis to treat real surface boiling. In this case the vapour, instead of being artificially supplied, was considered to arise from the evaporation of the microlayer. (Incontrovertible evidence for the existence of microlayers had, to Snyder's satisfaction, become available over the years.) The differential equations derived for the

artificial bubble could thus be used, but terms for the transient conduction in the heating surface and for the kinetics of microlayer evaporation had to be added. The conductive resistance of the microlayer itself was neglected, and its thickness generally chosen as sufficient to preclude dry-out. The major unknowns in the model thus were the kinetic coefficients of evaporation and condensation and the eddy diffusivity of heat transfer, ϵ_h .

The coupled equations were solved numerically for the conditions of Gunther's surface boiling experiment [74] and for slight variations on these. The kinetic coefficients were assumed equal to unity and values of ϵ_h were adjusted for the best agreement between the solution and Gunther's experimental growth and collapse data. Extensive observations resulted from these numerical experiments; the two most directly relevant to our argument are:

- (1) The value of ϵ_h required for agreement was no longer equal to ϵ_m , but to approximately a tenth of this value. Several arguments were advanced to explain this discrepancy. The most interesting of these, based on comments by Tong [19] suggested that a low value of ϵ_h might be correct for real surface boiling: the presence of a dispersed phase (bubbles) might intermittently obstruct the free path of turbulent eddies between adjacent layers of the continuous phase. This Robin and Snyder noted, is 'in drastic conflict with the bubble agitation explanation of nucleate boiling'.
- (2) For a typical theoretical bubble the heat transmitted through its interior was approximately 30 times the latent heat required to form one bubble volume of vapour, that is $q_{LH}^i \gg q_{LHvis}^i$.

Finally Robin and Snyder checked whether their mechanism could account for the total surface-boiling heat flux. For

a particular data point of Gunther's the quantity $(q/A)_T/Nf$, that is, the heat removal required per bubble so as to accommodate the entire heat flux, was $3,82 \times 10^{-3}$ J. Under the same conditions a typical theoretical bubble could remove $5,45 \times 10^{-3}$ J. The mass transfer mechanism as envisaged by Robin and Snyder is thus capable of accommodating the entire heat transfer in high-flux surface boiling, and the corresponding area-of-influence heat flows are given approximately by

$$q_T^i = q_{ML}^i = q_{CONDENS}^i = q_{LH}^i.$$

Detracting features of Robin and Snyder's analysis are, firstly, the uncertainties surrounding the values of the kinetic coefficients, secondly, the lack of agreement between the values of the eddy diffusivities ϵ_m and ϵ_h , and thirdly the limitation to a hemispherical bubble geometry throughout the bubble life. Increased bubble sphericity, arising from increased surface tension forces, would reduce the microlayer area and hence the contribution of the mass transfer mechanism.

In summary: the work of Snyder and Robin and of Bankoff and his co-workers suggests strongly, but not conclusively, that in highly subcooled boiling, and particularly at high heat fluxes and with forced convection, the latent heat flow $q_{LH}^i (= q_{LHvis}^i + q_{CONDENS}^i)$ is of roughly the same magnitude as q_T^i .

2.6.4 Bubble Coalescence and Latent Heat Transport Estimates in Saturated Boiling

In highly subcooled boiling small isolated bubbles are observed on the heating surface. The visual impression of low latent heat transport is, however, of dubious significance because of recondensation on the upper bubble surfaces. A radically different bubble pattern, uncomplicated by recondensation, occurs in saturated boiling. Bubbles grow to a much larger size and at moderately high heat fluxes

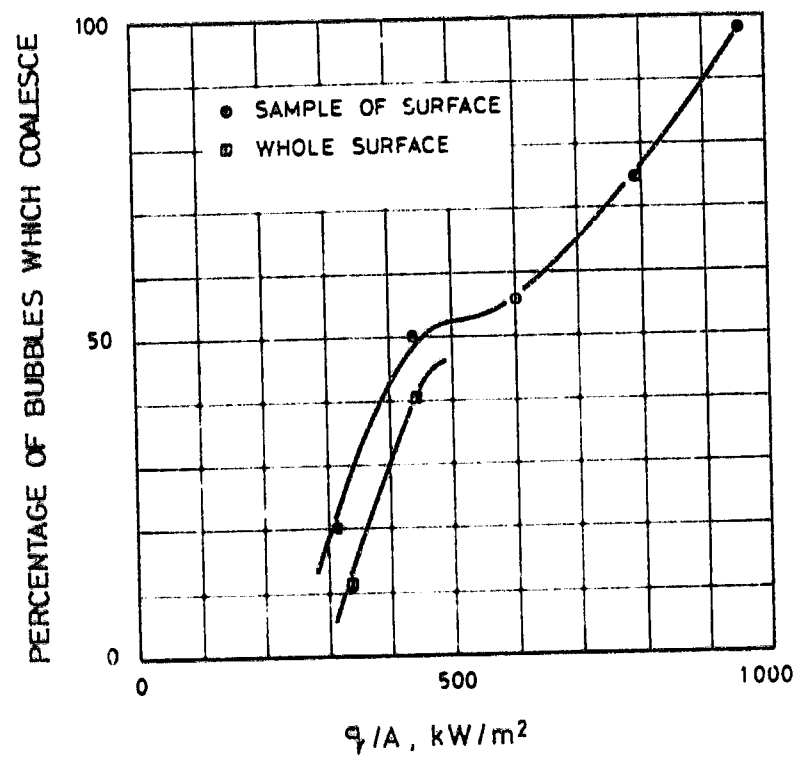


FIG. 2.15 PERCENTAGE BUBBLE COALESCENCE vs
HEAT FLUX. SÉMÉRIA [79]
 (Water, flat horizontal heater, atmospheric pressure)

start to coalesce into large attached vapour masses. The simple observation of such coalescence has forced several workers [37, 79-81] into a common-sense acceptance of the idea of a significant latent heat transport contribution.

Figure 2.15 reproduced from Séméria's work of 1962 [79] shows lateral bubble coalescence measurements on water boiling off a flat surface at atmospheric pressure. Coalescence increases with increasing heat flux until, in the vicinity of the critical flux (approximately 10^6 W/m²), all bubbles coalesce prior to departure. The majority of coalescences, Séméria showed, involved several, rather than two, bubbles, so that at high fluxes the surface was covered by coalesced vapour masses an order of magnitude larger in diameter than isolated bubbles. Experiments at pressures up to 10^4 kN/m² [80] showed the same trend.

Séméria argued [80] that with increasing surface coverage by such sluggish coalescences the liquid agitation near the surface must become progressively weaker. Nevertheless the slope of the boiling curve ($\log (q/A)$ vs $\log \Delta T_{\text{sat}}$) is not lowered by the onset of coalescence. He concluded that the major heat transfer mode from the thermal layer into the liquid bulk must thus be direct vaporisation 'at the foot' of attached vapour, that is, latent heat transport.

In 1963 Gaertner [81] presented a more detailed photographic study of bubble coalescence. Figure 2.16a shows his boiling curve for water at atmospheric pressure on a horizontal polished copper heater; Figure 2.16b reproduces his sketches of the bubble patterns.

In the region of isolated bubbles ($q/A = 33,1$ to 145 kW/m²) and within each area of bubble influence the heat transfer from the thermal layer to the liquid bulk was ascribed to a combination of latent heat transport (via microlayer vaporisation) and bubble-induced convection.

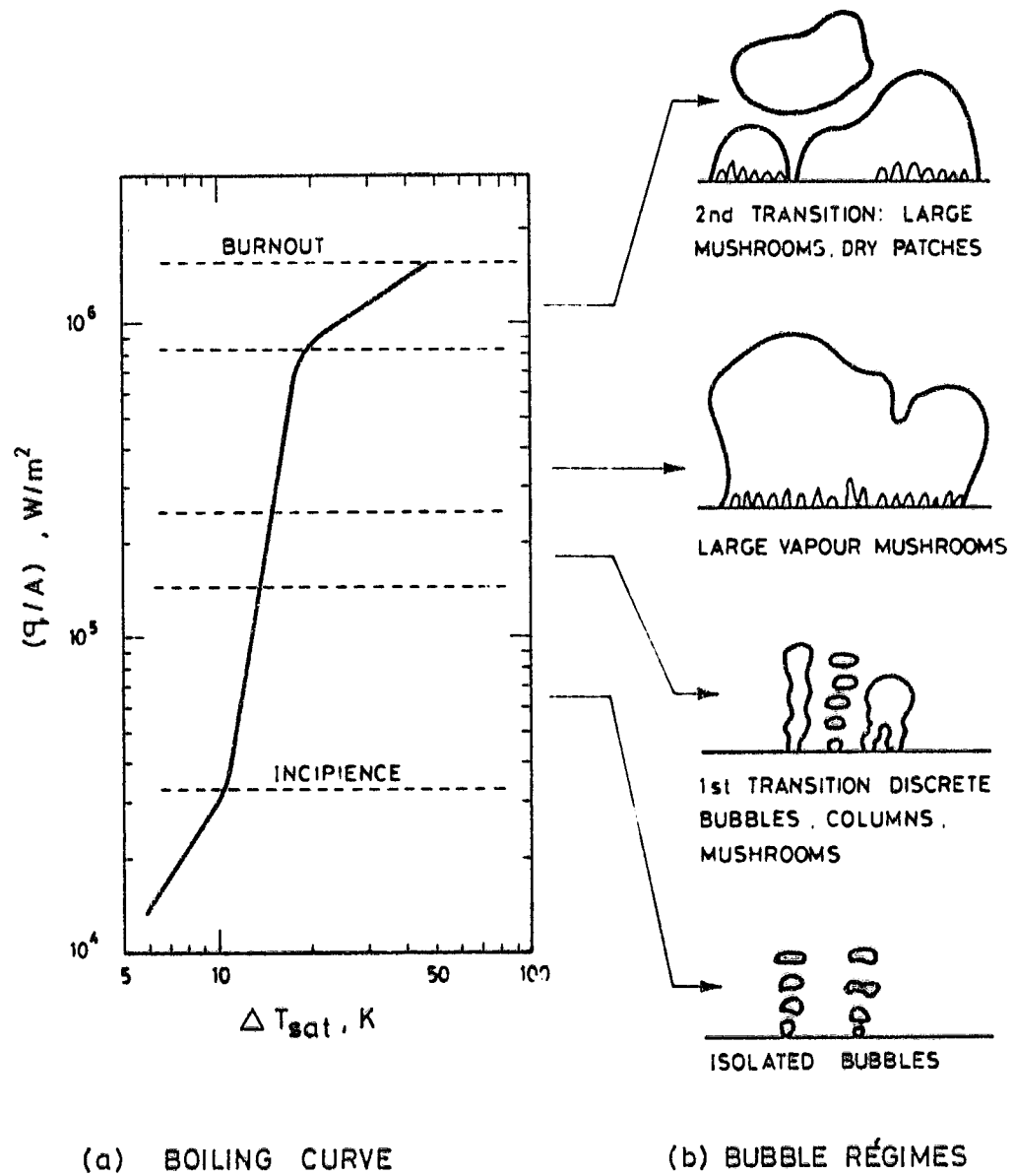


FIG. 2.16 WATER BOILING ON HORIZONTAL COPPER PLATE AT ATMOSPHERIC PRESSURE. GAERTNER [81]

In the first transition region ($q/A = 145$ to 252 kW/m^2) vertical and lateral bubble coalescence set in. For the area portions supporting vapour columns and mushrooms Gaertner concluded that the thermal-layer-to-bulk heat transfer took place predominantly by latent heat transport around the bases of the vapour stems. With increasing heat flux the concentration of coalescence regions on the surface increased.

In the vapour mushroom region ($q/A = 252$ to 820 kW/m^2) the heating surface was totally covered by huge vapour aggregates, each attached to the surface by numerous stems. When one mushroom detached (by breaking at the stems) a new one was immediately formed in its place. Under such conditions, Gaertner argued, the total boiling heat flux must be supplied by latent heat transport with vaporisation occurring around the bases of the vapour stems and possibly also at the bottom surface of the mushroom caps.

In the second transition region ($q/A = 820$ to 1550 kW/m^2) Gaertner postulated the appearance of dry surface patches within the vapour mushrooms. The predominant heat transfer mechanism was thus the same as in the normal mushroom region, but the area available for latent heat transport was reduced by the dry patches.

The latent heat flux ratio $(q/A)_{LL}/(q/A)_T$, as stated or implied by Gaertner, is sketched in Figure 2.17. The ratio, increasing from zero at incipience of boiling to unity in the vapour mushroom region, thus lies somewhere in the cross-hatched region. By way of comparison Figure 2.18 shows the same ratio for a thin wire heater, as calculated from the measurements of Rallis et al [1]. Despite the difference in coalescence behaviour (vapour columns are absent on thin wires and the high-flux vapour aggregates are much smaller) the two cases reveal a similar trend.

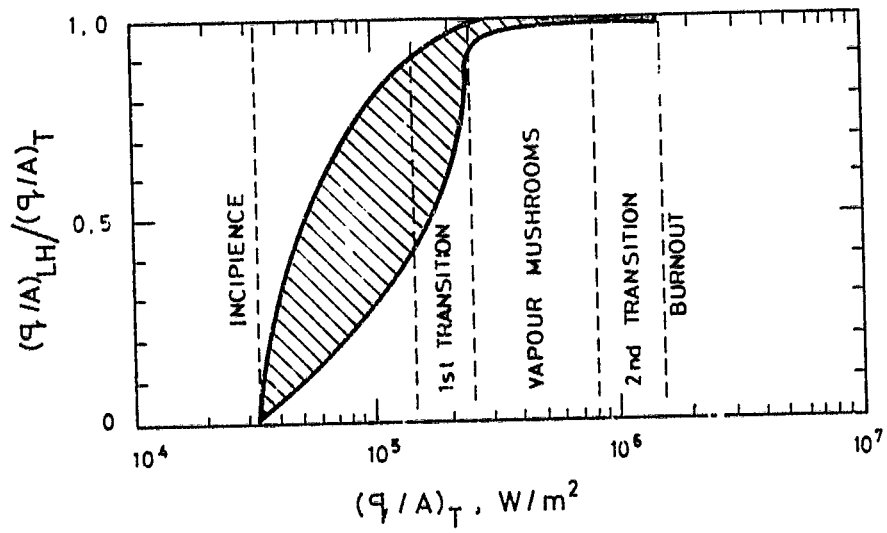


FIG. 2.17 CONTRIBUTION OF LATENT HEAT TRANSPORT, GAERTNER [81]
Water, plate heater, 100 kN/m^2

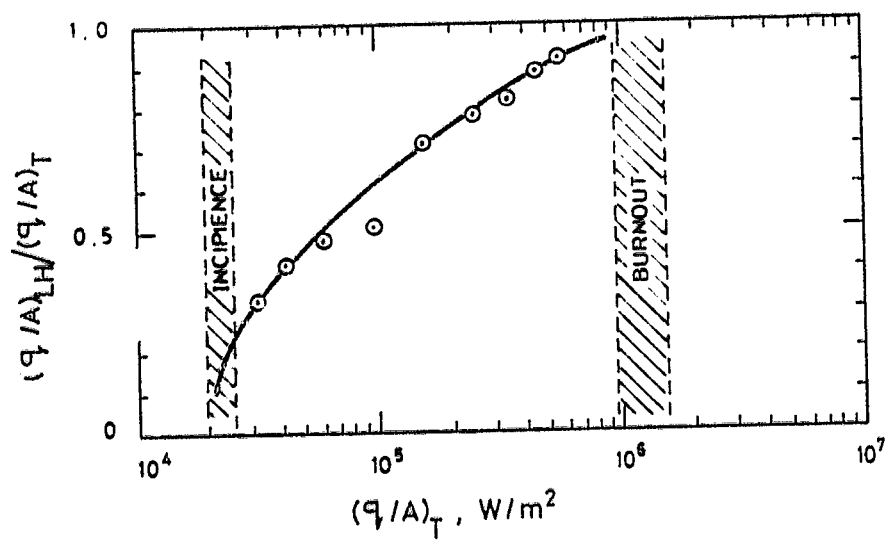


FIG. 2.18 CONTRIBUTION OF LATENT HEAT TRANSPORT
RALLIS ET AL. [1]
Water, thin-wire heater, 83 kN/m^2

The onset of lateral bubble coalescence, as already discussed in Section 2.6.1, implies the overlap of areas of bubble influence and the equality of the ratios $(q/A)_{LH}/(q/A)_T$ and q_{LH}^i/q_T^i . At Gaertner's lowest coalescence heat flux, 145 kW/m^2 , Figure 2.17 shows this ratio to lie somewhere in the region of 0,4 to 0,9. Thus Gaertner's work implies that at moderate heat fluxes the latent heat contribution to the total area-of-influence heat flow is significant and possibly predominant.

The review now turns to direct measurements of latent heat transport in saturated boiling.

2.6.5 Measurements of Rallis and Jawurek; Water Boiling from Thin Wire

The accuracy of the latent heat transport measurements of Rallis et al [1] are, as already mentioned, open to some doubt because of imperfections of photographic technique. New measurements, resulting from attempts at improved photography were presented by Rallis and Jawurek in 1964 [2]. The full study is submitted as a part of this thesis in Appendix 2. The extensive sections dealing with bubble dynamics and nucleation are, however, not relevant at this point, and only the latent heat transport results are given below.

The supposedly improved photographic technique (described in Appendix 2) provided 16 successive photographs of the bubble field at approximately 85 frames per second. This taking speed is now recognised as too low and almost certainly precluded the recording of some large low-frequency bubbles (see discussion of Figure 9 in Appendix 2). The reported values of latent heat transport must thus be considered as too low. It should be remembered, however, that the study, together with that of Rallis et al constituted pioneer

work which alerted the boiling field to the hitherto unrecognised significance of latent heat transport.

The liquid studied was water at saturation temperature under an ambient pressure of 83,4 kN/m²; the heater was a nickel wire of 0,51 mm diameter.

The use of thin wire heaters in latent heat transport studies, while reducing the generality of results, facilitates photography and, to some extent, interpretation: Firstly, a 'flat' and unobscured bubble pattern is produced which requires a small depth of field; secondly, the continuous vapour columns occurring on plate heaters [29, 81] are apparently suppressed. Thus, in saturated boiling, the latent heat transport can be determined by evaluating the terms entering into equation 2.49 below, where all bubble parameters are unambiguously obtainable from ciné records.

$$(q/A)_{LH} = \rho_V \lambda \sum_{i=1}^n (f_i V_{di}) / A \quad (2.49)$$

Equation 2.49 was rewritten as

$$(q/A)_{LH} = \rho_V \lambda N' \overline{fV}_d \quad (2.50)$$

where $\overline{fV}_d$ was defined as

$$\overline{fV}_d = \frac{\sum_{i=1}^n (f_i V_{di})}{n} \quad (2.51)$$

that is, as the arithmetic mean over all bubble sources, n , (present on area A) of the product of frequency and departure volume. With this definition equations 2.49 and 2.50 are consistent. Jawurek had previously shown [82] (see also Appendix 2) that the replacement of $\overline{fV}_d$ in equation 2.50 by the product of the arithmetic mean of frequencies and the

arithmetic mean of volumes, although frequently practised, destroys this consistency and yields a heat flux that is not necessarily the latent heat transport.

Measurements extended well into the lateral bubble coalescence region. Bubble departure volume could thus only be defined as the volume of a vapour globule leaving the surface, irrespective of whether the globule arose from one or more nucleation sites. Thus N' , the number of bubble sources per unit area, was necessarily defined as the number of regions per unit area from which vapour globules emanated, irrespective of whether such regions contained one or more nucleation sites. Only in the region of isolated bubbles is the bubble source concentration, N' , equal to the nucleation site concentration, N .

Rallis and Jawurek's boiling curve ($\log (q/A)$ vs ΔT_{sat}) and the latent heat flux given by equation 2.50 are shown in Figure 2.19. The natural convection data were correlated and extrapolated by an equation of the form $(q/A)_{\text{NC}} = \text{const } (\Delta T_{\text{sat}})^{5/4}$ (see Section 2.6.7 and [83]). The latent heat flux ratio $(q/A)_{\text{LH}}/(q/A)_{\text{T}}$ is plotted against $(q/A)_{\text{T}}$ in Figure 2.20. The latent heat contribution increases with increasing flux and at the critical flux (not measured, but for the wire used generally greater than 950 kW/m^2 , see [1]) it alone might represent the total flux.

Since the extent of an area of bubble influence is not known for thin wires, no estimate of $q_{\text{LH}}^i/q_{\text{T}}^i$ is possible at low heat flux. Appendix 2, however, shows lateral bubble coalescence to have set in at a heat flux of 195 kW/m^2 ; at that stage the entire surface may be considered as occupied by areas of bubble influence and $q_{\text{LH}}^i/q_{\text{T}}^i$ is equal to $(q/A)_{\text{LH}}/(q/A)_{\text{T}}$, that is, 0.51 (see Figure 2.20). At the highest measured heat flux, 494 kW/m^2 , $q_{\text{LH}}^i/q_{\text{T}}^i$ is 0.73.

The experimental error bounds shown in Figure 2.20 reflect

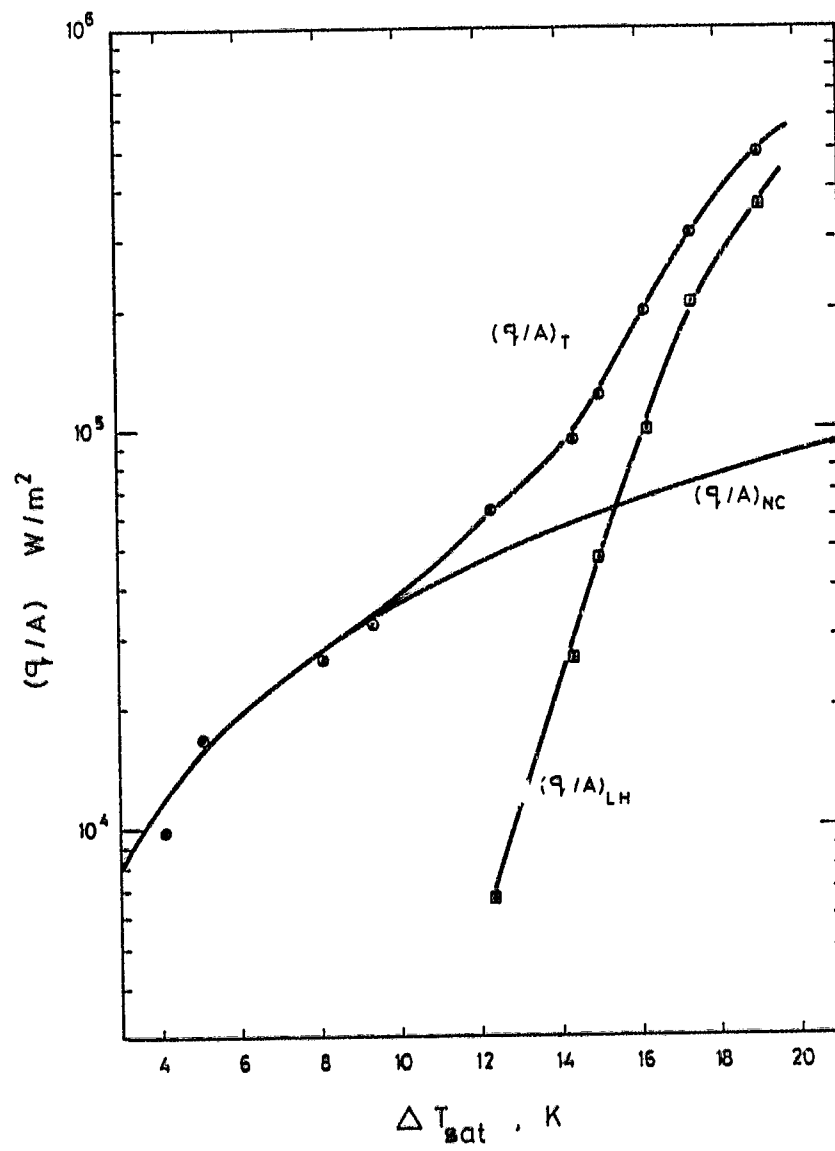


FIG. 2.19 BOILING CURVE AND LATENT HEAT TRANSPORT
 RALLIS AND JAWUREK [2]
 Water, thin wire, 83 kN/m^2

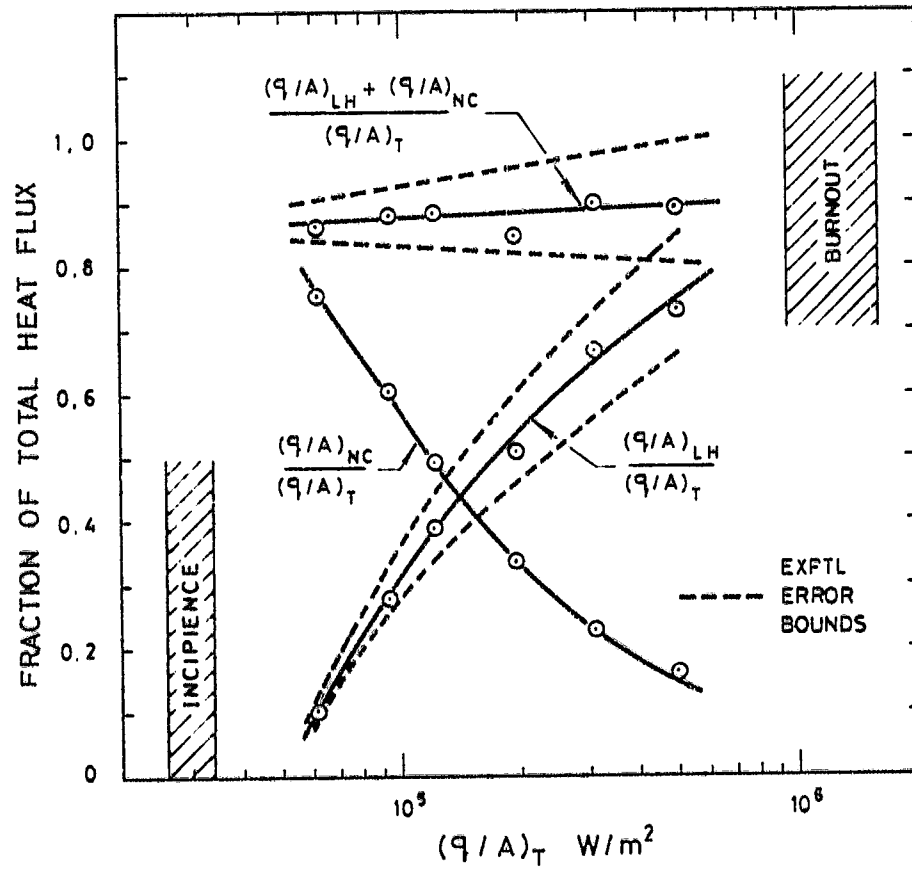


FIG. 2.20 LATENT HEAT TRANSPORT CONTRIBUTION
 RALLIS AND JAWUREK [2]
 Water, thin wire, 83 kN/m^2

uncertainties in the measurements of bubble volume and frequency (see Appendix 2). Errors arising from the total omission of large low-frequency bubbles are excluded. Thus it may be concluded that the latent heat transport contribution to the total area-of-influence heat transfer, q_{LH}^i/q_T^i , is greater than 0,51 at moderate heat flux and greater than 0,73 at high heat flux.

Finally, Figure 2.20 shows the sum of the latent heat flux $(q/A)_{LH}$ and of the extrapolated natural convection flux $(q/A)_{NC}$ as a fraction of the total heat flux. The ratio is of the order of 0,9 throughout. The sum $(q/A)_{LH} + (q/A)_{NC}$ is also shown in Figures 2.12 and 2.13 for the data of Rallis et al [1] and generally equals or exceeds $(q/A)_T$ throughout the flux range. For the two studies taken together the ratio $[(q/A)_{LH} + (q/A)_{NC}] / (q/A)_T$ may be taken as approximately equal to unity.

It would appear then, that the time-mean area-of-influence heat flow due to the intermittent bubble-induced convection, that is q_{BC}^i , is of the same magnitude as the natural convection heat flow were it to operate undisturbed over the entire area of bubble influence.

2.6.6 Measurements of Novaković et al; Ethanol Boiling on Mercury Surface

In 1966 Novaković et al [84] presented latent heat transport measurements for ethanol boiling on the surface of a pool of mercury under atmospheric pressure.

Line (1) in Figure 2.21 shows the total heat flux versus temperature difference; the dotted portion at high flux was taken from data previously reported for the same system by Novaković and Stefanović [85]. Line (2) shows the latent heat flux $(q/A)_{LH}$ as obtained from high-speed cine records. In the region below the knee of curve (1) the latent heat

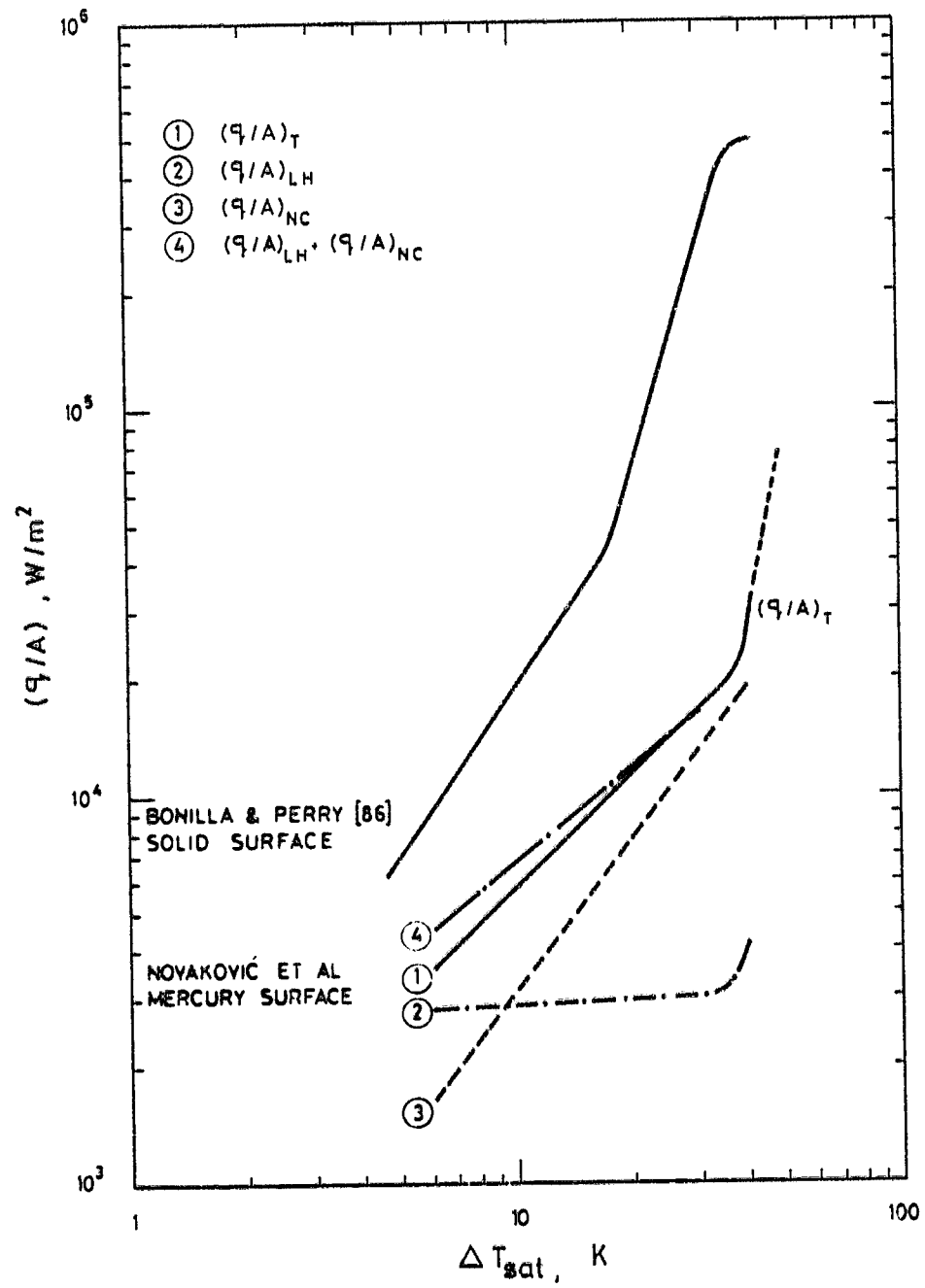


FIG. 2.21 ETHANOL BOILING ON MERCURY SURFACE AT
ATMOSPHERIC PRESSURE. NOVAKOVIĆ ET AL [84]

fraction $(q/A)_{LH}/(q/A)_T$ falls with increasing heat flux. This finding is contrary to those reported in the foregoing sections and calls for a closer examination of the experiment.

For purposes of comparison Figure 2.21 also shows a conventional ethanol boiling curve obtained by Bonilla and Perry ([86], Figure 12) on a chromium-plated copper disc heater. Because of the superficial similarity between such natural convection data and the low-flux portion of the mercury-ethanol data, the boiling occurring in this region was termed 'convective boiling'.

The chief features of convective boiling were: (1) nucleation sites which remained fixed in number ($N = 0,32 \text{ cm}^{-2}$) irrespective of increasing surface temperature, and (2) a volumetric vapour flow rate per site which increased only slightly with increasing temperature difference ΔT_{sat} . These immediately account for the nearly flux-independent latent heat transport. Neither feature, however, prevails in nucleate boiling on normal solid surfaces:

- (1) The nucleation site concentration, N , normally increases rapidly with increasing wall temperature, T_w , as illustrated by Gaertner's correlation [87]

$$N = N_0 \exp (-K^*/T_w^3) \quad (2.52)$$

where N_0 is a surface-liquid-dependent constant and K^* , as Anderson et al [88] suggest, approximates to a universal constant of value $3,305 \times 10^9 \text{ K}^3$. Thus an increase of ΔT_{sat} from 6 K to 30 K (the limits of Novaković's convective boiling), that is, an increase in T_w from $84,4^\circ\text{C}$ to $108,4^\circ\text{C}$, would, with a solid surface, cause N to increase by a factor of $3,6 \times 10^5$. (Surfaces drilled with artificial nucleation cavities, for example [46], are, of course, special cases).

- (2) The volumetric vapour flow rate per nucleation site

normally increases substantially with increasing temperature difference, ΔT_{sat} , an observation apparently first made by Rallis and Jawurek [2]. Cole [50], correlating a large body of data, has shown the product fV_d to be approximately proportional to the square of the Jakob number ($= \rho_L C_{L, \text{sat}} \Delta T_{\text{sat}} / \rho_V \lambda$) and thus to ΔT_{sat}^2 . Accordingly an increase in ΔT_{sat} from 6 K to 30 K would, with a solid heater, lead to a 25-fold increase in the volumetric vapour flow rate per site.

The 'convective boiling' of Novaković et al is thus a special case and untypical of normal nucleate boiling. Nevertheless it may be shown that the measurements of Figure 2.21 imply area-of-influence heat flows essentially identical to those found for solid heaters in Sections 2.6.1 to 2.6.5.

Line (3) in Figure 2.20 shows the undisturbed natural convection heat flux, which, according to Novaković, was given by the relation

$$\text{Nu} = 0,135 (\text{Gr} \cdot \text{Pr})^{1/4} \quad (2.53)$$

that is, the correlation for turbulent natural convection from a horizontal plate ([7], p. 527). The sum of this flux and of the latent heat flux $(q/A)_{\text{LH}}$ is shown in line (4) in Figure 2.21. Together they equal or exceed slightly the total heat flux, a most satisfying agreement with previous findings. Thus it may be concluded as before, that the time-mean heat flow due to the intermittent bubble-induced convection, q_{BC}^i , differs not greatly from that occurring in steady-state natural convection.

The distribution of the total area-of-influence heat flow, q_T^i , between q_{BC}^i and q_{LH}^i can, in the present case, be estimated more reliably than for the thin-wire studies. Firstly, equation 2.18 is recalled:

$$(q/A)_T = (q/A)_{NB} + (q/A)_{NC} (A_{NC}/A_T) \quad (2.18)$$

Here $(q/A)_{NB} = Nq_T^i$, that is, the heat flux referred to total area due to nucleate boiling alone, $(q/A)_{NC}$ is the flux due to natural convection when it alone is operative, and the last term is the area fraction available for such undisturbed convection. Next, from equation 2.46e, we have:

$$(q/A)_{NB} = Nq_T^i = Nq_{LH}^i + Nq_{BC}^i = (q/A)_{LH} + (q/A)_{BC} \quad (2.54)$$

By combining equations 2.18 and 2.54 we obtain the ratio $q_{LH}/q_T^i (= (q/A)_{LH}/(q/A)_{NB})$ as follows:

$$q_{LH}^i/q_T^i = (q/A)_{LH} / \left[(q/A)_T - (q/A)_{NC} (A_{NC}/A_T) \right] \quad (2.55)$$

where the area fraction is given by

$$A_{NC}/A_T = 1 - NA_i \quad (2.56)$$

In the experiment of Novaković et al the nucleation site concentration was fixed at $N = 0,32 \text{ cm}^{-2}$. For a flat heating surface the diameter D_i of the area of bubble influence, A_i , may, according to Hsu and Graham [43] be taken as equal to two bubble departure diameters, that is, $2 \times 1,9 \text{ mm}$ for the study of Novaković. Equation 2.56 then gives an area fraction of 0,964. With this value and the heat fluxes shown in Figure 2.20, equation 2.55 may be evaluated. The resulting values of q_{LH}^i/q_T^i range from 1,38 at $\Delta T_{\text{sat}} = 6 \text{ K}$ to 0,78 at 30 K. Values greater than unity reflect either measurement errors or an error in the diameter in the area of bubble influence. The former seems more likely since the replacement of D_i with $3D_d$, rather than $2D_d$, lowers the values of q_{LH}^i/q_T^i only slightly to range from 1,33 to 0,67.

It seems safe to conclude then that in the 'convective boiling' region the area-of-influence heat transfer from the thermal layer to the bulk liquid took place predominantly

by latent heat transport, with q_{LH}^i/q_T^i generally exceeding 0,7. This again agrees with the findings of Sections 2.6.1 to 2.6.5.

Towards the knee of Novaković's boiling curve ($\Delta T_{sat} = 30$ to 40 K) a process similar to nucleate boiling apparently set in. Nucleation sites, bubble departure volumes and frequencies all increased with increasing temperature difference and the latent heat flux rose sharply. Beyond $\Delta T_{sat} = 40$ K the mercury surface became unstable and bubble observations became impossible.

2.6.7 Measurements of van Stralen: Water Boiling from Thin Wire

In 1967 van Stralen [65] presented latent heat transport measurements which corroborated those of Rallis et al [1] and of Rallis and Jawurek [2] and raised some additional noteworthy points.

His test liquid was water at saturation temperature under atmospheric pressure; the heater was a nickel wire of 0,20 mm diameter. His boiling curve (taken from Figure 1, Part III, of his paper) is shown in Figure 2.22 on linear co-ordinates; that of Rallis and Jawurek is included for comparison. At the three data points shown van Stralen took ciné records at 6000 frames per second and evaluated the mass velocity of vapour departing from the surface. The corresponding heat flux, that is $(q/A)_{LH}$, is represented by the lengths of the dotted lines in Figure 2.22. The natural convection data were apparently extrapolated by an equation of the form $(q/A)_{NC} = a \Delta T_{sat}^b$, with a and b constants.

Before discussing the latent heat transport results some remarkable features of the boiling curve should be noted.

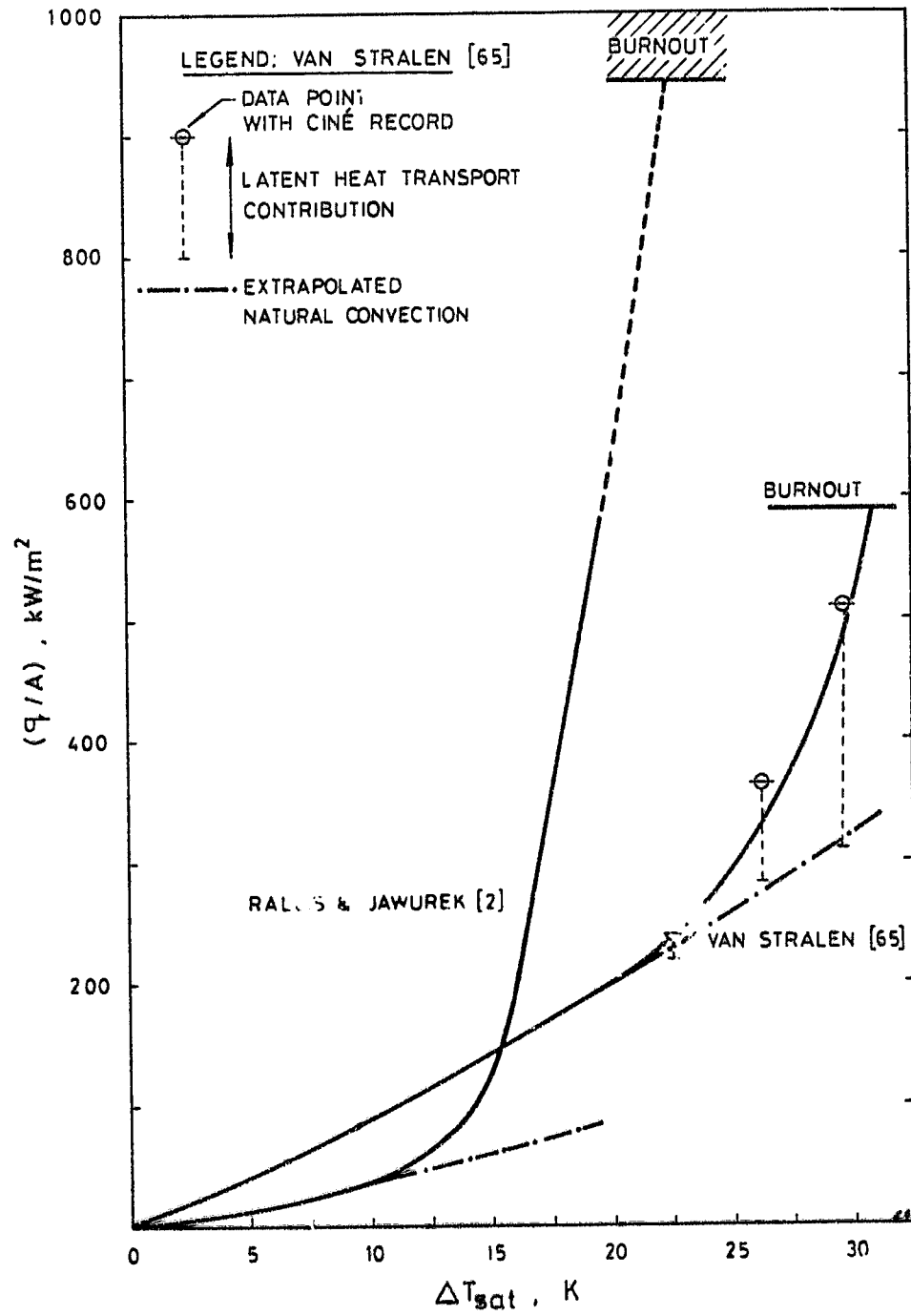


FIG. 2.22 BOILING CURVE AND LATENT HEAT TRANSPORT
VAN STRALEN [65]
 Water, ultra-thin wire heater, atmospheric pressure.

Firstly, boiling started at the unusually high superheat of 20 K. This may be attributed to a very smooth heating surface, that is, a surface marked by unusually small vapour-nucleating imperfections or cavities.

Secondly, the critical heat flux (burnout) had the very low value of 590 kW/m^2 . McAdams [89], by comparison, using an identical wire, obtained a value of 1000 kW/m^2 . This difference illustrates that on heaters of very low thermal capacity burnout is a local phenomenon [90]: A high local vapour production with local vapour blanketing can cause a premature burnout irrespective of the mean vapour flow over the entire heater.

Thirdly, van Stralen's natural convection coefficients are exceptionally high. In Figure 2.25 his data, together with those of Rallis and Jawurek, are compared with McAdams' empirical correlation ([51], p. 176) for natural convection from horizontal cylinders. At a value of $\log(\text{Gr} \cdot \text{Pr})$ of 1,0 van Stralen's value of $\log \text{Nu}$ is seen to be 0,4 instead of the expected 0,15; that is, his heat transfer coefficients were high by a factor of 1,78. This augmentation must be attributed to additional convection from the heated floor of his boiling tank (see Appendix 1, Part III of his paper). Rallis and Jawurek took precautions to minimise stray convection from their auxiliary tank heater (see Appendix 2); nevertheless their coefficients exceeded the correlation values by some 10 per cent.

In Figure 2.24 van Stralen's three data points are replotted as the ratios of the latent heat flux and of the extrapolated natural convection flux to the total heat flux. With allowance for the peculiarities of the experiment these results are in agreement with those of Rallis et al [1] and of Rallis and Jawurek [2]. The low value of the ratio $(q/A)_{\text{III}}/(q/A)_T$ at burnout (0,45) merely reflects the low

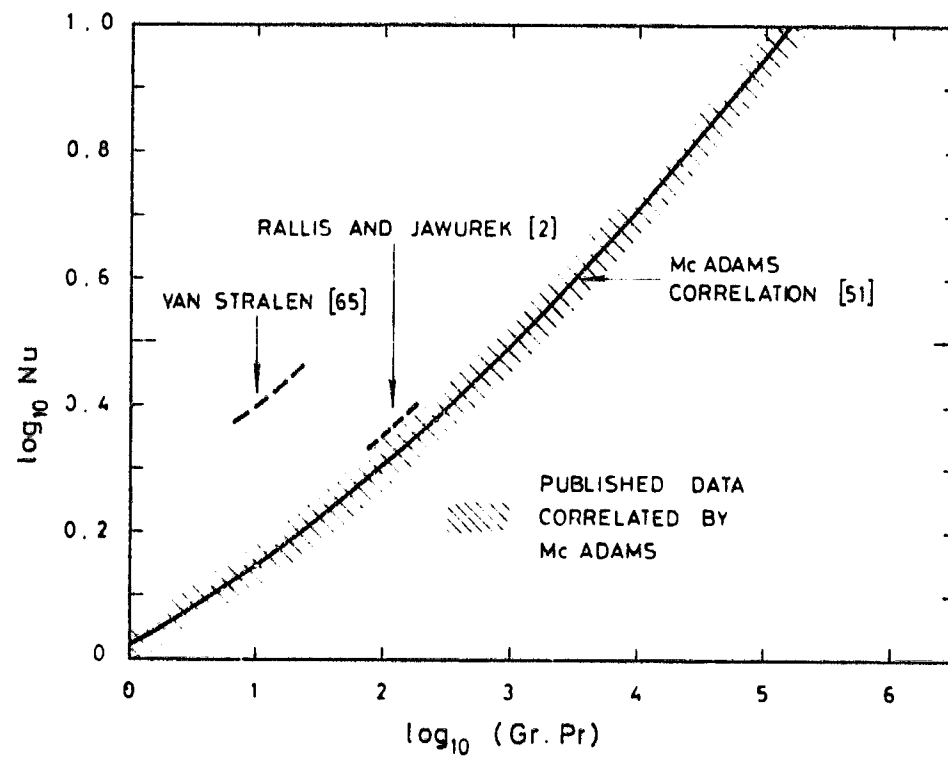


FIG. 2.23 NATURAL CONVECTION DATA COMPARED
WITH Mc ADAMS CORRELATION

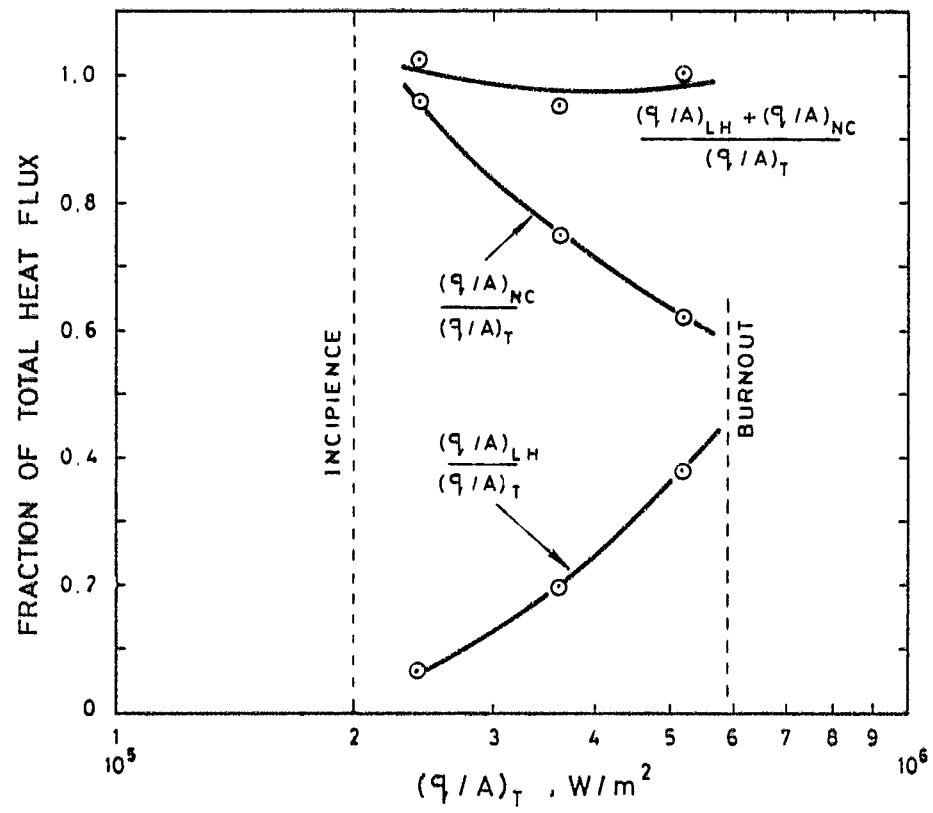


FIG. 2.24

LATENT HEAT TRANSPORT CONTRIBUTION
 VAN STRALEN [65]

Water, ultra-thin wire, atmospheric pressure.

burnout heat flux. Had burnout occurred at about 1000 kW/m^2 as usual, then, as a rough extrapolation in Figure 2.22 will show, the ratio $(q/A)_{\text{LH}}/(q/A)_T$ would most likely have been in the region of 0,8 to 1,0, as previously found.

Further, Figure 2.24 shows, again in agreement with previous findings, that the sum of the latent heat flux $(q/A)_{\text{LH}}$ and of the extrapolated natural convection flux $(q/A)_{\text{NC}}$ constitutes the total heat flux.

Estimates of the heat flow ratio q_{LH}^i/q_T^i are complicated by the unknown magnitude of the area of bubble influence on thin-wire heaters and by the absence of bubble coalescence data.

However, as a rough test, the lowest latent heat data point in Figure 2.22, that is, the point corresponding very nearly to the incipience of boiling, may be analysed by equation 2.55 with the area correction A_{NC}/A_T omitted. The simplified equation

$$q_{\text{LH}}^i/q_T^i = (q/A)_{\text{LH}} / \left[(q/A)_T - (q/A)_{\text{NC}} \right] \quad (2.57)$$

thus ignores the areas of bubble influence unavailable for natural convection. Although these are few at the onset of boiling an overestimate of q_{LH}^i/q_T^i must result. Further, the denominator in equation 2.57 is, for the onset of boiling, a difference between two nearly equal numbers, a situation conducive to magnified errors in q_{LH}^i/q_T^i .

The value of q_{LH}^i/q_T^i for van Stralen's data point is 1,54. In view of the potential errors this may be taken as a rough confirmation of earlier findings.

It may thus be cautiously concluded that in van Stralen's experiment, as in previous studies, the area-of-influence heat transport from the thermal layer into the bulk took

place predominantly by latent heat transport. The remaining bubble-induced convection could be approximated by natural convection.

2.6.8 Measurements of Schwartz and Mannes; Water Boiling at Reduced Gravities

Latent heat transport data at reduced gravities were obtained by Schwartz and Mannes, apparently in 1967, and reported in a review article by Siegel [91].

The low-gravity conditions, of duration 8 to 10 seconds, were attained in an aircraft executing controlled trajectories.

The test liquid was water at saturation temperature and atmospheric pressure. The heater was a horizontal, electrically heated metal ribbon.

The photographically determined latent heat transport results are shown in Figure 2.25 with the data points (which showed strong scatter) omitted. Added to the figure are the critical heat fluxes, $(q/A)_c$ for each acceleration. The value at terrestrial gravity is that given by Zuber's correlation [92]; values at lower gravities were estimated from the well-tested relation [91]:

$$(q/A)_{c(g)}/(q/A)_{c(g_c)} = (g/g_c)^{1/4} \quad (2.58)$$

No data on natural convection, bubble dynamics and nucleation were presented by the review. Figure 2.25, however, contains sufficient information for a rough assessment of the area-of-influence latent heat contribution q_{LH}^i/q_T^i .

At the highest heat flux measured in each of the reduced-gravity runs, 54.6 kW/m², the ratio $(q/A)_{LH}/(q/A)_T$ ranged

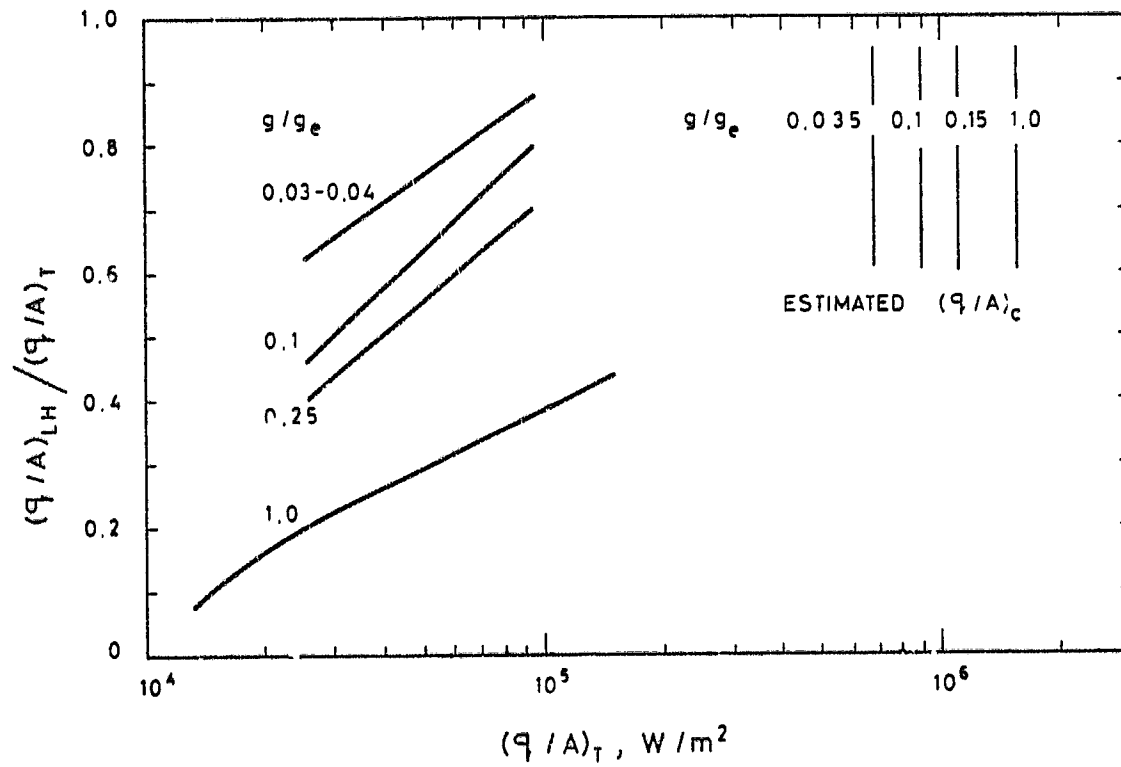


FIG. 2.25 LATENT HEAT TRANSPORT CONTRIBUTION AT REDUCED GRAVITIES. SCHWARTZ AND MANNES [91]
 Water, metal ribbon, atmospheric pressure.

from 0,70 to 0,43 for g/g_e values of 0,25 to 0,035. If in each case the heater was entirely occupied by areas of bubble influence then we have $q_{LH}^i/q_T^i = 0,70$ to 0,88 for $g/g_e = 0,25$ to 0,035. Despite the low heat flux (8,5 to 14 per cent of $(q/A)_c$) such surface coverage would appear possible owing to the rapid increase of bubble departure diameter with reducing gravity [93, 94]. If, on the other hand, the heater still supported areas of undisturbed natural convection, then the ratio q_{LH}^i/q_T^i was even larger. It would appear then, that our previous conclusions of a large or predominant contribution of q_{LH}^i to q_T^i also holds at reduced gravities.

For the test at terrestrial gravity lateral bubble coalescence may be assumed to have set in at a flux of about 200 kW/m². Then, by previous arguments $(q/A)_{LH}/(q/A)_T$ equals q_{LH}^i/q_T^i , which a small extrapolation in Figure 2.25 gives as approximately 0,5. This result is in agreement with the value of 0,51 obtained (probably by underestimation) by Rallis and Jawurck [2] at the same heat flux.

2.6.9 Measurements of Judd and Merte; Freon Boiling at Normal and Multi-g Accelerations

In 1972 Judd and Merte [95], in evaluating a number of published heat transfer models, compared the predicted heat fluxes against new data on Freon boiling at varying levels of subcooling and centrifugal acceleration. Their data contained determinations of the visible latent heat flux $(q/A)_{LHvis}$, which lend themselves to estimates of the ratio q_{LH}^i/q_T^i .

The test liquid, Freon 113, boiled at a pressure of 57,4 kN/m² on an oxide-coated, electrically-heated glass disc. High-speed ciné photography from beneath the transparent heater furnished the bubble measurements entering into the following expression for the visible latent heat flux:

$$(q/A)_{LHvis} = \rho_V \lambda \left[(\pi/6) D_{max}^3 \right] FN \quad (2.59)$$

where D_{max} was the maximum diameter of attached bubbles and, on the assumption of a spherical bubble geometry, the term in square brackets was the maximum bubble volume.

For purposes of separating the boiling and non-boiling heat transfer components Judd and Merte invoked an expression similar to equation 2.18, namely

$$(q/A)_{NB}' = (q/A)_T - (q/A)_{NC} (A_{NC}/A_T)_{inst} \quad (2.60)$$

where $(q/A)_{NB}'$ was a modified heat flux due to the nucleate boiling process alone, differing from $(q/A)_{NB}$ of the present treatment by the replacement of A_{NC}/A_T with the instantaneous area fraction $(A_{NC}/A_T)_{inst}$, where

$$(A_{NC}/A_T)_{inst} = (1 - A_i N_{inst}) \quad (2.61)$$

and N_{inst} is the instantaneous concentration of attached bubbles per unit heater area.

The natural convection heat flux was obtained from the relation

$$Nu = 0,160 (Gr. Pr)^{1/3} \quad (2.62)$$

where the constants had been verified at varying accelerations. The area of influence was given by

$$A_i = 3 (\pi/4) D_{max}^2 \quad (2.63)$$

that is, with D_i assumed equal to $3/4 D_{max}$, a choice which the authors had justified previously [66].

The combination of equations 2.59 to 2.63 gave the ratio $(q/A)_{LHvis}/(q/A)_{NB}'$ as follows:

$$(q/A)_{LHvis}/(q/A)_{NB}' = (q/A)_{LHvis}/\left[(q/A)_T - (q/A)_{NC}\{1-3(\pi/4) D_{max}^2 N_{inst}\}\right] \quad (2.64)$$

Judd and Merte's values of this quantity are shown as a function of subcooling and acceleration in Figure 2.26.

It will be recalled that at bulk saturation the visible latent heat flux $(q/A)_{LHvis}$ is equal to the total latent heat flux $(q/A)_{LH}$ and, further, that the flux ratio $(q/A)_{LH}/(q/A)_{NB}'$ is equal to the area-of-influence ratio q_{LH}^i/q_T^i . Thus at $\Delta T_{sub} = 0$ the ratio $(q/A)_{LHvis}/(q/A)_{NB}'$ in Figure 2.26 is a measure of q_{LH}^i/q_T^i . More precisely, it is an overestimate of q_{LH}^i/q_T^i since the area fraction $(1 - A_i N_{inst})$ of Judd and Merte's is larger than the area fraction $(1 - A_1 N)$ available for permanently undisturbed natural convection. Judd and Merte's treatment, in fact, implies that during the waiting period an area of bubble influence supports both natural convection and bubble-associated heat transfer. The present treatment, in accordance with definition, considers all area-of-influence heat transfer as bubble-associated.

Despite this difference Figure 2.26 immediately shows that at multi-g accelerations latent heat transport ceases to dominate the total area-of-influence heat transfer: At $g/g_c = 10$ the latent heat fraction q_{LH}^i/q_T^i is smaller than 0,25 and at $g/g_c = 100$ it would appear to be smaller than 0,08.

Under terrestrial gravity ($g/g_c = 1$) Figure 2.26 shows the ratio $(q/A)_{LH}/(q/A)_{NB}'$ as 0,78 independently of heat flux. Sufficient

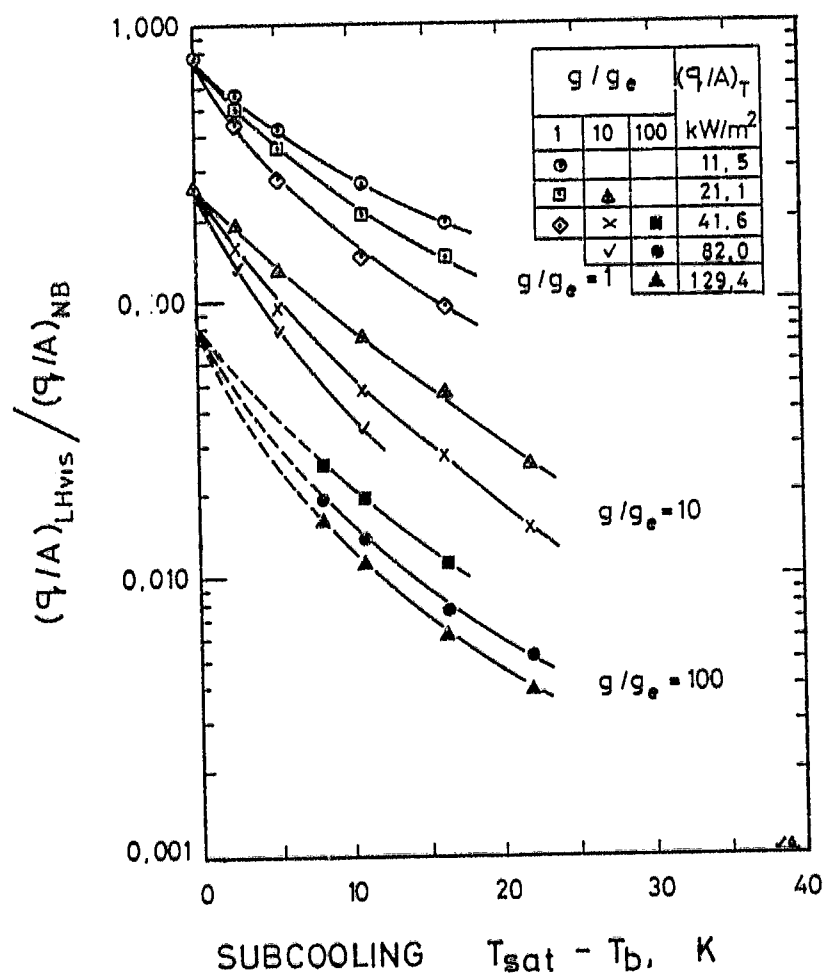


FIG. 2.26 VISIBLE LATENT HEAT TRANSPORT vs
SUBCOOLING AT VARIOUS ACCELERATIONS.
JUDD AND MERTE [95]
 Freon 113, coated glass plate heater,
 57,4 kN/m^2

information is given in Judd and Merte's paper to correct this ratio to q_{LH}^i/q_T^i as given by equation 2.65:

$$q_{LH}^i/q_T^i = (q/A)_{LH}/(q/A)_{NB} = (q/A)_{LH} / \left[(q/A)_T - (q/A)_{NC} \{1 - 3(\pi/4) D_{max}^2 N\} \right] \quad (2.65)$$

The correction is small and is given in detail in Appendix 3. The resulting values of q_{LH}^i/q_T^i are 0,67; 0,72 and 0,78 at total heat fluxes of 11,5; 21,1 and 41,6 kW/m². Thus once again the area-of-influence heat transport from the thermal layer into the bulk is found to be largely due to latent heat transport.

Judd and Merte originally presented Figure 2.26 as an assessment of Rallis and Jawurek's 'latent heat transport model' [2]. The assessment, however, misrepresents the model by ascribing to it the predominance of visible latent heat transport irrespective of subcooling. In actual fact Rallis and Jawurek referred exclusively to saturated boiling, that is, to total latent heat transport and, tacitly, to terrestrial gravity. Under such conditions the results of the two studies are essentially in agreement.

2.6.10 Microlayer Studies and Latent Heat Transport

The thin liquid films or microlayers, first postulated by Snyder [12] and Edwards [13] have since the pioneer work of Moore and Mesler [67] been the subject of extensive experimentation. Local surface temperature measurements [67, 96 - 102], scale deposition techniques [103] and especially optical methods [10 - 111] have firmly demonstrated the existence of microlayers for a wide variety of surface-liquid combinations at atmospheric and subatmospheric pressure.

The local surface temperature measurements beneath and in the vicinity of bubbles led to estimates of the heat flow due to microlayer evaporation, q_{ML}^i , and to the remaining surface-to-liquid heat flows in the area of bubble influence, q_{LA}^i and q_{LW}^i . While doubt has been cast [61, 112] on the accuracy in some surface temperature studies and others dealt with specialised forms of boiling (e.g. boiling from low thermal capacity heaters, boiling with single, individually-triggered bubbles) a fairly unified set of conclusions has nevertheless emerged from them. These are: (1) that the microlayer heat fluxes exceed the mean surface heat fluxes by factors of 6 to 20 [67, 96 - 98], (2) that at high bubble concentrations microlayer evaporation dominates the total boiling heat transfer [67, 97], (3) that under many conditions the microlayer heat transfer, q_{ML}^i , is a significant or even the predominant contributor to bubble growth [98 - 102] and (4) that the remaining surface-to-liquid heat transfer processes in the area of bubble influence, q_{LA}^i and q_{LW}^i , are of minor significance compared with microlayer evaporation [96, 97, 99, 100, 102].

A full review of these studies (and of one optical study which apparently contradicts them [104]) is given in Chapter 4. The present remarks are merely intended to show that the microlayer approach has largely confirmed the latent heat transport measurements and indeed has gone beyond them in identifying microlayer evaporation as a possible key contributor to latent heat transport.

2.7 SUMMARY AND CONCLUSIONS

This chapter has attempted to provide, by way of an interpretive overview of the literature, a broad outline of the mechanism of nucleate pool boiling heat transfer. More precisely, it has attempted to identify and to classify quantitatively the chief groups of transfer processes which

together constitute the boiling mechanism. The treatment was largely confined to saturated pool boiling in the presence of isolated bubbles.

The experimental studies available for review were mostly conducted at terrestrial gravity and at atmospheric or lower pressures. The validity of the conclusions given below is thus generally limited to these conditions.

1. It was shown that the region of isolated bubbles each bubble has associated with it an 'area of bubble influence', that is, a limited portion of the heating surface the heat transfer from which is intensely influenced by the action of the bubble. As an approximation, bubble action, and thus the boiling process, was considered as entirely limited to such areas of influence. The remaining undisturbed surface was considered to support natural convection.

2. Each area of bubble influence was shown to support a complex pattern of primary transfer processes at the heating surface, and subsequent redistributions of energy in the fluid phases. Mean rates of heat transfer, q^i , were defined for each transfer process or related group of processes. The chief features of their breakdown and interrelation were:

- (a) q_T^i , the total area-of-influence heat transfer rate is essentially identical to q_L^i , the primary heat transfer from the surface into the adjacent liquid.
- (b) q_L^i is resolved into: firstly, q_{LW}^i , the heat transferred to the liquid during the waiting period; secondly, q_{ML}^i , the heat transferred to the liquid microlayer and causing microlayer evaporation; and thirdly, q_{LA}^i the heat transferred into the liquid annulus surrounding the attached bubble.

- (c) q_{LW}^i and q_{LA}^i build up the thermal layer and are redistributed into q_{BWE}^i , the heat transferred due to evaporation at the walls of attached bubbles (micro-layer excluded), and q_{BC}^i , the heat transferred from the thermal layer into the bulk by bubble-induced convection.
- (d) q_{ML}^i and q_{BWE}^i together constitute the latent heat transport q_{LH}^i , that is, the heat transfer due to evaporation at all surfaces of the attached bubble.
- (e) The primary heat transfer processes, q_{LW}^i , q_{ML}^i and q_{LA}^i thus manifest themselves partly as latent heat transport and partly as bubble-induced convection:

$$q_T^i = q_{LH}^i + q_{BC}^i$$

- (f) In subcooled boiling q_{LH}^i is redistributed into $q_{CONDENS}^i$, that portion of q_{LH}^i which recondenses from attached bubbles, and q_{LHvis}^i , the remaining portion of q_{LH}^i which visibly detaches from the surface as vapour.

3. The review showed that the concept of negligible latent heat transport, $q_{LH}^i \ll q_T^i$, derives mainly from subcooled boiling studies in which not q_{LH}^i , but q_{LHvis}^i was estimated and found to be negligible. Neglect of the term $q_{CONDENS}^i$, loose semantics and arbitrary extrapolation led to the widespread application of the concept to all conditions of nucleate boiling.

4. With q_{LH}^i considered negligible, bubble-induced convection, q_{BC}^i , became generally accepted as the predominant thermal-layer-to-bulk heat flow (i.e. $q_{BC}^i \approx q_T^i$). The review dealt with some of the many resulting mechanistic suggestions and heat transfer models for q_{BC}^i and for the related primary heat flows q_{LW}^i and q_{LA}^i . In the most widely accepted of these, the numerically successful 'bulk convection'

models, q_{LW}^i and q_{LA}^i were approximated by transient conduction into the liquid, q_{ML}^i was neglected, and q_{BC}^i was ascribed to the periodic stripping-away of the entire transient-conduction layer by departing bubbles.

5. The review of experimental studies on the thermal boundary layer, however, indicated (at least for saturated and near-saturated boiling) that q_{BC}^i , while significant, does not constitute the predominant fraction of q_T^i . The primary heat transfers q_{LW}^i and q_{LA}^i , on the other hand, were found to be convective and controlled by bubble movement: The convection velocities of q_{LW}^i appeared to arise from the wake flow of departing bubbles; the convection velocities of q_{LA}^i emerged as the resultants of the wake flow of the previous bubble and the outward flow induced by the growth of the attached bubble.

6. Next, the review dealt with latent heat transport in the area of bubble influence. For saturated boiling both indirect arguments and bubble measurements indicated that the previously neglected latent heat transport is in fact the major mode of heat transfer from the thermal layer into the bulk. A summary of the experimental latent heat transport determinations is given in Table 2.1. The measurements cover a wide range of test conditions, heating surfaces and fluids. Those conducted at terrestrial and reduced gravities all indicate values of q_{LH}^i/q_T^i greater than 0,5 and in most cases greater than 0,7. At multi-g accelerations, however, the contribution appears to fall off.

For the case of subcooled boiling it was shown that indirect and theoretical estimates of the term $q_{CONDENS}^i$ suggest strongly, if not conclusively, that $q_{LH}^i (= q_{LHvis}^i + q_{CONDENS}^i)$ is of comparable magnitude to q_T^i .

TABLE 2.1

EXPERIMENTAL DETERMINATIONS OF LATENT HEAT TRANSPORT IN
 AREA OF BUBBLE INFLUENCE

SOURCE		LIQUID	HEATER	PRESSURE kN/m ²	GRAVITY g/g _e	HEAT FLUX	q_{LI}^i/q_T^i
Rallis et al [1]	1961	water	thin wire	83	1	moderate - high	0,76 - 0,93
		ethanol	thin wire	83	1	high	1,0
Rallis & Jawurek [2]	1964	water	thin wire	83	1	moderate - high	(0,51 - 0,73)
Novaković et al [84]	1966	ethanol	mercury pool	101	1	low	0,78 - 1,0
van Stralen [65]	1967	water	thin wire	101	1	incipience	1,0 (?)
Schwartz & Mannes [91]	1967	water	flat metal ribbon	101	0,035	moderate	0,88
					0,1	moderate	0,8
					0,25	moderate	0,7
					1,0	moderate	0,5
Judd & Merte [95]	1972	Freon 113	coated glass plate	57	1	low	0,67 - 0,78
					10	low	<0,25
					100	-	<0,08 (?)

7. For saturated boiling at terrestrial gravity the total boiling heat flux was shown to be approximated closely by the sum of the latent heat flux and of the undisturbed natural convection heat flux.

$$(q/A)_T = (q/A)_{LH} + (q/A)_{NC}$$

Thus the heat flow q_{BC}^i , due to the intermittent bubble-induced convection is of approximately the same magnitude as the natural convection heat flow were it to operate undisturbed over the entire area of bubble influence.

8. Finally, by way of a preview of Chapter 4, it was indicated that microlayers exist under a wide variety of boiling conditions and that the microlayer evaporation heat flow q_{ML}^i is possibly a major contributor to the latent heat transport q_{LH}^i .

CHAPTER 3A LATENT-HEAT-BASED CORRELATION OF SATURATED
NUCLEATE POOL BOILING HEAT TRANSFER3.1 INTRODUCTION3.1.1 Basic Approach and Outline

In this chapter the mechanistic conclusions reached in Chapter 2 are applied to the problem of correlating and predicting heat transfer rates during saturated nucleate pool boiling.

Such boiling, Chapter 2 showed, exhibits several classes of bubble behaviour. The simplest of these, occurring at low heat fluxes, is characterised by isolated bubbles, that is, by bubbles so distributed on the heating surface that each is essentially unaffected by its neighbour. The present analysis is initially restricted to this case.

Each isolated bubble strongly influences the heat transfer over its surrounding 'area of bubble influence'. Outside the areas of influence the heating surface is assumed to be undisturbed by bubble action and to support natural convection.

Under such conditions the requirements of a heat transfer correlation are:

- (1) to predict the heat transfer within an area of bubble influence,
- (2) to sum this over all areas of bubble influence, and

- (3) to predict the heat transfer from the remaining non-boiling area.

The correlation of the area-of-influence heat transfer should ideally be based on the primary surface-to-liquid heat transfer processes, that is, on the heat flows q_{LW}^i , q_{ML}^i and q_{LA}^i . These processes are shown to be extremely complex, and amenable to analysis only if severely simplifying assumptions are made. An examination of the bulk convection correlations illustrates this difficulty.

An alternative approach is suggested. It is founded upon the experimental findings (Chapter 2) that (1) the primary q_L^i heat flows manifest themselves predominantly as latent heat transport, q_{LH}^i , and (2) that the remainder, the bubble-induced convection heat flow q_{PC}^i , may be approximated by steady-state natural convection operating over the entire area of bubble influence.

The resulting model is analysed without recourse to further mechanistic assumptions. Coupled with a method of surface characterisation (similar to that of Mikic and Rohsenow [47]) this leads to a new and realistic heat transfer correlation. Comparisons with experimental data are presented. These indicate that the correlation allows the prediction of boiling curves q/A versus ΔT_{sat} at any pressure, provided that at least one boiling curve for the same surface, or preferably the same surface-liquid combination, is available as a reference.

3.1.2 Ideal Treatment of Area of Influence Heat Transfer

Ideally the correlation of heat transfer in the area of bubble influence should be based on the primary transfer processes at the heating surface, that is on q_{LW}^i , q_{ML}^i and q_{LA}^i . These processes are complex and insufficiently investigated.

For example, the mechanism of q_{LW}^i would appear to be intermittent convection arising from the wake flow of the departed bubble (see Section 2.4). The radial velocity varies throughout the waiting period, while simultaneously the surface temperature of the heater (recovering from microlayer evaporation) varies both with time and position (see Fig. 3.1). The convection velocities governing q_{LA}^i are more complex; they are the resultants of the wake flow of the previous bubble and the outward flow induced by the growth of the attached bubble. Concerning q_{ML}^i , Part 2 of this thesis shows that there is as yet no agreement on initial microlayer thickness or on the flow pattern within it.

Attempts at the mechanistically detailed correlation of each of these processes would at present appear to be unprofitable. Nevertheless such correlations have been attempted, the most noteworthy being the 'bulk convection' correlations.

3.1.3 Bulk Convection Correlations

The numerically successful boiling correlations of Han and Griffith [42] and Mikic and Rohsenow [47] offer remarkable examples of a simplified treatment of the area-of-influence heat transfer. In both cases all q_L^i processes were approximated by cyclic transient conduction into the liquid.

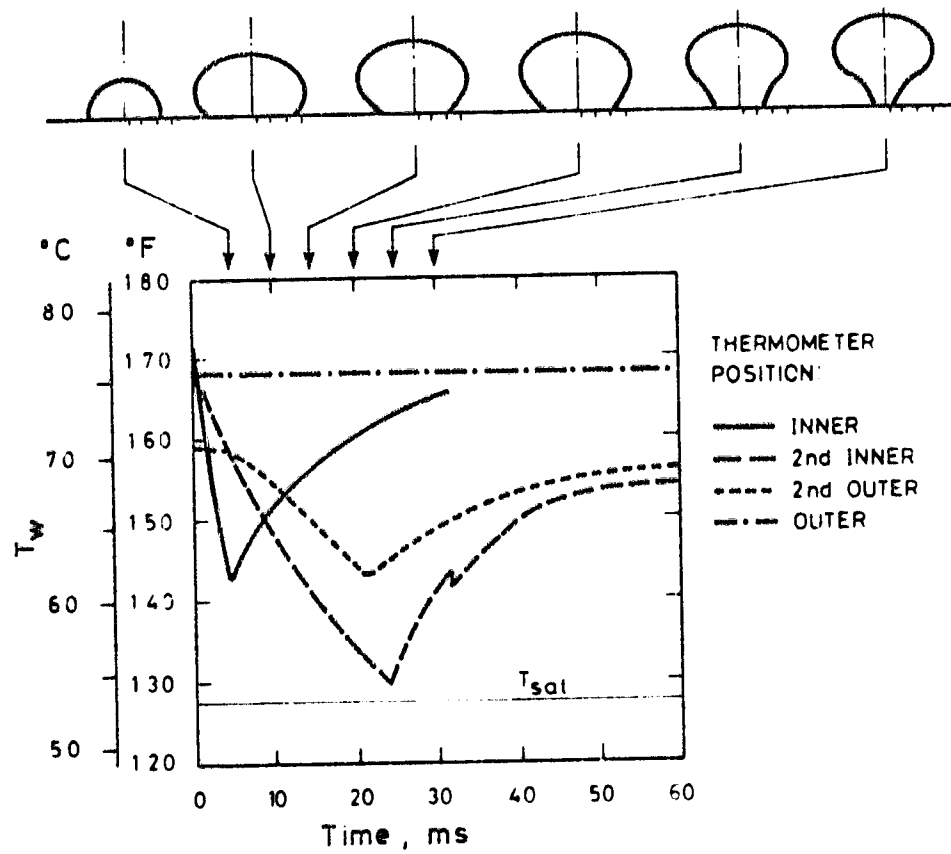
The conduction cycles (fully described in Section 2.3) were each subject to:

Initial conditions:

$$T = T_{w(\text{mean})} \quad \text{at } x = 0, t = 0 \quad (3.1a)$$

$$T = T_{\text{sat}} \quad \text{at } x = 0, t = 0 \quad (3.1b)$$

(a) BUBBLE PROFILES vs TIME
Marks on bubble base show position of surface thermometers.



(b) LOCAL SURFACE TEMPERATURE OF HEATER

FIG. 3.1 TEMPERATURE FLUCTUATION ON HEATER SURFACE
COOPER AND LLOYD [99]

Liquid : Toluene
Surface : Glass
Pressure : 13,8 kN/m²
Mean flux : 47,3 kW/m²
 ΔT_{sub} : 7,8 K

Boundary conditions:

$$T = T_{w(\text{mean})} \text{ at } x = 0, \quad t > 0 \quad (3.1c)$$

$$T = T_{\text{sat}} \text{ at } x = \infty, \quad t > 0 \quad (3.1d)$$

where $t = 0$ corresponds to bubble departure, $x = 0$ demarkates the heater-liquid interface and the region $x > 0$ is the liquid.

With these conditions and with the liquid considered infinite in the x direction, the solution of the one-dimensional conduction equation is

$$q/A = k_L \Delta T_{\text{sat}} / (\pi \alpha t)^{1/2} \quad (3.2)$$

The two bulk convection models differ in the detailed application of equation 3.2. In the simpler model, that of Mikic and Rohsenow [47], transient conduction as given by equation 3.2 extends over the entire bubble area, that is, the bubble contact area is negligible. The transients repeat at the frequency of bubble departure, f , and hence the mean area-of-influence heat flux is

$$(q/A)_{\text{mean}}^i = f \int_0^{1/f} k_L \Delta T_{\text{sat}} / (\pi \alpha t)^{1/2} dt \quad (3.3)$$

The remaining derivation of the correlation is given in Section 2.3.4.

The area-of-influence energy flows and transfer mechanisms implied by this model may be summarised as:

$$q_T^i = q_L^i = (q_{LW}^i + q_{LA}^i + q_{LBB}^i)_{\text{con}} = q_{BC}^i \quad (3.4)$$

where

q_{LBB}^i (LBB = LIQUID, BUBBLE BASE) is the hypothetical mean

heat transfer into the liquid over the area actually occupied by the bubble base,

and

Subscript ' ∞ con' indicates pure conduction into the semi-infinite liquid.

The scheme given by equation 3.4 and the associated boundary conditions of equation 3.1 differ severely from reality. When applied together, however, they are mutually corrective.

Surface microthermometry studies (see, for example, Fig.3.1) show that, over a substantial portion of the area of influence, the heater temperature is not $T_{w(\text{mean})}$, as stated by equation 3.1a and 3.1c, but is considerably lower. Schlieren and liquid microthermometry studies (see Section 2.4.1) indicate that the temperature of liquid rushing to the surface after bubble departure is not T_{sat} , as stated by equation 3.1b, but is higher. The model thus overestimates considerably the ΔT driving force available for the assumed conduction. Through neglect of the bubble base area (see term q_{LBB}^i , equation 3.4) the model furthermore overestimates the area available for such conduction. Clearly these overestimates correct for the omission of the microlayer term, q_{ML}^i , and of the convective effects in q_{LW}^i and q_{LA}^i .

Similar arguments apply, with minor modifications, to the more elaborate model of Han and Griffith [42]. The success of the bulk convection correlations must thus be attributed, at least partially, to the cancellation of unrealistic approximation errors. The most disturbing of these, namely the neglect of microlayer evaporation, was forced upon the model by adherence to the widely accepted notion of negligible latent heat transport.

3.1.4 Latent Heat Transport in Area of Bubble Influence

It was shown in Chapter 2 that the concept of negligible latent heat transport is of dubious validity in subcooled boiling and is apparently invalid in saturated boiling.

Bubble measurements in saturated boiling in fact showed that at terrestrial and reduced gravities latent heat transport generally accounts for at least 70 per cent of the total area of influence heat transfer, that is

$$q_{LH}^i \geq 0,7 \quad q_T^i = 0,7 (q_{LW}^i + q_{ML}^i + q_{LA}^i) \quad (3.5)$$

The validity of this conclusion is now assumed to extend to all conventional pure liquids (excluding liquid metals), to all heating surfaces and to all pressures supporting normal nucleate boiling. The first two extensions are well supported by the data in Chapter 2 (see Table 2.1). The extension to high pressure is a working hypothesis.

Since latent heat transport q_{LH}^i is dependent on bubble departure, volume and frequency, and since empirical expressions for these parameters are available in the literature, equation 3.5 forms a practical basis for the development of a heat transfer correlation.

3.2 DERIVATION OF NEW CORRELATION

An expression is developed below, relating latent heat transport, summed over all areas of bubble influence, to ΔT_{sat} , T_{sat} , physical properties and heater surface characteristics. The remaining area of influence heat transfer (30 per cent or less of the total) and the non-boiling heat transfer outside the areas of influence are then dealt with in a simultaneous natural-convection approximation.

3.2.1 Expression for Latent Heat Transport

For saturated isolated bubble boiling we have

$$q_{LH}^i = \rho_V \lambda f \bar{V}_d \quad (3.6)$$

or summing over all nucleation sites (that is, areas of bubble influence) per unit area,

$$(q/A)_{LH} = N \rho_V \lambda \bar{V}_d \quad (3.7)$$

where $\bar{V}_d$ is the mean volumetric vapour flow rate per nucleation site.

Expressions for N and $\bar{V}_d$ are now needed.

(a) Nucleation Site Concentration, N

Nucleation generally proceeds from vapour entrapped in microscopic surface cavities [113, 114, 115, 116]. The mouth radius of potentially active cavities, that is, cavities capable of entrapping vapour [117], will, for a certain surface-liquid combination, have some frequency distribution, for example, as in Figure 3.2. Under given conditions a range of these cavities will be active, as shown.

An approximate nucleation criterion (a slight modification of that of Griffith and Wallis [48]) may be stated as follows:

$$r_{min}^* = 2\sigma T_{sat} / \rho_V \lambda \Delta T_{sat} \quad (3.8)$$

This equation has been discredited in the past, largely because of its failure to predict boiling-nucleation from artificial cavities machined into heating surfaces [118, 119, 120]. For boiling from naturally occurring cavities,

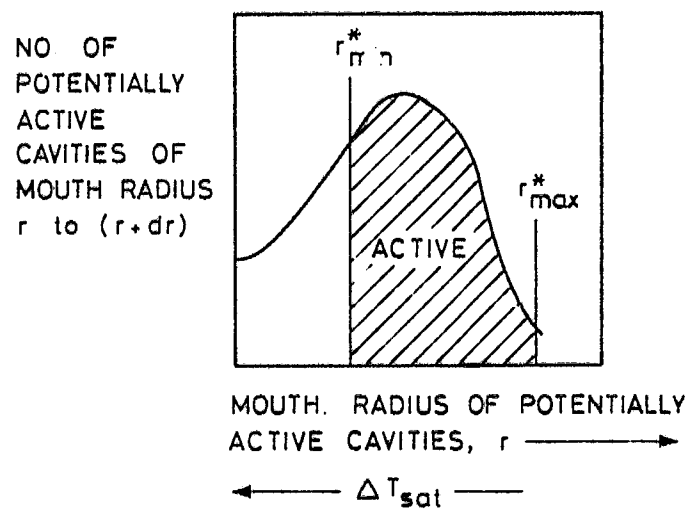


FIG. 3.2 HYPOTHETICAL SIZE DISTRIBUTION OF
NUCLEATION CAVITIES

however, indirect tests [3, 48] show that $r_{\min}^*$ is at least proportional to the terms on the right-hand side and the recent direct measurements of Shoukri and Judd [121] indicate the equation to be exact (see Appendix 4). The more elaborate nucleation criteria of Hsu [122] and Han and Griffith [42] offer little advantage over equation 3.8 and are in fact based on questionable assumptions as discussed in Appendix 4.

Equation 3.8 gives the mouth radius of the smallest cavity that will be active for a saturated liquid at a particular pressure and wall superheat. Thus, as ΔT_{sat} is raised from zero, the largest potentially active cavity on the surface is the first to become active ($r_{\max}^* = r_{\max}$). Further increase in ΔT_{sat} causes activation of progressively smaller cavities, while all larger cavities remain active. Thus N , the total number of active cavities per unit area, is given by the integral with respect to $r_{\min}^*$ of Figure 3.2, or, in other words, N versus $r_{\min}^*$ is the cumulative cavity size distribution (see also [3] and Appendix 2).

This cumulative distribution is now approximated by the simple power function

$$N = N_0 (r_{\max}/r_{\min}^*)^m \quad (3.9)$$

where $r_{\max}$ is the mouth radius of the largest potentially active cavity and N_0 is the concentration of $r_{\max}$. Figures 3.3 and 3.4 illustrate normal and lognormal distributions and the success over their lower range of the power function approximation.

An equation of the form

$$N = \text{const.}/(r_{\min}^*)^m \quad (3.10)$$

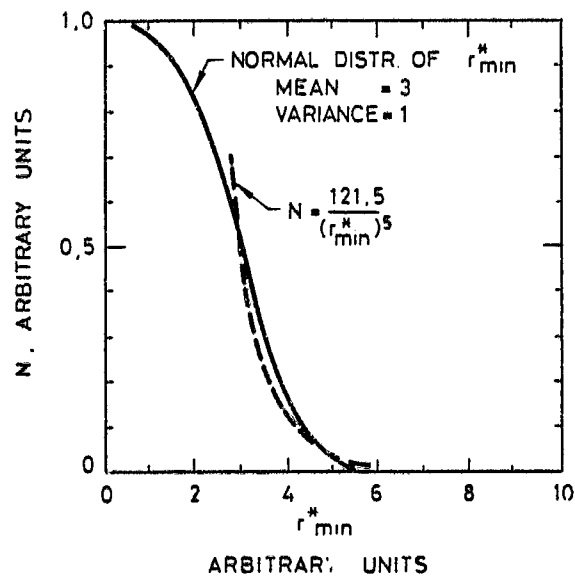


FIG. 3.3 NORMAL DISTRIBUTION OF $r_{\min}^*$ APPROXIMATED BY POWER FUNCTION.

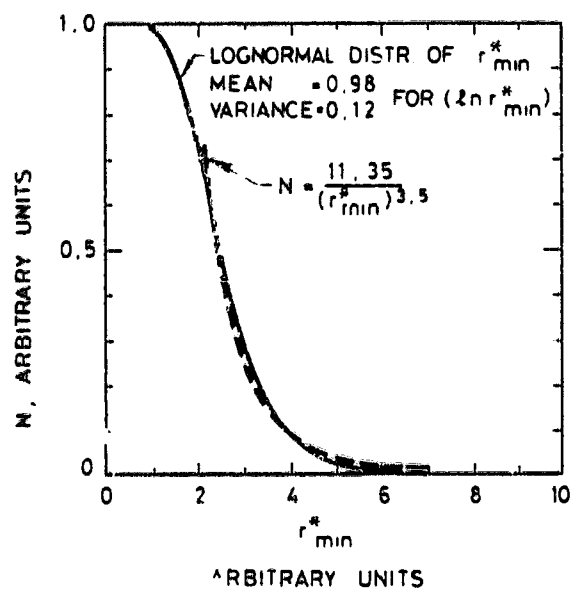


FIG. 3.4 LOGNORMAL DISTRIBUTION OF $r_{\min}^*$ APPROXIMATED BY POWER FUNCTION

was apparently first proposed by Brown on empirical grounds (see [47]) and found to hold quite well.

Combining equations 3.8 and 3.9, we obtain

$$N = N_o (r_{\max})^m (\rho_V \lambda \Delta T_{\text{sat}} / 2 \sigma T_{\text{sat}})^m \quad (3.11)$$

Essentially the same relation was obtained by Mikic and Rohsenow [47] from equation 3.8 and Brown's equation 3.10.

Equation 3.11 predicts a linear relation between $\log N$ and $\log (\Delta T_{\text{sat}})$. Figure 3.5, showing the data of Anderson [123] and Jawurek [124], supports this prediction. Further, for a particular surface-liquid combination (i.e. constant N_o , $r_{\max}$ and m) $\log N$ should vary linearly and independently of pressure with $r_{\min}^*$ (the reciprocal of the last bracketed term in equation 3.11). It would even seem possible that this relationship should be near-independent of liquid, that is, that the distribution of potentially active cavities should depend essentially on surface only. This is approximately borne out by Figure 3.6, showing the data of Anderson [123], all for the same coated glass surface, and by Figure 3.7, showing the data of Jawurek [124], all for the same platinum wire heater. (For the calculation of some unavailable physical properties of Anderson's test liquids, see Appendix 5). The correlation with respect to pressure is poor in Figure 3.6. This, however, seems to be inherent in Anderson's data, see Figure 3.5, which shows N versus ΔT_{sat} to be independent of pressure. This unusual result implies either a physical change in the surface properties of his coated glass, or alternatively, an inverse pressure dependence of surface vapour-entrapment. In Figure 3.7 the correlation with respect to pressure does approximately hold.

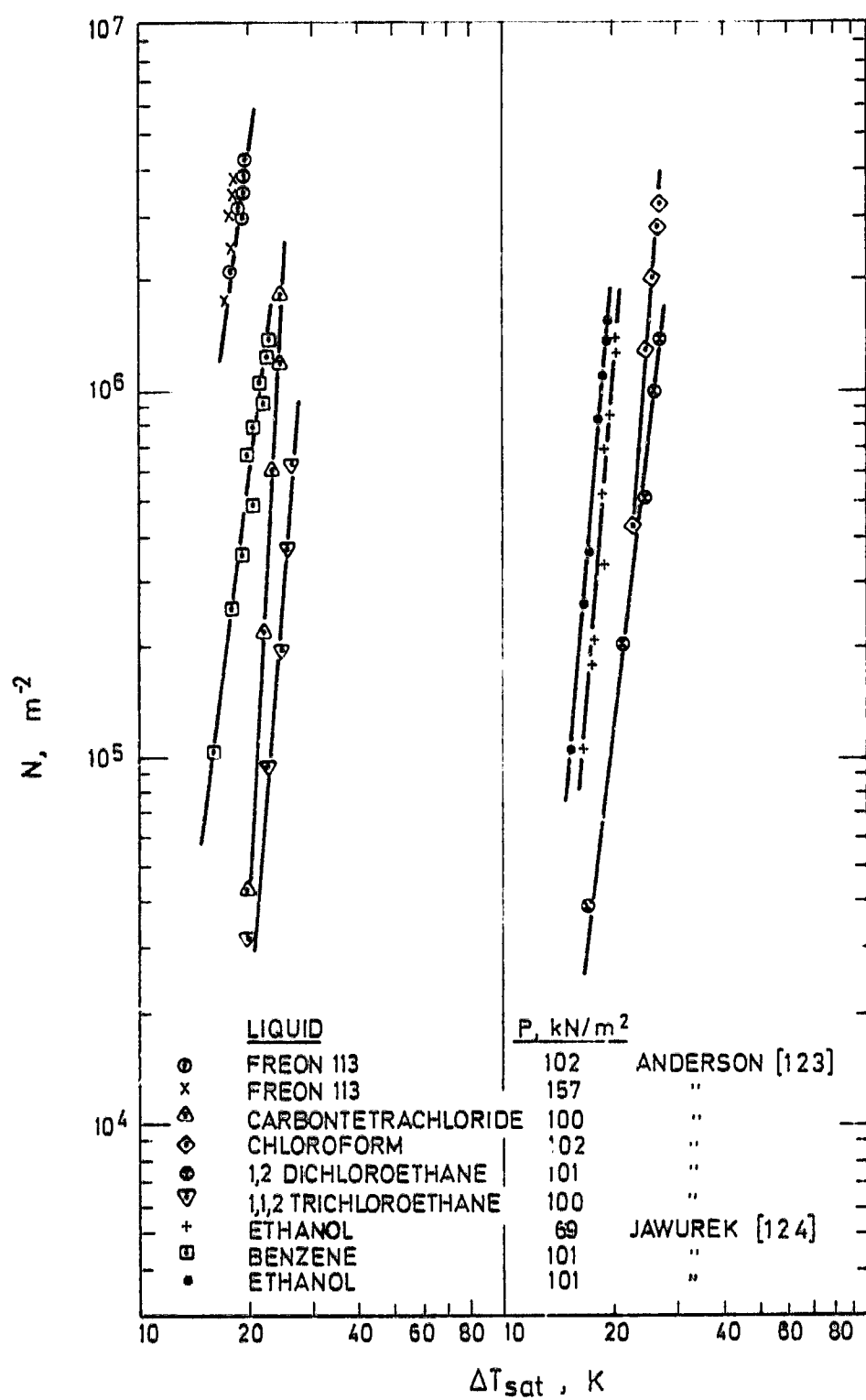


FIG. 3.5 N VERSUS ΔT_{sat} , DATA OF ANDERSON [123] AND JAWUREK [124]

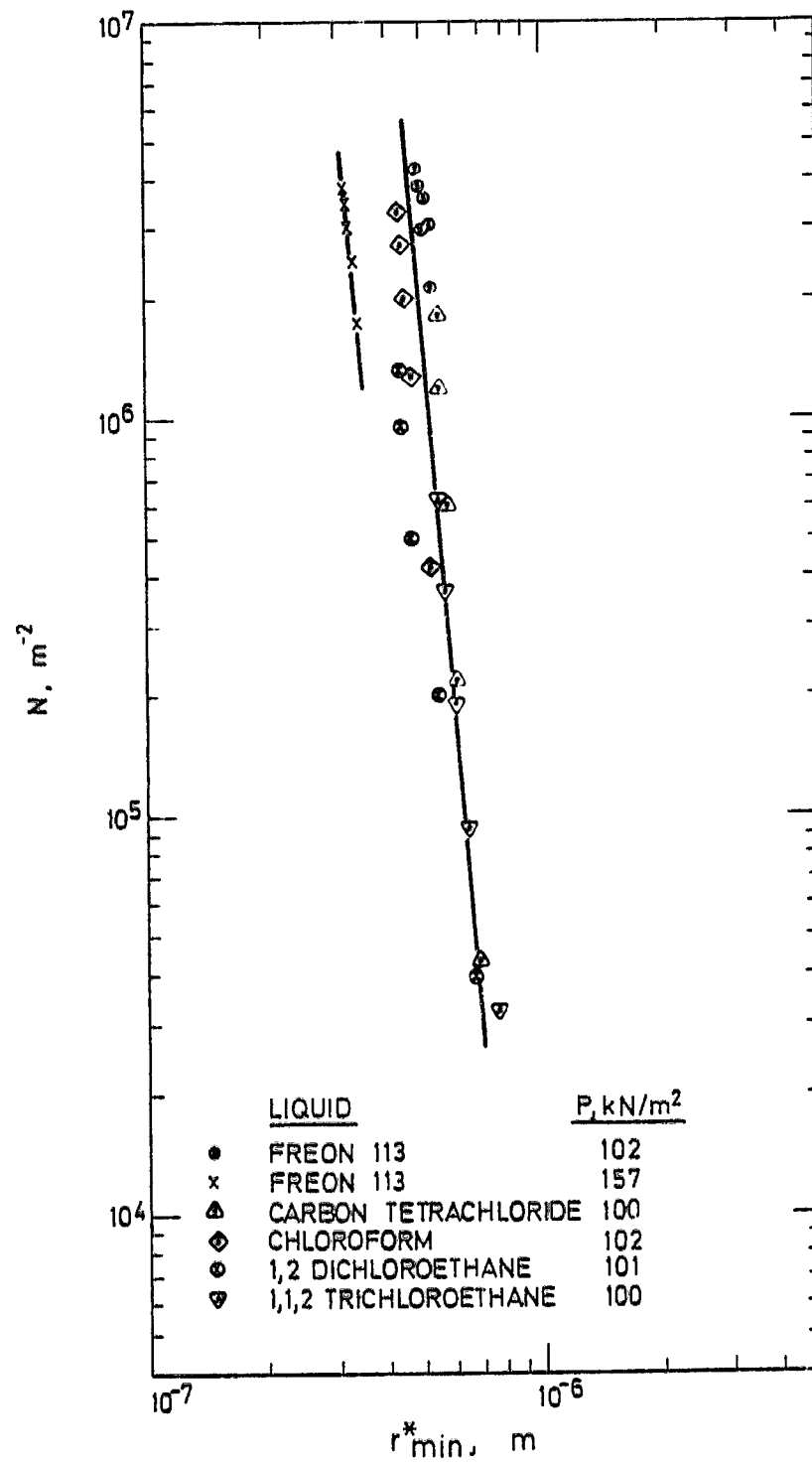


FIG. 3.6 N VERSUS $r_{\min}^*$ FOR COATED GLASS SURFACE
FROM DATA OF ANDERSON [123]

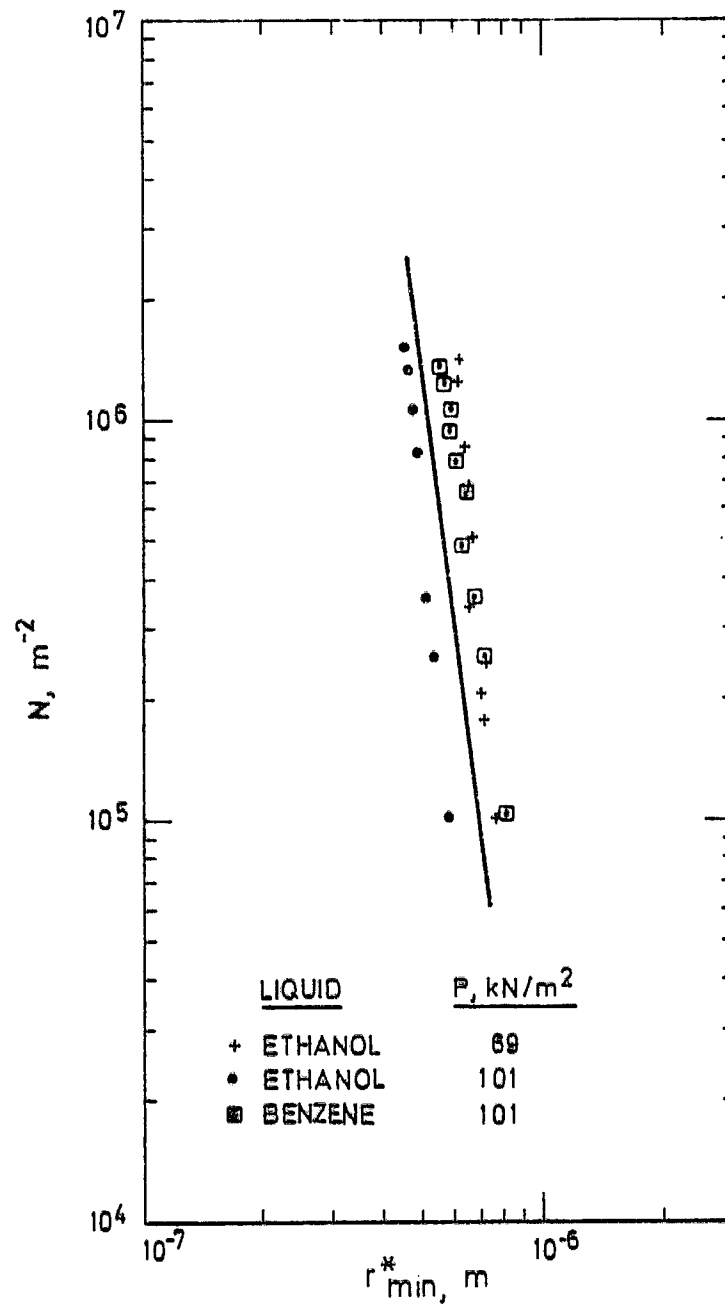


FIG. 3.7 N VERSUS r^*_{min} FOR PLATINUM SURFACE,
FROM DATA OF JAWUREK [124]

These tests only confirm proportionality between $r_{\min}^*$ and the right-hand side of equation 3.8. A proportionality constant might be necessary for numerical correctness. This constant should be carried through to equation 3.11 which can then be written in final form as

$$N = K_N (\rho_V \lambda \Delta T_{\text{sat}} / \sigma T_{\text{sat}})^m \quad (3.12)$$

K_N and m , which are generally unknown, now characterise nucleation from a surface-liquid combination or, at slightly increased risk, from a surface irrespective of liquid.**

(b) Mean Volumetric Vapour Flow Rate per Nucleation Site, $\bar{FV}_d$

A large number of expressions have been proposed, mostly on semi-empirical grounds, for the product of bubble frequency and bubble departure volume (or diameter). Extensive reviews are given by Ivey [125] and Cole [50].

** An abbreviated version of this analysis (and of the entire boiling correlation) was published in 1974 [4]. In 1975 Shoukri and Judd [121] proposed a similar treatment of nucleation and furthermore claimed that in the work of Anderson, Judd and Merte [88] there appeared a test of the nucleation correlation apparently identical to that given in Figure 3.6. No such test can, however, be found in the paper, the study dealing exclusively with Gaertner's nucleation criterion [87]. Wherever Judd's equivalent of Figure 3.6 appeared (possibly in discussion following the paper) it was, and remains, unavailable in the present, independent study.

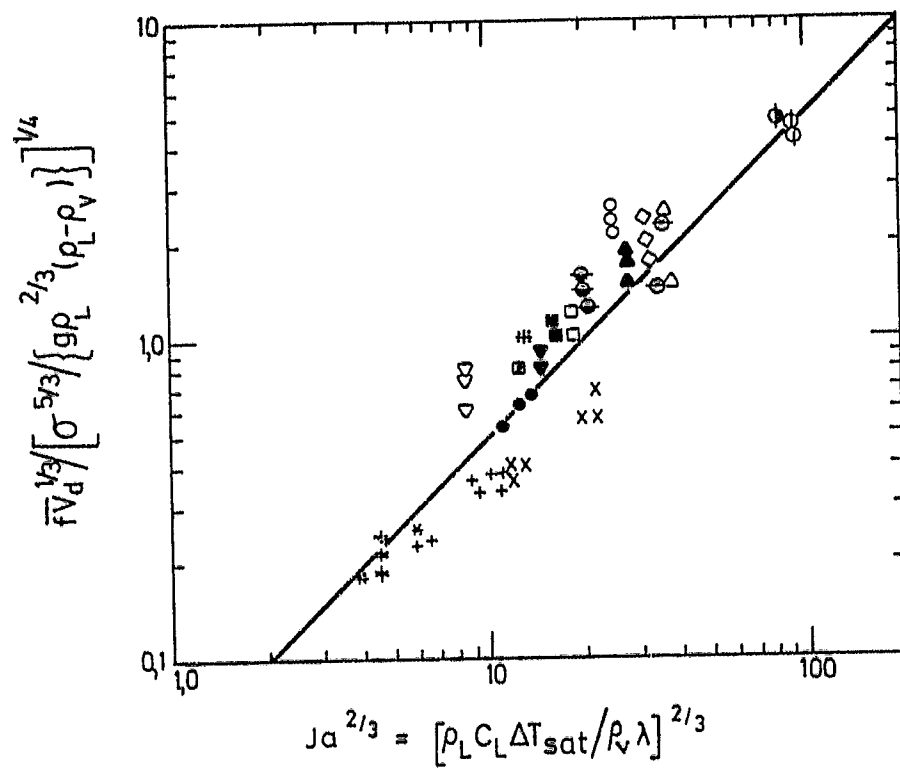
The following correlation, developed by Cole [50], appears to have the widest validity:

$$\overline{fV_d} = K_V \left[\sigma^{5/3} / \{g\rho_L^{2/3} (\rho_L - \rho_V)\} \right]^{3/4} \left[\rho_L C_L \Delta T_{\text{sat}} / \rho_V \lambda \right]^2 \quad (3.13)$$

K_V is a dimensionless constant ($= 1.25 \times 10^{-4}$), and the last bracketed term is the Jakob number.

Figure 3.8, reproduced from Cole [50], compares equation 3.13 with experimental values. (For convenience of plotting, the cube roots of dimensionless groups are shown). The scatter probably arises largely from the neglect in equation 3.13 of bubble contact angle, nucleation site concentration, active cavity size and bubble growth rate, all of which are known to affect bubble departure size and/or frequency [31, 119, 120, 126, 127, 128]. Further, the definition of means in $\overline{fV_d}$ differed among the various investigators shown in Figure 3.8. Thus the values of Cole [50] are the means of successive $\overline{fV_d}$ products from single nucleation sites, those of Rallis and Jawurek [2] are arithmetic means of $\overline{fV_d}$ products from all active sites, and those of Perkins and Westwater [129] are the products of $\overline{f}$ and $\overline{V_d}$, each being individually averaged values over all sites. Finally, the data points of Nickelson and Preckshot [130] and of McFadden and Grassmann [131] are the results of some minor estimates by Cole, the original data having been incompletely reported.

Nevertheless equation 3.13 correlates with some success the data for a wide range of liquids at pressures ranging from low subatmospheric to slightly above atmospheric. The high-pressure bubble departure data of Séméria [132], Wanninger [133] and Tolubinsky and Ostrovsky [134] could not be correlated because either frequency or temperature difference were not reported.



COLE [50]	$P, \text{ kN/m}^2$	NICKELSON & PRECKSHOT [130]
○ ACETONE	39,6	* CARBON TET. 152 kN/m^2
◇ CARBON TET.	18,4	+ CARBON TET. 101 kN/m^2
△ METHANOL	17,9	x CARBON TET 50,7 kN/m^2
▲ METHANOL	27,2	
□ METHANOL	40,5	McFADDEN & GRASSMANN [131]
■ METHANOL	52,9	* NITROGEN 101 kN/m^2
■ METHANOL	72,0	
▼ N-PENTANE	89,9	PERKINS & WESTWATER [129]
▽ N-PENTANE	101	* METHANOL 101 kN/m^2
○ WATER	6,87	
◇ WATER	8,87	RALLIS & JAWUREK [2]
● WATER	26,0	• WATER 83,4 kN/m^2
● WATER	48,0	

FIG. 3.8 COLE'S CORRELATION FOR BUBBLE DEPARTURE [50]

Inherent in equation 3.13 is the following equation for bubble departure diameter [50]:

$$D_d = K_D \left[\sigma / \{g(\rho_L - \rho_V)\} \right]^{1/2} \left[\rho_L C_L \Delta T_{sat} / \rho_V \lambda \right] \quad (3.14)$$

Cole [50] has tested equation 3.14 against the low-pressure data of the investigations shown in Figure 3.8 and, with $K_D = 4 \times 10^{-2}$, has obtained excellent agreement. It is believed that if ΔT_{sat} values were available, the high-pressure data referred to above [132, 133, 134] would also be accommodated by equation 3.14. This belief is based on the successful correlation of these data by the departure diameter equation of Cole and Rohsenow [49]:

$$D_d = \text{const.} \left[\sigma / \{g(\rho_L - \rho_V)\} \right]^{1/2} \left[\rho_L C_L T_{sat} / \rho_V \lambda \right]^{5/4} \quad (3.15)$$

Equation 3.15 is identical with equation 3.14 except that the last term is a modified Jakob number, Ja^* (with T_{sat} replacing ΔT_{sat}), of exponent $5/4$ instead of 1. Since the fractional variation with pressure is essentially the same for Ja and $(Ja^*)^{5/4}$, see Figure 3.9, equation 3.14 can be expected to hold at high pressure. This, in turn, lends some confidence to the extension of equation 3.13 to high pressure.

At constant pressure, equation 3.13 predicts an increase of $\overline{TV}_d$ with increasing ΔT_{sat} ; this is consistent with the measurements of Rallis and Jawurek [2] and the confirmatory evidence of Preckshot and Denny [128]. Division of equation 3.13 by equation 3.14 squared, however, leads to

$$\overline{TV}_d = \text{const.} \left[\sigma g(\rho_L - \rho_V) / \rho_L^2 \right]^{1/2} \quad (3.16)$$

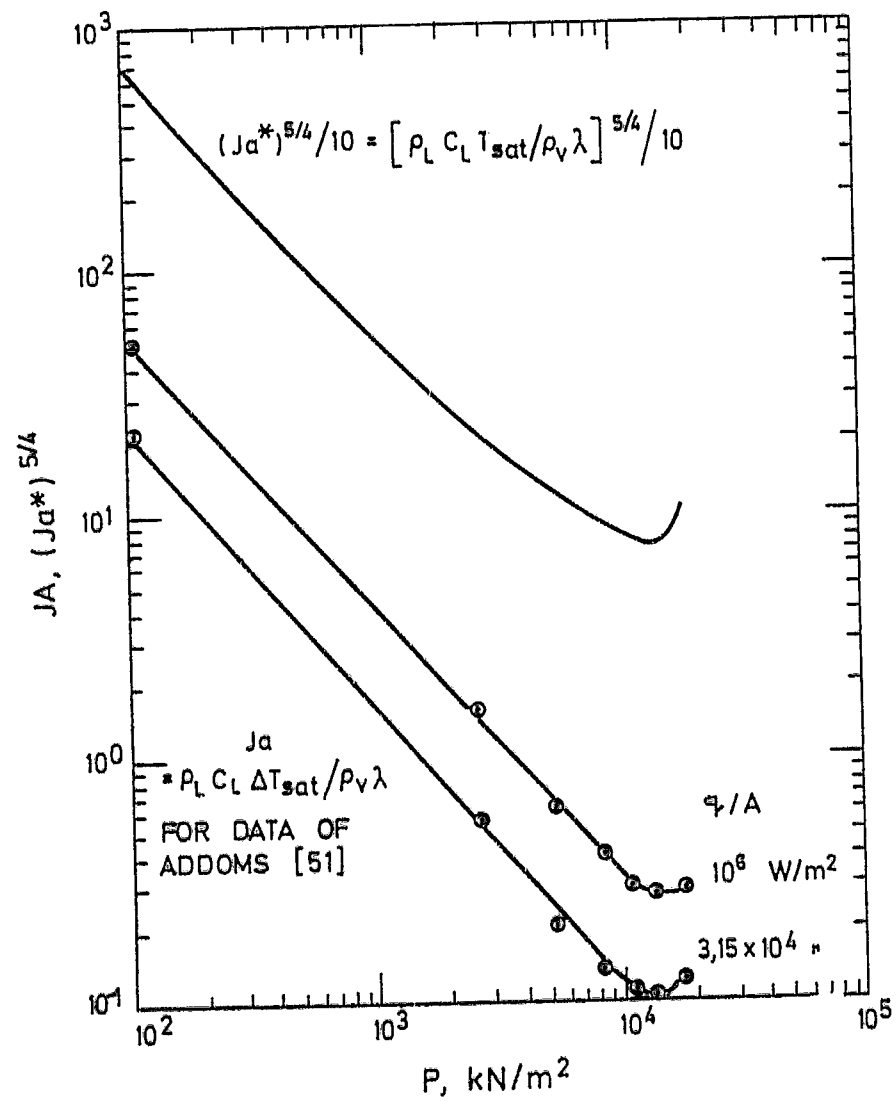


FIG. 3.9 PRESSURE VARIATION OF JAKOB NUMBER AND
MODIFIED JAKOB NUMBER, FOR WATER.

which, at constant pressure, is consistent with the well-known approximation

$$fD_d = \text{const.} \quad (3.17)$$

Equation 3.13 together with equation 3.14, thus offers an attractively rounded-off description of bubble departure.

(c) Final Expression for Latent Heat Transport

Substitution of equations 3.12 and 3.13 into equation 3.7 leads to the following expression for latent heat transport:

$$(q/A)_{LH} = K_N K_V \rho_V \left[\frac{\lambda \rho_V \Delta T_{\text{sat}}}{T_{\text{sat}}} \right]^m \left[\frac{\sigma^{5/3}}{g_L^{2/3} (\rho_L - \rho_V)} \right]^{3/4} \left[\frac{\rho_L C_L \Delta T_{\text{sat}}}{\rho_V \lambda} \right]^2 \quad (3.18)$$

which may be rearranged to give

$$(q/A)_{LH} = K \frac{\lambda^{(m-1)} \rho_V^{(m-1)} \rho_L^{3/2} C_L^2}{T_{\text{sat}}^m \sigma^{(m-5/4)} g^{(3/4)} (\rho_L - \rho_V)^{3/4}} \Delta T_{\text{sat}}^{(m+2)} \quad (3.19)$$

where the constant K (having dimensions of $\text{length}^{(m-2)}$) and the exponent m characterise a particular surface-liquid combination or perhaps even a particular surface independent of liquid.

3.2.2 Expression for Total Heat Flux

The total heat flux in isolated bubble boiling is given by the relation

$$(q/A)_T = (q/A)_{NB} + (q/A)_{NC} (\Lambda_{NC}/\Lambda_T) \quad (3.20)$$

where $(q/A)_{NB}$ is the heat flux (referred to total area) due to nucleate boiling from the areas of bubble influence, $(q/A)_{NC}$ is the flux due to natural convection when it alone is operative, and the last term is the area fraction available for such undisturbed natural convection.

From the breakdown of the area-of-influence heat transfer given in Chapter 2 we have

$$(q/A)_{NB} = Nq_T^i = N(q_{LH}^i + q_{BC}^i) = (q/A)_{LH} + (q/A)_{BC} \quad (3.21)$$

and hence via equation 3.20

$$(q/A)_T = (q/A)_{LH} + (q/A)_{BC} + (q/A)_{NC}(A_{NC}/A_T) \quad (3.22)$$

The summation of equation 3.5 over all nucleation sites gives the magnitude of the latent heat flux as follows:

$$(q/A)_{LH} \geq 0,7 (q/A)_{NB} \quad (3.23)$$

The remaining 30 per cent or less of the total area-of-influence heat transfer, that is, the bubble-induced convection contribution, was shown in Chapter 2 to be transferred at a mean rate approximately equal to that of undisturbed natural convection. Thus

$$(q/A)_{BC} = (q/A)_{NC} (1 - A_{NC}/A_T) \quad (3.24)$$

where the last term is the fraction of the total area occupied by areas of bubble influence.

By substitution of equation 3.24 into equation 3.22 the total heat flux is given simply by

$$(q/A)_T = (q/A)_{LH} + (q/A)_{NC} \quad (3.25)$$

The treatment in equation 3.25 of bubble-induced convection and of the natural convection area fraction is a rough approximation only. This, however, does not critically affect the overall accuracy since, firstly, latent heat transport and not bubble-induced convection, is the predominant contributor to the total area-of-influence heat transfer (see equation 3.23), and secondly, the natural convection ratio $(q/A)_{NC}/(q/A)_T$ falls rapidly with increasing heat flux so that the precise value of A_{NC}/A_T becomes unimportant.

In Chapter 2 equation 3.25 was arrived at directly from four independent sets of experiments [1, 2, 65, 84] and was shown to hold within ± 30 per cent.

The combination of equations 3.25 and 3.19 gives our final correlation:

$$(q/A)_T = K\psi\Delta T_{sat}^{(m+2)} + (q/A)_{NC} \quad (3.26)$$

where

$$\psi = \frac{(\lambda\rho_V)^{(m-1)} \rho_L^{3/2} C_L^2}{T_{sat}^m \sigma^{(m-5/4)} g^{3/4} (\rho_L - \rho_V)^{3/4}} \quad (3.27)$$

and $(q/A)_{NC}$ is obtained from standard correlations.

3.3 APPLICATION OF CORRELATION

Boiling heat fluxes can be predicted by equations 3.26 and 3.27, provided that at least one boiling run $(q/A)_T$ versus ΔT_{sat} is available for the surface-liquid combination under study. $(q/A)_{NC}$, as obtained experimentally or by calculation, is subtracted from $(q/A)_T$ and the difference, $K\psi\Delta T_{sat}^{(m+2)}$, is plotted against ΔT_{sat} on log-log paper. The best straight line is fitted to the lower (isolated bubble) portion of the data and the slope $(m+2)$ is established. With m known, K

is obtained from equations 3.26 and 3.27. Thus all terms in equations 3.26 and 3.27 are known and $(q/A)_T$ can be calculated at any other pressure for the same surface-liquid combination. At slightly increased risk the prediction can be extended to other liquids boiling on the same surface.

3.4 TESTING OF CORRELATION

The correlation scheme outlined above is now tested against the experimental results of Addoms (see McAdams [51]) for water boiling at high pressures, and against those of Bonilla and Perry [86] for ethanol at low pressures. Both sets of data are presented as sets of boiling curves $(q/A)_T$ versus ΔT_{sat} . For testing data in this form, equation 3.26 is rewritten as

$$(q/A)_{T-NC} = (\Lambda \cdot \Delta T_{sat})^{(m+2)} \quad (3.28)$$

where

$$\Lambda = \psi^{\frac{1}{m+2}} \quad (3.29)$$

and ψ is given by equation 3.27. Thus for one surface-liquid combination a log-log plot of $(q/A)_{T-NC}$ versus $\Lambda \cdot \Delta T_{sat}$ should accommodate data at all pressures on a single straight line of slope $(m+2)$.

3.4.1 Test Against High-pressure Water Data of Addoms

Figure 3.10, taken from Figure 14.14 in McAdams [51], shows the data of Addoms for water boiling from a 0.61 mm diameter platinum wire. The natural convection fluxes (three sample curves are shown) were calculated from an equation of the form

$$Nu = \text{const.} (Gr.Pr)^n \quad (3.30)$$

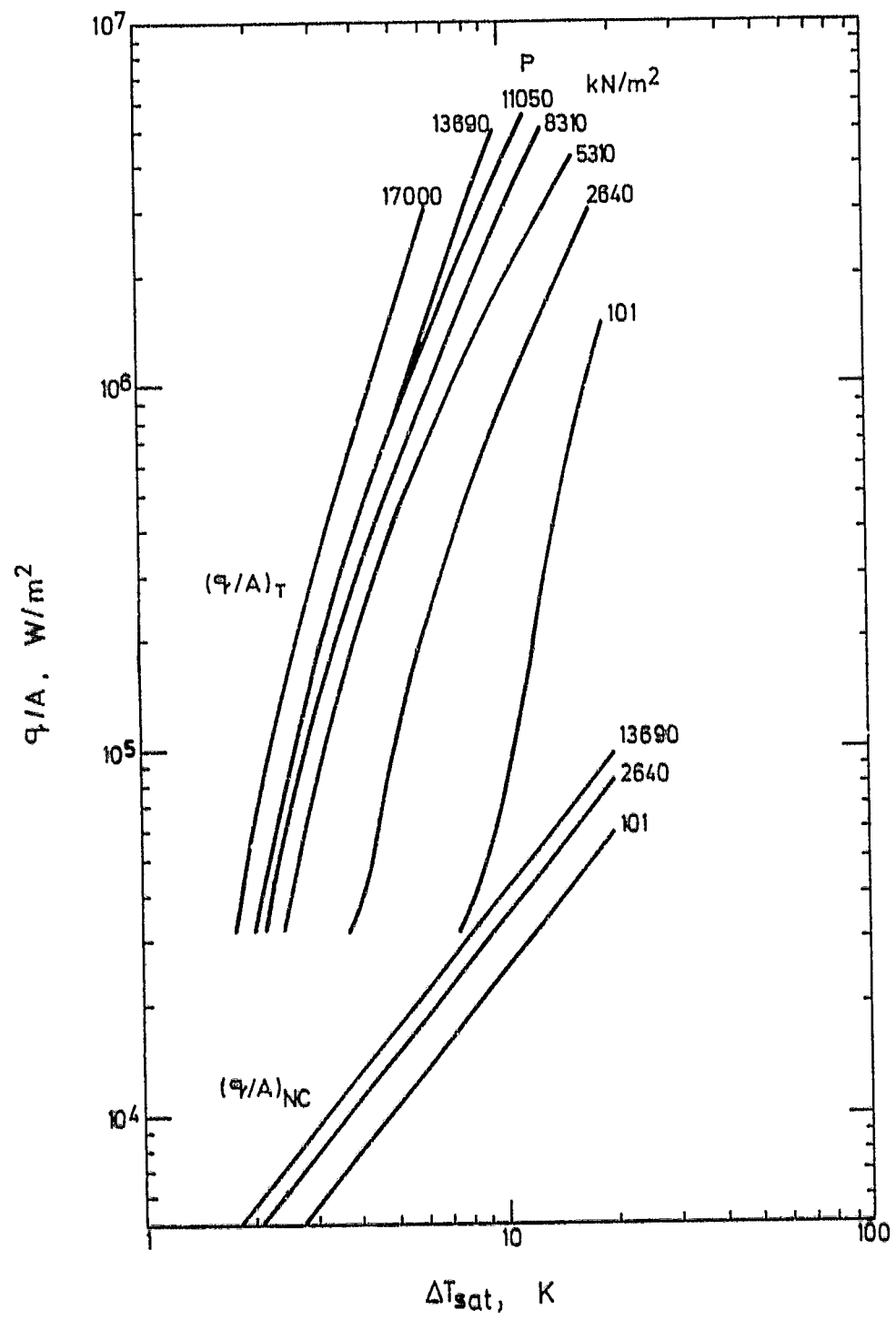


FIG. 3.10 WATER BOILING AT HIGH PRESSURES, DATA OF
ADDOMS [51]

which for the conditions of Addoms runs ($Gr.Pr. = 10^2$ to 10^4) may be written as [135]

$$(q/A)_{NC} = 0,53 (k_L^3 \rho_L^2 g \beta C_L / D_o \mu)^{1/4} \Delta T_{sat}^{5/4} \quad (3.31)$$

where D_o is the wire diameter. Equation 3.31, while only approximately valid for $Gr.Pr < 10^3$, is acceptable since the contribution of $(q/A)_{NC}$ to $(q/A)_T$ is small. For the same reason physical properties in equation 3.31 were evaluated at an overall mean film temperature for each pressure.

Figure 3.11 shows $(q/A)_{T-NC}$ versus ΔT_{sat} . The run at 17 000 kN/m² ($P = 0,77 P_{crit}$) is excluded because some physical properties become unreliable and equation 3.13 for $\overline{TV_d}$ probably no longer holds (see Fig. 3.9). Mikic and Rohsenow [47], in correlating Addoms' data, similarly omit this run. The mean slope of the lower halves of the curves in Figure 3.11 is taken as 4. Thus $m = 2$ and equations 3.27, 3.28 and 3.29 give

$$(q/A)_{T-NC} = (\Lambda \Delta T_{sat})^4 \quad (3.32)$$

where

$$\Lambda = \left[(\lambda \rho_V \rho_L^{3/2} C_L^2) / \{ T_{sat}^2 (\sigma g)^{3/4} (\rho_L - \rho_V)^{3/4} \} \right]^{1/4} \quad (3.33)$$

The data points inserted into Figure 3.11 are now processed by equations 3.32 and 3.33, with physical properties [51, 136] evaluated at T_{sat} . Figure 3.12 shows the correlation to be satisfactory. Since $m = 2$, the constant K is dimensionless.

Figure 3.12 further shows that our correlation scheme continues to hold at high heat fluxes beyond the end of the isolated bubble region, and thus beyond the limit of validity of our model. This can be explained as follows: In the presence of lateral bubble coalescence, equation 3.7 must

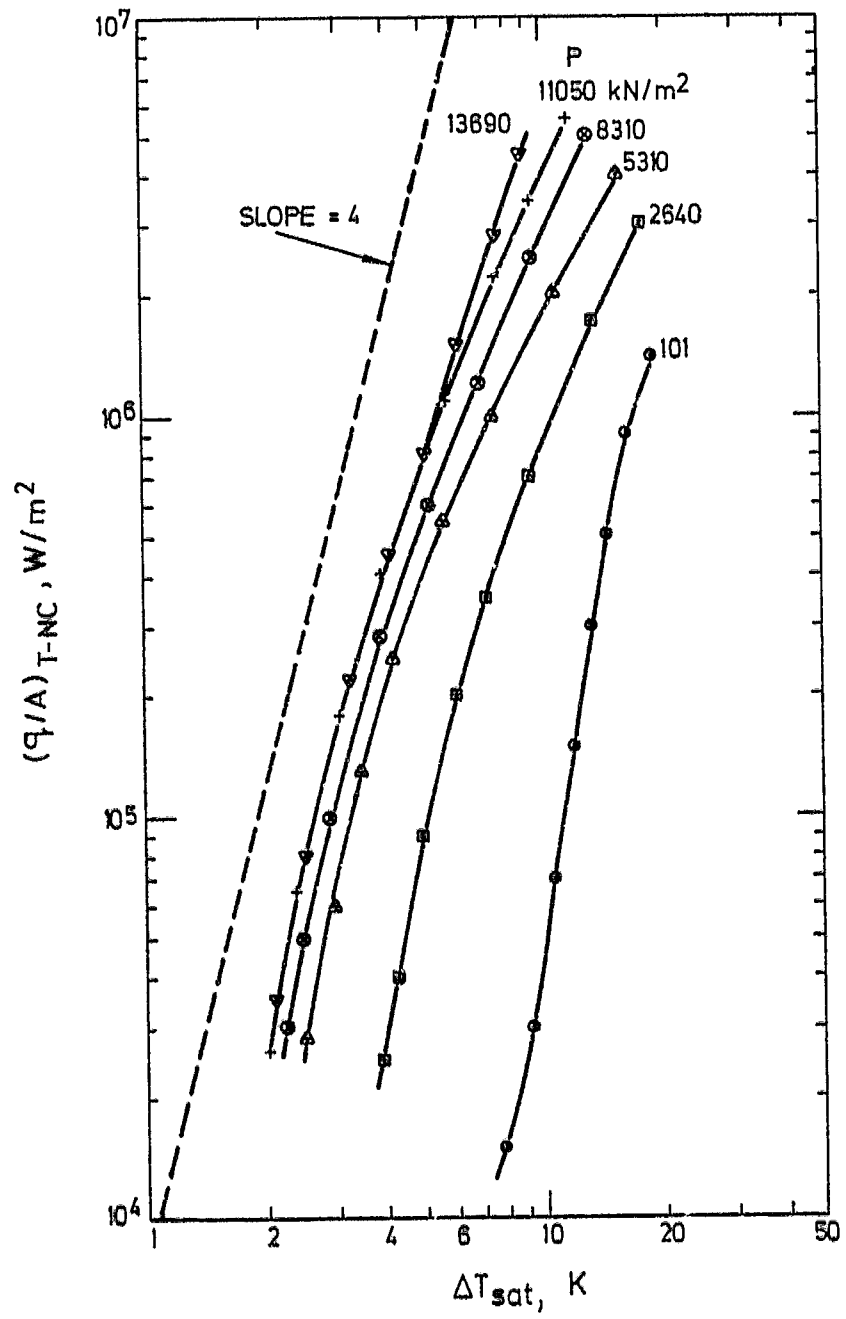


FIG. 3.11 TOTAL FLUX MINUS NATURAL CONVECTION FLUX
FOR DATA OF ADDOMS [51]

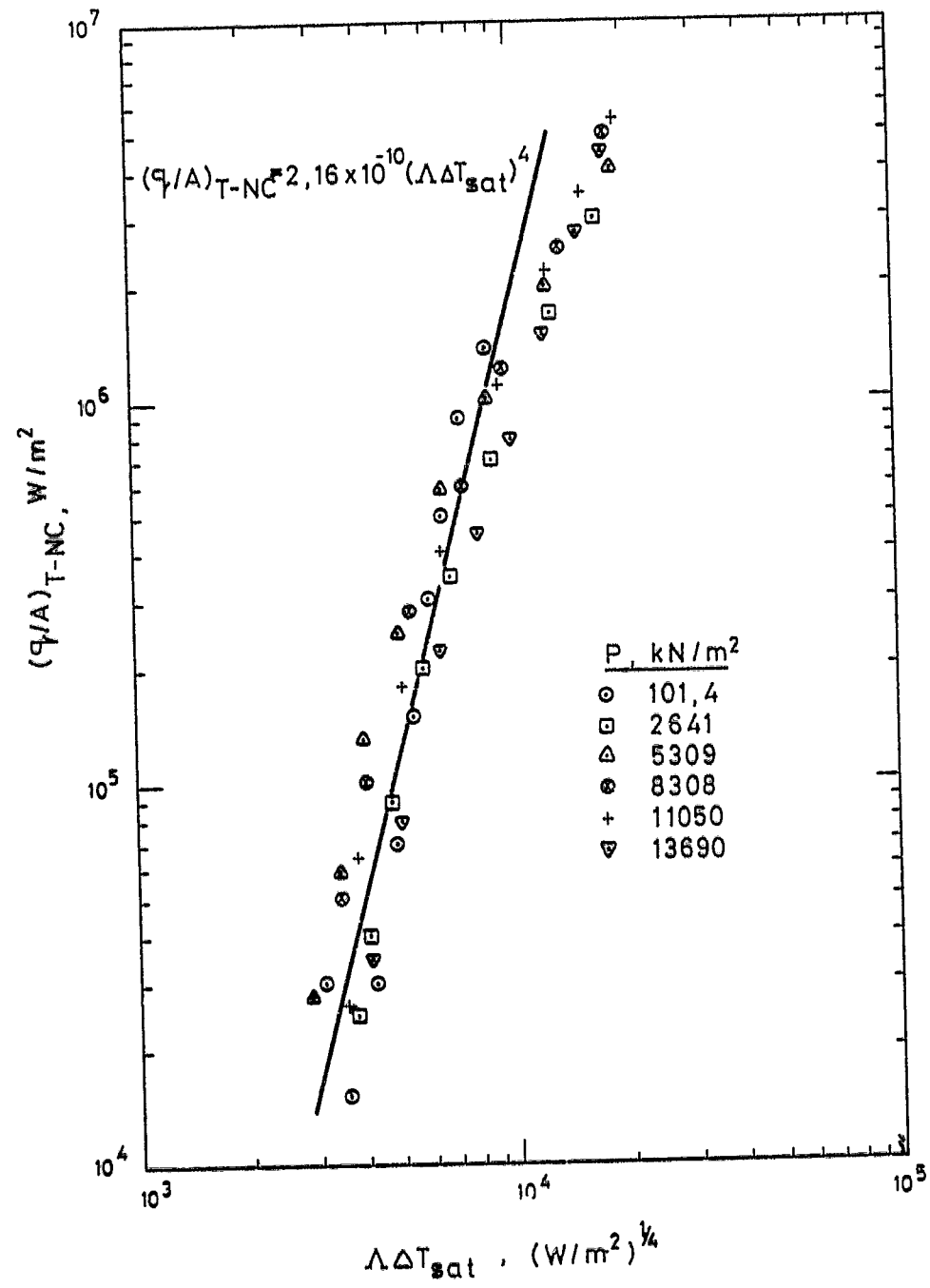


FIG. 3.12 NEW CORRELATION APPLIED TO DATA OF ADDOMS [51]

be modified to read

$$(q/A)_{LH} = N' \rho_V \lambda \overline{V}_d \quad (3.34)$$

where f and V_d now refer to bubbles arising from one or more nucleation sites and N' is the concentration of such bubble sources. Now $\overline{V}_d$ increases more rapidly with ΔT_{sat} for coalescence bubbles than for isolated bubbles (see Fig. 11b in Appendix 2), and N' (with increasing coalescence) increases more slowly than N . The two effects at least partially cancel each other, and thus the correlation scheme remains approximately valid.

As the critical heat flux is approached, massive vapour patches form on the heating surface [81]; the model breaks down altogether and the correlation equations cease to hold.

3.4.2 Test Against Low-pressure Ethanol Data of Bonilla and Perry

Figure 3.13, taken from Bonilla and Perry's Figure 10 [86] shows data points for ethanol boiling on a horizontal, chrome-plated copper disc 60,6 mm in diameter. No natural convection measurements were reported for these runs. The necessary convective fluxes were thus calculated from equation 3.30, which for the prevailing turbulent conditions ($Gr.Pr = 10^9$ to 10^{10}) becomes [51, 135]

$$(q/A)_{NC} = C_c (k_L^2 \rho_L^2 g \beta C_L / \nu)^{1/3} \Delta T_{sat}^{4/3} \quad (3.35)$$

The constant C_c was determined from Bonilla and Perry's data, using those ethanol runs which included natural convection measurements (Figs. 2, 3, 4, 5, 9, 11 and 12 in [86]). A mean value for C_c of 0,9 resulted. This procedure was advisable, since with horizontal plate heaters the constant C_c can take on a wide range of values, depending on minor

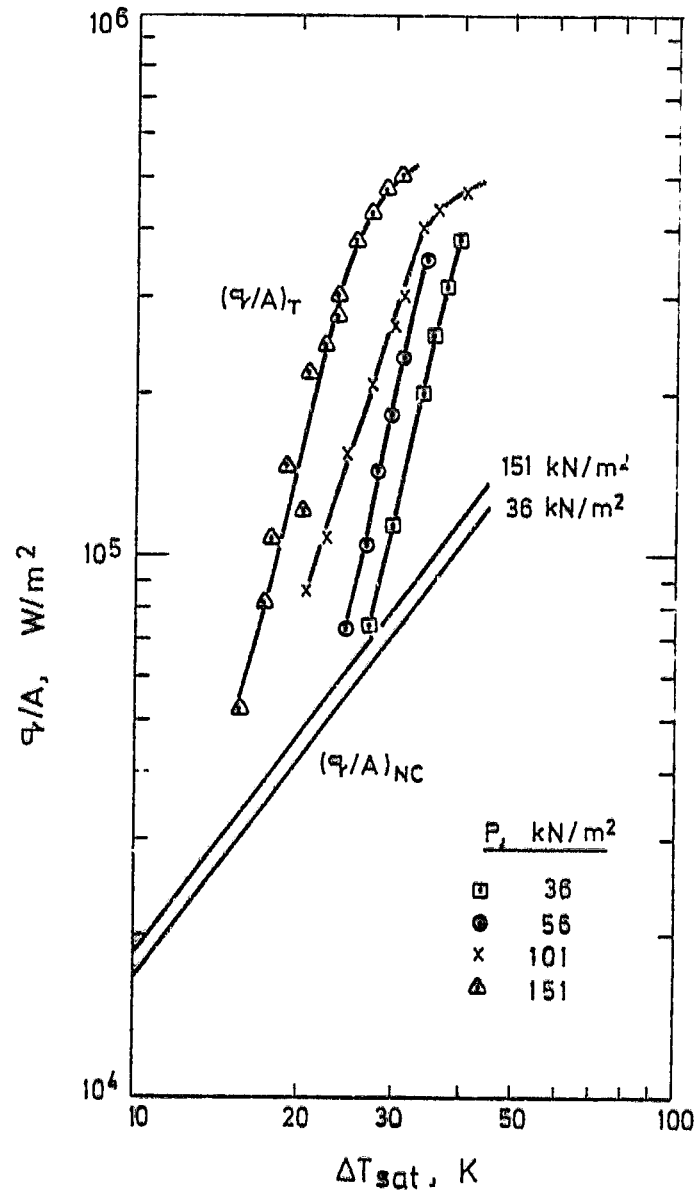


FIG. 3.13 ETHANOL BOILING AT LOW PRESSURES, DATA OF BONILLA AND PERRY [86]

system disturbances, see, for example, [46]. The value of 0,9 is, in fact exceptionally high (cf the 'standard' value of 0,14 [135]); it appears to be due to extraneous bubbles formed at the heater gasket in Bonilla and Perry's apparatus. Figure 3.13 shows two sample curves of equation 3.35, with $C_c = 0,9$.

Figure 3.14 shows $(q/A)_{T-NC}$ versus ΔT_{sat} . With the mean slope of the straight lines interpreted as 6, we have $m = 4$. Thus from equations 3.27, 3.28 and 3.29 the correlation is

$$(q/A)_{T-NC} = K (\Lambda \cdot \Delta T_{sat})^6 \quad (3.36)$$

where

$$\Lambda = \left[\{ (\lambda \rho_V)^3 \rho_L^{3/2} C_L \} / \{ T_{sat}^{11/4} g^{3/4} (\rho_L - \rho_V)^{3/4} \} \right]^{1/6} \quad (3.37)$$

Figure 3.15 shows the data points correlated by equations 3.36 and 3.37. Physical properties [86, 137] were evaluated at T_{sat} . Over the lower half of the flux range the correlation is again acceptable.

The whole procedure was repeated with the slope in Figure 3.14 interpreted as 7, instead of 6, that is $m = 5$. No significant spread in the correlated points resulted. These tests and the test against Addoms' data with $m = 2$ show that our correlation scheme is not fortuitously successful for certain values of m , but appears to have general validity.

3.5 CONCLUDING COMMENTS

1. Ideally, the correlation of nucleate boiling heat transfer should be based on the primary surface-to-liquid heat transfer processes in the area of bubble influence. These processes are complex and incompletely understood.

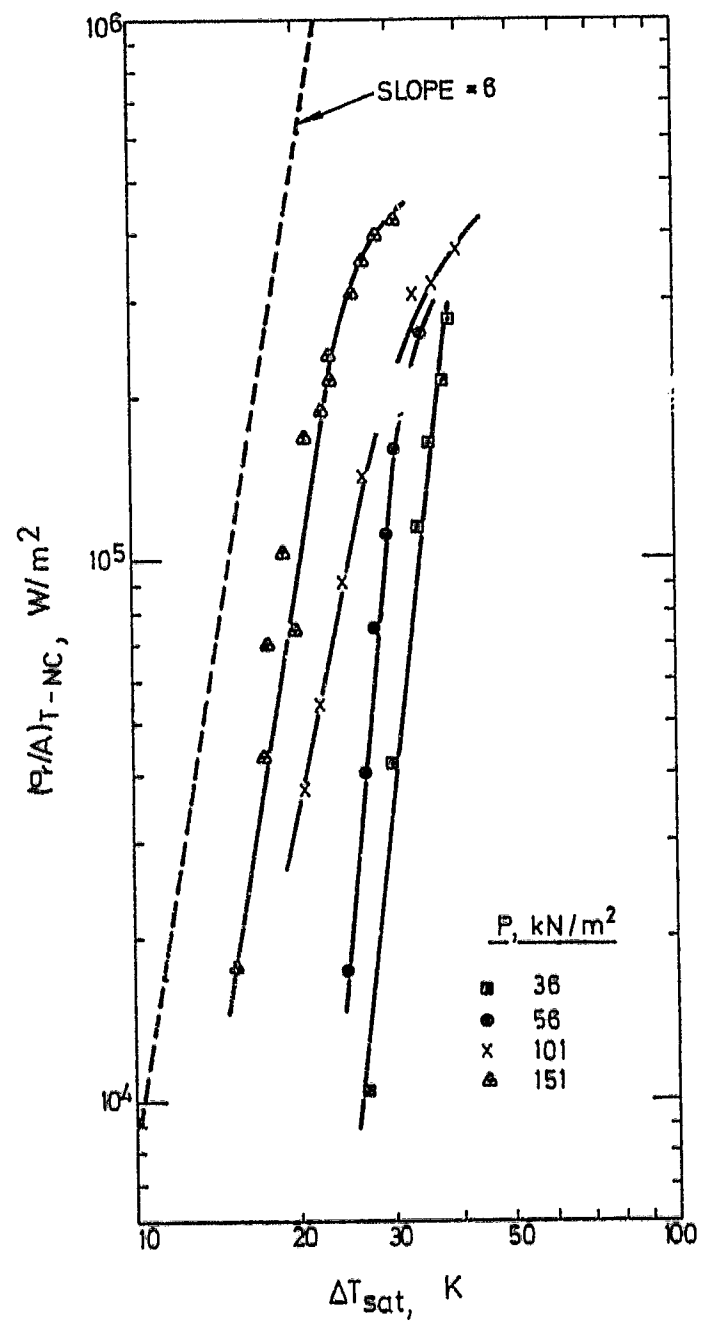


FIG. 3.14 TOTAL FLUX MINUS NATURAL CONVECTION FLUX
FOR LOW-PRESSURE ETHANOL DATA OF BONILLA
AND PERRY [86]

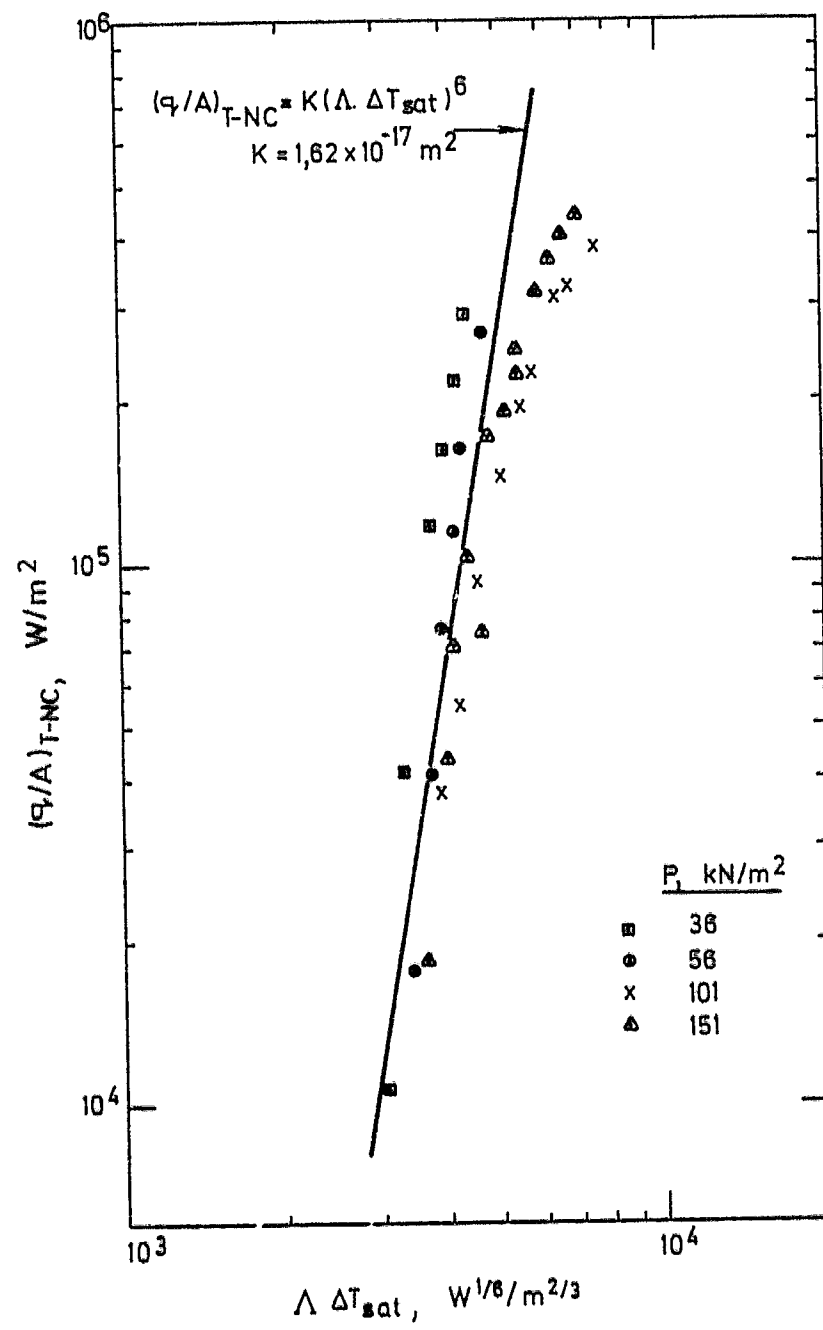


FIG. 3.15 NEW CORRELATION APPLIED TO DATA OF BONILLA AND PERRY [86]

Thus their models have generally become amenable to analysis only when simplified to the point where they no longer relate to reality. The bulk convection correlations, despite their numerical success, illustrate this difficulty.

2. An alternative approach is outlined in this chapter. Its basis is the experimental finding that all primary area-of-influence heat transfer processes manifest themselves firstly, and predominantly, as latent heat transport, and secondly as bubble-induced convection, where bubble-induced convection may be closely approximated by natural convection. The resulting model can be analysed without recourse to further mechanistic assumptions.

3. A new correlation of heat transfer during saturated pool boiling has resulted. The correlation equations have been tested successfully against two sets of experimental data, one for water at high pressures and one for ethanol at low pressures. While more extensive tests are necessary, it appears that the correlation allows the prediction of boiling curves q/A versus ΔT_{sat} at any pressure, provided at least one boiling curve for the same surface, or preferably the same surface-liquid combination is available as a reference. This feature and the underlying method of surface characterisation are essentially identical to those in Mikic and Rohsenow's correlation [47].

4. Although derived in terms of isolated bubbles (i.e. low heat flux), the correlation remains approximately valid throughout the linear range of log-log boiling curves. Its validity is probably restricted to terrestrial gravity and to one-component, conventional (e.g. non-metallic) liquids.

130.

PART 2

EXPERIMENTAL STUDY OF MICROLAYER EVAPORATION

CHAPTER 4FORMULATION OF PROBLEM4.1 INTRODUCTION AND OUTLINE

Part 1 of this thesis, and Chapter 2 in particular, provided a broad overview of the heat flows associated with an area of bubble influence. The primary surface-to-liquid heat flows, q_L^i , were found to be so redistributed as to manifest themselves predominantly as latent heat transport, q_{LH}^i , that is, as evaporation at all surfaces of attached bubbles. Latent heat transport by virtue of bubble departure (or departure and recondensation) thus emerged as the chief mode of heat transport from the immediate vicinity of the heater into the liquid bulk. Further, latent heat transport was shown, in Chapter 3, to be an effective basis for the derivation of a new pool boiling heat transfer correlation.

With the importance of the overall latent heat transport thus demonstrated, Part 2 of this thesis takes a more detailed view of evaporation into attached bubbles. In Chapter 2 two contributors to such evaporation were distinguished. The first of these, namely evaporation at the side walls and dome of a bubble has been very extensively studied. The second, and comparatively recently discovered contributor, namely microlayer evaporation, is less understood and forms the subject of the remainder of this study.

The aim of Part 2 of this thesis is to further the understanding of the microlayer and, in particular, to assess its evaporation contribution to the total latent heat transport.

In order to formulate an appropriate experimental program

this chapter provides a review of previous microlayer research. The review reveals the existence of only one set of direct measurements of microlayer geometry [104]. The bubbles were, however, studied under abnormal conditions (flattened to facilitate interferometry) and their growth rate was not recorded.

The review thus points to a need for synchronised records, at various conditions of normal boiling, of:

- (1) microlayer shape and thickness versus time, and
- (2) overall bubble growth and detachment.

For freely-growing bubbles direct measurements of microlayer geometry should, it is argued, be obtainable from high-speed ciné interference photography from beneath and through a transparent heating surface. The synchronisation of these records with bubble growth records in profile should be achievable by image-reflection and by photographing both views of the bubble simultaneously and on the same film.

It is concluded that this novel but complex combination of techniques should be applied to the study of sufficiently large, and thus low-pressure bubbles at bulk saturation, at various levels of subcooling and at various heat fluxes.

4.2 REVIEW OF PREVIOUS MICROLAYER STUDIES

4.2.1 Local Surface Temperature Measurements

In 1956 Snyder [12] postulated the mass transfer mechanism for subcooled boiling (see Section 2.2). Basic to his mechanism were the formation and evaporation of thin liquid films beneath attached bubbles. The existence of such

microlayers, he suggested, could be verified by temperature measurements at the base of bubbles, with a sharp drop in temperature signifying the onset of microlayer vaporisation.

Temperature signals consistent with the microlayer hypothesis were first obtained by Moore and Mesler in 1961 [67] by use of a surface thermocouple of very small dimensions and time constant. Hsu and Schmidt [138], in the only previous specific study of local surface temperature, employed too coarse a thermocouple.

The thermocouple of Moore and Mesler consisted of a Chromel tube and a concentric oxide-insulated Alumel centre wire, all press-fitted into a Nichrome heating strip with the thermocouple ends flush and at right angles with the top surface of the strip and the junction established with a nickel plating 0,001 mm thick. The thermocouple sensed the average temperature over a surface annulus of 0,13 mm inner diameter and 0,03 mm width. Its time constant was of the order of 1 ms [68, 138a]. The heating strip, 1,6 mm thick and 6,4 mm wide, was insulated at the back and sides, heated by direct current and immersed in a pool of water at saturation temperature under atmospheric pressure. Boiling tests were conducted at mean heat fluxes of 426 to 637 kW/m².

The thermocouple output was linked to an oscilloscope and typically gave temperature traces as shown in Figure 4.1. Moore and Mesler could explain these traces (assuming them to reflect the true surface temperature) only in terms of a microlayer, with the sharp drop (some 16 K in 1 to 2 ms) corresponding to microlayer evaporation and the slower rise corresponding to the recovery of the dry or bubble-free surface. Figure 4.2 shows the surface temperature of an infinitely thick slab in pure conduction subjected first to the extraction of a constant heat flux and then to its sudden cessation. The similarity of this theoretical

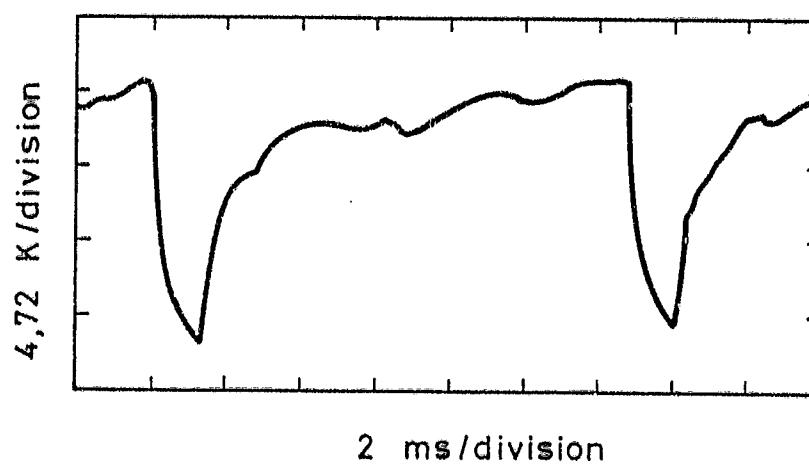


FIG. 4.1 SURFACE MICROTHERMOCOUPLE OUTPUT.
MOORE AND MESLER. [67]
Water at saturation, atmospheric pressure,
heat flux 637 kW/m^2 , Nichrome ribbon heater.

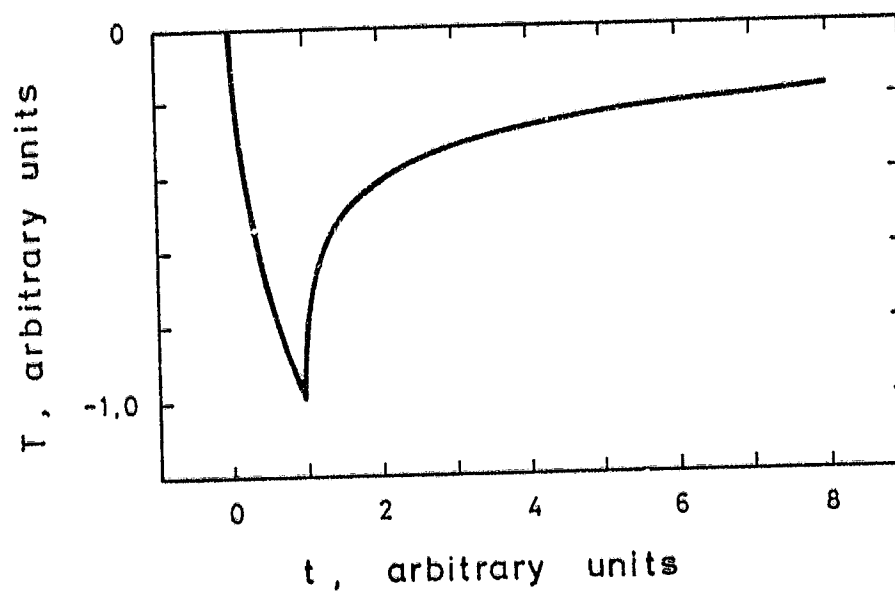


FIG. 4.2 SURFACE TEMPERATURE RESPONSE OF INFINITELY THICK SLAB, INITIALLY AT $T = 0$, TO EXTRACTION OF CONSTANT SURFACE HEAT FLUX DURING $0 < t < 1$

temperature response with the experimental results of Figure 4.1 were interpreted as strong support of the microlayer phenomenon.

By integration over the temperature dips Moore and Mesler obtained the heat extraction per unit area due to microlayer evaporation. From these the initial microlayer thicknesses were estimated as 2,0 to 2,3 μm and the mean microlayer evaporation heat fluxes as 2080 to 4130 kW/m^2 , that is, as approximately six times the overall surface heat flux. Finally, on the assumption that at the high heat fluxes studied the entire surface behaved like the thermocouple sites, microlayer evaporation was estimated as contributing 70 to 90 per cent of the total boiling flux. This implied that the latent heat transport contribution was at least as large.

To appreciate the courageousness of Moore and Mesler's claims it should be remembered, firstly, that no visual proof was available that the measured temperature dips occurred during bubble growth, and, secondly that in 1961 the microlayer postulate of Snyder was, where remembered at all, unacceptable since it ran counter to the entrenched concept of negligible latent heat transport.

In 1964 Rogers and Mesler [96] extended the experiment of Moore and Mesler by fitting an artificial nucleation site to the thermocouple centre wire and by taking single flash photographs at one preselected point along each of the temperature traces. The photographs showed that the temperature dips indeed occurred during bubble growth and not, as had been suggested, during the inrush of colder liquid following bubble departure. The amplitudes of the temperature dips (4,4 to 10,6 K) were smaller than those of Moore and Mesler, possibly owing to the lower overall heat fluxes (61 to 161 kW/m^2) or to some slight changes in the design of the thermocouple. No evaporation heat fluxes or

microlayer thicknesses were reported.

The first records of local surface temperature synchronised with high-speed ciné views of the bubble profile were obtained by Hendricks and Sharp in 1964 [97]. In order to avoid the discontinuity in the heater, present in the studies of Mesler, a microthermocouple, of junction dimensions 0,04 mm, was spot-welded to the bottom of a 0,03 mm thick, electrically-heated Nichrome boiling surface. With this assembly water was boiled under a pressure of 5,6 kN/m² (saturation temperature 35 °C), at overall heat fluxes of 85 to 104 kW/m² and at bulk subcoolings of 6 to 7 K. Under such conditions large, isolated and essentially hemispherical bubbles were formed, in some cases above or slightly displaced from the thermocouple junction. Temperature dips of amplitudes up to 43 K, that is, three times as large as those of Moore and Mesler, and of duration 10 to 20 ms were measured. The dips were shown not only to correspond to bubble growth, but to start at the instant that the bubble perimeter passed over the thermocouple junction. Integration over the temperature dips gave estimated microlayer evaporation heat fluxes some 20 times as large as the mean overall heat fluxes; each flux surge ceased as the bubble reached maximum diameter. On the assumption that at this stage the microlayer was completely evaporated (this was uncertain because of the low heat capacity of the ultra-thin heater) initial microlayer thicknesses were estimated as being of the order of 2 µm. Finally, the records revealed a small but measurable flux contribution due to bubble departure and collapse (i.e. a small q_{LW}^i), but no measurable contribution due to the movement of liquid surrounding an attached bubble (i.e. a negligible q_{LA}^i).

Bonnet, Macke and Morin [98] in 1965 provided local surface temperature records and synchronised bubble ciné picture(s) for water boiling at saturation and atmospheric pressure

from an oil-heated stainless steel tube. Heat fluxes were not measured but varied from incipience of boiling to incipience of bubble coalescence. The thermocouple junction was roughly hemispherical, 0,25 mm in diameter, and brazed and polished so as to be continuous with the outer tube surface. (The thermocouple wires were led away internally through the tube wall.) Bubbles grew naturally on the thermocouple junction, probably because of the brazing spot's local increase in thermal conductivity over the surrounding stainless steel.

A set of results for isolated bubble boiling is shown in Figure 4.3. Note that the temperature dip occurs in some 8 ms (cf 2ms in Moore and Mesler's study) and that at the end of the dip the bubble is still attached; thus the microlayer has probably dried up. The dotted curve shows the response (and recovery) of surface temperature of an infinitely thick slab to a constant flux extraction, the theoretical curve being adjusted for best fit to the experimental temperature record. By assuming the local wall material to be stainless steel, the value of this microlayer evaporation heat flux was estimated as 1156 W/m^2 . From a rough estimate of the average bubble attachment area the enthalpy due to microlayer evaporation was calculated, and found to account for 98 per cent of the heat of formation of a detaching bubble. The corresponding microlayer thickness was of the order of $4 \text{ }\mu\text{m}$. Similar estimates for a second, smaller bubble gave comparable results.

In 1966 Johnson, de la Peña and Mesler [139], reporting on the interrelation of bubble growth rate and bubble shape, showed spherical bubbles to have the lowest growth rate, oblate bubbles to have intermediate rates and hemispherical bubbles (having the largest base contact area) the highest growth rates. Continuing from this study, Hospeti and Mesler [101] attempted to estimate the microlayer contribution to total bubble growth for each such class of bubble.

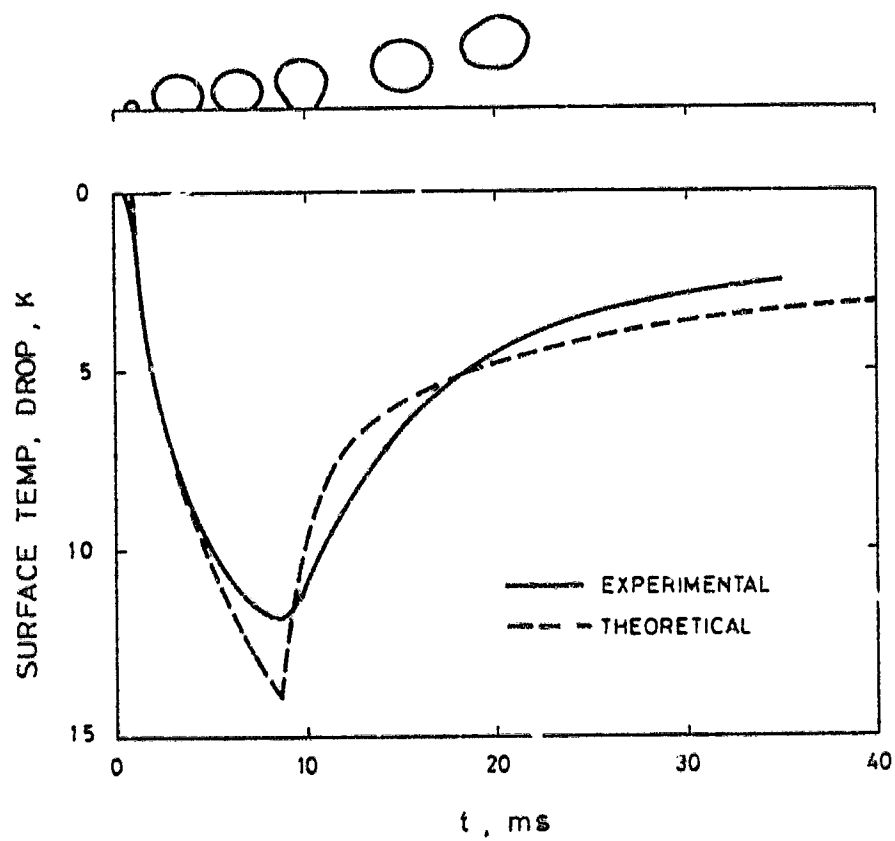


FIG. 4.3 HEATER SURFACE TEMPERATURE AT BUBBLE BASE
vs BUBBLE HISTORY. BONNET ET AL. [98]
 Water at saturation, atmospheric pressure,
 stainless steel tube heater.

The test liquid was water, boiling at saturation temperature and atmospheric pressure from a flat metal heater.

Spherical bubbles were classified as having a ratio of major to minor axis of 1 to 1,4, oblate bubbles as 1,4 to 1,7 and hemispherical bubbles as 1,7 to 2,0. From temperature records, bubble ciné pictures and transient conduction estimates, similar to those described above, they found that for spherical bubbles the contribution of microlayer evaporation to the departure bubble volume was 1,3 to 17 per cent, for oblate bubbles 5,4 to 42,9 per cent, and for a hemispherical bubble 50 to 100 per cent. Estimated initial microlayer thicknesses were 0,2 to 1,3 μm for spherical bubbles, 1,1 to 2,3 μm for oblate bubbles, and 3,6 μm for the only hemispherical bubble.

Further surface temperature measurements, offering at least qualitative agreement with the studies reviewed above, have been reported by Morin [140], Roemer [141] and Fontana [102].

Objections have, however, been raised [61, 112] to both the technique and the interpretation of all temperature measurements so far discussed. Bonnet et al [61], after testing and analysing theoretically a whole range of surface thermocouples, concluded that even tiny embedded instruments severely disturb the heat flow in their vicinity and that the measured temperatures are not generally the true undisturbed surface temperatures. They thus cautioned against estimating surface heat fluxes and microlayer thicknesses from such measurements. The wide differences in temperature records between the various studies on water conducted under essentially the same conditions, might thus be due to differences in measurement error. Madsen [112], going one step further, suggested on theoretical grounds that the surface-mounted thermocouples measure the variations in the liquid temperature of the thermal layer, rather

than the heater surface temperature. He thus rejected the microlayer concept altogether, a view subsequently supported by van Stralen [63, 65]*.

4.2.2 Temperature Measurements of Cooper and Lloyd

Refined temperature measurements, free from most of the objections just discussed, were presented by Cooper and Lloyd [99, 100] in 1966 and 1969.

Their measuring system consisted of an array of miniature germanium resistance thermometers [144], each fitted with copper leads, all vacuum-deposited onto a glass or ceramic heating disc. Each thin-film thermometer was 0,1 mm square, 0,25 to 0,5 μm thick and had a time constant of the order of 10^{-8} s. Four such thermometers were arranged in a straight line, spaced 1,5 mm from each other and linked to recording equipment; a fifth, spaced 0,5 mm from its neighbour, served to trigger a bubble when a short current pulse was passed through it.

The glass or ceramic disc formed the floor of a boiling tank and was electrically heated from below at known input rates. The bare thermometers necessitated non-conducting test fluids; toluene and isopropyl alcohol were used. System pressures were varied from 6,9 to 13,8 kN/m^2 , overall heat fluxes from 22,7 to 47,3 kW/m^2 and bulk subcooling from 0 to 7,8 K. Under such conditions essentially hemispherical bubbles up to 30 mm in diameter were formed.

For such bubbles the thermometers gave the temperature

*Van Stralen [142, 143], in his most recent work (1975), has apparently changed his mind.

of the heater-fluid interface at and near the bubble base as a function of time and position. The sensing area was small in relation to bubble size, the response (including that of the recorders) rapid compared with the surface temperature transients, and the perturbation of heat flow in the solid insignificant. The thickness of the thermometers (0,25 to 0,5 μm), however, was significant compared with previous estimates of microlayer thickness (0,2 to 4 μm , see Section 4.2.1).

In some runs high-speed ciné films were taken simultaneously and in synchronisation with the temperature records.

A sample set of results is shown in Figure 4.4. Test conditions are given in the figure caption. The temperature records of Figure 4.4(b) were interpreted as follows: during the initial temperature drop (stage 1), each commencing as the bubble base passed over a thermometer, the microlayer evaporated; after dry-out the temperature of the surface recovered (stage 2); during the second, smaller temperature drop (stage 3) cooler liquid returned as the bubble base retracted, and during the final temperature recovery (stage 4) the liquid thermal layer was built up again. In some cases stage 1 was followed immediately by stage 4, thus the microlayer had not evaporated completely.

Figure 4.4(c) shows the local heat fluxes from the heater into the fluid. The curves resulted from numerical integration of the one-dimensional conduction equation using as boundary conditions the temperature-time records of Figure 4.4(b) and the known flux input at the bottom face of the heater. The area under the curves of Figure 4.4(c) gives the heat transferred during the time interval considered. Thus for cases where the microlayer evaporated completely, and on the assumption of a stagnant layer, integration over the rising portion of the initial flux peak led to estimates of the initial microlayer thickness, d_0 .

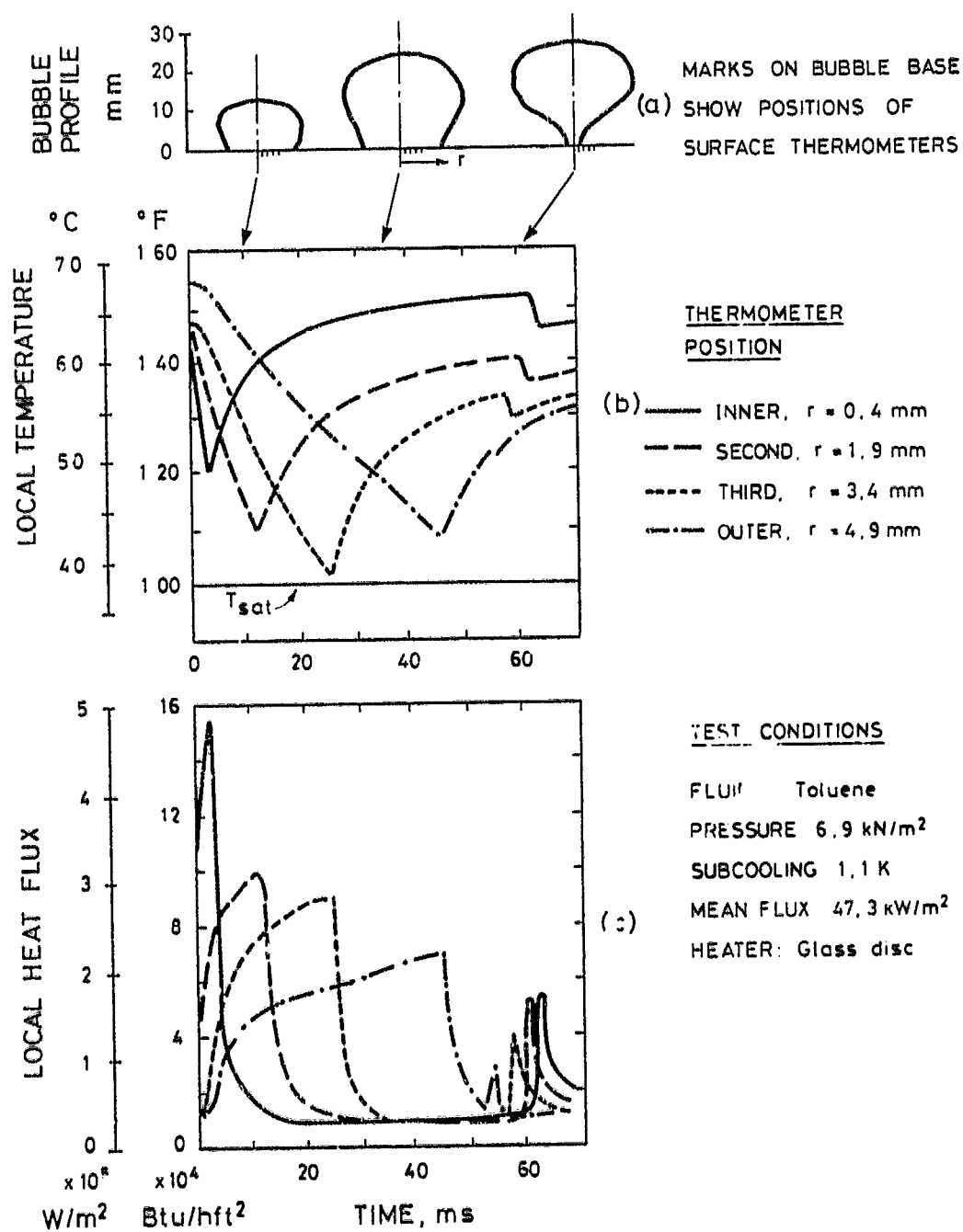


FIG. 4.4 SURFACE TEMPERATURE AND CALCULATED SURFACE
HEAT FLUX. COOPER AND LLOYD [99]

The d_o values resulting from the data of Figure 4.4 are shown in column 2 of Table 4.1. A second estimate of d_o was obtained from considerations of heat conduction in the liquid microlayer. The temperature of the bottom of the layer was given by Figure 4.4(b), that of the evaporating face was assumed as T_{sat} , and the gradient through the layer was taken as linear. The resulting estimates are shown in column 3 of Table 4.1. A third estimate, applied mainly to incompletely evaporated layers, resulted from the combined solution of the conduction equation for both the solid and the fluid. Starting with an assumed value of d_o , the wall temperature variations were obtained numerically, and the d_o values refined till best agreement between the calculated and measured temperatures was reached. The agreement between the three estimates of d_o was within ± 50 per cent in the worst cases and in general considerably better.

TABLE 4.1

INITIAL MICROLAYER THICKNESS, d_o , FROM DATA OF FIGURE 4.4

RADIUS (bubble centre to thermometer) mm	d_o FROM HEAT CONDUCTION	
	in solid μm	in liquid μm
0,4	3,89	6,25
1,9	9,86	10,8
3,4	15,6	13,6
4,9	22,4	21,6

From an approximate solution of the one-dimensional Navier-Stokes equation and a number of simplifying assumptions

Cooper and Lloyd were able to derive the following relationship for initial microlayer thickness:

$$d_0 = \text{const.} (vt_g)^{\frac{1}{2}} \quad (4.1)$$

where t_g is the time of bubble growth up to the point considered and the constant, from theory, is of the order of 0,8. For the experimentally observed microlayers the constant ranged from 0,5 to 1,0.

Analysis of microlayer evaporation, using either the measured temperature records or equation 4.1, and comparison with experimental bubble growth showed that the microlayer was essentially the sole source of vapour evolution into attached bubbles. It was stressed that this finding might not be general and referred strictly to the hemispherical bubbles under study.

In a subsequent paper Cooper [184] gave a general analysis of the conditions for domination of bubble growth by the microlayer. For non-metallic liquids on highly-conducting metal heaters these conditions are: (1) that the change in saturation temperature due to bubble inertia stresses is much less than the wall superheat, that is, that the vapour saturation temperature within the bubble is appreciably lower than the temperature at the heater-microlayer interface, and (2) that the bubble inertia force is considerably greater than the surface tension and gravity forces, that is, that the bubble is essentially hemispherical and does not lift off under buoyancy while small. In practice and at typical wall superheats these conditions are satisfied by many non-metallic liquids at subatmospheric pressure. When the conditions fail the vapour contribution from the thermal layer becomes significant. For water at saturation and typical wall superheats, for example, the microlayer predominates at 10 kN/m² pressure and is insignificant at 1000 kN/m².

In addition to estimates of the microlayer heat flow q_{ML}^i , the temperature records of Cooper and Lloyd also provide some assessments of the two remaining primary area-of-influence heat flows q_{LW}^i and q_{LA}^i . Figure 4.4 and many similar records show (1) the q_{LW}^i heat transfer, due to surface quenching following bubble departure, to be small compared with q_{ML}^i , and (2) q_{LA}^i , the heat transfer due to movement of liquid surrounding the attached bubble to be undetectable. Exploratory runs with steady boiling from an artificial nucleation site, however, revealed a small decrease in wall temperature over a region of diameter two or three bubble departure diameters.

A second microlayer study using thin-film resistance thermometry on a glass heater was reported by Foltz and Mesler [145] in 1970. The heater carried a single deposited platinum film some 0,5 mm wide, 0,5 to 6 mm long and less than 1 μ m thick; its time constant was approximately 1 ms. Water was boiled at saturation temperature at pressures of 6,9 to 27,6 kN/m² with natural steady nucleation from the thermometers. The resulting bubbles were large compared with the thermometer. From the experimental temperature records initial microlayer thicknesses, d_0 , were deduced by a method similar to the third method of Cooper and Lloyd, that is, from considerations of conduction in both the heater and the microlayer. The thickness was, however, refined iteratively for best agreement between the calculated and measured time required for completion of evaporation. This method, Foltz and Mesler claimed, was less prone to errors than the matching of temperatures practised by Cooper and Lloyd. The resulting microlayer thicknesses, of necessity treated as uniform with radius, were in the range of 3,3 to 9,4 μ m, that is, about half those of Cooper and Lloyd's.

4.2.3 Scale Deposition Technique

Ingenious estimates of microlayer thickness were obtained by Hospeti and Mesler [103] from the study of scale deposits formed beneath the base of bubbles.

Their test liquid was a saturated solution of calcium sulphate in water, tagged with radio-active sulphur-35. Calcium sulphate has a decreasing solubility with increasing temperature and is notorious for the formation of tenacious scale; it was thus assumed that no deposited scale would redissolve or become detached.

The solution was boiled at saturation temperature and atmospheric pressure from a Chromel heater at fluxes ranging from 14,9 to 65,0 kW/m². From photo-electric counts of the number of bubbles formed at a particular site, and from β -emission measurements of the mass of solute in the resulting scale, the mean mass of liquid evaporated per microlayer was estimated. On the assumption of stagnant microlayers, mean thicknesses of 0,48 to 2,59 μ m were found. These values are in agreement with the thermocouple-based estimates (see Section 4.2.1) of 0,2 to 4,0 μ m for water at atmospheric pressure. No trend of thickness with heat flux could be detected.

4.2.4 Optical Reflection Technique

Visual evidence for the existence of microlayers was first obtained in 1964 by Sharp [104] and Torikai et al [105] from observations of the reflection of light at the bubble base.

Sharp's paper dealt primarily with interference measurements of microlayer thickness; these are discussed in the following section. The reflection experiment here described was intended as a preliminary qualitative demonstration of

microlayer effects.

His heating surface was a glass disc forming the floor of a boiling vessel and heated from below by a jet of hot air. The test liquids, methanol and water, were boiled at (undisclosed) subatmospheric pressures to promote large bubbles. The boiling surface was illuminated from beneath by a collimated, plane-polarised beam of white light incident at Brewster's angle for the air-glass interface. This arrangement led to reflectances such that the microlayer appeared as a bright area against a grey background (the bulk liquid) and the dry spot appeared as a dark central area. An increase in heater surface temperature led to an increase in the dried-up area fraction of the microlayer. The contribution of microlayer evaporation to total bubble growth, q_{ML}^i/q_{LH}^i , however, could not be established since the microlayer thickness was unknown.

In the studies of Torikai et al [105] and of Torikai and Yamazaki [106,107] the heating surface was a stannic-oxide-coated, transparent and electrically heated glass plate. On this surface water was boiled at atmospheric pressure, at bulk subcoolings of 0 to 50 K and at heat fluxes of 30 to 1800 kW/m². The surface was illuminated obliquely from beneath so that the smooth upper surface of the microlayer gave specular total internal reflection and the rougher stannic oxide surface in the dry patch gave incomplete diffuse reflection. Wet (bright) and dry (dark) portions of the bubble base, as well as the bubble dome outline were clearly distinguishable. The microlayers were observed throughout the heat flux range. Their thickness and hence the ratio q_{ML}^i/q_{LH}^i were again unobtainable. The studies, however, contributed extensive information on the heater area fraction covered by microlayers and on microlayer aspects of burnout.

More recently, van Ouwkerk [110, 111] employed a technique

similar to that of Torikai and observed microlayers and dry spots with n-heptane, benzene and carbon tetrachloride boiling on glass and Perspex heaters at pressures of 16 to 30 kN/m²

Finally, Katto and Yokoya [108, 109], in the course of a study of burnout, obtained photographs of microlayers and dry spots on an opaque copper heater. Differences in reflectance of wet and dry bubble base areas were observed through the tops of bubbles flattened between the heater and a closely adjacent optical assembly. Burnout, it was concluded, occurred when detaching vapour masses prevented the re-supply of liquid to regions of evaporated microlayer.

4.2.5 Interference Measurements of Sharp

Direct optical measurements of microlayer geometry versus time were obtained by Sharp [104] for the special case of bubbles flattened between the heating surface and a parallel observation window. The measurements were based on ciné records of interference fringes produced in the manner of Newton's rings at the microlayer.

Sharp's heating surface was the air-heated glass disc already discussed: the test liquid was water, the system pressure approximately 10 kN/m² (saturation temperature 46 °C) and the bulk subcooling approximately 17 K. Heat fluxes from the glass disc could not be measured, but were guessed as roughly 40 kW/m². Under such conditions large isolated bubbles grew and were flattened between the heater and the observation window. The heater-to-window spacing was of the order of 5 mm. The microlayer was illuminated from above, that is, through the window and vapour, by a collimated, nearly monochromatic and nearly normally incident beam of light. Reflections from the top and bottom of the microlayer interfered to produce concentric light and dark fringes as in the case of 'Newton's rings by reflection' [146].

The refractive index of the glass, n_1 , was such that a bright central spot represented a microlayer thickness of zero; subsequent bright fringes corresponded to thicknesses of $1\lambda/2n_1$, $2\lambda/2n_1$, $3\lambda/2n_1$, ... and dark fringes to $1\lambda/4n_1$, $3\lambda/4n_1$, $5\lambda/4n_1$, ... all multiplied by a small correction factor for the departure from normal incidence. The rapidly shifting fringe patterns were recorded by high-speed ciné photography. Bubble growth was not determined.

The only set of results reported by Sharp is reproduced in Figure 4.5. From these thickness-versus-time records, and on the assumption of a stagnant (non-replenished) layer, Sharp calculated the mean microlayer evaporation heat flux as 120 kW/m^2 , that is, some three times the overall flux from the heater. This rather low flux augmentation factor was attributed to the thermal properties of the glass heater.

Microlayer thickness measurements did not extend beyond the second fringe of maximum reflectance, that is, beyond $0,42 \text{ } \mu\text{m}$, see Figure 4.5. A further bright circle of radius some two to three times that of the second maximum is, however, discernible in Sharp's photographs. If this circle is interpreted as the third fringe of maximum reflectance, then a sudden reduction in the slope of the microlayer beyond a thickness of $0,42 \text{ } \mu\text{m}$ results; see Figure 4.6 curve (a). If, on the other hand, the outer circle is interpreted as the periphery of the bubble base, then either a totally flat outer microlayer profile is obtained, see Figure 4.6 curve (b), or it must be assumed that fringes of higher order than the second maximum, for some reason (for example, insufficient monochromaticity), had become invisible.

For purposes of estimating the maximum possible contribution of microlayer evaporation to bubble growth, the last

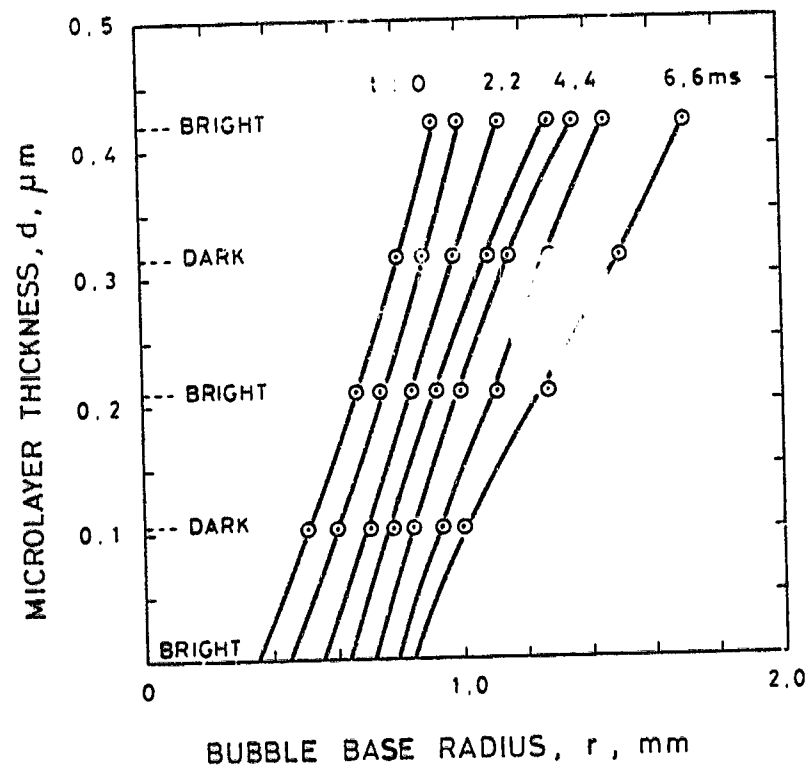


FIG. 4.5 MICROLAYER PROFILE vs TIME. SHARP [104]
 Test liquid: Water. Pressure: approx. 10 kN/m^2
 Subcooling: approx. 17 K. Bubble flattened
 between glass heater and parallel window;
 spacing approx. 5 mm.

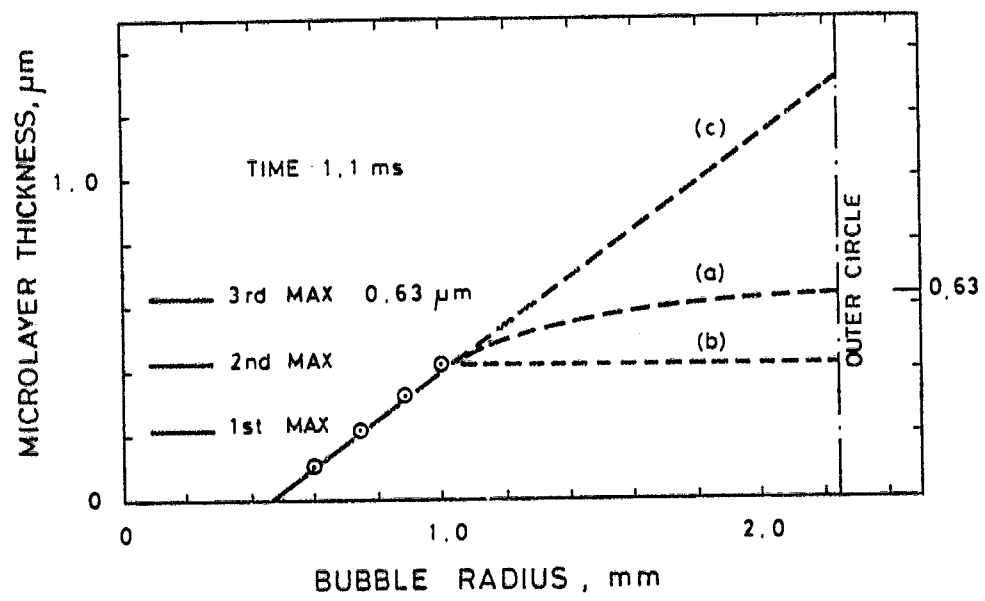


FIG. 4.6 POSSIBLE MICROLAYER PROFILES FROM DATA
OF SHARP [104]

possibility is accepted and the microlayer profile is assumed to continue linearly, at the same slope as in the visible portion, to the outer edge; see Figure 4.6 curve (c). For the bubble at 1,1 ms the outer circle had a radius of approximately 2,2 mm. Linear extrapolation to this radius, as shown in Figure 4.6, gives the corresponding microlayer edge thickness as 1,3 μm . Sharp's photographs show that at 1,1 ms the bubble apex had just touched the observation window. Up to that point the bubble had grown freely and, because of the low pressure, probably hemispherically. Thus Cooper and Lloyd's relation for initial microlayer thickness, d_0 , namely equation 4.1, should be applicable.

The equation predicts a d_0 of 20 μm , a thickness drastically larger than the experimental value of 1,3 μm . If, instead of the extrapolation treatment, the bright circle at radius 2,2 mm is interpreted as the third fringe of maximum reflectance, then the experimental d_0 becomes 0,63 μm and the discrepancy becomes even larger.

With linear extrapolation of the visible profile the reduction in microlayer volume between 0 and 1,1 ms can be calculated as 0,0011 mm^3 . The corresponding volume of saturated vapour is approximately 16 mm^3 . From Sharp's photographs and on the assumption of hemispherical geometry, the increase in bubble volume between 0 and 1,1 ms is 52 mm^3 . At a subcooling of 17 K the visible bubble growth probably accounts for less than one fifth of the total vapour supplied to the bubble. Thus the ratio of the microlayer contribution to the total latent heat flow appears to have been less than 16/250, that is less than 6 per cent, during the period considered. This estimate again contrasts sharply with those obtained from surface thermometry studies.

The confined bubbles of Sharp's experiment clearly constitute a special case. Nevertheless, the results, and particularly those for the first millisecond of growth, do

point to a microlayer behaviour in normal boiling that differs considerably from that deduced indirectly from surface thermometry. Direct measurements, by interferometry, on microlayers beneath freely-growing bubbles would appear to be called for.

4.2.6 Theoretical Predictions of Microlayer Thickness

Equations predicting the initial microlayer thickness have been derived by Cooper and Lloyd [100], Olander and Watts [147] and van Ouwerkerk [110], all on the following common basis:

- (1) Bubbles grow hemispherically according to the relation

$$R = C_1 t^{1/2} \quad (4.2)$$

Thus in the main body of the liquid (Region 1, Figure 4.7) a radial flow resembling inviscid flow is set up with the liquid velocity at the bubble surface essentially equal to dR/dt .

- (2) On the heater the liquid is at rest and in a thin adjacent region the radial flow is retarded by viscous forces so that a laminar boundary layer is formed (Region 2, Figure 4.7). The boundary layer flow is governed by the Navier-Stokes equation.

- (3) When the velocity of the bottom edge of the bubble dome fractionally exceeds the local boundary layer velocity (that is, when the bubble 'overtakes' the liquid) the microlayer is formed. Within the microlayer the liquid rapidly comes to rest. Since the bubble edge advances at dR/dt which is also the local freestream velocity the microlayer thickness is analogous to the displacement thickness of the boundary layer.

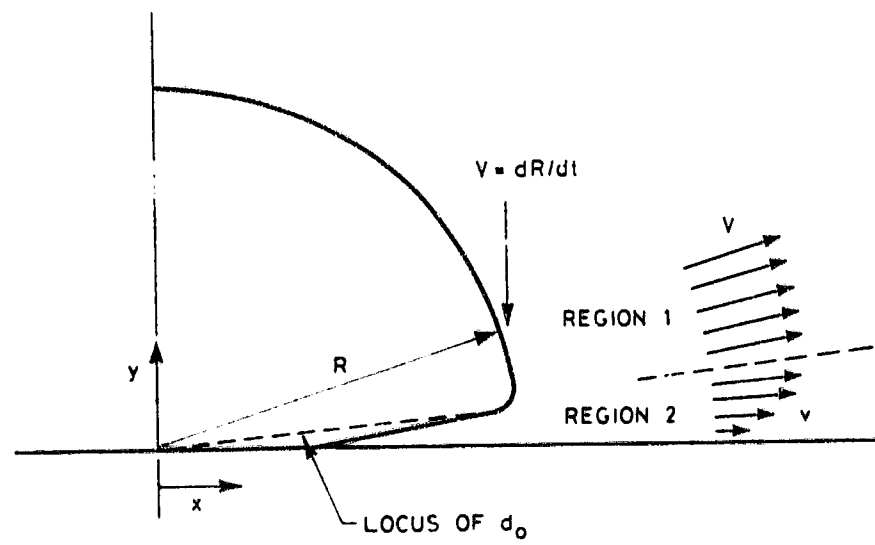


FIG. 4.7 MICROLAYER AND LAMINAR BOUNDARY LAYER
IN HEMISPHERICAL BUBBLE GROWTH

The derivations of the three equations differ in the degree of simplification of the Navier-Stokes equation, in the detailed treatment of the flow near the microlayer edge and in the treatment of the flow retardation within the microlayer. Despite this the equations differ only by a constant, as shown below.

The equation of van Ouwerkerk [110] was presented in the form

$$d_0 = 0,9 R / Re_b^{1/2} \quad (4.3)$$

where d_0 is the initial microlayer thickness, that is, the thickness undiminished by evaporation, at a bubble radius R . The bubble Reynolds number was given by

$$Re_b = (R \cdot dR/dt) / \nu_L \quad (4.4)$$

or from equation 4.2

$$Re_b = C_1^2 / 2\nu_L \quad (4.5)$$

that is, Re_b was independent of time. The constant 0,9 in equation 4.3 resulted from an elaborate solution of the Navier-Stokes equation.

Since d_0 is analogous to the displacement thickness of boundary layer theory and the Reynolds number is constant, equation 4.3 would be expected to be of the same form as expressions for the displacement thickness d^* of the laminar boundary layer in steady flow over a flat plate. This is indeed the case, the various solutions for d^* being [148]

$$d^* = \text{const. } x / Re_x^{1/2} \quad (4.6)$$

Substitution of equations 4.2 and 4.5 into equation 4.3 gives

$$d_o = 1,27 (v_L t_g)^{\frac{1}{2}} \quad (4.7)$$

where t_g is the bubble's time of growth to reach radius R .

The equation of Cooper and Lloyd [100], derived on the basis of a plane bubble front (that is, plane in the z direction in Figure 4.7) and with a simpler solution of the Navier-Stokes equation, is

$$d_o = 0,8 (v_L t_g)^{\frac{1}{2}} \quad (4.8)$$

With the bubble front treated as spherical the constant apparently rose to 1,12. Cooper and Lloyd accepted the lower value on the grounds of better agreement with their experiments.

Olander and Watts [147] presented their equation in the form

$$d_o = (\pi v_L)^{\frac{1}{2}} R / 4C_2 \quad (4.9)$$

where

$$C_2 = (3\alpha_L / \pi)^{\frac{1}{2}} \{ (T_w - T_{sat}) \rho_L C_L / \rho_V \lambda \} \quad (4.10)$$

Equation 4.10 resulted from Scriven's solution [149] for bubble growth in an isothermal field, applied to a hemispherical bubble growing according to the relation

$$dR/dt = C_2 t^{\frac{1}{2}} \quad (4.11)$$

Integration of equation 4.11 and substitution into equation 4.9, however, leads to

$$d_o = 0,89 (v_L t_g)^{\frac{1}{2}} \quad (4.12)$$

All three microlayer equations are thus reducible to the form

$$d_o = \text{const.} (v_L t_g)^{\frac{1}{2}} \quad (4.13)$$

For the d_o values deduced from Cooper and Lloyd's experiments (see Section 4.2.2) the constant in equation 4.13 took on values of 0,5 to 1,0 with a mean of 0,69. Cooper and Lloyd's theoretical value of 0,8 is thus in closest agreement, followed by that of Olander and Watts at 0,89 and that of van Ouwerkerk at 1,27.

Olander and Watts' prediction in original form (incorporating Scriven's growth relation), that is, equations 4.8 and 4.9, gives d_o values differing from the experimental value of Cooper and Lloyd [99] by +35 to -10 per cent.

The only other data suitable for testing the thickness predictions, that is, data of microlayer thickness as a function of radius and time, are provided by the direct measurements of Sharp. For a growth time t_g of 1,1 ms (the limit of the period of undistorted growth) equation 4.13 predicts a thickness d_o ranging from 20 μm (for Cooper and Lloyd's constant) to 32 μm (for van Ouwerkerk's constant). Since Sharp's ciné frame marked '0 ms' already shows a bubble the true growth time for the bubble marked '1,1 ms' could have been up to 100 per cent larger and the predicted d_o values could be as high as 28 to 45 μm . The experimental microlayer edge thickness, d , while not known precisely (see Figure 4.6), was between 0,63 and 1,5 μm . The initial edge thickness, d_o (the thickness prior to evaporation), may be estimated from Figure 4.5 as having been approximately 0,1 μm larger, that is, 0,73 to 1,4 μm .

The original equation of Olander and Watts, equation 4.9, when applied to the conditions of Sharp's experiment with physical properties evaluated at T_{sat} , and with the bubble radius at $t_g = 1,1$ ms taken as 2,2 mm, gives

$$d_o (T_w - T_{sat}) = 77,8 \text{ } \mu\text{m K} \quad (4.14)$$

The wall superheat was not measured in Sharp's experiment, but can be guessed to have been in the region of 10 to 30 K. The corresponding predicted d_o values are 2,6 to 7,8 μm which again differ considerably from Sharp's measurements.

In summary: The theoretical predictions of Cooper and Lloyd [100], Olander and Watts [147] and van Ouwerkerk [110], all of the form of equation 4.13, are in good to fair agreement with Cooper and Lloyd's thicknesses as deduced from thermometry, but are in total disagreement with Sharp's direct measurements. Olander and Watts' original equation incorporating Scriven's bubble growth relation, similarly agrees with Cooper and Lloyd's measurements and disagrees, but not too drastically, with Sharp's measurements.

4.3 SUMMARY AND DEFINITION OF PROBLEM

A review of previous microlayer studies has been given. This has shown that local surface temperature measurements, scale deposition techniques and optical methods have firmly established the existence of microlayers for a wide variety of surface-liquid combinations at atmospheric and subatmospheric pressures.

The local surface temperature measurements beneath and in the vicinity of bubbles have led to estimates of the heat flow due to microlayer evaporation, q_{ML}^i , and of the remaining surface-to-liquid heat flows in the area of bubble influence, q_{LA}^i and q_{LW}^i . The estimates of q_{ML}^i have in turn led to estimates of microlayer thickness. While doubt has been cast on the accuracy in some surface temperature studies, and several dealt with specialised forms of boiling (e.g. boiling from ultra-thin heaters, boiling with single individually-triggered bubbles) a fairly unified set of

conclusions has nevertheless emerged from them. These are: (1) that microlayer evaporation heat fluxes exceed the mean surface heat fluxes by factors of 6 to 20, (2) that under many conditions the microlayer heat transfer q_{ML}^i is a significant or even the predominant contributor to bubble growth, and (3) that the remaining surface-to-liquid heat transfer processes in the area of bubble influence, q_{LA}^i and q_{LW}^i , are of minor significance compared with microlayer evaporation.

All surface temperature studies, except those of Cooper and Lloyd [99, 100], employed single temperature sensors. From the output of these, and on the assumption of a stagnant microlayer, the mean thicknesses of evaporated microlayer liquid were estimated. In Cooper and Lloyd's studies multi-sensor thermometry and more advanced heat flow analyses (also based on the assumption of a stagnant layer) led to estimates of initial microlayer thickness as a function of bubble radius and time.

Theoretical equations for microlayer formation, the review showed, are all essentially identical and predict initial microlayer thicknesses which are in agreement with the values deduced from Cooper and Lloyd's measurements.

Only one set of direct measurements of microlayer geometry, that of Sharp [104], based on ciné-interferometry, was available for review.* In this study bubbles were flattened between the heating surface and a closely adjacent observation window. Bubbles thus grew under abnormal conditions.

*This was the situation at the commencement of the present study. Two further interference studies [150, 151], initiated in response to the published preliminary results of the present work [152], have since become available. These studies will be dealt with in Chapter 9.

Growth rate was not recorded. Nevertheless the interferometry results, and especially those for the first millisecond of (unrestricted) bubble growth, did point to a microlayer behaviour in normal boiling differing considerably from that deduced from surface thermometry. Initial microlayer thickness, in particular, differed by an order of magnitude from that predicted theoretically or deduced from the experiments of Cooper and Lloyd.

In order to clarify this situation the specific objectives of the present study were defined as follows:

- (1) To obtain, for freely-growing bubbles under various conditions of normal pool boiling, direct measurements of the microlayer shape and thickness as a function of time.
- (2) To record simultaneously and in synchronisation with the microlayer data, the profile views of growing and detaching bubbles.
- (3) To establish from such data the contribution of microlayer evaporation to bubble growth.
- (4) To test the theoretical equations for initial microlayer thickness against the direct measurements.

For the measurement of microlayer geometry versus time, interferometry (in the manner of Newton's rings) coupled with high-speed ciné photography seemed the only suitable technique. To satisfy the requirement of unrestricted bubble growth, interferometry was required from beneath and through a transparent heating surface. The synchronisation of the microlayer records with the bubble profile views should, it was felt, be achieved by image reflection such that the camera would see and record both bubble views side by side and on the same frame of film. Since large bubbles are necessary for interferometry, boiling at sub-atmospheric pressures was indicated.

CHAPTER 5BOILING APPARATUS5.1 GENERAL REQUIREMENTS

To achieve the objectives stated in Chapter 4 a boiling apparatus was required having the following capabilities:

- (1) To support normal pool boiling, both saturated and subcooled, with the formation of large isolated bubbles,
- (2) To offer an undistorted view of the bubble attachment area from beneath and through a transparent heating surface;
- (3) To offer an unobscured and undistorted profile view of bubble growth and detachment;
- (4) To permit the control and measurement of subatmospheric system pressure, of mean heat flux and of liquid bulk temperature;
- (5) To permit the measurement of mean surface temperature.

Such an apparatus was built and is described in the remainder of this chapter. A number of uncommon techniques, employed in the preparation of the heating surface, are given in some detail.

5.2 HEATING SURFACE

5.2.1 Specific Requirements

A transparent heating surface was required permitting the uniform transfer and measurement of low to medium heat fluxes (of the order of 10 to 100 kW/m²). The surface needed to be sufficiently sturdy to form the floor of a boiling tank operating at reduced pressure and to withstand the erosive action of bubbles and the corrosion of test liquids. The heat transfer was to cause no violent optical disturbance of the air below the surface. Nucleation of large isolated bubbles was required at or near the centre of the surface.

5.2.2 Electrically Conducting, Transparent Films on Glass

The above requirements could, it was felt, best be met by a plate glass surface having deposited on it a thin, electrically conducting, transparent film. The film was to be in contact with the test liquid and heated by direct current.

With a film of appropriate dimensions uniform current densities could be maintained and the heat flux calculated from measurements of the electrical input.

Natural convection heat losses from the bottom of a reasonably thick glass substrate would be low, thus minimising the optical disturbance of the air below.

Types of Film

Holland [153], reviewing electrically conducting transparent films on glass, lists the following as the most important types:

- (1) Vacuum-deposited cadmium oxide films. These are stated to have a poor stability to temperature gradients and were not investigated.
- (2) Vacuum-deposited metal oxide/gold two-layer films and metal oxide/gold/metal oxide sandwich films. A number of sandwich films of the types $\text{Bi}_2\text{O}_3/\text{Au}/\text{Bi}_2\text{O}_3$ and $\text{In}_2\text{O}_3/\text{Au}/\text{SiO}_2$ were prepared (by vacuum evaporation followed by air-baking) and tested as heating surfaces. Resistances of the order of 100 ohms/square and acceptable optical transmittances were attainable; boiling tests, however, revealed rapid film pitting and burn-out. These films were thus ruled out.
- (3) Chemically deposited stannic oxide films. Trial surfaces of stannic oxide on glass were prepared by the vapour deposition technique described in the following section. The films possessed outstanding adherence and abrasion resistance, optical transmittances of better than 90 per cent, resistances of the order of 100 ohms/square and excellent stability as heating surfaces in preliminary boiling tests.

Similar films have been successfully used in the boiling studies of Torikai [105, 106] and apparently also those of Kirby and Westwater [154] and Hsu [155].

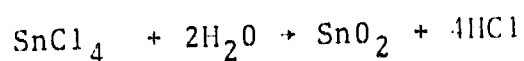
Stannic oxide films on glass were thus adopted for the present study.

5.2.3 Stannic Oxide Films on Glass

Preparation

The preparation of transparent conducting stannic oxide films, as patented by McMasters [156], consists of spraying the liquid stannic chloride, dissolved in organic solvents,

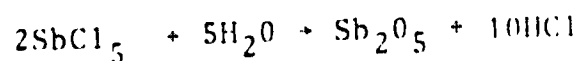
onto a glass surface heated almost to its softening point. In the presence of atmospheric moisture the chloride is hydrolysed as follows:



The breakdown of the organic carriers produces reducing conditions at the glass surface so that non-stoichiometric, that is, oxygen-deficient and thus conducting stannic oxide is deposited.

The conduction mechanism and methods of controlling film resistance are discussed by Aichison [157] and in patents [158, 159]. The presence of iron oxide (Fe_2O_3) or indium oxide (In_2O_3) in the film reduces conductivity, and antimony oxide (Sb_2O_5) increases conductivity.

The spraying solution used in the present study consisted of equal parts by volume of stannic chloride, absolute ethyl alcohol and glacial acetic acid. The stannic chloride was enriched with antimony chloride (by some 10 per cent by volume) so that antimony oxide was co-deposited with the stannic oxide.



The techniques of pre-cleaning, spraying and HCl corrosion prevention of the glass followed those given by Aichison [157] and in patents [156, 160].

Successful films as already described, with optical thicknesses ranging from one half to one and a half wavelengths, could be produced.

The deposition technique was, however, difficult to control, messy, and owing to the corrosive and poisonous chemicals involved, somewhat hazardous.

Commercial Film: 'Electropane'

Glass plate, ready-coated with conducting stannic oxide is manufactured as the commercial product 'Electropane' by the Libbey-Owens-Ford Glass Co. (of Toledo, Ohio).

Owing to information and supply difficulties this material was obtained only on completion of preliminary boiling tests. 'Electropane' surfaces were, however, employed in all formal tests.

The 'Electropane' coating had a nominal optical thickness of half a wavelength ($\lambda/2$), a refractive index of 2.0 and a resistance of 70 ohms/square. The substrate was plate glass of thickness 3 mm and refractive index 1.523.

The material was supplied as 150 mm square sheets. Coating thickness, as judged from interference colour, varied slightly from sheet to sheet and also gradually on each sheet. The optical thickness required in this study was $\lambda/2$, where λ was 437 nm, the wavelength of the interferometry beam used. Preliminary tests (see Appendix 6) established which interference colour corresponded to the required coating thickness. Circular boiling discs, approximately 40 mm in diameter, were cut out from correctly coloured areas in the sheets.

5.2.4 Electrical Contacts: Indium Soldering

Fired-on silver current contacts for 'Electropane' have been patented by the Libbey-Owens-Ford Glass Co. [161]. Attempts to duplicate such contacts, using platinum rather than silver, were not very successful. Indium soldering [162, 163] was found to be an entirely reliable and much simpler method of attaching contacts.

Belser [162, 164], investigating the wetting action of the metal indium (melting point 155°C) on a large range of materials, found that in the absence of a flux and at certain temperatures, indium formed excellent solder joints with several metal oxides and ceramics (for example, aluminium oxide and glass).

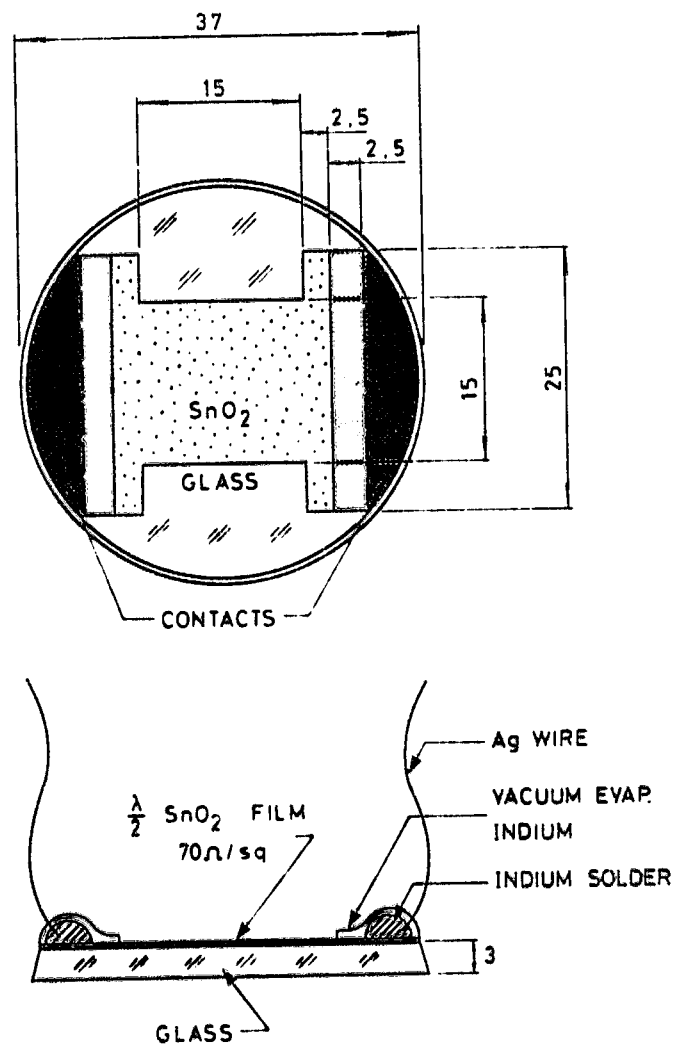
Belser's techniques were applied to stannic oxide films and similarly excellent bonds were obtained.

The heating disc, with edges ground to final size (see Figure 5.1) was ultrasonically cleaned in ethanol and methyl chloride. A miniature soldering iron, of bit diameter 2 mm and controlled via a variable transformer, was set to a temperature slightly above the melting point of indium. With a small blob of metal applied to the edge of the film, rubbing action of the bit, at gradually increasing soldering temperatures, suddenly established wetting. The molten metal, initially globular, could now be spread thinly over the contact area. The temperature was then lowered and a thicker indium film built up.

Silver current leads, 0,5 mm in diameter and pre-coated with indium, were then soldered into the contacts.

With practice contacts with reasonably straight edges could be achieved. However, accurate definition of the indium/stannic oxide contact line was achieved by vacuum-evaporating a further layer of indium slightly beyond the soldered contacts (see Figure 5.1). During vacuum-evaporation the central portion of the film was covered by a mask.

Utmost cleanliness was essential for the adherence of the evaporated layer. Thus after soldering, the surface was again ultrasonically cleaned (this also served as a solder adherence test), and then cleaned by ion-bombardment [153]



Dimensions: mm

FIG. 5.1 HEATING SURFACE

at 3 kV and 50 mA for 10 minutes. The indium was evaporated from a tungsten filament following standard techniques [153].

5.2.5 Configuration of Heating Film

The circular heating discs were initially entirely coated with stannic oxide. A parallel-sided heating strip with uniform energy dissipation in the film was, however, required. Further, to prevent excessive nucleation along the indium/stannic oxide contact line, the contact length was to exceed the width of the heating strip, see also [105]. The required H-shaped heater geometry (see Figure 5.1) was achieved by chemical removal of the stannic oxide film.

The removal technique, an adaptation from a patent by Tarnopol [165], consisted of painting the stannic oxide film with a reacting mixture of dilute hydrochloric acid and powdered zinc. This mixture reduced the oxide film to metallic tin which in turn was dissolved by the acid. The heating disc was masked with adhesive PVC tape, cut to the same shape as the required heating film, and the reacting zinc/acid mixture applied with a small brush to the bare portions of the film.

The distribution of energy dissipation in the H-shaped film was determined from a conducting sheet analogue [166, 167]. An enlarged model of the film was cut out from graphite-coated paper (Teledeltos paper), contacts were attached along the 'legs' of the letter H, and the equipotential lines determined with a Servomex (type EP 144) field plotter. Results are shown in Figure 5.2. Over the central portion of the heating strip the distribution is seen to be uniform.

5.2.6 Artificial Nucleation Site

Nucleation of large isolated bubbles was required at the centre of the heating surface.

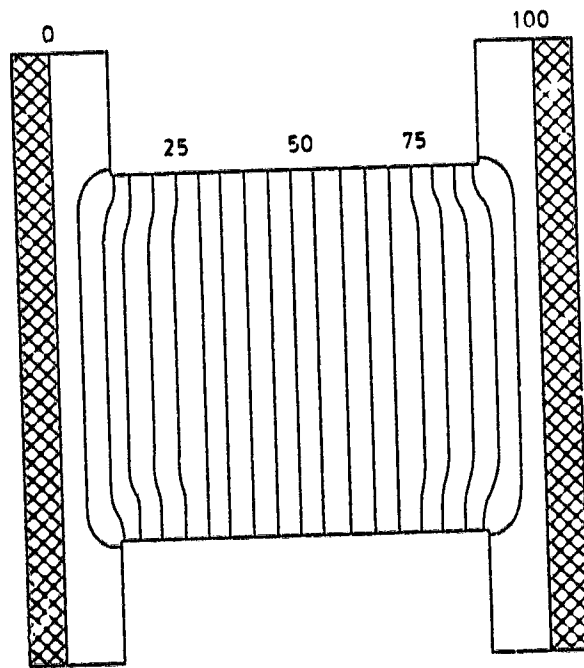


FIG. 5.2 EQUIPOTENTIAL LINES FOR HEATING FILM

From the nucleation theory discussed in Chapter 3 it may be seen that a vapour-entrapping, heated cavity, larger than the largest naturally occurring cavity was required at the centre of the surface. Such an artificial cavity was produced as follows:

In the middle of the disc a small pit was scratched (with a diamond tool) through the stannic oxide layer and into the glass. The pit was again rendered electrically conducting by a vacuum-evaporated indium film. Deposition of this film, and of the film covering the electrical contacts, was carried out in one operation. To this end the mask covering the test section (and demarcating the edge of the contacts) had a 5 mm hole drilled in the middle. Indium deposited on the heater around the mouth of the nucleation pit was polished away with a mild abrasive (the household polish 'Brasso'), see Figure 5.3. Traces of the abrasive were removed by ultrasonic cleaning.

The heated cavities thus produced were active only when the correct geometry was achieved by chance, usually after numerous attempts.

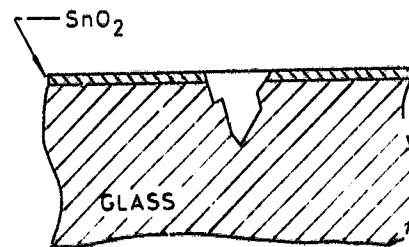
Successful surfaces, under subatmospheric pressure and at low heat flux produced large isolated bubbles and at higher fluxes vertically interfering bubbles and vapour columns.

Laterally interfering bubbles from naturally occurring nucleation sites were also obtainable.

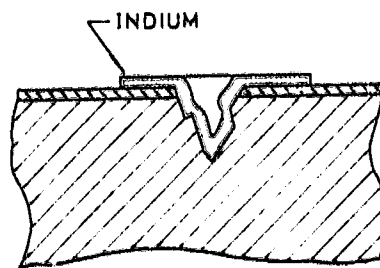
5.2.7 Electrolytic Corrosion

The heating surface assembly operated satisfactorily with several organic test liquids. With water, however, the contacts were destroyed by electrolytic corrosion.

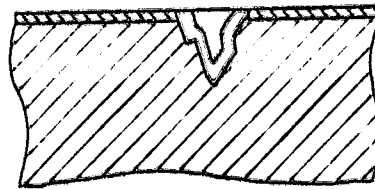
Protection of the contacts was attempted as follows:



CAVITY SCRATCHED
THROUGH GLASS
AND OXIDE LAYER



INDIUM EVAPORATED
INTO AND AROUND
CAVITY



EXCESS INDIUM
POLISHED AWAY.
CAVITY REMAINS
COATED

FIG. 5.3 PREPARATION OF ARTIFICIAL
NUCLEATION CAVITY

The evaporated indium film was replaced with aluminium and on this a high-resistance, non-porous anodic oxide film [168, 169] was electroformed; this in turn was coated with a film of epoxy resin.

While some improvement resulted, it was now found that the stannic oxide film itself was electrolytically attacked. A similar attack is reported by Kirby and Westwater [154].

It was thus decided to revert to bare all-indium contacts and to limit the study to organic test liquids.

5.3 BOILING SYSTEM

5.3.1 Requirements

A boiling vessel was required such that the heating surface described constituted its floor and could be viewed unobscured and undistorted both from beneath and in profile.

Within the tank subatmospheric pressure at various levels was to be maintained and measured.

Test liquid temperatures needed to be controlled and measured, both at saturation and at various levels of sub-cooling.

The vessel was to be corrosion-resistant to a large range of organic test liquids.

5.3.2 Overall Description

The boiling system is shown schematically in Figure 5.4.

The heating surface was mounted on a hollow plug which fitted into the bottom of a cylindrical glass tank. The tank was connected via a non-reflux condenser and a

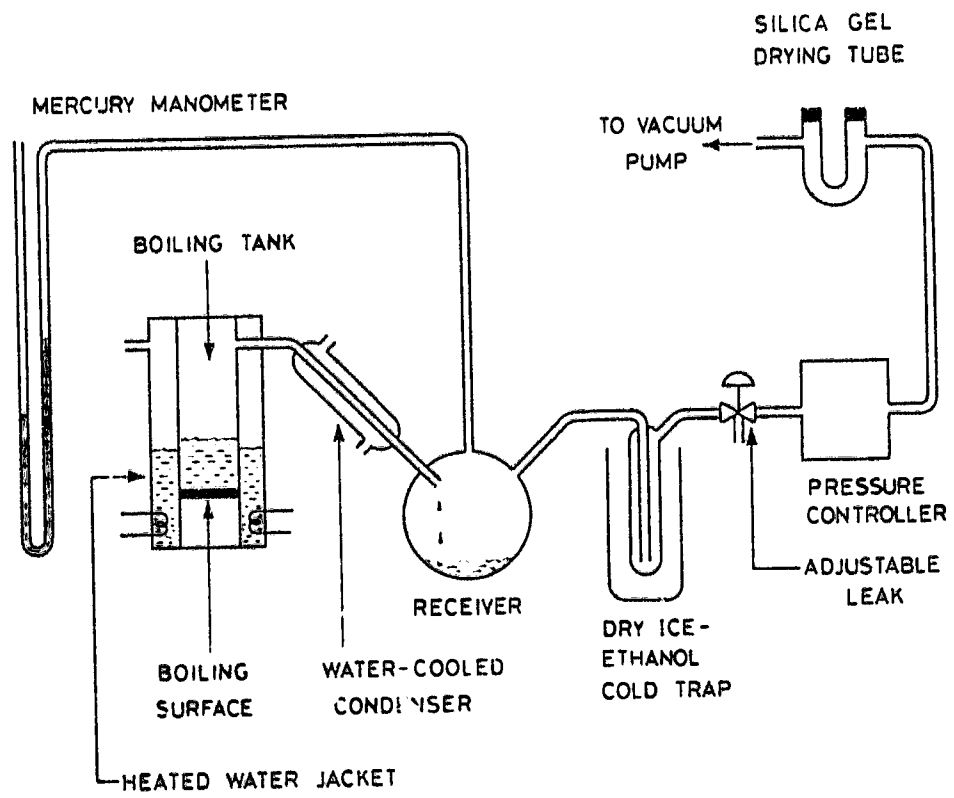


FIG. 5.4 BOILING APPARATUS
(Optics not shown)

receiver to a pressure control system. This consisted of a Cartesian manostat [170] protected by a dry ice/ethanol cold trap on the tank side, and a silica gel drying tube on the other. A small air leak into the system was provided via a needle valve. Vacuum was drawn by a water jet pump. A mercury manometer, connected to the receiver measured the system pressure.

A flat-sided, glass-walled water jacket, maintained at atmospheric pressure, surrounded the boiling tank. This provided a constant-temperature bath and eliminated distortion of the bubble profile view. Water jacket temperature was maintained by variac-controlled heaters and measured by a mercury thermometer.

Inner tank temperature was measured by thermocouples sealed through the plug walls and by a thermometer sealed through the tank lid. A variac-controlled platinum heating coil was suspended above the heating surface from two electrodes sealed through the tank lid. When desired, precise saturation temperature could be maintained in the inner tank by additional boiling from this coil.

Details of the main system components are provided in subsections 5.3.3 and 5.3.4 below.

5.3.3 Mounting of Heating Surface

The mounting plug and heating surface are shown assembled in Figure 5.5.

The plug body and the clamping ring were constructed in stainless steel. O-rings were of Viton.

Copper current leads 0,8 mm in diameter were sealed and insulated with epoxy resin into 1,5 mm holes through the plug wall.

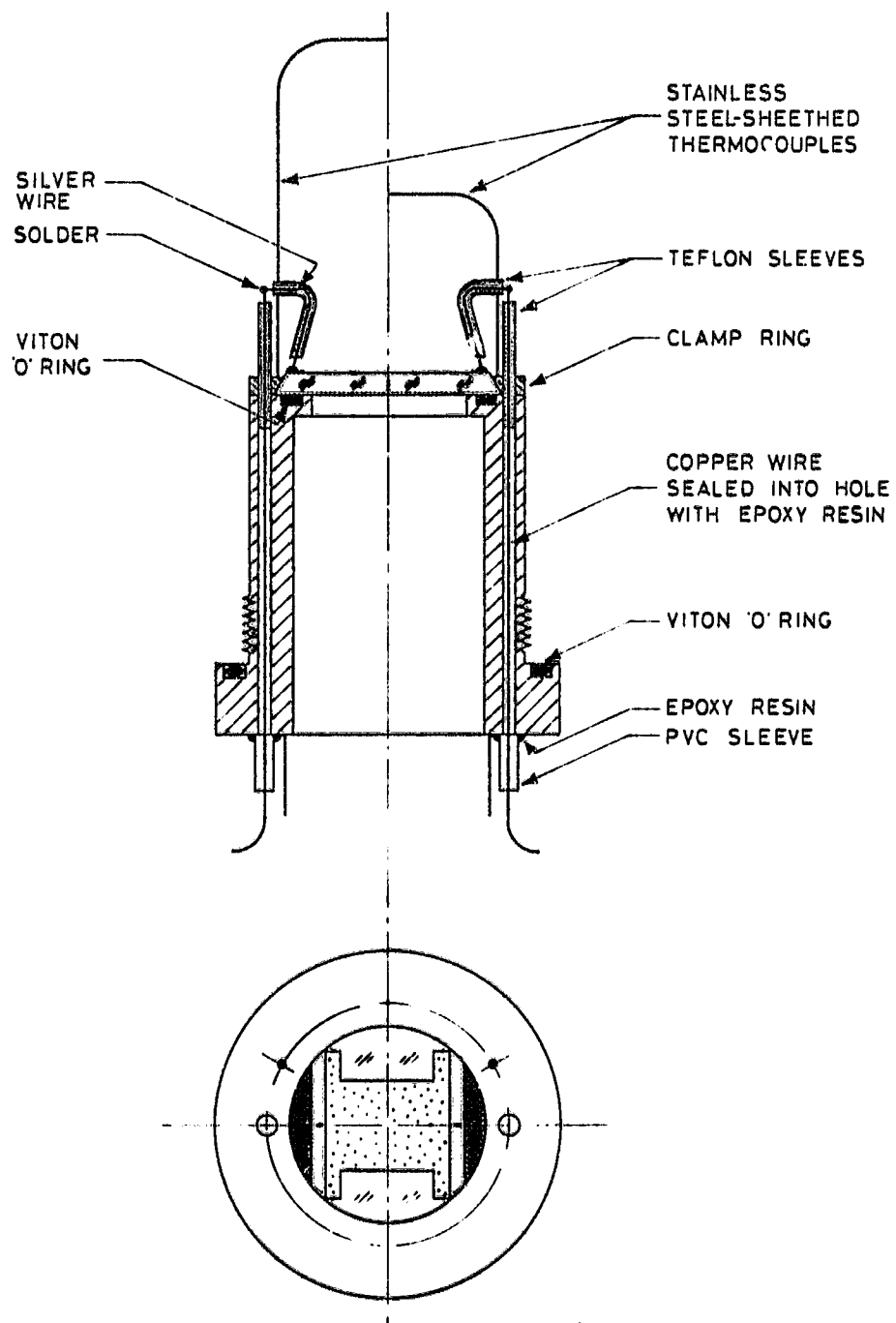


FIG. 5. 5 MOUNTING OF HEATING SURF 5.

Towards the top of the holes the resin was replaced with silicone rubber sealant (General Electric RTV 102), and above that with tightly-fitting Teflon sleeving. The sleeving virtually excluded contact between the test liquid and the sealant, and, in extending beyond the plug, insulated the current leads from the clamping ring and reduced contact with the test liquid. The silver current leads to the heating surface were similarly Teflon sleeved.

Two stainless steel-sheathed thermocouples, of sheath diameter 0,5 mm, were sealed through the plug walls in a similar manner.

The mounted heating surface protruded slightly beyond the clamping ring, thus offering an unobscured profile view of the bubble attachment area.

5.3.4 Tank and Water Jacket

The boiling tank consisted of a glass tube, of internal diameter 56 mm and wall thickness 2 mm, sealed with Viton O-rings into two stainless steel blocks, see Figure 5.6. The blocks were secured by two stainless steel tie-rods.

The mounting plug and heating surface assembly screwed into the lower block and against a Viton O-ring seal. The liquid in the narrow annular space between the mounting plug and the tank wall was essentially stagnant; thus the heating surface could be considered as the true floor of the tank.

The stainless steel tank lid was sealed by a Viton O-ring and carried a mercury thermometer and two electrodes for the auxiliary boiling coil.

A port in the upper tank block and a Teflon-gasketed

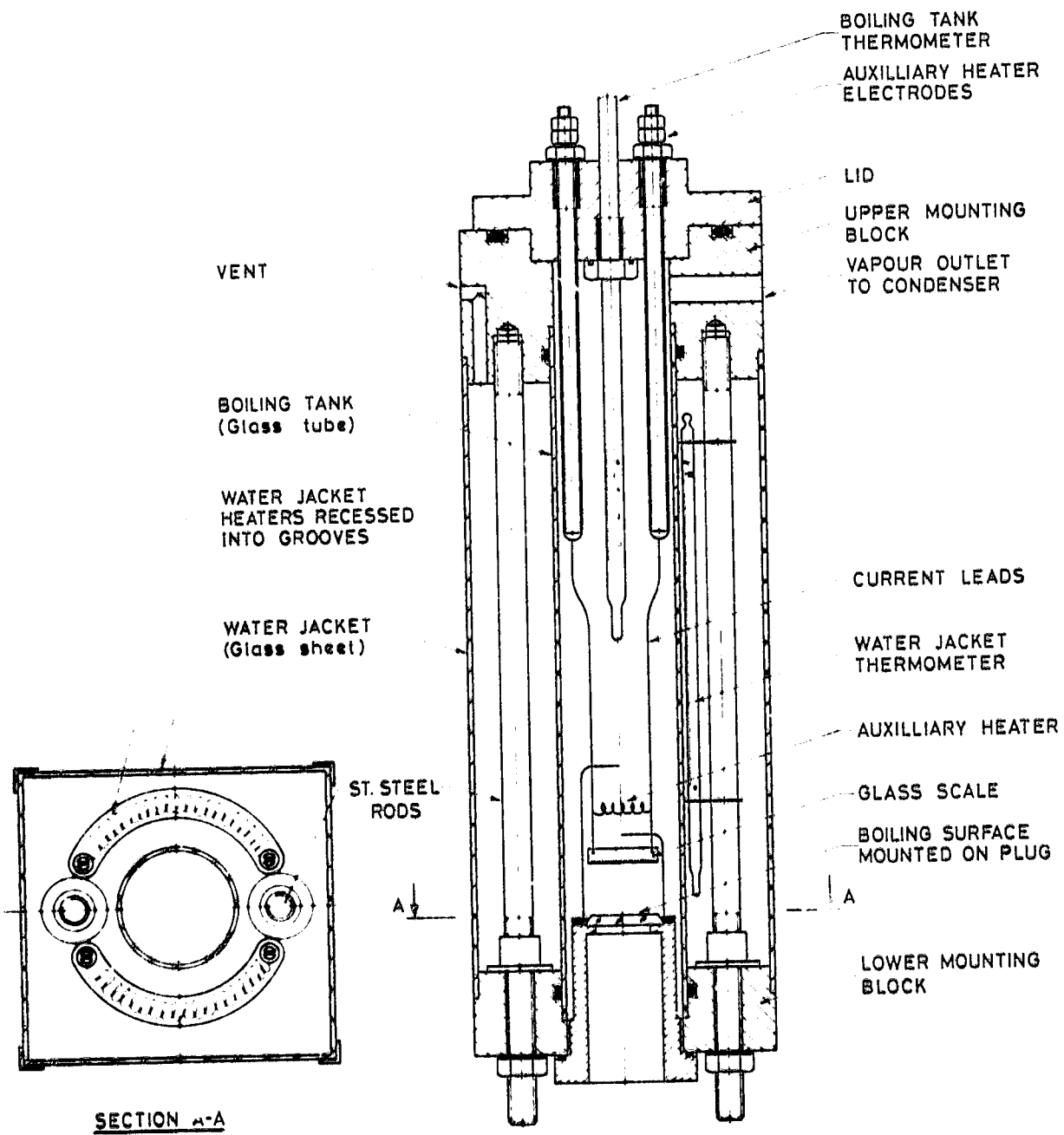


FIG. 5.6 BOILING VESSEL ASSEMBLY

'Quickfit Visible Flow' joint connected the boiling tank to the condenser.

The water jacket surrounding the cylindrical tank consisted of four sheets of glass, 2 mm thick, attached by 'underwater grade' adhesive tape to the upper and lower tank blocks. A vent in the upper block maintained atmospheric pressure in the jacket.

Two Nichrome coils, the jacket heaters, were recessed into silicone rubber-insulated grooves in the lower tank block.

5.4 CONTROL AND MEASUREMENTS (NON-OPTICAL)

This section discusses the control and measurement of mean heat flux, system pressure and bulk liquid temperature, and the measurement of mean surface temperature of the heating surface.

5.4.1 Mean Heat Flux from Boiling Surface

The electrical system of the heating surface is shown in Figure 5.7.

A constant-voltage transformer (C.I.T.E. Type 250 FS) was connected across the insufficiently 'clean' mains supply. Supply voltages ranging from 200 to 240 V could be stabilised to 220 V with deviations of less than 1 per cent.

Connected to the constant-voltage transformer was a rectifier power pack which controlled the input to the heating surface. A continuously variable voltage of 0 to 180 V could be supplied at a maximum current of 2 A. Voltage ripple during boiling tests was less than 0.05 per cent. The instrument was designed and supplied by the Instrumentation Division of the Atomic Energy Board.

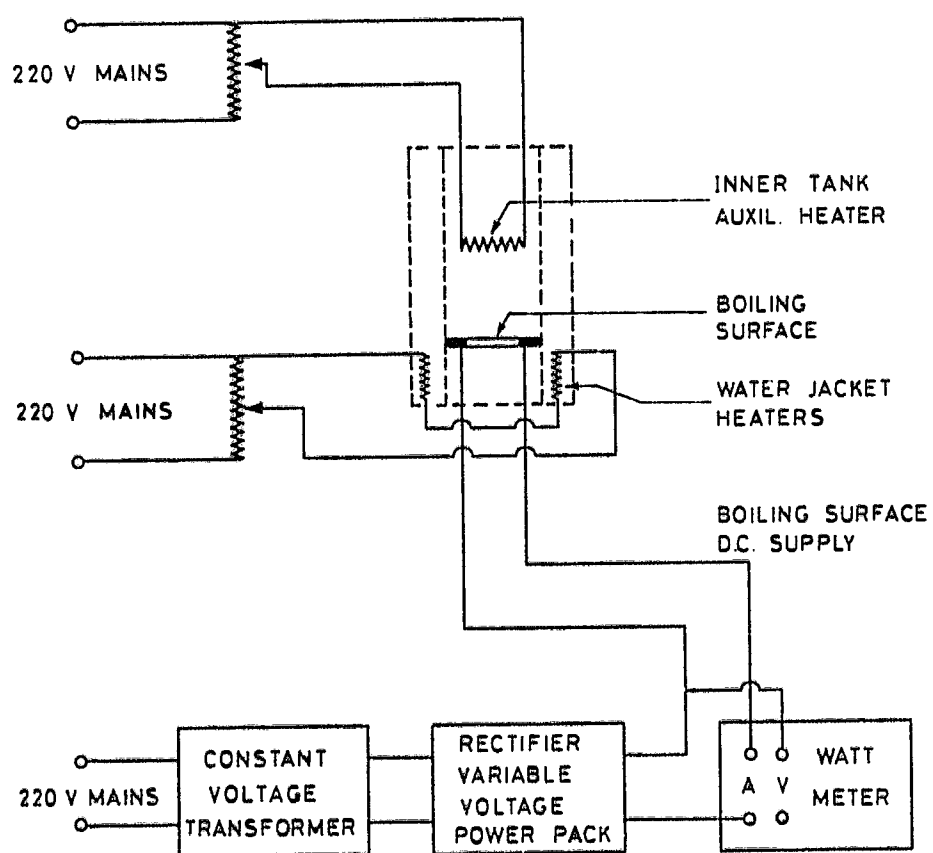


FIG. 5.7 ELECTRICAL SYSTEM

Power dissipation was measured by a precision wattmeter (Goertz, Type 444381), newly calibrated by the manufacturers. With the instrument connected as shown in Figure 5.7, a correction for leads resistance was necessary to obtain the power dissipation in the heating film; see Appendix 7. Radial conduction losses and heat losses from the bottom of the heating disc were considered negligible; see also Appendix 7. Thus the power dissipated in the heating film was entirely transferred to the liquid. For the central, uniformly heated section of the film the mean heat flux could then be calculated. The accuracy of this determination was estimated as ± 7 per cent. Details are given in Appendix 7.

5.4.2 System Pressure

Pressure control was provided by a Cartesian manostat (Edwards, Model 6). The instrument consisted of a tube closed on top, floating, in the manner of a diving bell, in a pool of mercury. Air at the required control pressure could be trapped inside the diving bell. The outside of the bell was connected to the boiling system. As the system was evacuated the bell rose and closed the exhaust nozzle. Leakage into the system, provided via a needle valve, tended to depress the bell. At steady state the exhaust nozzle was nearly closed with the rate of evacuation balancing the rate of leakage and system outgassing; see also [170].

The instrument permitted operation at any pressure between $0,5 \text{ kN/m}^2$ and atmospheric, with fluctuations of less than $0,1 \text{ kN/m}^2$.

Absolute system pressure was obtained as the difference between atmospheric pressure, measured by a Fortin barometer, and the system gauge pressure measured by a U-tube mercury manometer. Both instruments were corrected to 0°C and to

standard gravitational acceleration in the usual manner [171].

5.4.3 Bulk Temperature of Liquid

In runs at saturation temperature steady state conditions were maintained in the tank by boiling from the auxiliary heater.

In subcooled boiling runs the initial test liquid temperature was set by the temperature of the water jacket. Once boiling in the inner tank was started the bulk temperature rose with the duration of the run. The temperature of interest was the one prevailing during the optical observations. These observations occupied approximately one second; the temperature rise during this interval was negligible.

The liquid bulk temperature was measured by two stainless steel-sheathed Chromel-Alumel thermocouples (Solderm Thermocoax, Type 2ABAc05TI) and plotted on a multipoint recorder (Leeds Northrop Speedomax W). The thermocouple tips were situated 30 mm and 50 mm above the centre of the heating surface; this prevented interference with attached bubbles. Except in the case of periodic vapour envelopment of the lower thermocouple in subcooled boiling, the two temperature readings were identical.

The thermocouples had a sheath diameter of 0,5 mm and junctions, insulated from the sheath, of 0,2 mm. Their time constant, when plunged from air at room temperature into water at 100 °C, was 35 ms. This response was more than sufficient to follow the slow liquid temperature rise during subcooled boiling runs.

A precision mercury thermometer, with the bulb arranged 100 mm above the heating surface, provided an additional temperature measurement.

From the agreement between the thermometer and thermocouple temperatures, and from consideration of the scatter in the recorder output, the accuracy of the reported bulk liquid temperatures was estimated as $\pm 0,5^{\circ}\text{C}$.

5.4.4 Mean Surface Temperature of Heater

It was initially attempted to obtain the mean heater surface temperature by using the stannic oxide film itself as a resistance thermometer. Preliminary tests, however, showed that the coefficient of resistivity of the film was of the order of $1,5 \times 10^{-4}^{\circ}\text{C}^{-1}$, (cf platinum: $4 \times 10^{-3}^{\circ}\text{C}^{-1}$). With errors in voltage drop and current measurements of $\pm 0,2$ per cent, that is, errors in resistance of $\pm 0,4$ per cent, the resulting error in the surface temperature would have been of the order of $\pm 26^{\circ}\text{C}$. In addition, the film resistance at a particular temperature appeared to drift with heating; for example, a surface calibrated before and after some 10 minutes of boiling showed a resistance increase of 1 in 500. The method was thus ruled out.

Instead, the temperature of the bottom of the boiling disc was measured. A further Solder thermocouple of 0,5 mm sheath diameter was pressed firmly against the disc as shown in Figure 5.8 and attached with adhesive masking tape. Appendix 8 shows that this method of attachment essentially ruled out axial conduction losses in the thermocouple. Appendix 8 further shows that the contact resistance between the thermocouple tip and the glass surface was negligible and that the convection losses from the bottom of the surface and from the thermocouple sheath (acting as an extended surface) were similarly negligible. Thus the measured temperature corresponded closely to the mean boiling surface temperature. The accuracy is estimated in Appendix 8 as $\pm 5^{\circ}\text{C}$.

The masking tape and thermocouple obscured the bubble

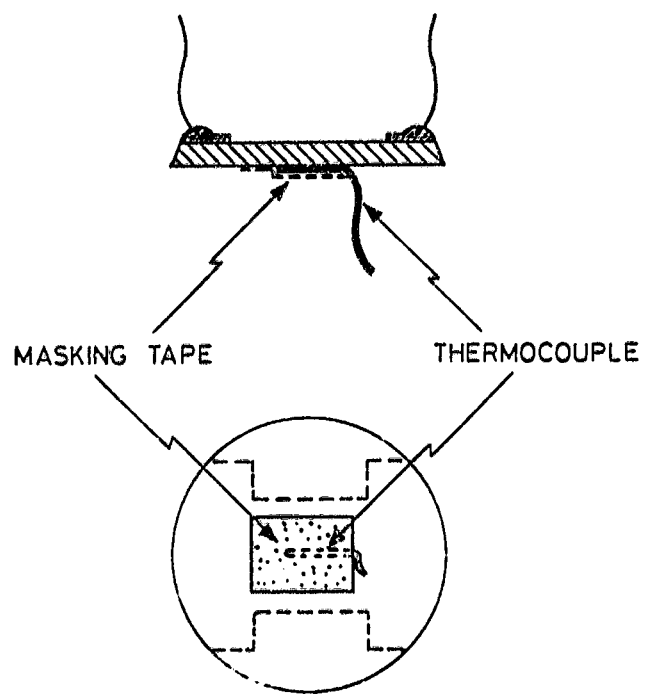


FIG. 5.8 ATTACHMENT OF THERMOCOUPLE
ON BOTTOM OF HEATER

185.

view from beneath. Thus for each run the temperature measurements were carried out immediately on completion of the optical observations.

CHAPTER 6OPTICS OF MICROLAYER INTERFEROMETRY AND
OPTICAL APPARATUS6.1 OUTLINE

The first part of this chapter (section 6.2) deals with the optics of interference by amplitude division at a microlayer in contact with the heating surface described in Chapter 5.

First, the required basic concepts are introduced. There follows a detailed analysis of all reflectances and of the fringe contrasts which occur in the system (consisting of the glass substrate, the stannic oxide film, the liquid microlayer and the surrounding 'infinite' liquid) under near-perfectly collimated and monochromatic, normally incident illumination. The analysis shows that only a limited range of liquid refractive indices gives fringes of the required high contrast. The test liquids methanol and ethanol are found to be suitable for the present study. For these liquids the microlayer thickness is given as a function of fringe order. Regions of evaporated microlayer, and the surroundings of the fringe pattern are shown to be dark and to have a reflectance equal to that of the fringe minima. Reflections from the lower surface of the glass heater are shown to be undesirable and the effect of a magnesium fluoride anti-reflection coating is analysed.

The remainder of the chapter (section 6.3) describes the complete optical apparatus.

Details of the interferometry beam system are provided and

the effects of imperfect collimation and imperfect monochromaticity are analysed. The conventional illumination of the bubble in profile, the mirror system bringing together the two bubble views, and the high-speed camera are described.

6.2 OPTICS OF MICROLAYER INTERFEROMETRY

6.2.1 Basic Concepts

(a) Optical Thickness

In the study of thin films it is desirable to express thicknesses in terms of wavelengths.

Light having a wavelength λ in a vacuum has a wavelength $\lambda' = \lambda/n$ in a film of refractive index n . The thickness of this film could thus be given in terms of λ' , the 'internal wavelength'. It is common optical practice to use the more convenient term 'optical thickness', defined as follows:

$$\text{Optical thickness, } d' = nd \quad (6.1)$$

The statement 'optical thickness = $x\lambda'$ ' means that the film has a thickness such that it represents a path of x wavelengths within it ($d = d'/n = x\lambda'/n = x\lambda$).

The stannic oxide (SnO_2) films in the present study had an optical thickness of $\lambda/2$ for $\lambda = 437 \text{ nm}$.

(b) Optical Path Difference

For normal incidence the optical path difference, $\bar{x}$, in a film is simply twice its optical thickness, that is,

$$\bar{x} = 2nd \quad (6.2)$$

For the SnO_2 films used, $\bar{x} = \lambda$. The path of a ray of light reflected from the top of the film thus differed from that reflected from the bottom by one internal wavelength.

(c) Reflectance

If I is the intensity of light waves then the reflectance, ρ , is given as follows:

$$\rho = I_{\text{reflected}} / I_{\text{incident}} \quad (6.3)$$

(d) Contrast

For a system of interference fringes, that is, for maxima and minima of reflectance, the contrast, κ , is defined, (see Françon [172] p 31) as follows:

$$\kappa = (I_{\text{maxima}} - I_{\text{minima}}) / I_{\text{maxima}} \quad (6.4)$$

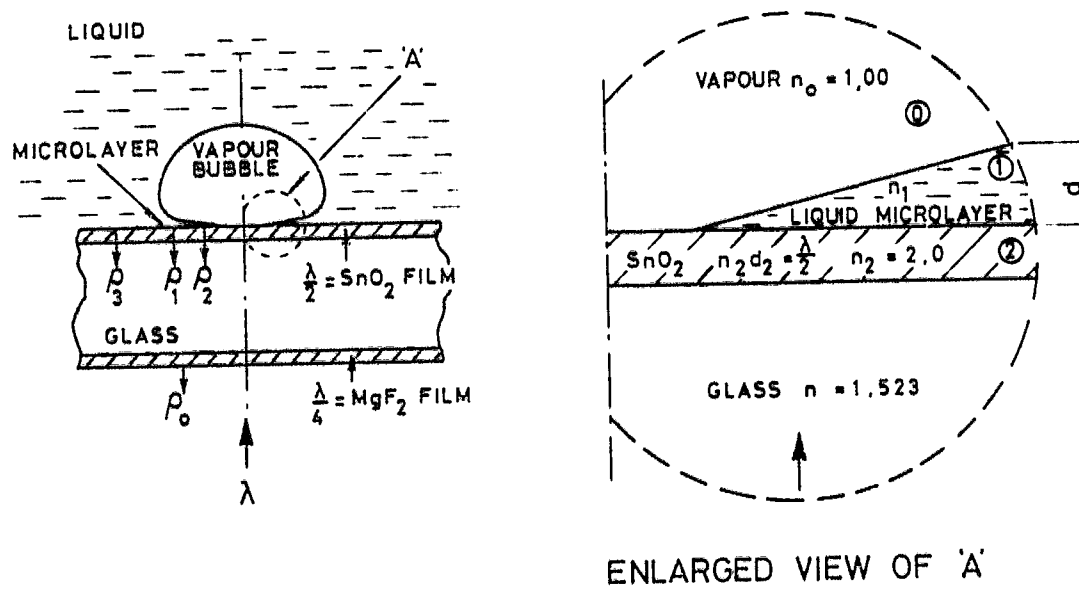
thence also

$$\kappa = (\rho_{\text{maxima}} - \rho_{\text{minima}}) / \rho_{\text{maxima}} \quad (6.5)$$

The maximum contrast ($\kappa = 1.0$) is clearly the ideal condition for the measurement of film thicknesses by interference methods.

6.2.2 Optical Analysis

Consider a bubble with a partially evaporated microlayer attached to the heating surface, as shown in Figure 6.1. The system is illuminated from below by a normally incident, nearly parallel and nearly monochromatic beam of light. Reflectances from the top and bottom of the microlayer interfere to produce localised fringes similar to Newton's rings [172].



n = REFRACTIVE
 ρ = REFLECTANCE
 d = THICKNESS

FIG. 6.1 OPTICS OF MICROLAYER INTERFEROMETRY

The reflectances of the various regions of this system are evaluated below and the fringe contrast is calculated.

(a) Microlayer and $\lambda/2$ SnO_2 film

See Figure 6.1 b. Two extreme cases of reflectance, dependent on microlayer thickness, can be recognised:

Case 1. Microlayer optical thickness is zero or an even multiple of a quarter wave, i.e.

$$n_1 d = 2m (\lambda/4), \quad m = 0, 1, 2 \dots \quad (6.6)$$

that is, the microlayer path difference $\bar{x} = 0, \lambda, 2\lambda \dots$. The reflectance is then given (see Vašíček [173] pp 124, 176) by

$$\rho_1 = \left[\frac{(n - n_0)}{(n + n_0)} \right]^2 \quad (6.7)$$

where refractive indices are defined in Figure 6.1.

Case 2. Microlayer optical thickness is an odd multiple of a quarter wave, i.e.

$$n_1 d = (2m + 1) (\lambda/4), \quad m = 0, 1, 2 \dots \quad (6.8)$$

that is, the path difference $\bar{x} = \lambda/2, 3\lambda/2, 5\lambda/2 \dots$. The reflectance is then given ([173] p 177)

$$\rho_2 = \left[\frac{(n_1^2 - nn_0)}{(n_1^2 + nn_0)} \right]^2 \quad (6.9)$$

and is dependent on n_1 , the refractive index of the liquid.

The effects of phase reversal, occurring on reflections at optically denser media [146], are inherent in equations 6.7 and 6.9. The refractive indices of the various media decide which case represents maxima of reflectance and which minima.

In case 1 ρ_1 is independent of the liquid refractive index and n_0 can be taken as 1,00 for all vapours. Thus for a given glass the value of ρ_1 is fixed. In the present case $n = 1,523$ (the value provided by the manufacturers) and thus $\rho_1 = 0,0429$ or 4,29 per cent.

In case 2 ρ_2 depends on the liquid refractive index n_1 , as shown in Figure 6.2.

For the range of liquid refractive indices $1,0 < n_1 < 1,523$ we have $\rho_1 > \rho_2$, that is, ρ_1 represents maxima of reflectance and equation 6.6 defines light fringes, while equation 6.8 defines dark fringes. For the range of refractive indices $1,523 < n_1 < 2,0$ the reverse holds. Thus, using equation 6.5, the fringe contrast can be calculated as a function of liquid refractive index, n_1 . Results are shown in Figure 6.3.

Clearly the test liquid must be chosen with care; as its refractive index, n_1 , tends towards that of the glass, n , the contrast tends towards zero and fringes disappear. If it were essential to study such a liquid (e.g. benzene, $n_1 = 1,50$) a different glass substrate of suitable refractive index would have to be found.

Maximum contrast ($\kappa = 1,0$) is achieved with $\rho_2 = 0$, that is with $n_1 = n^1 = 1,234$. Liquids with refractive indices in the vicinity of this value are [174]: nitrous oxide ($n_1 = 1,19$), hydrocyanic acid ($n_1 = 1,25$) and hydrogen chloride ($n_1 = 1,25$). These clearly had to be ruled out in our study. Operation in the region $n_1 > 1,523$, see Figure 6.3, appeared unprofitable. The highest achievable refractive indices (for practically usable liquids) appeared to be about 1,6 (e.g. carbon tetrachloride); the corresponding contrast is only 0,34. It is seen then that the refractive index of the heating surface's glass substrate imposed limitations on the liquids that could successfully be tested.

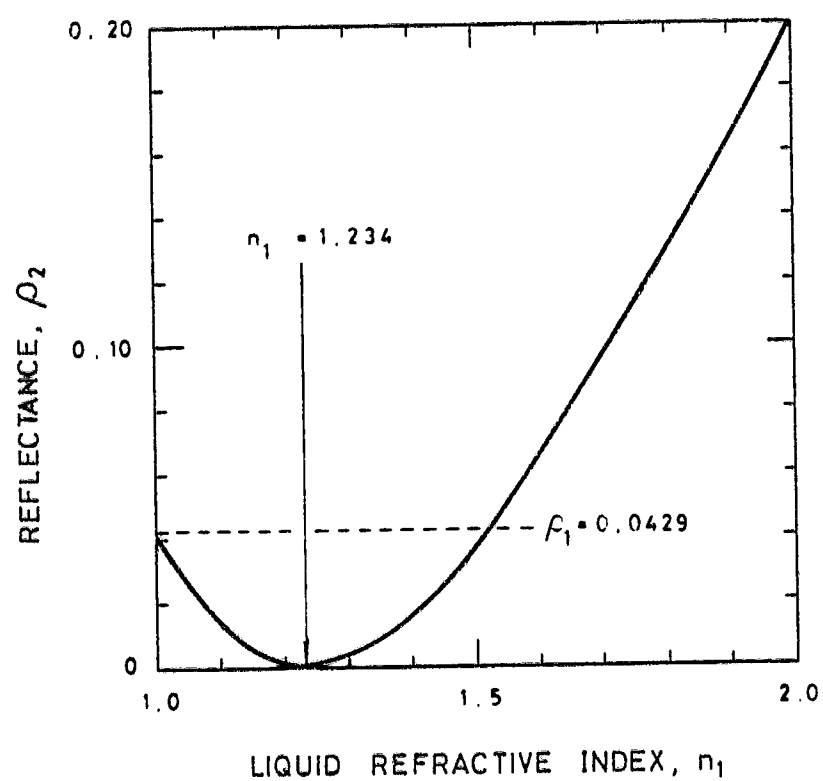


FIG. 6.2 FRINGE REFLECTANCES vs REFRACTIVE INDEX OF LIQUID

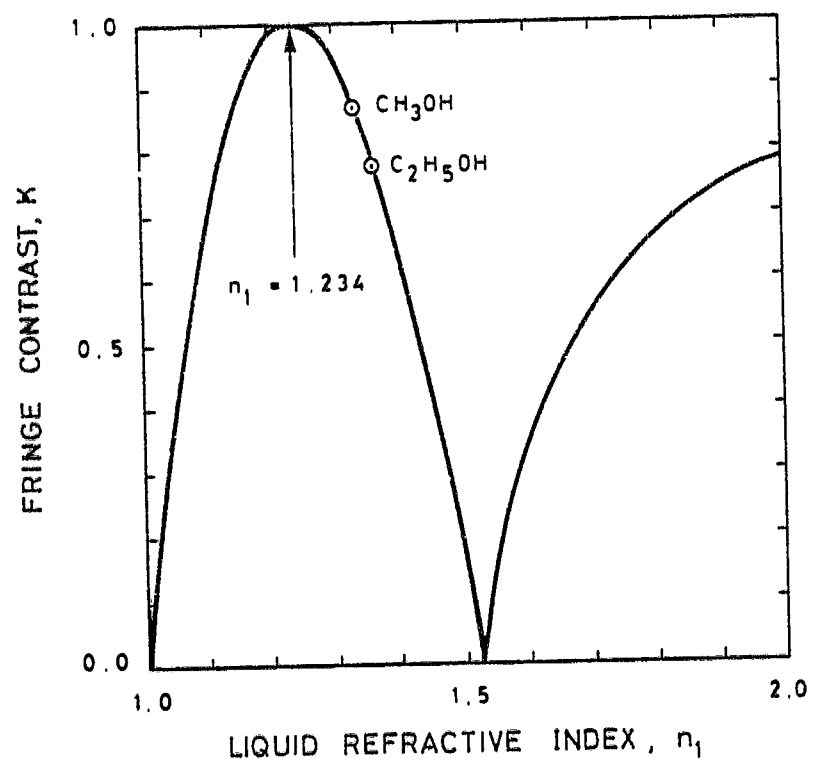


FIG. 6.3 FRINGE CONTRAST vs REFRACTIVE
INDEX OF LIQUID

The test liquids finally chosen were methanol and ethanol. Their fringe characteristics and contrast are shown in Table 6.1 and Figure 6.3.

TABLE 6.1

TEST LIQUIDS AND FRINGE CHARACTERISTICS

Liquid	Refr. index n_1	Reflectance, %		Contrast κ
		$\rho_1 = \rho_{\max}$	$\rho_2 = \rho_{\min}$	
methanol CH_3OH	1,33	4,29	0,56	0,87
ethanol $\text{C}_2\text{H}_5\text{OH}$	1,36	4,29	0,93	0,78

For these liquids the microlayer thicknesses are:

1. For bright fringes:

$$d = 0, \quad 1\lambda/2n_1, \quad 2\lambda/2n_1, \quad 3\lambda/2n_1 \dots \quad (6.10)$$

2. For dark fringes:

$$d = 1\lambda/4n_1, \quad 3\lambda/4n_1, \quad 5\lambda/4n_1 \dots \quad (6.11)$$

Thicknesses were thus given in steps of $\lambda/4n_1$ ($= 0,083 \mu\text{m}$ for methanol and $\lambda = 437 \text{ nm}$). This was the resolution of the measurements.

(b) Reflectance of bubble surroundings

The reflectance ρ_3 from the region of 'infinite' liquid surrounding the bubble (see Figure 6.1) is now evaluated. The optical system to be considered thus consists of one thin film of optical path difference $\bar{x} = \lambda$ (the SnO_2 film)

in contact with glass ($n = 1,523$) on one side and with methanol ($n_1 = 1,33$) on the other. The reflectance of such a system is given (see Vašíček [173] p 124) by

$$\rho_3 = \left[(n_1 - n_0) / (n_1 + n_0) \right]^2 \quad (6.12)$$

from which $\rho_3 = 0,0046$.

The reflectance of dark fringes (for methanol) is given in Table 6.1 as $\rho_2 = 0,0056$. The fringe pattern is thus surrounded by a dark field of reflectance indistinguishable from that of the dark fringes.

(c) Reflection from bottom of glass disc

The reflectance ρ_0 from the bottom of the glass disc (see Figure 6.1) is superimposed on the fringe pattern and reduces the overall contrast.

For bare glass ($n = 1,523$) in contact with air ($n_0 = 1,0$) the reflectance is given [173] as follows:

$$\rho_0 = \left[(n - n_0) / (n + n_0) \right]^2 \quad (6.13)$$

that is, $\rho_0 = 0,0429$ or 4,29 per cent.

The properties of the incident beam used in this study (see Section 6.3.3) were such that the glass disc was an optically 'thick' film (see [173] pp 60-63), that is, a film in which interference effects were no longer noticeable. Under such conditions the overall reflectance, ρ' , can be calculated in terms of the reflectance of the bottom of the disc, ρ_0 , and that of the top (that is, of the fringe pattern, ρ_1 or ρ_2) as follows (see [173] p 64):

$$\rho_1' = (\rho_0 + \rho_1 - 2\rho_0\rho_1) / (1 - \rho_0\rho_1) \quad (6.14)$$

$$\rho_2' = (\rho_0 + \rho_2 - 2\rho_0\rho_2) / (1 - \rho_0\rho_2) \quad (6.15)$$

For methanol one obtains $\rho_1' = 0,0823$ and $\rho_2' = 0,0481$. The overall contrast, κ' , is then

$$\kappa' = (\rho_1' - \rho_2') / \rho_1' = 0,42 \quad (6.16)$$

The reflection from the bottom of the disc thus reduces the fringe contrast from 0,87 to 0,42.

The suppression of the superimposed reflection was thus called for. The best means of achieving this, namely to grind and polish the bottom of the disc at a slight angle to the top surface [175], was not recognised at the time of experimentation. Instead, an anti-reflection coating was used.

For a coating or film of refractive index n_f and optical thickness $n_f d_f = \lambda/4$, in contact with the glass disc, the reflectance is ([173] p 127):

$$\rho_0 = \left[(n_f^2 - n n_0) / (n_f^2 + n n_0) \right]^2 \quad (6.17)$$

Thus for a perfect anti-reflection coating ($\rho_0 = 0$) it is necessary to satisfy the following condition:

$$n_f^2 - n n_0 = 0 \quad \text{or} \quad n_f = (n n_0)^{1/2} \quad (6.18)$$

In the present case this condition is: $n_f = 1,234$. Durable coatings of such refractive index are difficult to achieve in practice. The best substitute, a standard $\lambda/4$ magnesium fluoride coating was thus vacuum-evaporated onto the glass. The refractive index of such coatings is dependent on

evaporation conditions and on ageing, but is generally approximately 1,38 [153]. With this figure, equation 6.17 gives: $\rho_0 = 0,0123$ or 1,23 per cent (compare 4,29 per cent for untreated glass). For methanol the corresponding overall fringe contrast is 0,67.

This figure represents a good contrast, but is based on the assumption of ideal illumination. In practice, with imperfectly collimated and imperfectly monochromatic beams, the contrast is further reduced. It was required in the present study to minimise these effects, preferably to the point where 0,9 times the ideal contrast was maintained.

6.3 OPTICAL APPARATUS

6.3.1 General Requirements

Synchronised high-speed ciné records were required of the microlayer interference fringes and of the bubble profile view.

The interferometry beam, of sufficient collimation and monochromaticity, was required to impinge normally upon the bottom of the boiling surface and to illuminate its active portion, some 20 mm in diameter. The reflected beam was to be directed towards a ciné camera capable of speeds of the order of 1000 frames per second.

The synchronisation of the microlayer fringe records with the bubble profile records was to be achieved by photographing both bubble views side by side on the same frame of film. This required appropriate deflection of the two beams while maintaining their optical paths essentially identical in length.

6.3.2 Overall Description

A schematic view of the optical apparatus is given in Figure 6.4.

The interferometry beam was produced by a direct-current-operated mercury arc lamp, a collecting lens, a pinhole, a collimating lens and a monochromator filter. Mirror 1 deflected the beam to fall perpendicularly upon the bottom of the heating disc. A beam splitter directed the reflected beam towards a high-speed camera.

The bubble profiles were photographed by transmitted light from flood lamps.

The beam splitter and mirrors 2 and 3 were so arranged that the camera saw the bubble profile view and the microlayer view side by side with each occupying about half a film frame.

Components of the interferometry system and all mirrors were mounted on optical rails. The boiling tank was supported on legs fitted with levelling screws. Adjustable supports carried the camera and the flood lamps.

The assembled system is shown in Figure 6.5.

Various black screens and cloth drapes eliminated stray light and undesired reflections. The entire apparatus was housed in a room that could be darkened.

Detailed descriptions and analyses of the various system components are given below.

6.3.3 Interference Beam Light Source and Image at Pinhole

A light source was required having a high luminance and a

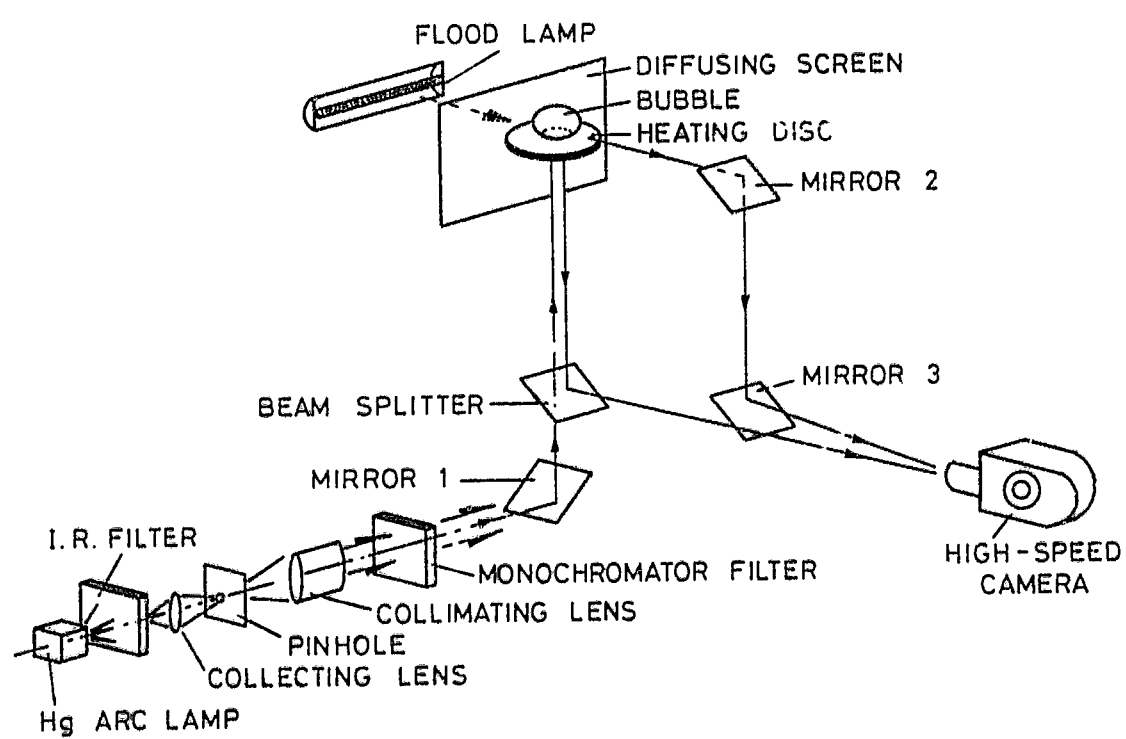


FIG.6.4 OVERALL OPTICAL SYSTEM

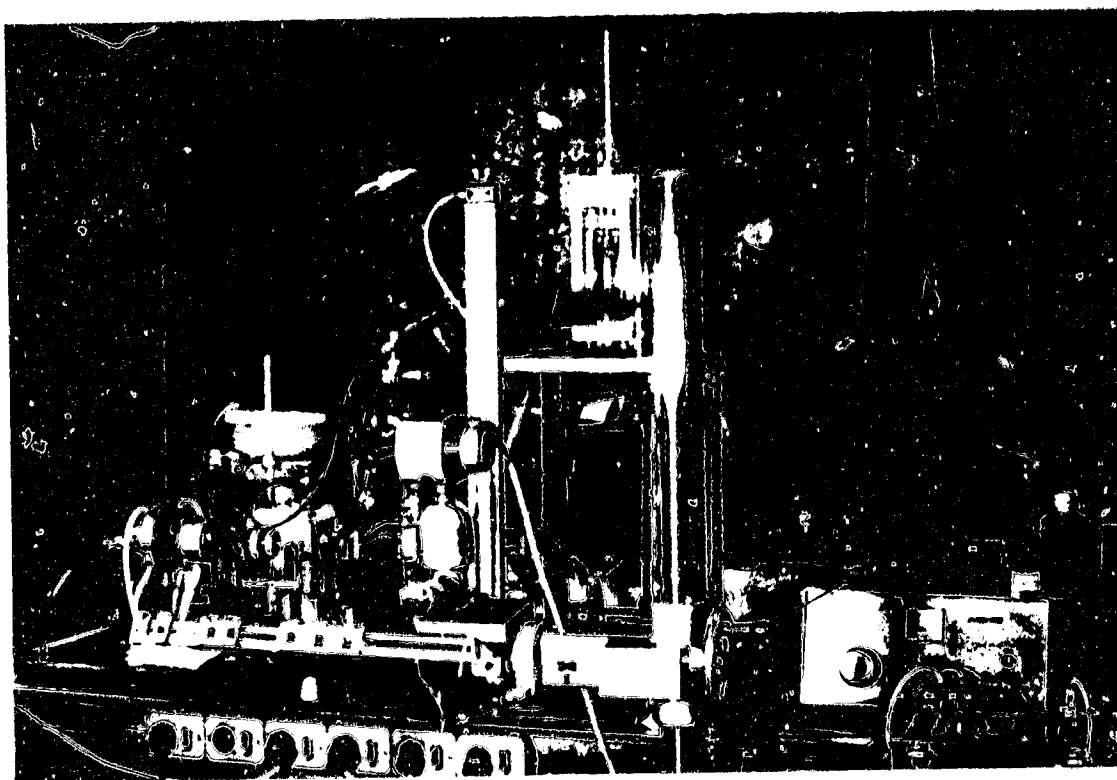


FIG. 6.5 GENERAL VIEW OF APPARATUS.

non-fluctuating output concentrated in narrow wavelength bands.

A direct-current-operated, 1 kW, high-pressure mercury arc lamp, the Philips SP 1000W, was used. The lamp was supplied complete with a compact metal housing carrying a built-in reflector, a borosilicate glass window and provisions for water cooling. The whole unit measured approximately 30 x 40 x 200 mm. Detailed descriptions are given in the manufacturer's literature [176, 177], and an early design is described by Bourne ([178] p166).

The lamp required a striking voltage of at least 700 V and operated normally at 500 V and 1,9 A. A suitable power supply was built by the Instrumentation Division of the Atomic Energy Board. Facilities were provided for operation at reduced output.

The arc had a luminance of $4,5 \times 10^8$ cd/m², that is, about one quarter that of the sun at zenith. The relative spectral energy distribution (including the effects of the glass window and the cooling water) is given in Figure 6.6; the high infra-red output is not shown. The peak at 437 nm (blue) was used in this study.

Cooling was provided by de-ionised water flowing at 5 litres per minute.

The collecting lens was a two-element condenser of focal length 50 mm and diameter 50 mm. It was placed midway between the arc and the pinhole, two focal lengths (100 mm) distant from each. Thus a real image of the arc, of unit magnification, was formed at the pinhole plate, with the pinhole itself becoming the effective light source.

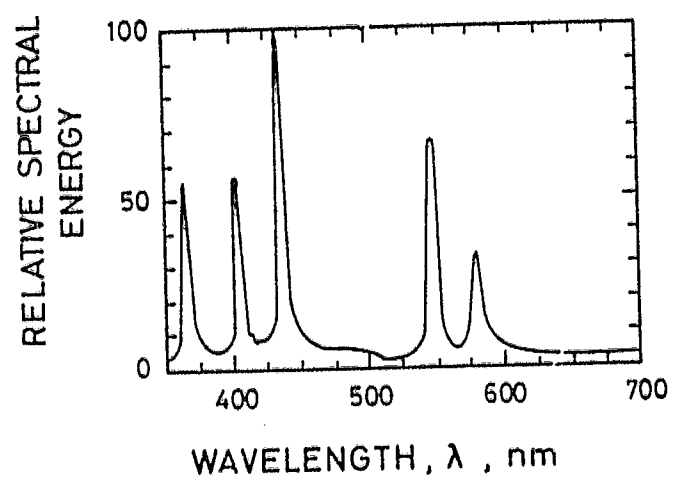


FIG. 6. 6 SPECTRAL ENERGY DISTRIBUTION OF
ARC LAMP
(Philips SP 1000 W)

6.3.4 Imperfect Collimation and Monochromaticity

Departure from perfect collimation and from perfect monochromaticity in the incident beam causes reduction in fringe contrast and ultimately fringe extinction [179]. It was required that this contrast reduction be kept as low as possible in the present study.

For purposes of analysis let the contrast for perfect illumination be unity. Now let κ_c be the contrast resulting from imperfect collimation only, and κ_m the contrast resulting from imperfect monochromaticity only. The requirements of the incident beam were tentatively stated as

$$\kappa_c \geq 0,95 \quad \text{and} \quad \kappa_m \geq 0,95 \quad (6.19)$$

Thus the combined reduced contrast, given by the product $\kappa_c \kappa_m$, was greater than or equal to 0.9.

(a) Collimation

It is assumed (1) that the collimating lens (in this study a 50 mm f/1,4 camera objective) was free from spherical aberration, and (2) that the pinhole was placed exactly at the principal focal point. Imperfect collimation (beam divergence) was then caused by the finite size of the pinhole only.

Let ψ be the angle subtended by the pinhole at the collimator; then for small pinholes

$$\psi \approx (\text{pinhole diameter})/(\text{collimator focal length}) \quad (6.20)$$

With d being the thickness of the thin film under study (in

the present case the microlayer) and n the film refractive index, let

$$\pi\psi^2 d/n\lambda = p \quad (6.21)$$

Fringe contrast is then given (see Françon [172] p 77) as follows:

$$\kappa_c = |2 \sin p|/(p + \sin p) \quad (6.22)$$

Equation 6.22 is plotted in Figure 6.7. As p increases from zero, the contrast diminishes; at $p = \pi$ the fringes disappear, then reappear with reversed and weaker contrast and again disappear at $p = 2\pi$ etc.

For the required contrast, $\kappa_c = 0,95$, equation 6.22 gives (very nearly) $p = \pi/4$ and thus from equation 6.21

$$\psi = 0,5 (n\lambda/d)^{1/2} \quad (6.23)$$

Equation 6.23 relates pinhole size with maximum film thickness for virtually undiminished contrast.

A pinhole of diameter 2,0 mm was used in the present study. Thus, from equation 6.20, the beam divergence, ψ , was 2/50 or 0,04 radians. With this value and a refractive index, n , of 1,33 (for methanol) equation 6.23 becomes

$$d = 208\lambda \quad (6.24)$$

or, in terms of optical thickness

$$nd = 277\lambda \quad (6.25)$$

Fringes of virtually undiminished contrast would thus have been visible up to a microlayer optical thickness of 277 wavelengths or 91 μm . Published estimates of microlayer

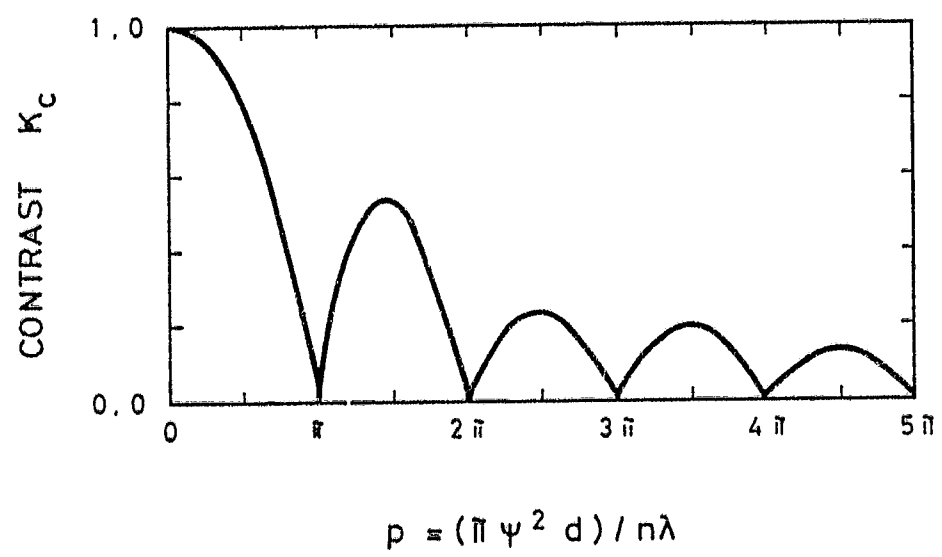


FIG. 6.7 EFFECT OF BEAM DIVERGENCE, ψ ,
ON CONTRAST.

thickness and the thicknesses measured in the present study were all considerably lower than this value. The collimation of the interferometry beam was thus more than adequate.

At this point one may justify the treatment (in Section 6.2.2) of the glass substrate as a thick film. For the substrate one has: $d = 3 \text{ mm}$, $n = 1,523$, and for the incident beam: $\lambda = 437 \text{ nm}$ and $\psi = 0,04 \text{ radians}$. Then from equation 6.21: $p = 22,7 = 7,23 \pi$. Extrapolation of Figure 6.7 shows that at $p = 7,5 \pi$ there occurs a maximum in the contrast; its value, from equation 6.22, is 0,09. Such a contrast is undetectable and hence the glass disc was a thick film.

(b) Monochromaticity

Interferometry beams, however 'monochromatic', have spectral distributions of finite bandwidths. The distributions can frequently be approximated by a Gaussian function [172, 180].

Let the wavelength corresponding to the peak of the distribution be λ_0 , let $\sigma = 1/\lambda$ and $\sigma_0 = 1/\lambda_0$; then in terms of the transformed variable $\sigma' = \sigma - \sigma_0$, the Gaussian distribution is given as

$$f(\sigma') = \exp - (\sigma'/a)^2 \quad (6.26)$$

see Figure 6.8. The distribution may be characterised by the σ' interval of $2a$, that is, the width for which $f(\sigma')$ has the value of $\exp(-1)$ or 0,37. The corresponding wavelength interval, $d\lambda$, is called the bandwidth. Hence

$$2a = 2\sigma' = 2(\lambda - \lambda_0) / \lambda \lambda_0 = d\lambda / \lambda \lambda_0$$

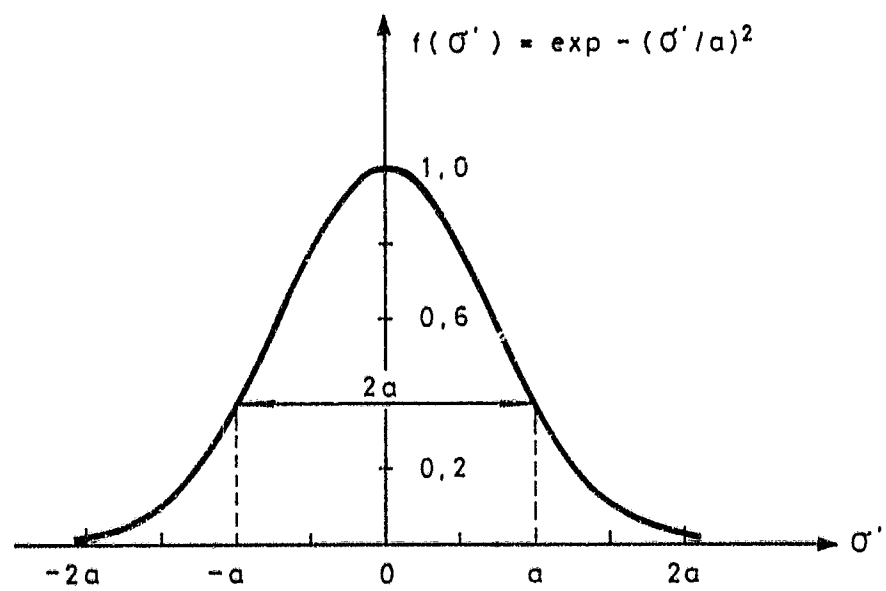


FIG. 6.8 GAUSSIAN DISTRIBUTION

or approximately

$$2a = d\lambda/\lambda_0^2 \quad (6.27)$$

For a beam of spectral distribution given by equation 6.26 the contrast κ_m is given in terms of 'a' and the optical path difference, $\bar{x}$, as follows (see Françon [172] p 97):

$$\kappa_m = \frac{2 \exp - (\pi a \bar{x})^2}{1 + \exp - (\pi a \bar{x})^2} \quad (6.28)$$

The equation is plotted in Figure 6.9.

Before applying equation 6.28 to the interferometry beam used in this study, the Gaussian approximation requires to be verified and the bandwidth to be determined. Figure 6.10 shows experimental values (measured on a Zeiss spectrophotometer) of transmittance, τ , versus wavelength for the interference monochromator filter used. (Details of the filter are given in Section 6.3.5). The dotted line in Figure 6.10 is the outline of the 437 nm emission band of the arc lamp, so transformed that its peak corresponds to a transmittance of 100 per cent. It is seen that the filter transmission alone determined the spectral distribution of the beam. The solid line in Figure 6.10 is a fitted Gaussian distribution having the same peak transmittance and bandwidth as the measured values. Since the filter had no secondary transmission bands the distribution of the interference beam was essentially Gaussian, and equation 6.28 for contrast was valid. The bandwidth $d\lambda$ was 18 nm.

For a contrast κ_m of 0,95 equation 6.28 gives

$$\pi a \bar{x} = 0,316 \quad \text{or approximately } 2a = 0,2/\bar{x}$$

which, substituting for bandwidth from equation 6.27, becomes

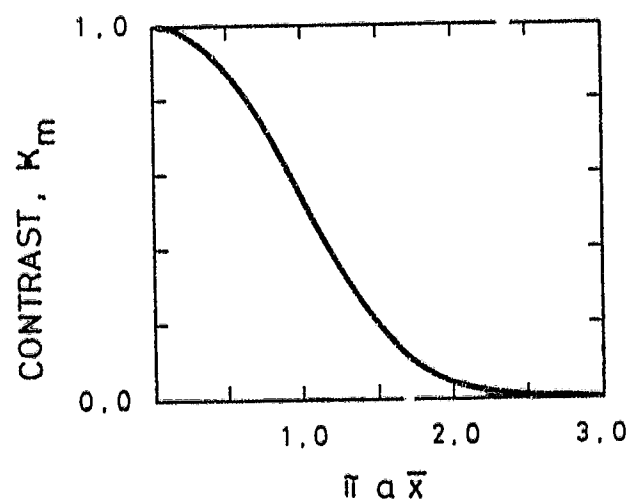


FIG. 6.9 EFFECT OF BANDWIDTH, $d\lambda$,
ON CONTRAST. $(2a = d\lambda / \lambda_0^2)$

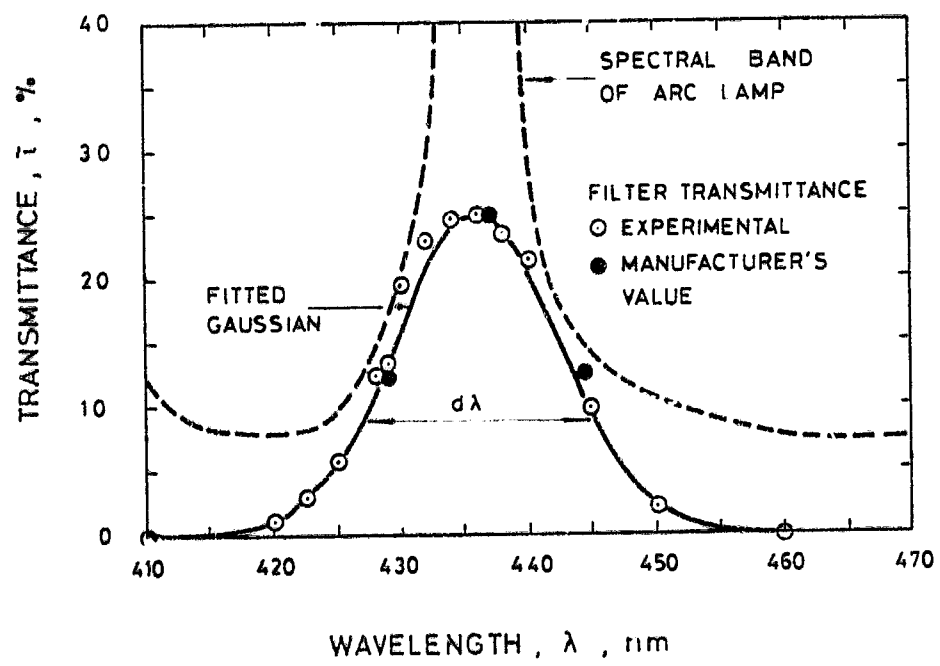


FIG. 6.10 SPECTRAL BAND OF ARC LAMP AND FILTER TRANSMITTANCE

$$d\lambda = 0,2\lambda_0^2 / \bar{x} \quad (6.29)$$

With a peak wavelength, λ_0 , of 437 nm (the manufacturer's value) and a bandwidth, $d\lambda$, of 18 nm equation 6.29 gives

$$\bar{x} = 4,85 \lambda_0 \quad (6.30)$$

Fringes of virtually undiminished contrast are thus predicted to be visible up to an optical path difference of some 5λ , that is, an optical thickness of 5λ or a microlayer thickness of approximately 1 μm . The microlayers actually measured in this study were thinner than 1 μm .

Thicker layers would, of course, still have been visible. Taking as the limiting contrast for fringe visibility Françon's value of 0,2 (see [172] p 97)* one obtains from equation 6.28

$$\pi a \bar{x} = 1,48$$

which, by a similar procedure to that used above, gives approximately

$$\bar{x} = 23\lambda_0 \quad (6.31)$$

Fringes are thus predicted to be visible up to a maximum optical thickness of $11,5\lambda$ or a maximum microlayer thickness of 3,8 μm . An alternative estimate of the maximum measurable thickness is presented in Appendix 9, and gives a value of 4,0 μm .

In some microlayer studies reviewed in Chapter 4, thicknesses

*Françon actually gives a value of 0,02. This is a misprint; his subsequent analysis shows the intended value to have been 0,2. To dispell all doubt, an alternative treatment of maximum visible thickness is given in Appendix 9.

of up to 50 μm were estimated. Such thicknesses would not have been measurable in the present study. Since the microlayers actually obtained were thinner than 1 μm , that is, thinner than the limit for virtually undiminished contrast, the monochromaticity of the beam was adequate. Had it been necessary to study thicker layers, a filter of narrower bandwidth would have had to be installed.

6.3.5 Interference Beam Filters and Mirrors

The interference beam's monochromator filter was of the double band interference type (manufacturer: Schott, Mainz; type DEPAL). The unit consisted of two identical metal-dielectric interference filters (the double filter arrangement leading to a steep-flanked transmission curve) and of additional coloured glass filters blocking the sideband transmissions. The filter's properties in the high transmission region are shown in Figure 6.10. The specified accuracy of the wavelength at peak transmission, 437 nm, was 0,5 per cent, or 2 nm. The peak transmittance was 25 per cent and the Gaussian bandwidth (occurring at 9 per cent transmittance) was 18 nm. At a transmittance of 0,025 per cent ($\tau_{\text{max}}/1000$) the width of the transmission curve was 60 nm. Secondary transmission bands were blocked to transmittances of less than 0,001 per cent. The filter was orientated at 90° to the collimated beam.

An infra-red absorption filter was placed between the arc lamp and the collecting lens. The filter was 3 mm thick, 50 mm in diameter, and mounted in a 10 mm thick water cell having borosilicate windows. The water, flowing at approximately 1 litre per minute, acted both as a coolant and as an auxiliary infra-red absorber (see [178] p 87). In the absence of the water cell the filter cracked within seconds of switching on the arc lamp.

The arc lamp's ultra-violet output was blocked by the

monochromator filter. In addition, a camera-objective's ultra-violet filter was fitted to the collimating lens.

Mirror 1 (see Figure 6.4), which deflected the interference beam towards the heating surface, was a high-quality front-surface mirror, 50 mm in diameter and mounted in an adjustable holder (manufacturer: Spindler and Hoyer, Göttingen; type 034 000).

The beam splitter (manufacturer: Balzer, Lichtenstein; type TF-MT-45) was of the multi-coating interference type. It was essentially free of absorption losses, and, under 45° incidence, transmitted 54 per cent and reflected 46 per cent of the incident beam. Its dimensions were 60 x 60 mm.

6.3.6 Bubble Profile: Illumination, Image Deflection and Distortion

A high-intensity floodlamp with the output concentrated on the bubble was required.

A 1000 W quartz-iodine lamp assembly (the Philips NK 17/00) was used. The unit consisted of a pencil-shaped filament tube and a housing carrying a cylindrical reflector (see Figure 6.4). The tube-to-reflector distance could be varied, thereby varying the spread of the output beam. Despite the rough 'hammer-finish' of the reflector quasi-parallel light was emitted.

With such illumination the optical system amounted roughly to a focused direct-shadowgraph system [181]. Thus, with n the liquid refractive index and co-ordinate axes x , y and z (z being in the direction of the incident beam), changes in illumination ($\Delta I/I$) roughly proportional to $\int (\partial^2 n / \partial x^2 + \partial^2 n / \partial y^2) dz$, appeared in the bubble field. With the variations in the liquid refractive index being due

to temperature variations, the temperature distribution around the bubble was thus visualised.

These shadowgraphs, while of interest and occasionally recorded, were generally undesirable, since the convection boundary layer on the heating surface obscured the bubble base. For quantitative runs then, the illumination was diffused by inserting a sheet of 'opal' Perspex between the flood lamp and the boiling tank. Light losses were high, and thus two auxiliary floodlamps (Philips Photolita SM, 250 W) were mounted above the main lamp.

Mirrors 2 and 3 (see Figure 6.4) deflected the bubble image such that it appeared alongside the microlayer view. The mirrors were of the front-surface type and 180 x 180 mm square (manufacturer: Optical Works, Ealing). The left-hand edge of mirror 3 was blackened to prevent stray reflections onto the microlayer image.

The cylindrical boiling tank was surrounded by a flat-sided water jacket to minimise lateral distortion of the bubble profile view. To check the distortion, a glass scale of known graduations was in some runs suspended centrally above the boiling surface and photographed through the water and the test liquid. No significant distortion could be detected.

6.3.7 High-speed Camera and Lens

A camera was required capable, firstly, of recording some 20 successive views of a bubble growth and departure cycle, and secondly, of providing a time base for these records.

Requirements were met by a Stålex 16 mm rotating-prism camera running at speeds of 500 to 1500 frames per second over a film length of 15 m. Several bubble cycles were generally recorded on each length of film. A miniature neon

lamp built into the camera housing and driven by mains frequency provided 100 timing marks per second along the edge of the film. A description of the Stålex, and a general discussion of rotating-prism cameras, is given by Saxe [182].

Camera speed was controlled by a unit specially manufactured for Stålex by John Hadland Photographic Instrumentation. The control unit caused the film to accelerate throughout the duration of a run, but more pronouncedly during the early stages. Thus the 15 m length of recording film was, on each film reel, preceded by a re-usable 15 m length of spliced-on leader. The acceleration during each bubble cycle was generally negligible.

The camera was fitted with a 105 mm f/2,8 Super Takumar Asahi Pentax lens. With this focal length and an image-to-object size ratio of 1/6, the object distance was a convenient 0,7 m.

6.3.8 Adjustment of System

(a) Lining-up

With the arc image focused on the pinhole, the first critical adjustment was the location of the collimator such that its back principal focus coincided with the pinhole. A plane mirror was placed in front of the collimator, approximately normal to the axis of the interferometry beam; the collimator was then moved until the reflected image of the pinhole was in focus on the pinhole plate.

The boiling tank was adjusted, using a spirit level, such that the heating surface was horizontal. Mirror 1 was then

set such that the collimated beam struck the heating surface vertically and thus normally. To this end the heating surface and lid of the boiling tank were removed, the beam was allowed to pass through the tank and to strike the ceiling, and verticality checked against a plumb-line.

The beam splitter was inserted and set at 45° to the horizontal.

The camera, approximately focused, was set with the film plane accurately vertical and (in the horizontal plane) normal to the beam reflected from the beam splitter. Normality could be checked from the concentricity of the outline of the heating disc and the bottom edge of its mounting plug. Simultaneously with these adjustments the image of the heating surface was brought into focus on one half of the film frame.

The bubble profile view, that is, the grazing view of the heating surface was now brought into focus on the remaining half of the frame. The optical path length of this view and of the interferometry view had to be maintained essentially identical. The necessary adjustment of mirrors 2 and 3 was achieved by laborious trial and error.

(b) Aperture and Depth of Field

From the point of view of depth of field the lowest possible lens aperture was desirable. The aperture (given by the focal length divided by the f-stop) is, however, the diameter of the largest parallel beam that can be passed by a lens. Thus with an active portion of the heating surface 15 mm in diameter, and with perfectly parallel illumination, an aperture of 15 mm would have been required. For a lens focal length of 105 mm this demanded a stop of $f/7$. With the slightly divergent beam, however, this adjustment

caused a sharp fall-off in illumination towards the edge of the field, and a larger aperture was necessary. A lens stop of $f/5$ was used in all tests.

With this setting and an image-to-object size ratio of $1/6$, the depth of field (conveniently obtainable from universal depth of field charts [183]), was 10 mm. Thus the difference in path length between the profile view and the microlayer view had to be maintained below 10 mm. This contributed to the difficulty in adjusting mirrors 2 and 3.

(c) Focusing

A focusing strip, consisting of a piece of film having a 'ground-glass' finish, was inserted into the camera film plane. Images formed on the strip were observed through a low-power telescope fitted to the camera. The nucleation pit on the heating surface acted as a focusing mark.

CHAPTER 7EXPERIMENTAL PROCEDURE AND DATA PROCESSING7.1 OUTLINE

The opening sections of this chapter deal with the experimental procedures which were applied to the boiling apparatus and to the optical system in the course of formal testing. The procedures involved the establishing of the desired boiling conditions the measurements of temperature, pressure and of power input to the heater, the readying of the optical system, and the gathering of the primary data, that is, the high-speed cine records of the microlayer interference pattern and of the bubble profile.

The remainder of the chapter describes the processing of the raw data. This consisted of the transcription of selected sequences of film frames onto enlarged prints, the establishing of the time base for the sequences, the analysis of each interference fringe pattern to give the complete microlayer geometry, the analysis of the bubble profiles to give bubble volumes and the establishing of certain overall bubble dimensions characterising the bubble shape.

The computation of the mean heat flux, the corrections of the temperature and pressure measurements and the determination of the saturation temperature and thus of subcooling are very briefly dealt with; details are available from Chapter 5 and Appendices 7, 8 and 13.

7.2 EXPERIMENTAL PROCEDURE7.2.1 Establishing of Boiling Conditions

At the start of a series of tests the boiling tank was

filled with the test liquid methanol or absolute ethanol of 'analytical reagent' (A.R.) grade. In the case of the hygroscopic ethanol, runs were conducted in rapid succession and the series completed in one day; liquid which had stood overnight was discarded. Fresh ethanol was supplied from a storage bottle fitted with a drying tube.

With the boiling tank sealed, the pressure was reduced to the ultimate vacuum of the water jet pump and the test liquid degassed by several minutes of boiling.

The desired test pressure was established via the needle valve leak and the Cartesian manostat pressure controller. The tank gauge pressure and the barometric pressure were noted.

Next, the required bulk liquid temperature was established. For saturated boiling runs this was achieved by adjusting the water jacket temperature (via the jacket heaters) to the vicinity of the test saturation temperature, and by boiling off the inner tank auxiliary heater. For subcooled runs the jacket temperature was set below the required test temperature to compensate for the bulk temperature rise during the subsequent boiling run.

With the liquid at steady temperature or, in the case of subcooled boiling, with the temperature slowly rising, the output recorded by the tank thermocouples were checked for agreement with the tank thermometer reading.

The power input to the heating surface was now raised to the required level and the wattage noted. The bubble behaviour was checked with a stroboscope. When found satisfactory and, in the case of subcooled boiling, when the required bulk temperature had been reached, the system was ready for the gathering of ciné records.

7.2.2 Final Preparations on Optical System

During the procedures outlined above the optical apparatus was in the standby condition; that is, the entire system was aligned, the arc lamp was at half power, one bubble profile flood lamp was switched on, the cooling water circuits running and the camera focused. The final preparations for the ciné run consisted of a last check on the camera focus and aperture, the removal of the focussing strip, the loading of the camera with a 30 m reel of film, the setting of the framing speed, the raising of the arc lamp to full power, the switching on of all flood lamps and the extinguishing of other lights.

7.2.3 Execution of Run

The camera was now fired. The film accelerated sharply over the 15 m length of leader and on the remaining unexposed length recorded the microlayer interference view, the bubble profile view and the timing marks.

The illumination was reduced to standby levels, the thermocouple recorder chart marked to indicate the point of ciné observations and the tank gauge pressure and the heater power input checked again.

A thermocouple was now attached to the lower surface of the heating disc in the manner described in Section 5.4.4. The output of this thermocouple, as shown in Appendix 8, gave an estimate of T_w , the mean temperature of the (upper) boiling surface. In subcooled boiling runs the power input to the heater was interrupted while the thermocouple was being fitted; this prevented the liquid bulk temperature, and thus T_w , from rising significantly above the levels which had been attained at the time of the ciné observations. The heater input was then raised to the previous reading and the thermocouple output recorded after quasi-steady conditions had again been reached.

The film take-up spool was removed, the leader detached and the remaining film immediately developed. Details of the photographic materials and of processing are given in Appendix 10.

7.3 DATA PROCESSING

7.3.1 Transcription of Optical Data

The developed negative film strip was examined via a projector permitting single frame and continuous, forward and reverse operation. Bubble cycles selected for study were transcribed onto prints to permit desk-top analysis.

First the entire cycle was printed onto one sheet. This permitted analysis of the timing marks and further gave the overall bubble behaviour at a glance. Figure 7.1 shows a sample print.

Next, each frame within the bubble cycle was printed on a separate sheet. The linear enlargement on the prints was approximately 30 with respect to the negative, or 5 with respect to life-size. Details of enlargement and print processing are given in Appendix 11.

Figure 7.2 shows a portion of a successful film on isolated bubble boiling. The two bright lines on either side of the interferogram are the indium heater contacts; reflections from the main body of the bubble caused the bright arc above the rings. The profile view shows the attached bubble, current leads and the lower thermocouple. Only one such print was made up for each bubble cycle; it allowed the linear enlargement to be determined from the known distance between the contacts and, in cases where a glass scale was fitted above the heater, the lateral distortion of the profile view to be checked. On the remaining standard prints the contacts and current leads were blanked off, and the interferogram and the profile

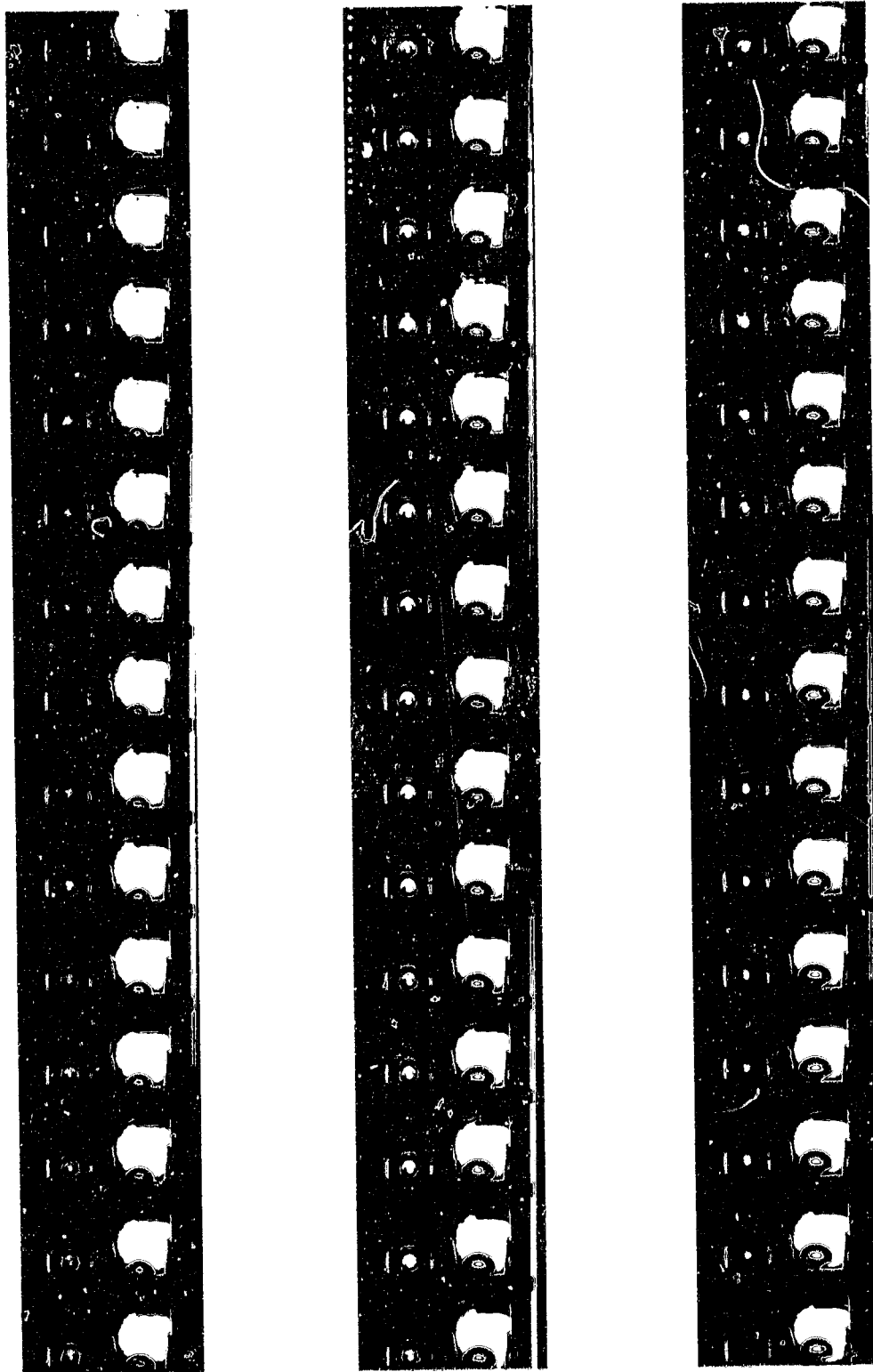


FIG. 7.1 OVERALL CINÉ RECORDS, MAIN CYCLE, RUN M10.

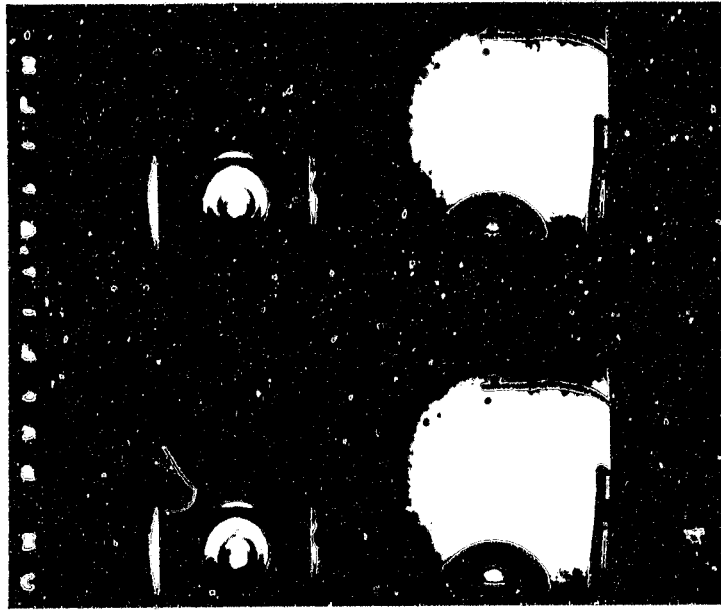


FIG. 7. 2 ENLARGED PORTION OF FILM, RUN M10.

view shifted closer together to occupy a convenient 20 x 25 cm sheet.

A bubble cycle generally occupied 20 to 50 frames. This number of standard prints, together with the prints for the determination of enlargement and of time base, constituted one bubble cycle file for analysis.

7.3.2 Time Base and Print Magnification

The time base for each bubble cycle was determined from the camera timing marks via a print of the type shown in Figure 7.1. The distance on the print between the start of successive timing marks was measured. With the timing lamp driven by mains frequency of 50 Hz, this length corresponded to a time interval of 0,01 s. The distance on the print between successive images of some fixed item (for example, the base of the heating surface) gave the length corresponding to one frame. The ratio of these lengths gave the number of frames per 0,01 s and thus the time interval between frames.

Several timing marks appeared in each bubble cycle, and several corresponding values of the framing rate were obtained. These showed no evidence of film acceleration during the cycle, but differed randomly by some 2 per cent. The short-term accuracy of the mains frequency was better than ± 1 per cent. The overall accuracy of the time interval between frames may thus be taken as ± 3 per cent.

The linear enlargement on the standard prints was determined from the distance between the images of the indium heater contacts, see Figure 7.2. The life-size distance between contacts (20,15 mm) was known to an accuracy of 0,01 mm from measurements on a profile projector. Repeated determinations of a print enlargement gave random errors of $\pm 0,5$ per cent. Measurements on the glass scale above the heater showed no evidence of lateral distortion of the bubble profile view. In view of possible uneven

shrinkage of the photographic paper the accuracy of the linear enlargement was taken as ± 1 per cent.

7.3.3 Microlayer Geometry: Analysis of Interference Patterns

Figure 7.3 shows an example of a microlayer interference pattern at the standard magnification used in analysis. The central light spot corresponds to a microlayer thickness of zero (that is, to the dry spot); subsequent light rings correspond to thicknesses of $\lambda/2n_1$, $2\lambda/2n_1$, $3\lambda/2n_1$...; dark rings to $\lambda/4n_1$, $3\lambda/4n_1$, $5\lambda/4n_1$ Microlayer thickness was thus given in increments of $\lambda/4n_1$, where λ is the wavelength of the interferometer beam (0,437 μm) and n_1 is the liquid refractive index.

Evaluation of the thickness increments required refractive indices at a wavelength of 0,437 μm and at appropriate microlayer temperatures. This problem is dealt with in detail in Appendix 12. It is shown, firstly, that the refractive indices may be suitably evaluated at the mean temperature $(T_w + T_{\text{sat}})/2$ for each run and, secondly, that for an overall error in microlayer thickness of ± 1 per cent, the refractive index for all methanol runs may be taken as 1,32 and that for all ethanol runs as 1,35. The thickness increments throughout this work are thus:

$$\text{methanol: } \lambda/4n_1 = 0,083 \mu\text{m}$$

$$\text{ethanol: } \lambda/4n_1 = 0,081 \mu\text{m}$$

The actual thicknesses corresponding to the various fringes are shown in Table 7.1.

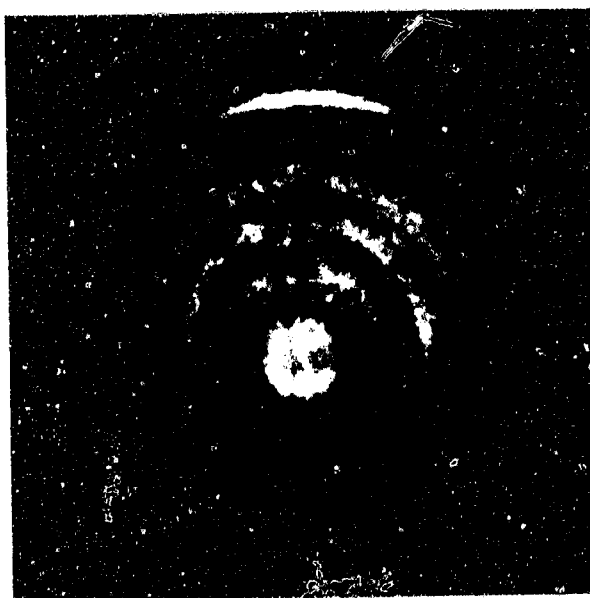


FIG. 7, 3 INTERFERENCE PATTERN, FRAME 837, RUN M10.
(At standard magnification for analysis)

TABLE 7.1

MICROLAYER THICKNESS versus FRINGE ORDER

Fringe order and reflectance	Thickness, d, μm	
	methanol	ethanol
1st maximum (spot)	0	0
1st minimum	0,083	0,081
2nd maximum	0,166	0,162
2nd minimum	0,249	0,243
3rd maximum	0,332	0,324
3rd minimum	0,415	0,405
4th maximum	0,498	0,486
4th minimum	0,581	0,567
5th maximum	0,664	0,648

With the fringes now representing known microlayer thicknesses, the diameter of each ring-shaped fringe was required. Two measurements in mutually perpendicular directions were taken off the enlarged print for each ring. Their mean, divided by twice the linear enlargement, gave the mean bubble base radius at which the microlayer thickness under consideration occurred. On non-circular, approximately elliptical, fringes the major and minor axes were measured. Fringe eccentricity (e.g. Figure 7.3) was ignored. For maxima of reflectance the measurements were taken on the brightest part of the fringe, for minima on the darkest part, or, when this was indeterminate, on a region half-way between two bright fringes.

In some patterns, for example in Figure 7.3, the outermost fringe could be seen to extend a little beyond its brightest

region. This outermost detectable edge was interpreted as the periphery of the microlayer and thus of the bubble base. Since the reflectance of the 'infinite' liquid surrounding the bubble was equal to that of the minima of reflectance (see Section 6.2.2) a dark fringe could have existed beyond this edge. Despite this potential error the edge measurements were taken, whenever possible, to provide at least a rough indication of the bubble base periphery. When the microlayer had dried out or lifted off, the periphery of the dry patch became the periphery of the bubble base.

Interference patterns of the type shown in Figure 7.3 presented few problems in interpretation. In some cases, however, the presence or absence of a fringe could not be ascertained without reference to previous or subsequent frames. In frame 830 in Figure 7.4, for example, two minima and a maximum could possibly be said to exist between the bright patch and the outermost fringe. This impression persists faintly in frame 832, but is absent in frame 834 and subsequent frames (see frame 837, Figure 7.3). In this and similar cases the questionable fringes were judged not to be real.

- The example of Figure 7.4 referred to the early stages of a bubble cycle. In the later stages fringe eccentricity, fringe merging and dry patch fragmentation complicated the analysis. In Figure 7.5, showing frame 860 from the same run as discussed above, the first (inner) maximum is seen as a crescent, apparently interrupted by a small dry spot, above the main dry patch; in the lower part of the pattern this maximum has merged with the dry patch. In the right-hand part of the pattern the next two maxima are separate, on the left they have merged. This interpretation could be verified by reference to previous frames; frame 837 in Figure 7.3 in fact already displays signs of decentering and a tendency toward fringe merging. The adopted method of analysing fringe dimensions, when applied to this and similar cases,



FIG. 7.4 INTERFERENCE PATTERNS
FRAMES 830, 832, 834; RUN M10.

230.



FIG. 7.5 INTERFERENCE PATTERN, FRAME 860, RUN M10.

led to a reported microlayer geometry which does not reflect the asymmetry actually present.

Similarly, non-circular or irregular dry patches were reported as symmetrical. In one extreme case, in the latter part of run M.13, the dry patch assumed such irregular shapes that no major and minor axes could be defined. The patch outline was thus traced onto graph paper, the area determined by the counting of squares, and the radius of the circle of equal area reported.

For reasonably circular fringes the diameter measurements (on the prints) had an accuracy of approximately ± 2 mm. The reported bubble base radii are thus accurate to some ± 10 per cent towards the inner edge of the microlayer, and (for large bubbles) to ± 5 per cent towards the outer edge. The accuracy of the microlayer thickness determinations was limited by the accuracy of the refractive indices (± 1 per cent) and by the accuracy of the wavelength of the incident beam ($\pm 0,5$ per cent, see Section 6.3.5). Thus, barring errors in the interpretation of fringe order, the accuracy of the reported thicknesses was $\pm 1,5$ per cent.

7.3.4 Bubble Volume

Bubble volume was determined by considering each bubble as composed of a spherical cap segment and a number of cylindrical discs. First, each bubble profile print was traced onto transparent graph paper. A circular arc was fitted by compass to the bubble cap and the remaining profile divided into rectangular strips of constant height as shown in Figure 7.6. Rotation of each area about its vertical axis of symmetry gave the corresponding (enlarged) volume increment. (On 'skew' bubbles the axes of symmetry did not coincide). The total life-size bubble volume was thus given as follows:

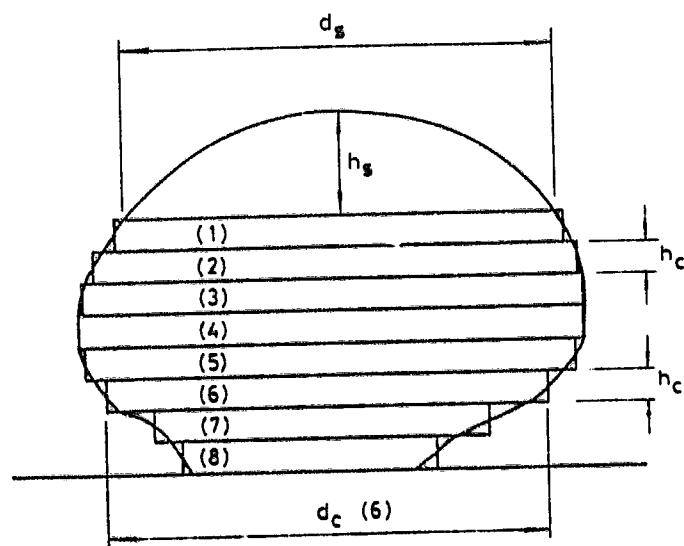


FIG 7.6 INCREMENTS OF BUBBLE VOLUME

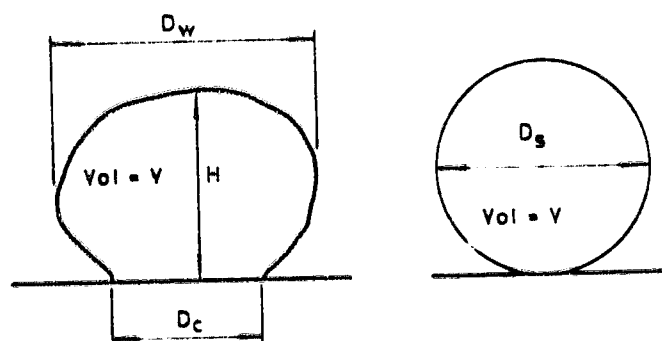


FIG. 7.7 DIMENSIONS CHARACTERISING BUBBLE SHAPE

$$V = \left[(\pi/6)h_s \{ (3/4)d_s^2 + h_s^2 \} + (\pi/4)h_c E d_c^2 \right] / E^3 \quad (7.1)$$

where E is the linear print enlargement and the remaining symbols are defined in Figure 7.6.

For the first few frames in each bubble cycle the bubble volume could be approximated by the spherical segment only or by the segment and one cylindrical disc. In the later stages, with large bubbles, 5 to 10 discs became necessary. A disc thickness (h_c in Figure 7.6) of 5 mm was found satisfactory on all enlarged profiles.

The accuracy of the reported bubble volumes cannot be assessed with much reliability. However, with measurement errors of 1 to 2 mm in the dimensions h_s , d_s , h_c and d_c , a graph paper accuracy of ± 0.1 per cent and an error of ± 1 per cent in the linear enlargement, the estimated error in the sum of the volume increments is, for a large bubble, approximately ± 6 per cent. The coarse fit of the increments and the local asymmetry might have increased the overall volume error to ± 12 per cent for large bubbles. For small bubbles the overall error might have been twice as large. In view of the effort involved in the incremental volume determinations this would appear a poor achievement. However, the accuracy is probably pessimistically assessed, and in any event, is a vast improvement on that achievable by the more common practice of approximating the bubble as a spheroid.

7.3.5 Dimensions Characterising Bubble Shape

In addition to the microlayer geometry and the bubble volume, some overall dimensions characterising the bubble shape were required. The following dimensions, shown in Figure 7.7, were considered suitable: the bubble height, H, the diameter at maximum bubble width, D_w , the base or contact area diameter, D_c , and the equivalent spherical diameter, D_s .

Bubbles having height, maximum diameter and spherical diameter all of similar magnitude, but small base diameter, could be classified as spherical. Bubbles having maximum diameter and base diameter of similar magnitude, but a height half that value, could be classified as hemispherical. Intermediate cases were termed oblate.

Bubble height and maximum diameter were obtained for each frame by measurement off the enlarged profile tracings and division by the linear enlargement. Bubble base diameter was determined as outlined in Section 7.3.3. The equivalent spherical diameter was calculated from bubble volume by equation 7.2 below.

$$D_s = (6V/\pi)^{1/3} \quad (7.2)$$

7.3.6 Heat Transfer Parameters

The remaining data processing consisted of the determination of the heat transfer parameters for each boiling run.

The mean heat flux was calculated from the electrical input to the heater and from various corrections as outlined in Chapter 5 and in Appendix 7. The mean heater surface temperature and the liquid bulk temperature were measured as outlined in Chapter 5 and Appendix 8; their difference gave the temperature driving force $(T_w - T_b)$. The bulk saturation temperature, T_{sat} , was calculated from the measured system pressure via a re-arranged Clausius-Clapeyron equation fitted to published vapour pressure data; details are given in Appendix 13. From these followed the bulk subcooling, $\Delta T_{sub} = (T_{sat} - T_b)$, and the saturation driving force, $\Delta T_{sat} = (T_w - T_{sat})$.

CHAPTER 8EXPERIMENTAL RESULTS8.1 AIMS AND OUTLINE

The aims of this chapter are: (1) to present, for various conditions of normal boiling, the detailed history of microlayer geometry and of overall bubble behaviour throughout one cycle, (2) to estimate the contribution of microlayer evaporation to the total bubble growth, and (3) to test the theoretical predictions of microlayer thickness against experimental values.

To this end seven isolated-bubble cycles, each from a different boiling run, were analysed exhaustively in the manner described in the last chapter. The cycles were selected, firstly, to cover a variety of boiling conditions, and secondly, to minimise errors and difficulties in interpretation, that is, to offer particularly clear fringe patterns.

Section 8.2 gives the boiling conditions for the seven tests. The system parameters (heat fluxes, temperatures and pressures), as well as the bubble frequencies and waiting times are listed and discussed. The overall boiling pattern is described. Details are given of the little-known phenomena of involution of departed bubbles and of the subsequent formation of secondary vapour columns and of bursts of secondary bubbles.

Section 8.3 summarises the accuracy of all data to be reported.

Section 8.4 presents the actual results. Histories of microlayer geometry, of bubble volume and of the dimensions

characterising bubble shape are all given in graphical form. The corresponding numerical data are given in Appendix 14.

Section 8.5 provides estimates of the contribution of microlayer evaporation to the maximum bubble volume. Since evaporating microlayers are found to be continuously replenished by in-flowing liquid, the apparent or visible microlayer contribution only is estimated.

Section 8.6 compares experimental and predicted microlayer thicknesses.

8.2 TEST CONDITIONS

8.2.1 Heat Flux, Temperature and Pressure

The heat flux, temperature and pressure conditions for the seven tests are shown in Table 8.1.

TABLE 8.1

TEST CONDITIONS

Run No. *	P kN/m ²	q/A kW/m ²	ΔT ($T_w - T_b$) K	ΔT_{sat} ($T_w - T_{sat}$) K	ΔT_{sub} ($T_{sat} - T_b$) K	T_w °C	T_{sat} °C	T_b °C
M10	25,6	70,0	47,0	36,8	10,2	72,0	35,2	25,0
M13	24,0	61,8	44,0	31,1	9,9	68,0	33,9	24,0
M14	33,3	61,9	28,5	28,1	0,4 (sat)	68,5	40,4	40,0
M19	33,3	61,9	52,3	32,1	20,2	72,5	40,4	20,2
M20	42,6	61,9	52,0	31,5	0,5 (sat)	77,0	45,5	45,0
E 8	25,4	30,4	34,0	31,1	2,6	79,0	47,6	45,0
E12	33,3	65,7	31,5	31,8	-0,3 (sat)	85,0	53,2	53,5

*M denotes test liquid methanol

E denotes test liquid ethanol

The pressure range of 24,0 to 42,6 kN/m² was covered. At lower pressures bubbles grew larger than the central portion of the heater; at higher pressures, particularly with subcooling, bubbles became too small for accurate analysis. The range of test heat fluxes was limited by incipience of boiling at the low-flux end and by the incipience of lateral bubble interaction at the high-flux end.

The first methanol run, Run M10, was conducted under conditions particularly favourable for analysis, that is, at slight subcooling which stabilised the main bubble and suppressed extraneous bubbles, and at low pressure which gave large bubbles. Run M13 was essentially a repetition of Run M10. Runs M14 and M19 were conducted at identical pressure and heat flux, but with the bulk saturated in M14 and highly subcooled in M19. Run M20 was a repetition of the saturation run M14, but at higher pressure. The ethanol run E12 was conducted under essentially the same conditions as the methanol run M14. Run E8 was conducted at low pressure, very low heat flux and slight subcooling. In this run the bubble did not, as in all other runs, appear from the central artificial nucleation site, but from a natural site nearer the edge of the heating film.

8.2.2 Overall Bubble Behaviour

Among the seven runs shown in Table 8.1, Run E8 only displayed the normal succession of bubble growth, departure, waiting time and formation of the next bubble. In all remaining runs the departure of the main (primary) bubble was followed by the emission, from the same nucleation site, of a burst of smaller (secondary) bubbles all of zero waiting time. The first secondary bubble was in most cases fairly large (up to half the volume of the primary bubble), became elongated into a vapour column, and penetrated the departed primary bubble. Further secondary bubbles tended (with occasional irregularities) to progressively smaller sizes, the last bubble

generally being of negligible volume compared with the primary bubble. After the elapse of a conventional waiting time this multiple bubble cycle was repeated.

Two waiting times may be defined for such multiple cycles: the primary waiting time, $t_{w(1)}$, that is, the time interval between the departure of a primary bubble and the nucleation of the next, and the secondary waiting time, $t_{w(2)}$, that is, the interval between the departure of the last secondary bubble and the nucleation of the next primary bubble. These times, together with the primary cycle period, $1/f$, are sketched in Figure 8.1 and are listed for all runs in Table 8.2.

TABLE 8.2

PERIOD AND WAITING TIME FOR MAIN CYCLE STUDIED
AND FOR PRECEDING AND FOLLOWING CYCLES*

Run	Cycle studied			Preceding cycle			Following cycle		
	$1/f$ ms	$t_{w(1)}$ ms	$t_{w(2)}$ ms	$1/f$ ms	$t_{w(1)}$ ms	$t_{w(2)}$ ms	$1/f$ ms	$t_{w(1)}$ ms	$t_{w(2)}$ ms
M10	282	256	120	265	229	74	248	204	124
M13	387	349	260	464	421	339	578	536	428
M14	391	348	325	85	52	29	353	315	244
M19	661	641	617	548	533	505	**	**	**
M20	635	605	565	629	596	546	146	111	104
E 8	702	669	669	760	729	729	**	**	**
E12	109	82	52	76	49	36	81	44	42

* Data are approximate only; based on frame-counts and on time base established for growth period of main cycle.

** Cycle beyond end of ciné film

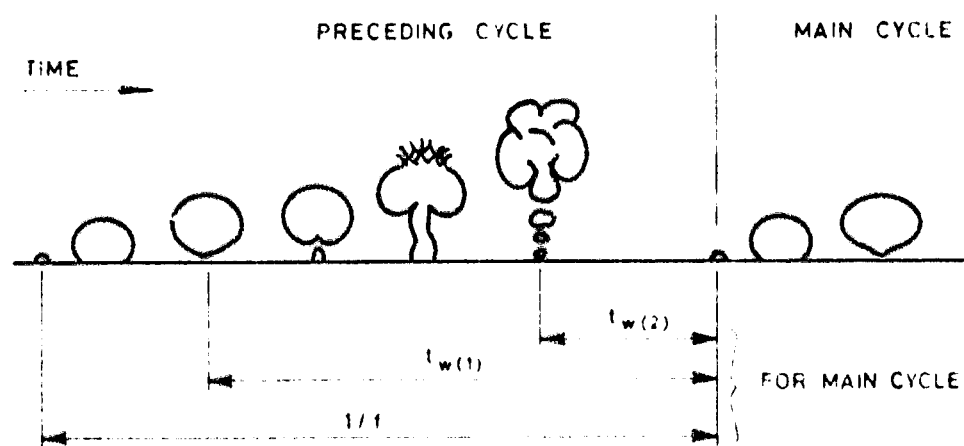


FIG. 8.1 WAITING TIMES AND FREQUENCY FOR MULTIPLE BUBBLE CYCLE

The first three data columns in Table 8.2 refer to the cycles studied; the remainder give data for the cycles preceding and following the main cycle. From the first column in each group it is seen that the primary bubble frequencies were extremely low, of the order of 1 to 10 s⁻¹. Comparison of the differences $\{1/f - t_{w(1)}\}$ and $\{t_{w(1)} - t_{w(2)}\}$ shows that the interval occupied by the emission of secondary bubbles was generally larger than the growth period of the primary bubbles. This was particularly the case in Runs M10 and M13. In these runs a large number of secondary bubbles, from 10 to 14 per cycle, were emitted, with the first and, in the case of Run M10, also the third, secondary bubble having a size comparable to that of the primary bubble. Further, the comparison of data along the rows of Table 8.2 reveals irregularities, severe in Runs M14 and M20, in the periods of successive multiple cycles.

Profiles versus time of all main bubbles studied are shown at life-size scale in Figure 8.2. The general shape is oblate.

8.2.3 Bubble Involution and the Formation of Secondary Bubbles

In addition to the formation of secondary vapour columns and bursts of secondary bubbles, a further peculiarity of bubble behaviour, namely the involution of the lower surface of departed primary bubbles, was observed throughout this study. Figure 8.3 shows this involution for the case of the single-bubble run, Run E8.

Such involution appears to be caused by the liquid jet formed in the wake of the departed primary bubble (see Section 2.4.1). Surface tension forces (tending to keep the bubble spherical) are low for the large bubble, thus permitting the wake-jet to deform and to penetrate the lower (and in some cases upper) bubble surface.

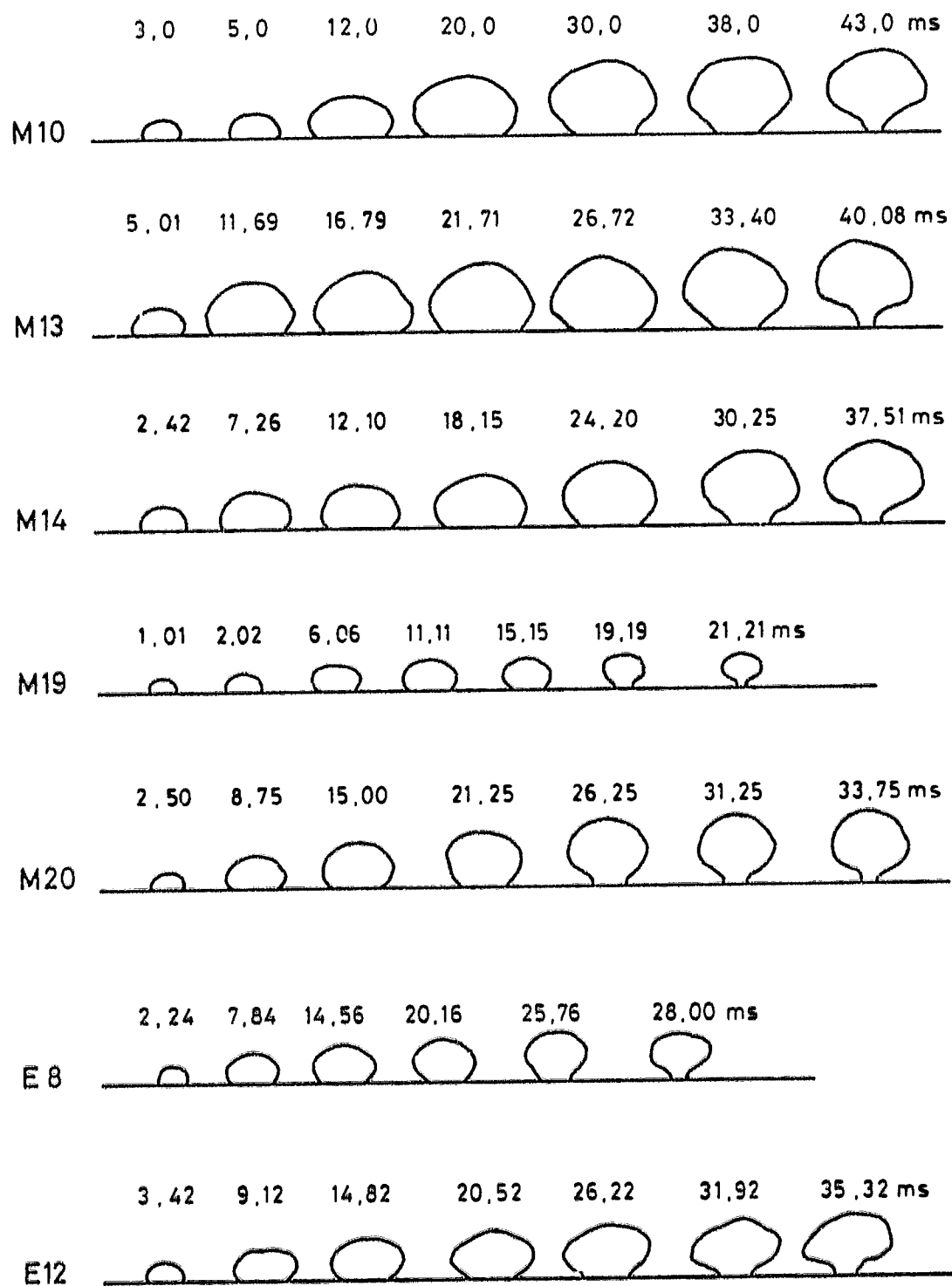


FIG. 8.2 BUBBLE PROFILES FOR MAIN CYCLES OF ALL RUNS
(Scale: life size)

242.



FIG. 8.3 BUBBLE INVOLUTION, RUN E8.
Interval between frames 4,48 ms

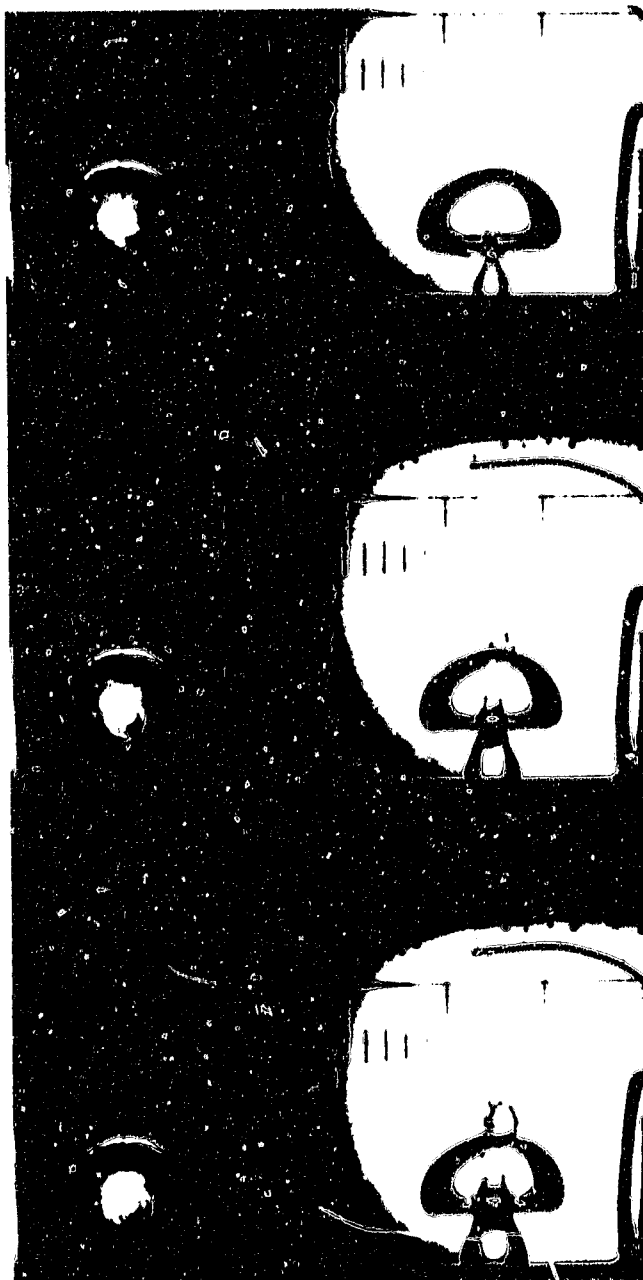


FIG. 8.4 VAPOUR COLUMN PENETRATING
PRIMARY BUBBLE. RUN M13.
Interval between frames 1,67 ms.

In the case where a fairly large secondary bubble is formed, it is elongated by the liquid jet into a vapour column which in turn penetrates the departed primary bubble. This case is illustrated in Figure 8.4. The jet furthermore causes the premature departure of the vapour column. Subsequent secondary bubbles are similarly torn off prematurely.

The formation of secondary bubbles may be explained by the prolonged presence of large dry patches beneath the attached primary bubbles. (The presence of such dry patches will be demonstrated in Section 8.4). The heater surface temperature, previously lowered by microlayer evaporation, recovers over the dry patches and, upon departure of the main bubble, causes immediate nucleation and rapid growth of a secondary bubble. This, in turn, may form a smaller dry patch of lower surface temperature, which may allow the formation of the next secondary bubble. This will grow more slowly than its predecessor and will be torn off at smaller size. By similar arguments subsequent secondary bubbles should become progressively smaller.

Bubble involution and the formation of secondary vapour columns (without further secondary bubbles) have been noted and explained in terms roughly similar to the above by van Stralen et al in 1975 [142]. Earlier mention of these phenomena and a photograph showing penetration of the primary bubble by a vapour column were given in 1969 in the publication of preliminary results of the present study [152].

8.3 ACCURACY OF REPORTED DATA

Before presenting the main results it would seem appropriate to collect together and to consolidate the estimates of data accuracy given at various points throughout this work. The resulting summary is presented in Table 8.3.

TABLE 8.3

ACCURACY OF EXPERIMENTAL DATA

QUANTITY MEASURED		ACCURACY			REFERENCE
		absolute	%	REMARKS	
Mean heat flux	q/A		7	over central heater area	Appendix 7
Mean surface temperature	T_w	5 °C	7	percentage refers to worst case	Appendix 8
Liquid bulk temperature	T_b	0,5 °C	2	percentage refers to worst case	
Liquid saturation temperature	T_{sat}	1 °C	3	percentage refers to worst case	Appendix 15
Overall temperature difference ($T_w - T_b$)	ΔT	5,5 K	9	percentage refers to worst case	
Wall superheat ($T_w - T_{sat}$)	ΔT_{sat}	6 K	10	percentage refers to worst case	
Subcooling ($T_{sat} - T_b$)	ΔT_{sub}	1,5 K	3	percentage refers to worst case	
Pressure	P	0,1 kN/m ²	0,1	percentage refers to worst case	
Time between frames	Δt		3		Section 5.4.2
Bubble base radius (locus of fringes)	r		5 10	for largest r for smallest r	Section 7.3.3
Microlayer thickness	d		1,5	barring errors of fringe interpretation	Section 7.3.3
Bubble volume	V		12-24	depending on size; probably better	
Bubble dimensions	D_w, H D_c		5		

8.4 RESULTS

A large body of results is reported in this section. Despite the variations in test conditions and the two test liquids covered, a very similar microlayer behaviour is demonstrated in all cases. Since this behaviour is generally unrecognised and since no other data on the complete history of microlayer geometry are available, the detailed presentation of the somewhat repetitive results is considered justified. The data are shown in abbreviated graphical form, that is, with a few exceptions, plotted for every second time-step only. A full listing of all data points is provided in Appendix 14.

8.4.1 Methanol Runs M10 and M13

Figure 8.5 shows the detailed microlayer history for Run M10. The radius from the centre of the bubble base to the outer edge of the microlayer, and base radii at which fixed microlayer thicknesses occurred, are plotted against time. Such plotting, although less graphically descriptive than that of successive microlayer profiles, was the only method of accommodating the entire history of the microlayer geometry on one figure. A positive slope for the locus of a particular thickness (r versus t) illustrates microlayer thinning, or, for the case of $d = 0$, dry patch growth; a negative slope illustrates microlayer thickening.

Figure 8.5 shows the microlayer edge to have advanced for the first 20 ms and to have retracted for the remaining 24 ms. Simultaneously the central dry patch grew for the first 36 ms and retracted rapidly during the remaining 8 ms. Microlayer thinning, demonstrated by the positive slope for $d = 0,083 \mu\text{m}$ was, surprisingly, restricted to the inner microlayer edge. Elsewhere, for $d \geq 0,166 \mu\text{m}$, local thickness increased or remained constant with time. The maximum microlayer thickness was of the order of $0,7 \mu\text{m}$. This value is very much lower than the estimates based on microthermometry (Section 4.2.2)

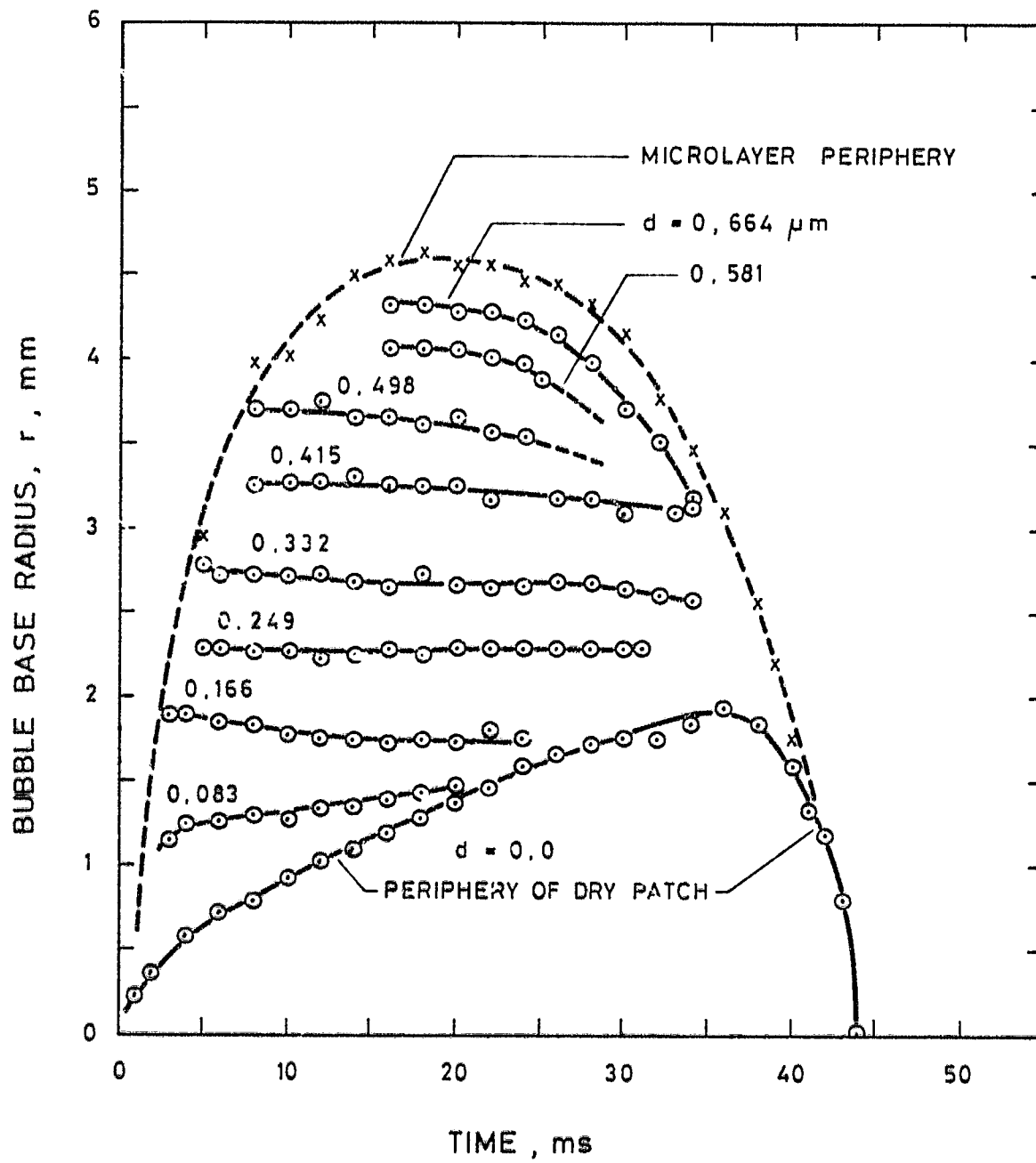


FIG. 8.5 MICROLAYER HISTORY. RUN M10

Test liquid: methanol
 $P = 25,6 \text{ kN/m}^2$
 $q/A = 70,0 \text{ kW/m}^2$
 $\Delta T_{\text{sub}} = 10,2 \text{ K}$

and of the same order as Sharp's interferometry measurements on flattened bubbles (Section 4.2.5).

Figure 8.6, a cross-plot from Figure 8.5, shows the microlayer profiles at selected times. (Data points are omitted. The vertical portions of the profiles emanate from the data point for the microlayer edge and denote the bubble side walls.) At 3 ms a microlayer of 2,1 mm radius has formed. At 7 and 16 ms the layer has grown to radii of 3,6 and 4,6 mm respectively, has become thinner near the inner edge and has thickened elsewhere. At 30 ms the inner microlayer edge has become blunt, that is, vertical, up to a thickness of 0,166 μm (note that in Figure 8.5 the curves for $d = 0,083$ and $d = 0,166 \mu\text{m}$ have merged with that for the dry patch); in the outer regions the microlayer has thickened considerably. Few data points are available at 51 ms (several fringes had become merged or indistinct), nevertheless the profile could be constructed with some confidence; the inner microlayer edge has dried out further and the outer regions have lifted dramatically.

It appears then that the microlayer was continuously replenished by liquid, with the rate of inflow exceeding the rate of evaporative depletion over all areas except the inner edge. In other words, while the microlayer evaporated, the bubble continuously lifted off from its base. In the latter part of the cycle, as the bubble tended towards a spherical shape a gross lift-off or curling-up of the microlayer edge (microlayer retraction) was superimposed upon the overall lift-off.

Figure 8.7 shows the equivalent spherical diameter and Figure 8.8 the bubble volume against time. The maximum volume, prior to collapse, is 1640 mm^3 . An estimate of the contribution to this volume by the apparent, visible microlayer evaporation may now be given. The portion visibly evaporated may be approximately taken as the inner tip of the wedge-shaped profile up to a radius of 2 mm and a thickness of

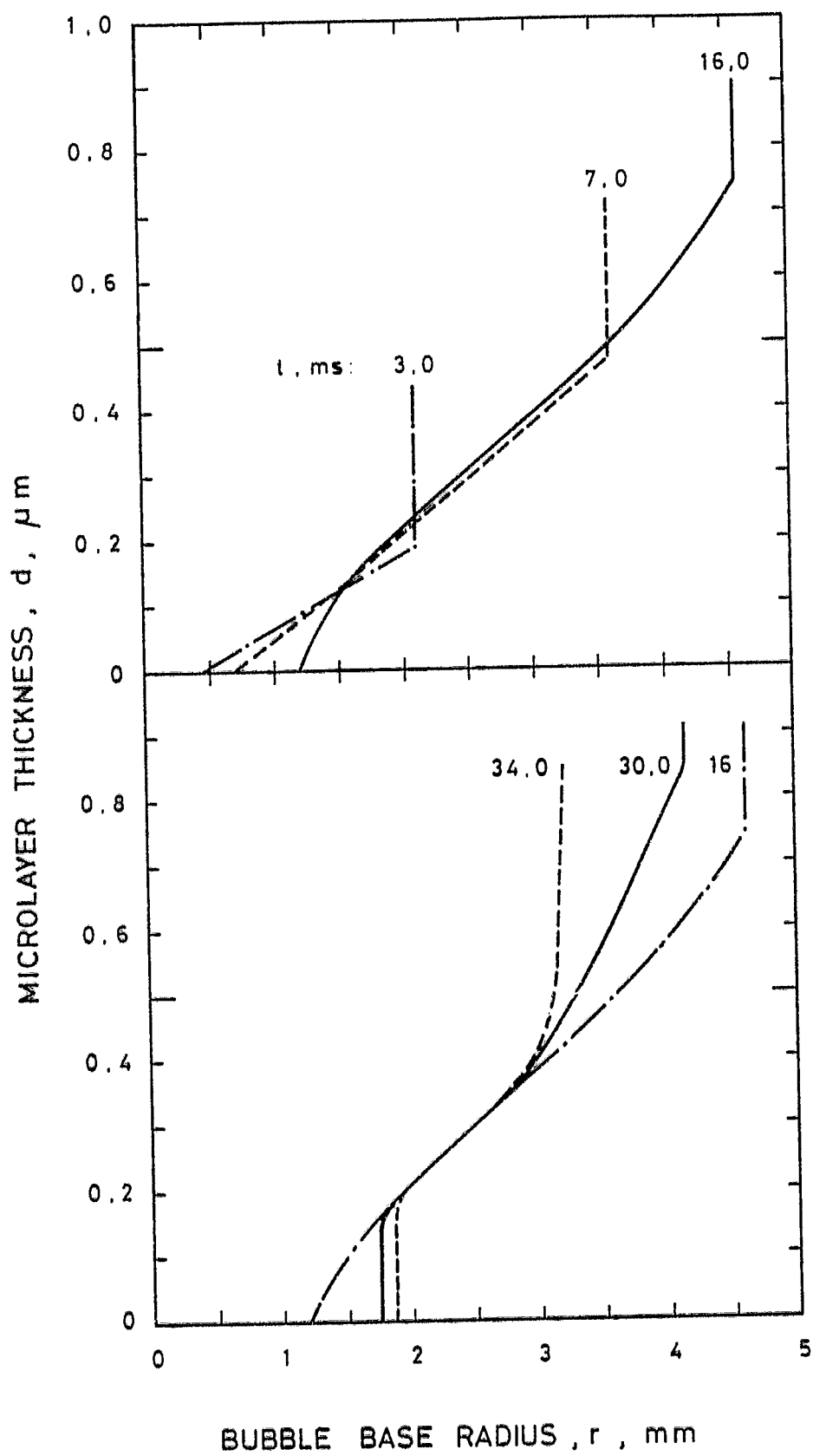


FIG. 8.6 MICROLAYER PROFILES. RUN M10

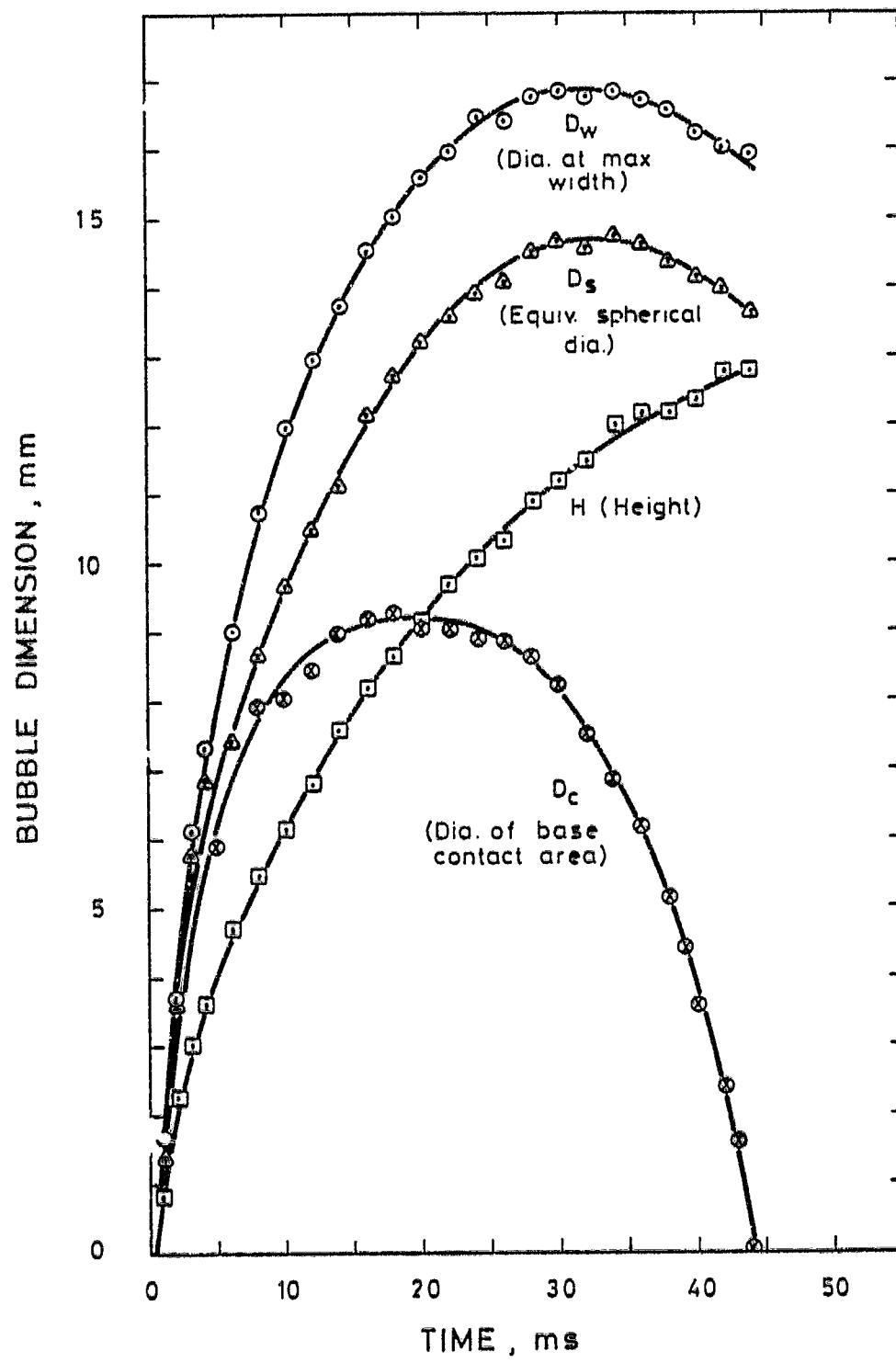


FIG. 8.7 BUBBLE DIMENSIONS. RUN M10.

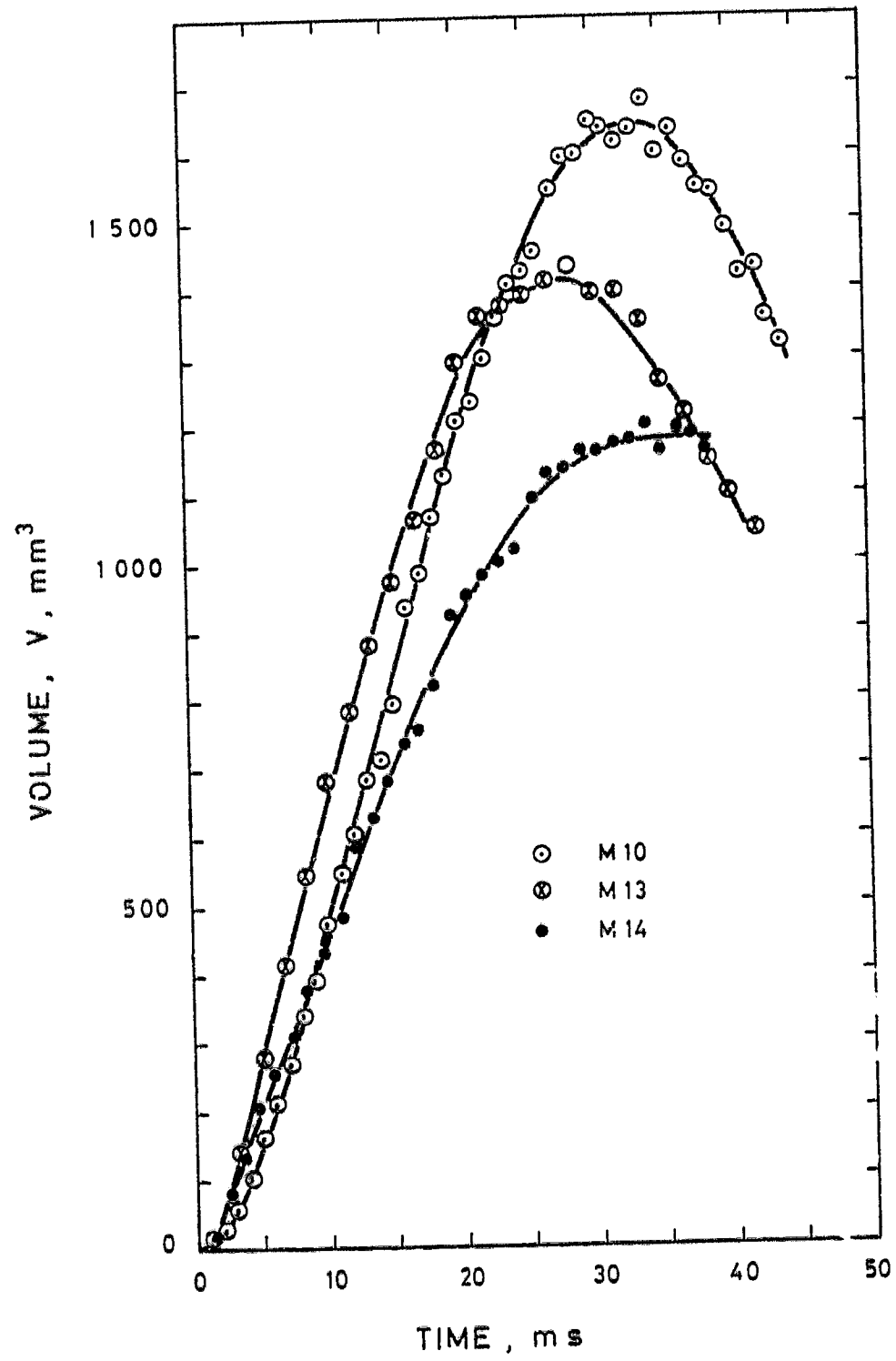


FIG. 8.8 BUBBLE VOLUME. RUNS M10, M13, M14.

0,2 μm , all rotated about the vertical bubble axis. The resulting liquid volume is 0,0017 mm^3 . (Details of this calculation are given in Section 8.5). The equivalent vapour volume is approximately 4 mm^3 , that is, 0,24 per cent of the maximum bubble volume. This contribution, while neglecting microlayer replenishment effects, is nevertheless amazingly low.

Figure 8.7 further gives the bubble dimensions characterising the overall bubble shape. The plot for D_c , the diameter of the bubble contact area (the microlayer), is simply a replot of the outer envelope of Figure 8.5. Since D_c and D_w , the maximum bubble width, diverge widely, except perhaps for the first few milliseconds of growth, the bubble shape cannot be taken as hemispherical. Thus the theoretical equations predicting microlayer thickness (Section 4.2.6) do not generally apply. Since H , the bubble height, and D_s , the equivalent spherical diameter, diverge widely at all stages except at departure, the bubble cannot be taken as spherical either, and is classed as oblate.

Run M13 was essentially a repeat of Run M10. Figures 8.9 and 8.10 again demonstrate the simultaneous microlayer evaporation and replenishment, with thinning limited to the inner microlayer edge. The maximum microlayer thickness is a little lower than in Run M10, namely 0,5 instead of 0,7 μm . The bubble dimensions and shape, shown in Figure 8.11, are very similar to those for Run M10, but with the tendency to bubble sphericity somewhat more pronounced. Since the bubble volumes, see Figure 8.8, and the microlayer profiles are comparable for the two runs, the contribution of the apparent (visible) microlayer evaporation to the total bubble volume is again extremely small.

8.4.2 Methanol Runs M14, M19 and M20

Runs M14 and M19 were conducted at identical heat fluxes and

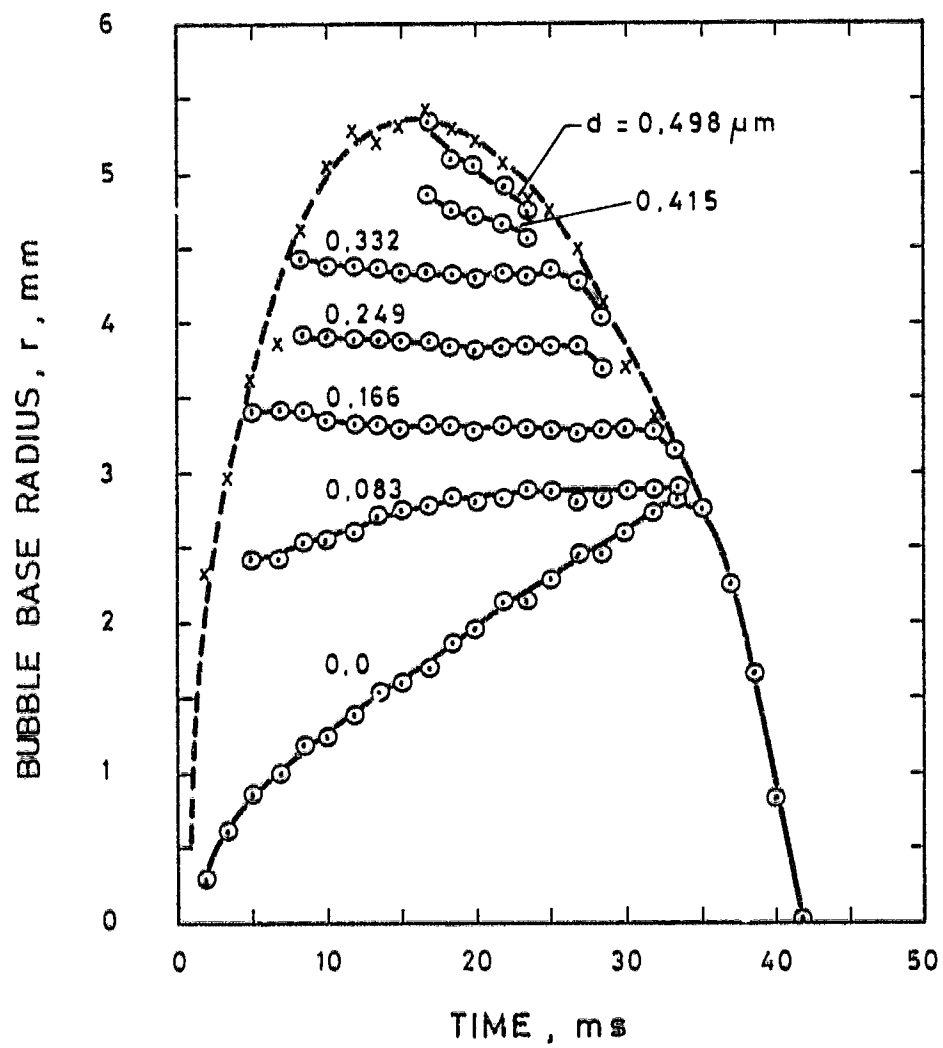


FIG. 8.9 MICROLAYER HISTORY. RUN M13.

Test liquid : methanol

$P = 24,0 \text{ kN/m}^2$

$q/A = 61,8 \text{ kW/m}^2$

$\Delta T_{\text{sub}} = 9,9 \text{ K}$

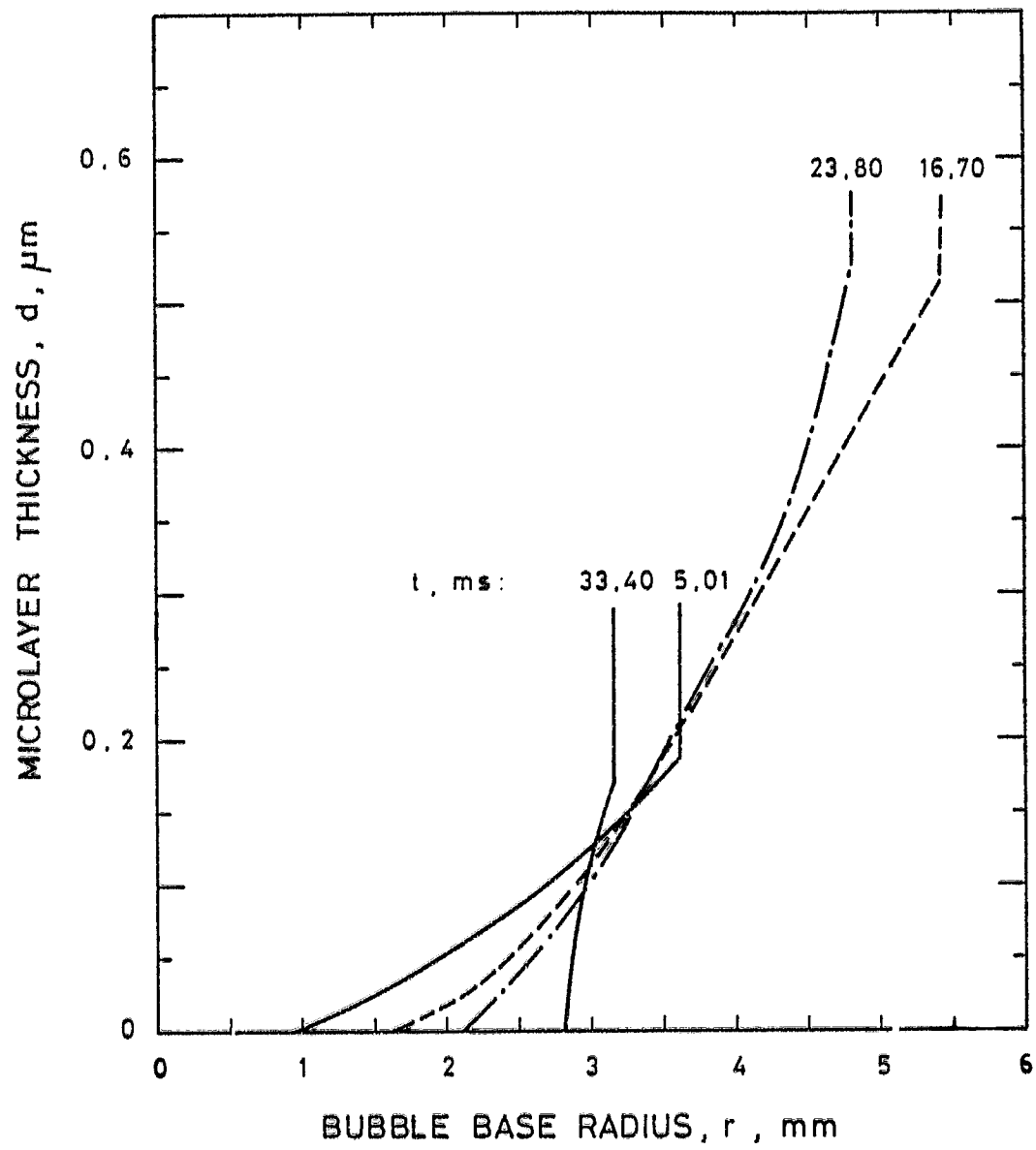


FIG. 8.10 MICROLAYER PROFILES RUN M13

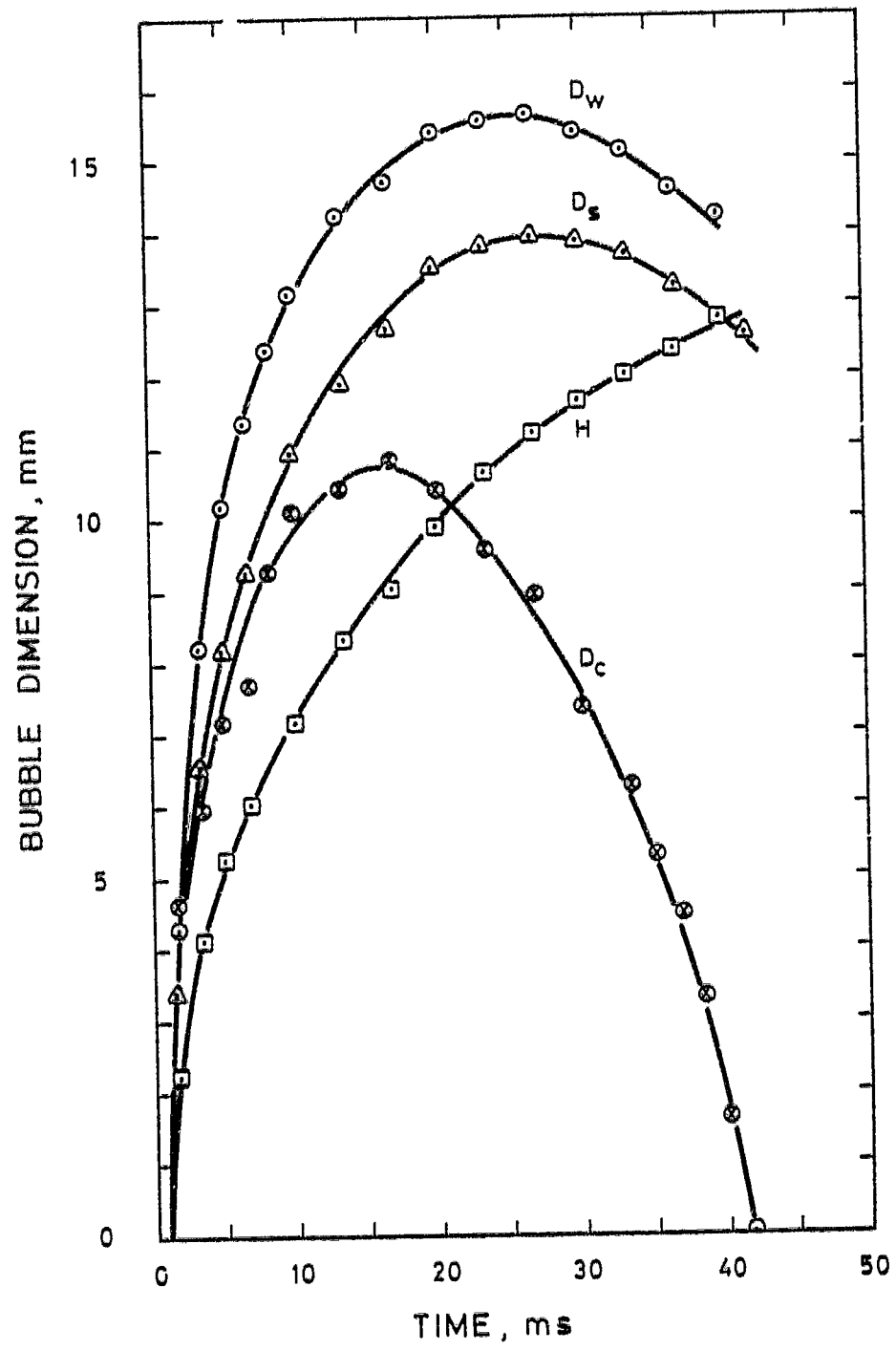


FIG. 8.11 BUBBLE DIMENSIONS. RUN M13.

pressures, but with the bulk saturated in M14 and highly subcooled in M19. Run M20 was a repetition at higher pressure of the saturation case M14.

Detailed microlayer histories are shown in Figures 8.12 and 8.13 for Run M14 and in Figures 8.15 and 8.16 for Run M19. (The locus of the microlayer periphery in Figure 8.15 is sketched in by hand, no data points being obtainable.) The two cases reveal a very similar microlayer behaviour. Microlayer thinning is again restricted to the inner edge, that is, to thicknesses of less than $0,166 \mu\text{m}$; over all remaining areas the thickness tends to increase progressively with time. The maximum thickness is of the order of $0,5 \mu\text{m}$ for M14 and $0,3 \mu\text{m}$ for M19. Since the maximum microlayer radii, see Figures 8.12 and 8.15, are $4,2 \text{ mm}$ for M14 and $2,5 \text{ mm}$ for M19, the two microlayers are approximately geometrically similar. The overall bubble dimensions, given in Figures 8.13 and 8.17, in fact show the entire bubble shape histories to be geometrically similar, with the subcooled bubble generally smaller by a factor of 0,6. Both bubbles were again pronouncedly non-hemispherical.

Run M20 was conducted under the same conditions of heat flux and bulk saturation as run M14, but at a pressure of $42,6$ instead of $33,3 \text{ kN/m}^2$. The comparison of bubble dimensions in Figures 8.14 and 8.21 shows the expected smaller bubble size at the higher pressure and, towards departure, a slight tendency to greater bubble sphericity. The microlayer history of Run M20 is shown in Figures 8.19 and 8.20. The maximum thickness is approximately $0,5 \mu\text{m}$, as in Run M14, and the radius to the microlayer edge is approximately 0,8 times that of Run M14, that is, reduced by the same ratio as the overall bubble dimensions. Further, Run M20 displays a more pronounced blunting of the inner microlayer edge and a less pronounced thickening of the outer regions. In fact, for growth times greater than 10 ms and thicknesses greater than $0,166 \mu\text{m}$ the local thickness remains constant

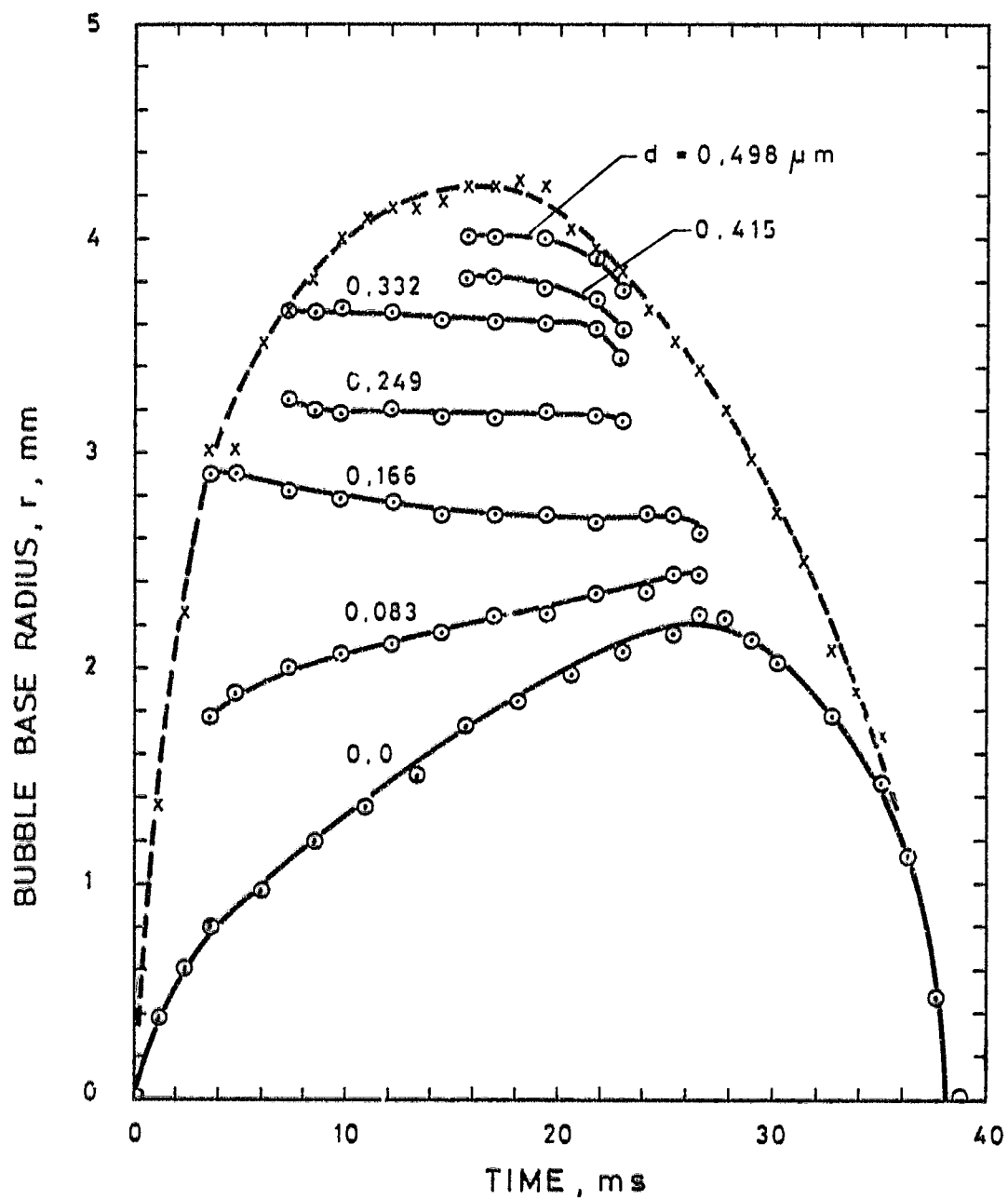


FIG. 8.12 MICROLAYER HISTORY. RUN M14.
 Test liquid: methanol
 $P = 33,3 \text{ kN/m}^2$
 $q/A = 61,9 \text{ kW/m}^2$
 $\Delta T_{\text{sub}} = 0,4 \text{ K i.e. bulk saturated}$

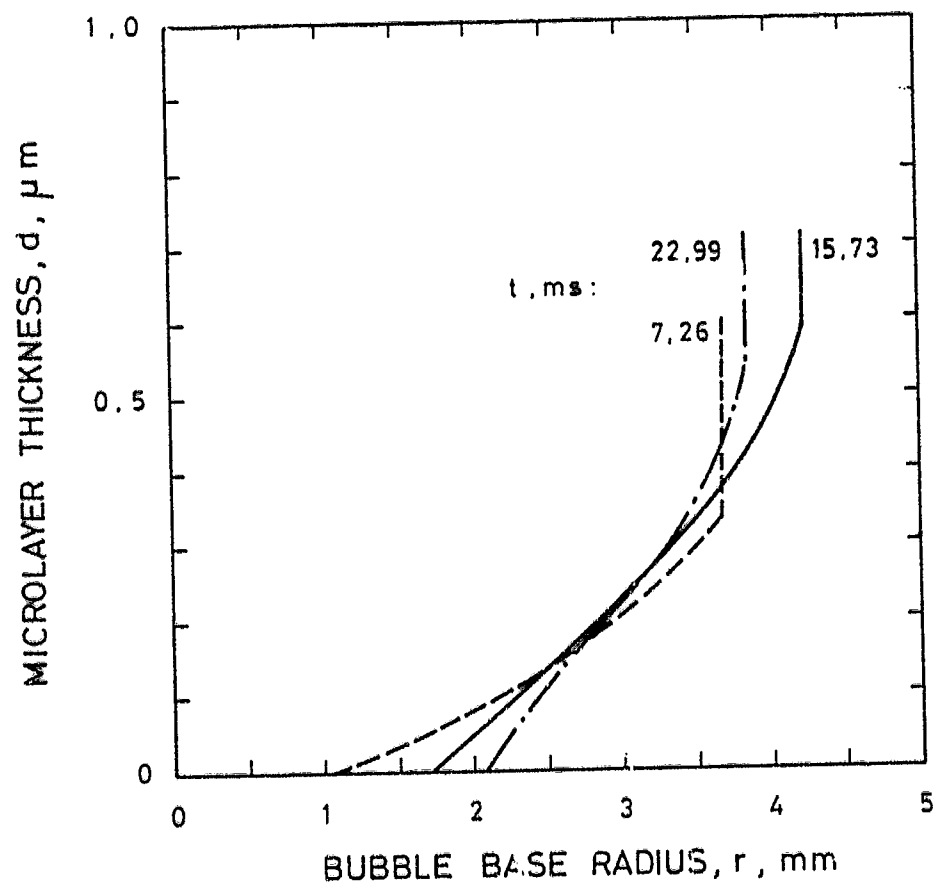


FIG. 8.13 MICROLAYER PROFILES. RUN M14.

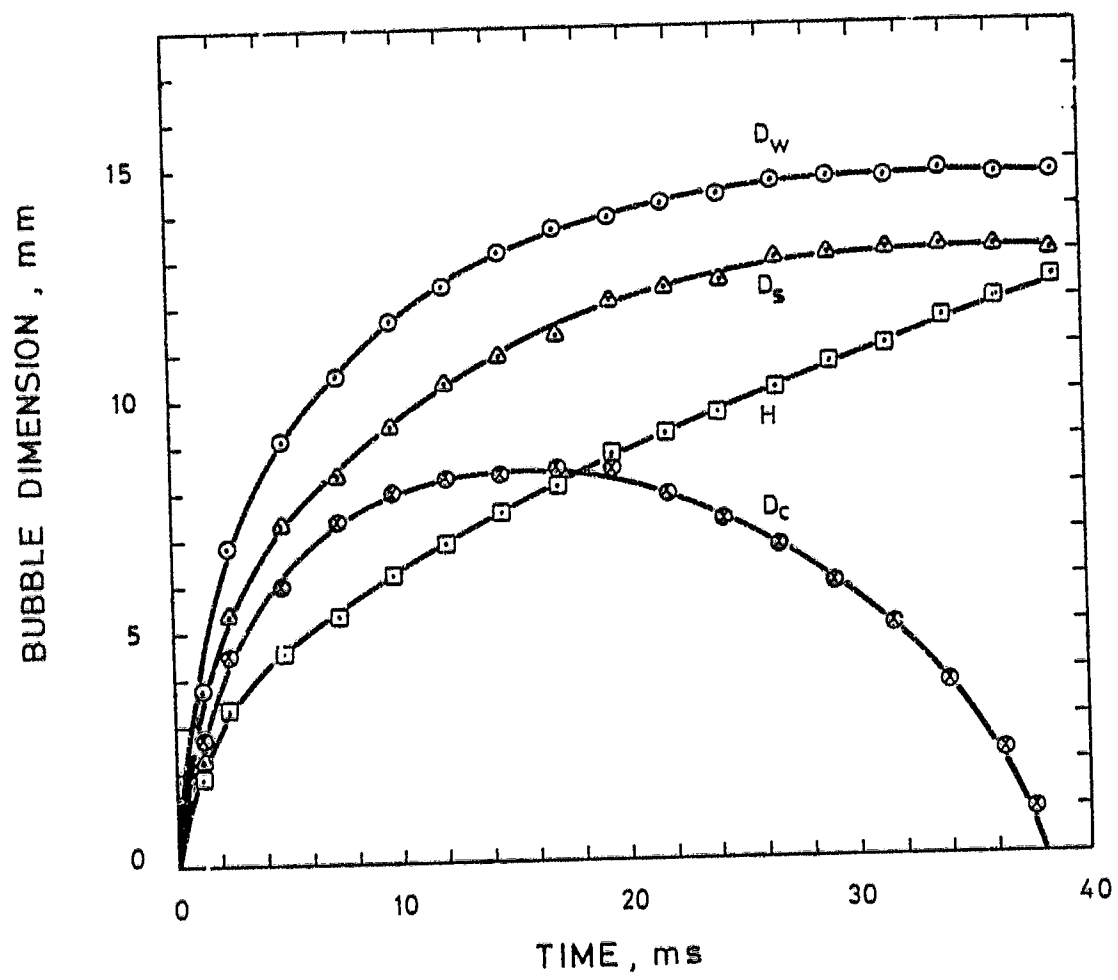


FIG. 8.14 BUBBLE DIMENSIONS. RUN M14.

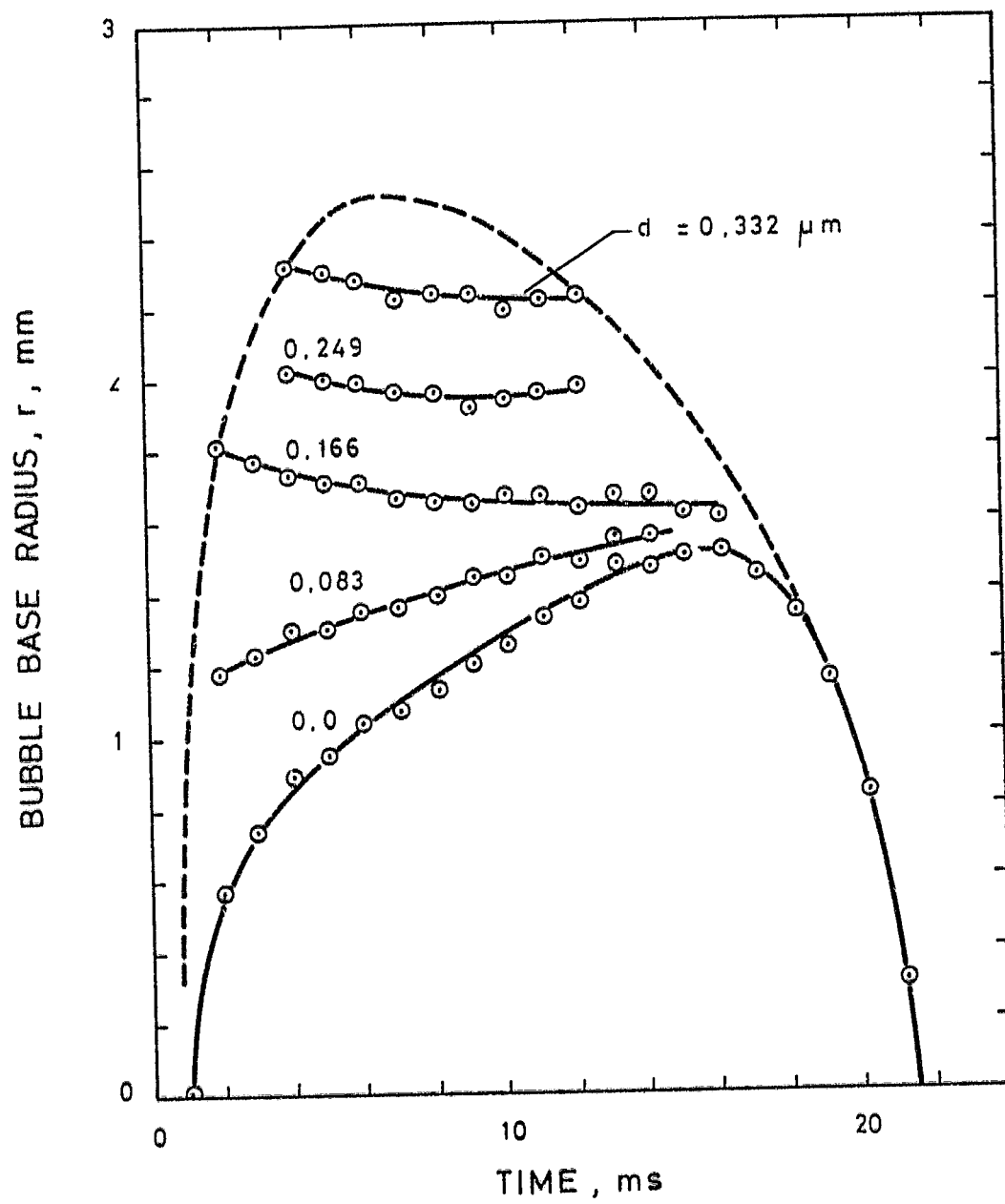


FIG. 8.15 MICROLAYER HISTORY. RUN M19.
 Test liquid : methanol
 $P = 33,3 \text{ kN/m}^2$
 $q/A = 61,9 \text{ kW/m}^2$
 $\Delta T_{\text{sub}} = 20,2 \text{ K}$

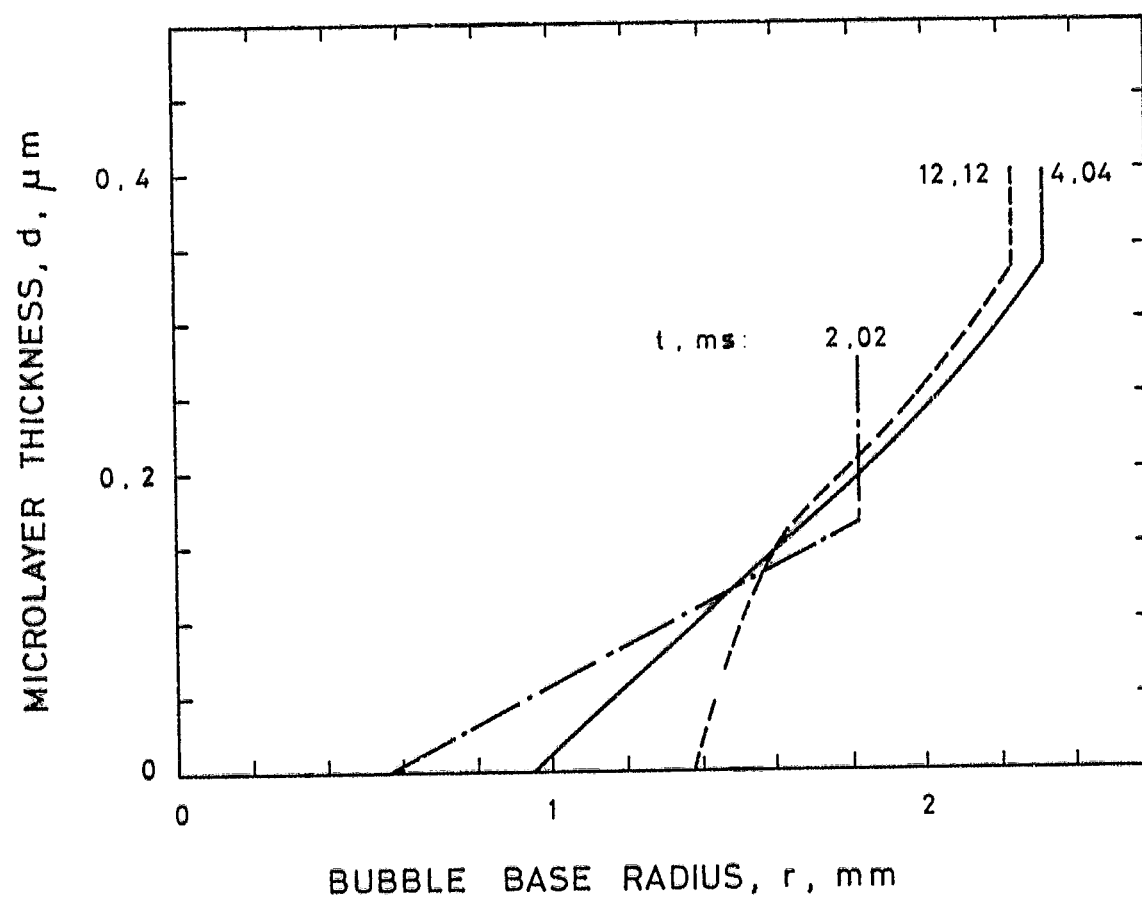


FIG. 8.16 MICROLAYER PROFILES. RUN M19.

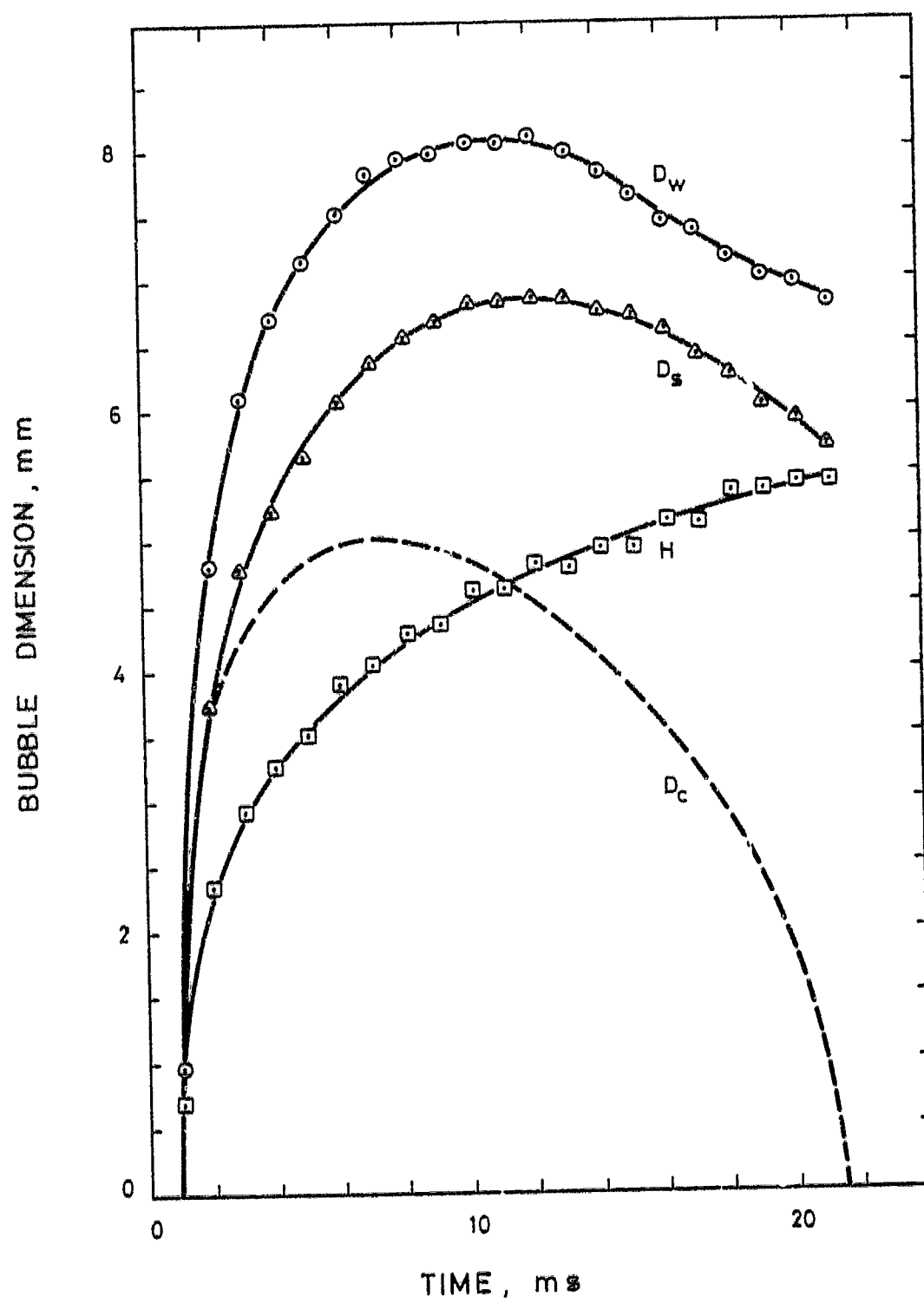


FIG. 8.17

BUBBLE DIMENSIONS. RUN M19.

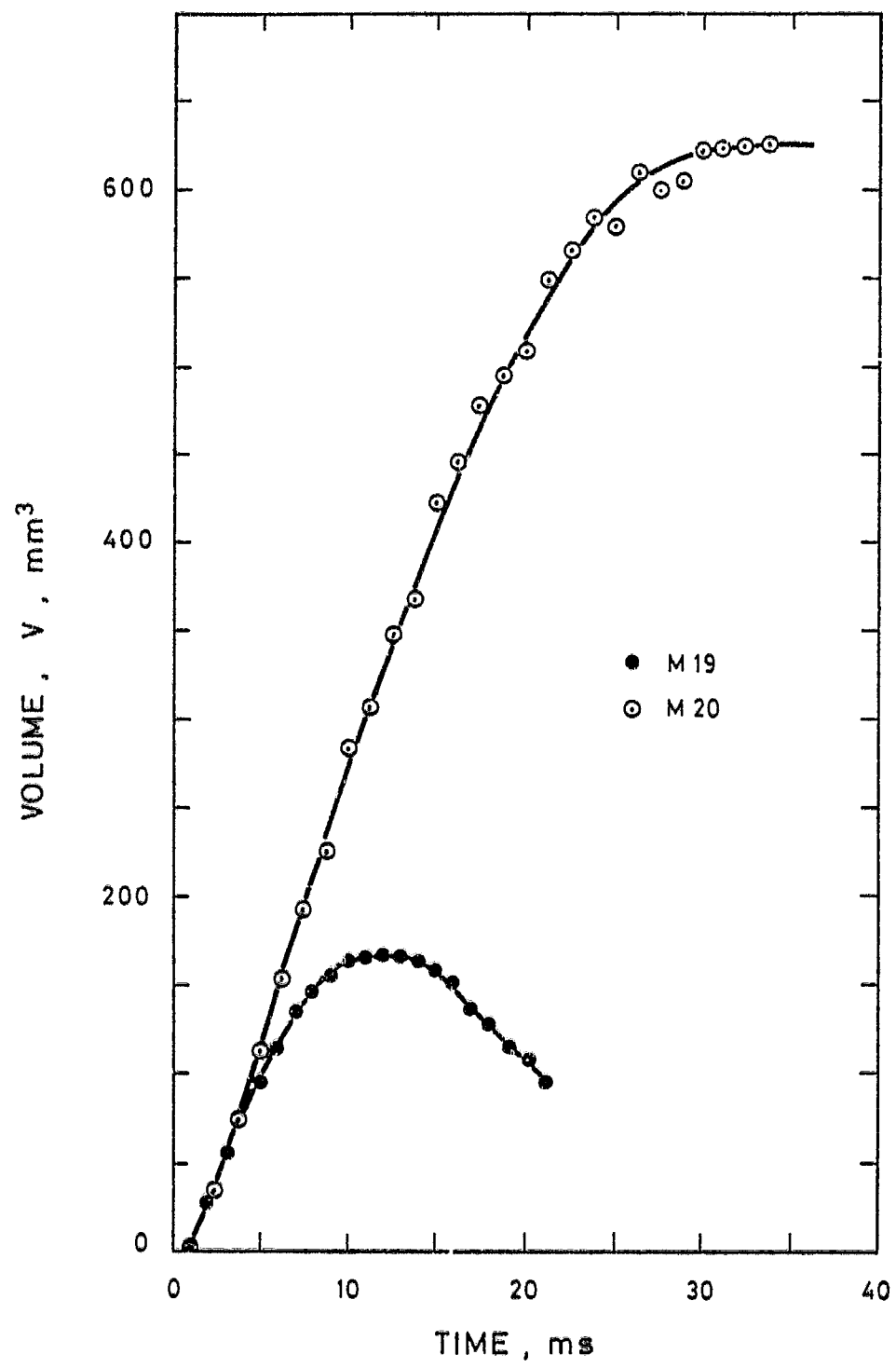


FIG. 8.18 BUBBLE VOLUMES. RUNS M19 AND M20.

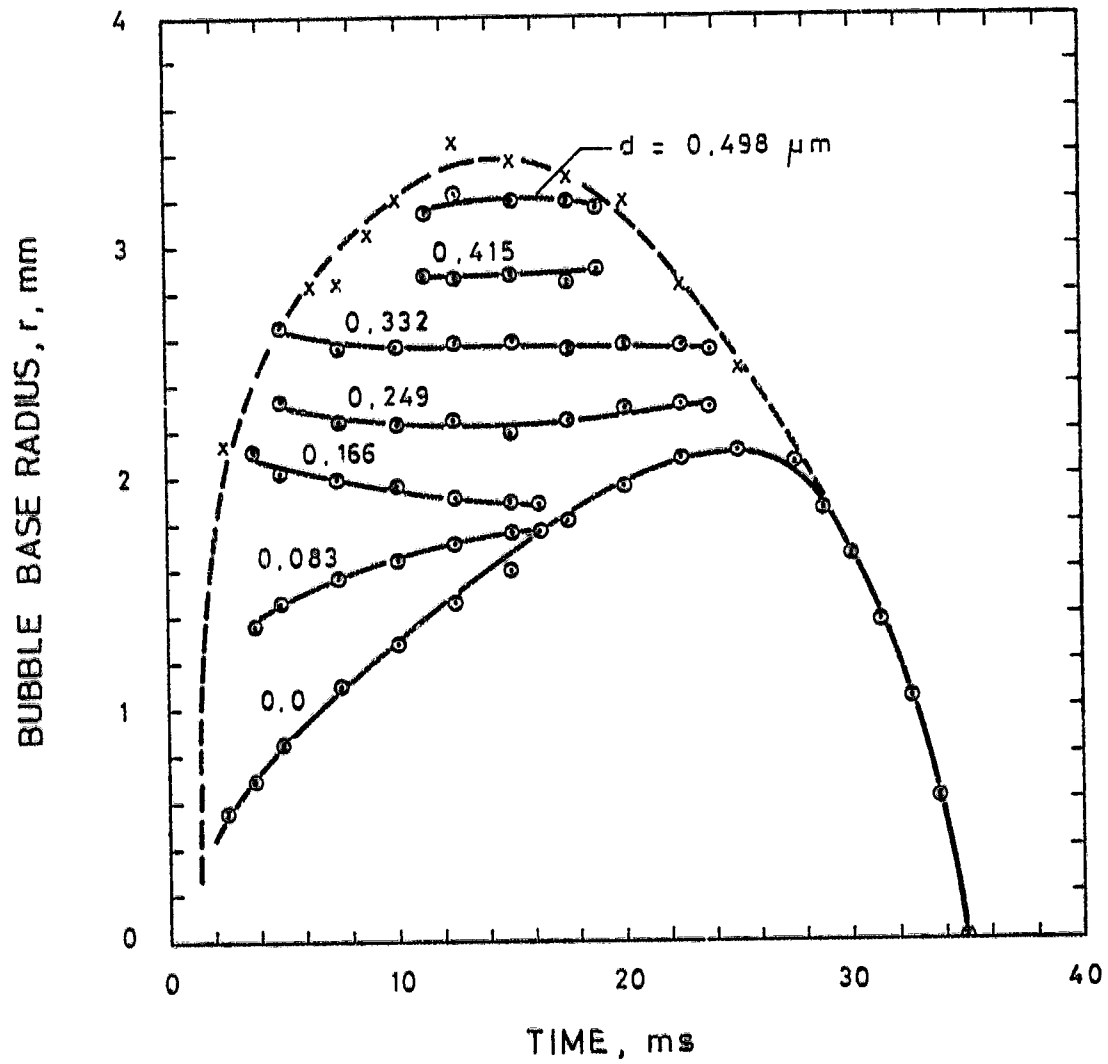


FIG. 8.19

MICROLAYER HISTORY. RUN M20.

Test liquid: methanol

 $P = 42,6 \text{ kN/m}^2$ $q/A = 61,9 \text{ kW/m}^2$ $\Delta T_{\text{sub}} = 0,5 \text{ K i.e. bulk saturated}$

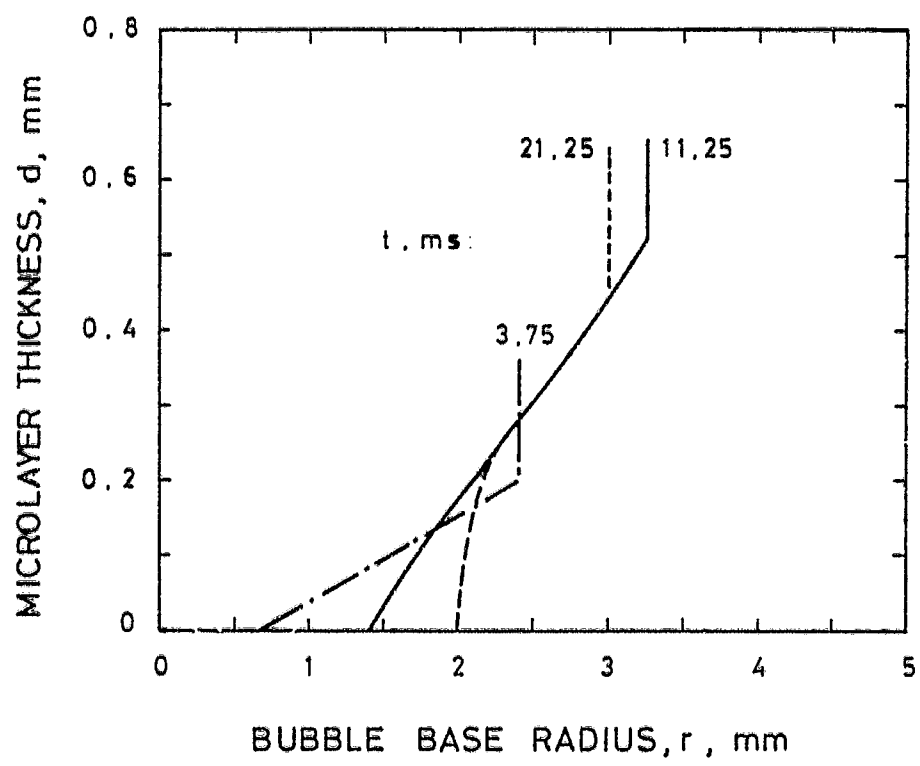


FIG. 8.20 MICROLAYER PROFILES. RUN M20.

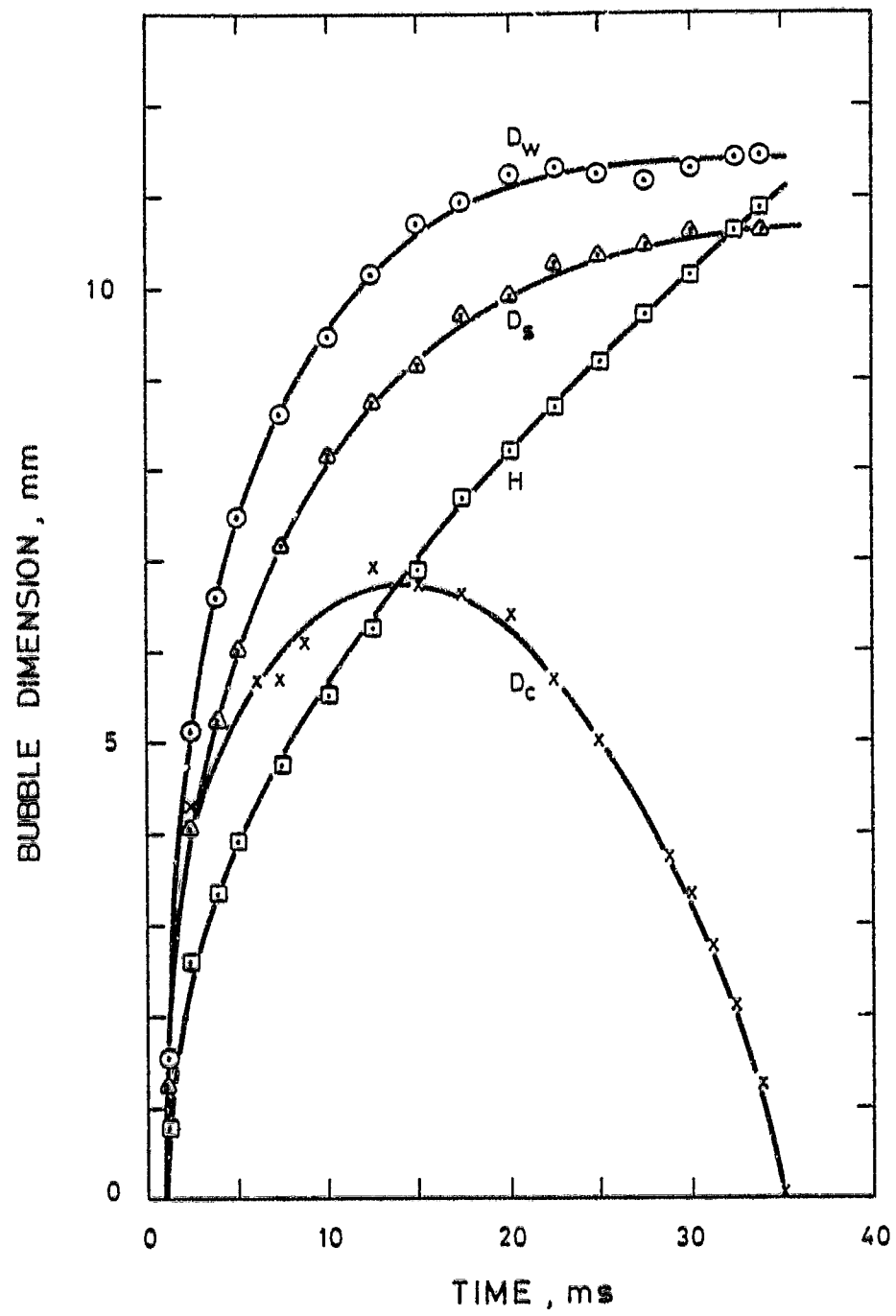


FIG. 8.21 BUBBLE DIMENSIONS. RUN M20.

with time. In view of the results of the previous runs this constant thickness must be interpreted as the dynamic equilibrium between evaporative depletion and in-flow replenishment.

Bubble volumes versus time for Runs M14, M19 and M20 are shown in Figures 8.8 and 8.18. The contributions by the visible microlayer evaporation to these volume is dealt with in Section 8.5. From the estimate on the results of Run M10, however, it is clear already at this stage, that the contributions are small.

8.4.3 Ethanol Runs E8 and E12

The microlayer behaviour observed with methanol was confirmed (by projection of the ciné films) in a considerable number of ethanol runs. Two of these, Runs E8 and E12, were analysed fully.

Run E8 was conducted under conditions of slight subcooling (2,6 K) and very low heat flux (30,4 kW/m²), and the saturation run E12 was virtually a repetition of Run M14. Results are shown in Figures 8.22 to 8.28. These confirm very clearly the chief features of the methanol microlayer behaviour, namely, the restriction of microlayer thinning to the inner layer edge, the progressive thickening over the remaining regions and the very low value of the maximum microlayer thickness. Bubble shapes were again oblate.

8.5 CONTRIBUTION OF VISIBLE MICROLAYER EVAPORATION TO BUBBLE VOLUME

The foregoing section has shown that evaporating microlayers are not, as is generally assumed, stagnant, but are continuously replenished by in-flowing liquid. Thus microlayer vapour evolution cannot be determined directly from measurements of the change of microlayer geometry with time.

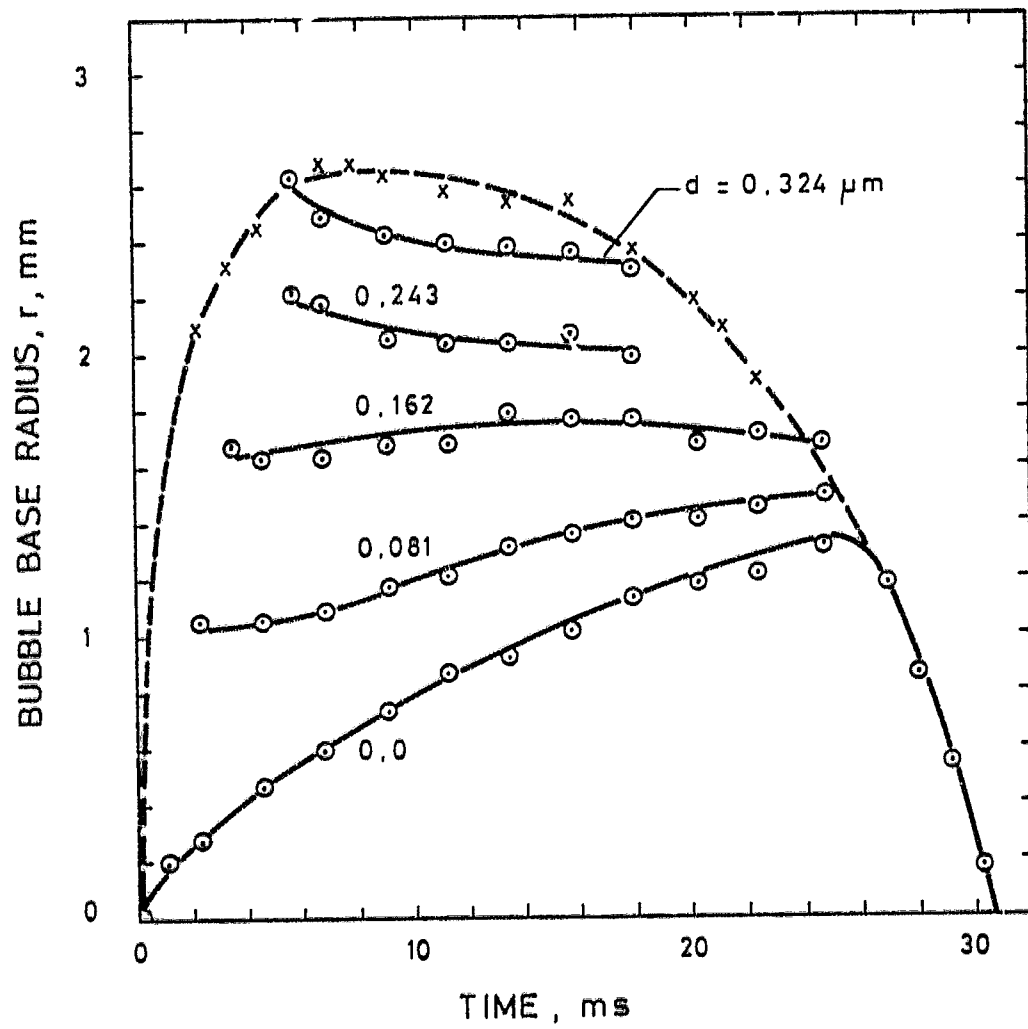


FIG. 8.22 MICROLAYER HISTORY. RUN E8.
 Test liquid: ethanol
 $P = 25,4 \text{ kN/m}^2$
 $q/A = 30,4 \text{ kW/m}^2$
 $\Delta T_{\text{sub}} = 2,6 \text{ K}$

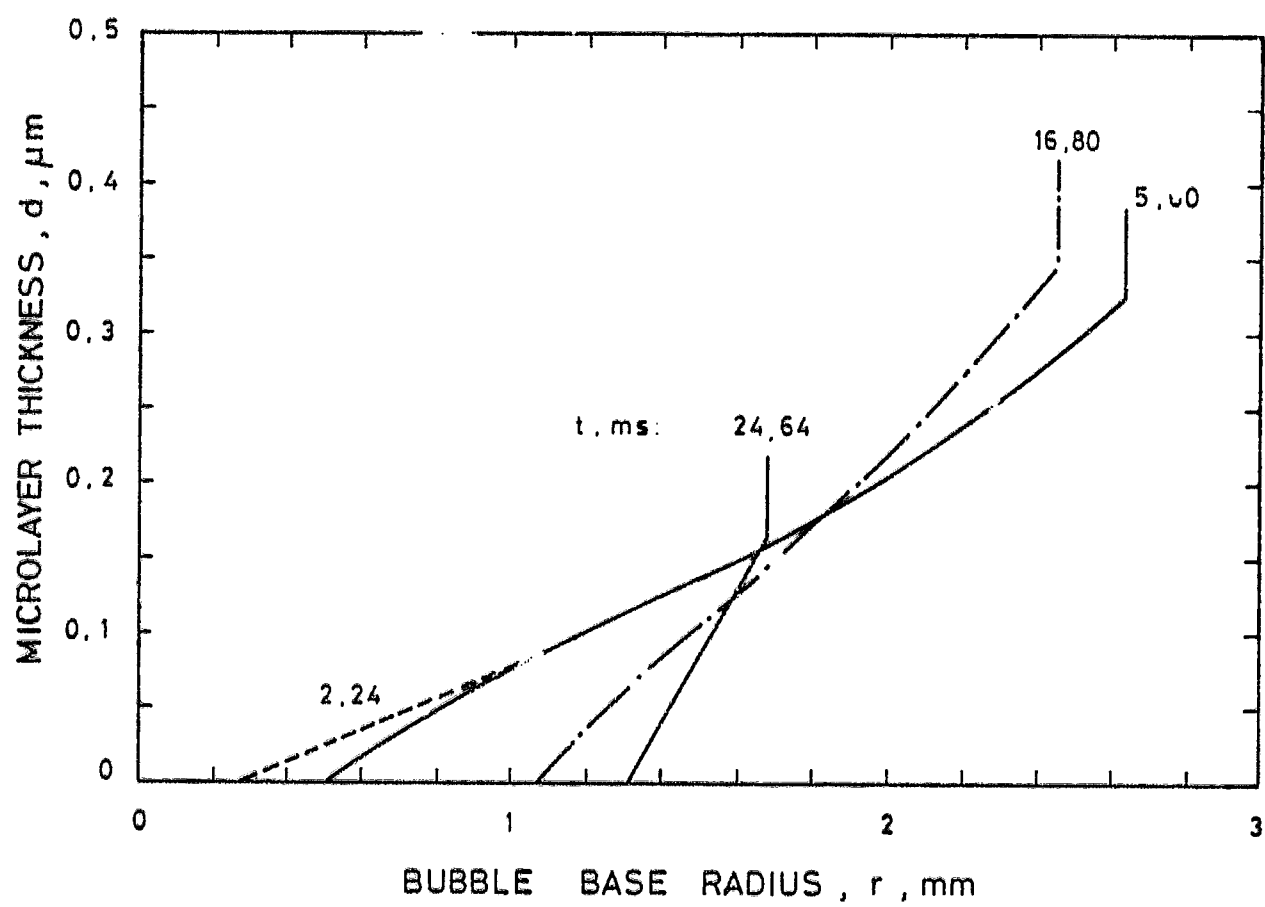


FIG. 8.23 MICROLAYER PROFILES. RUN E8.

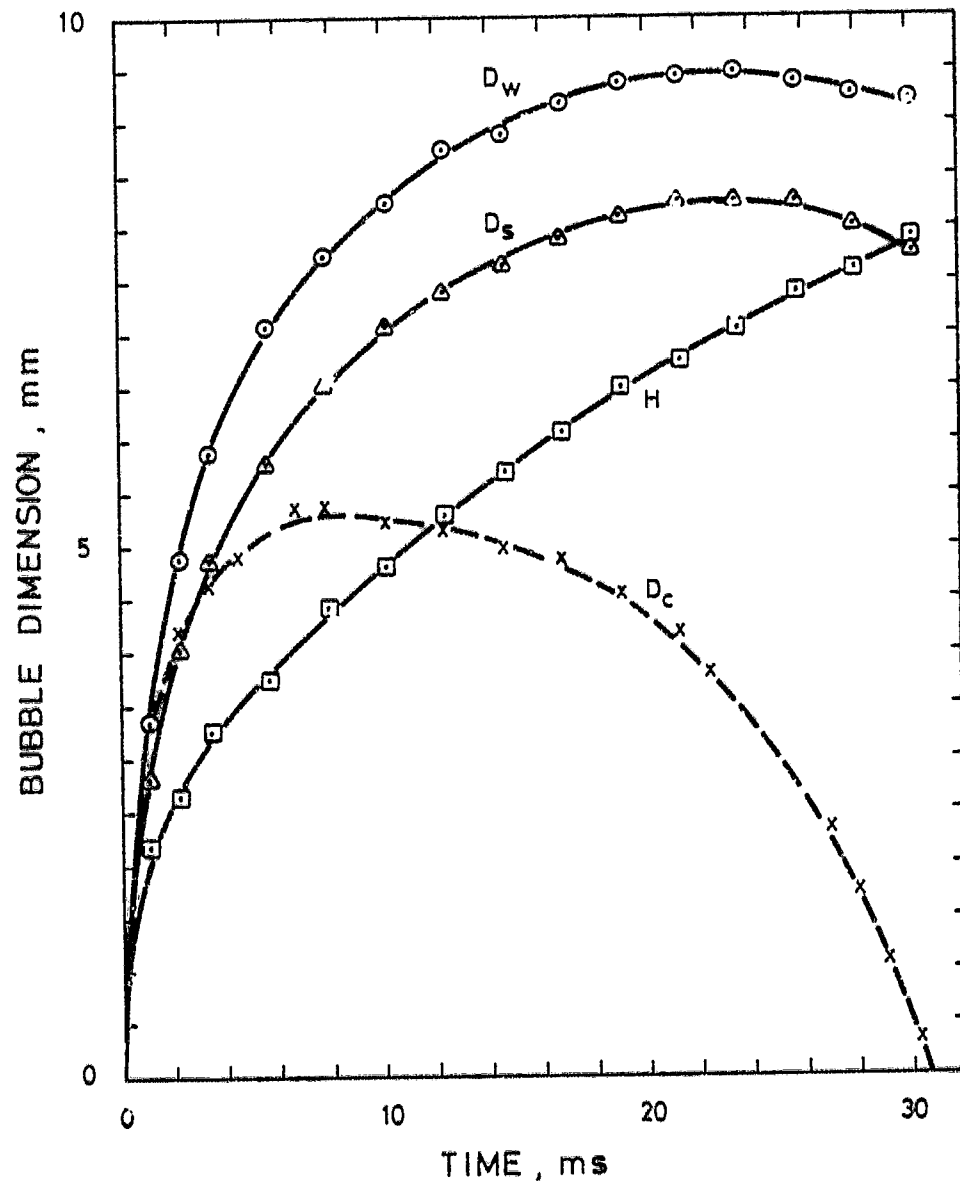


FIG. 8.24 BUBBLE DIMENSIONS. RUN E8.

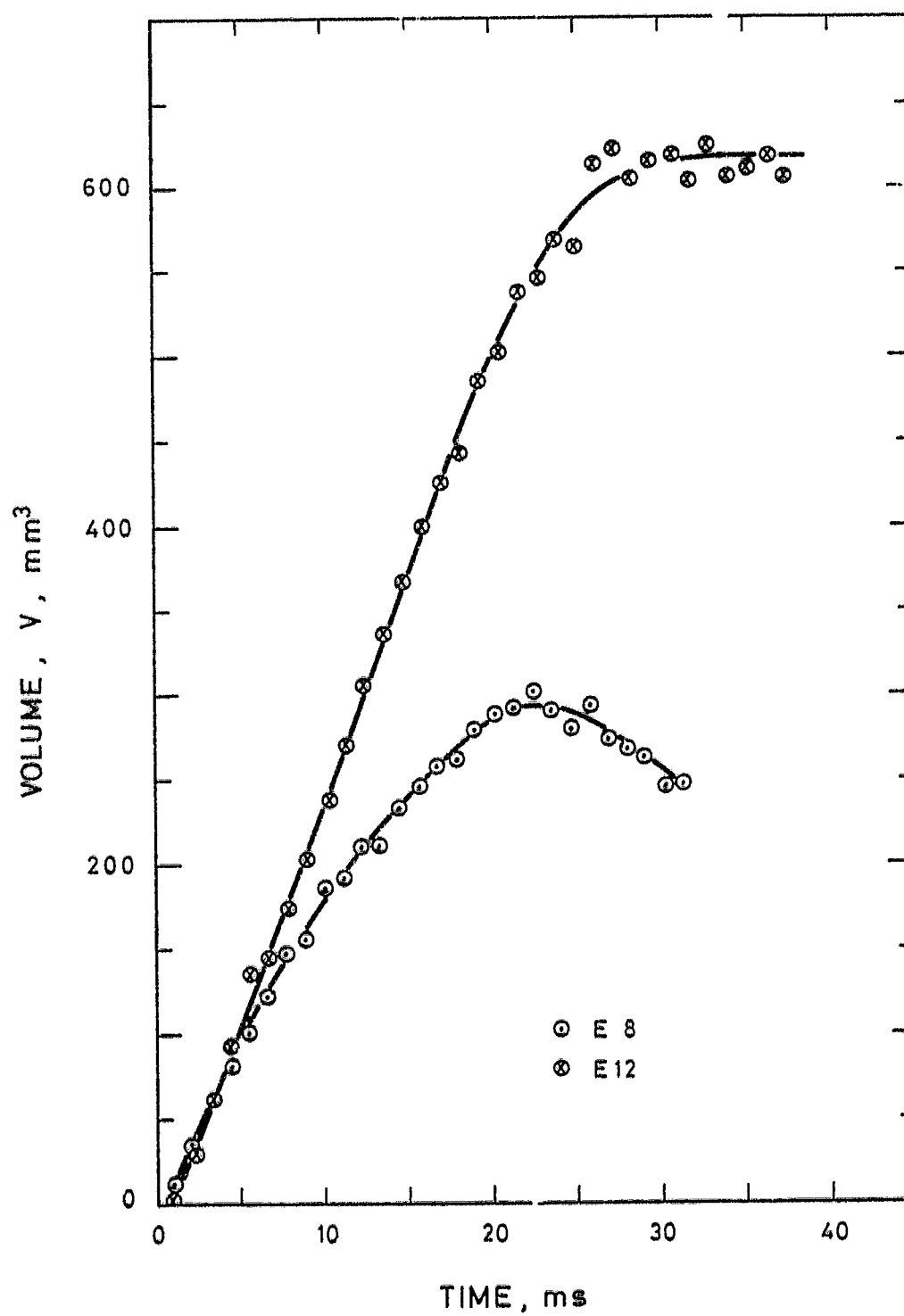


FIG. 8.25 BUBBLE VOLUME. RUNS E8 AND E12.

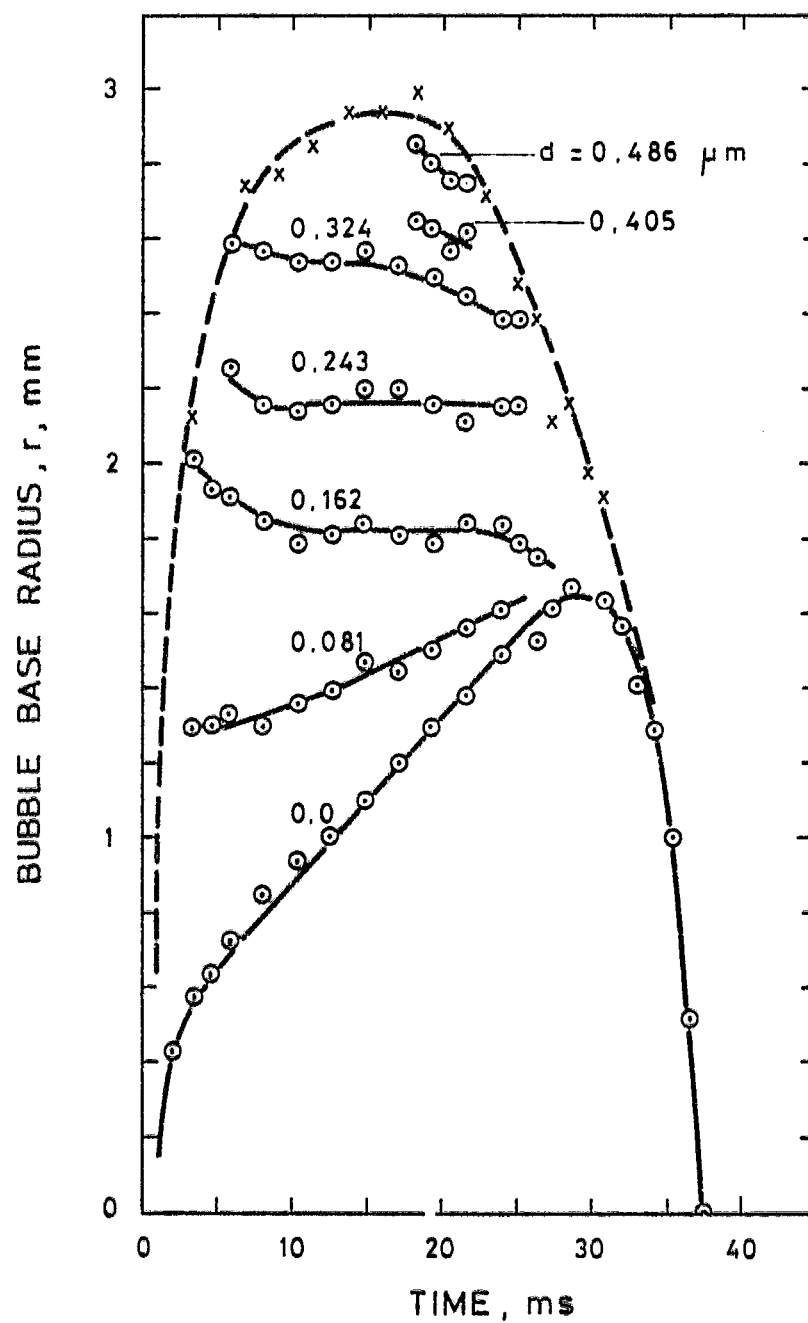


FIG. 8.26

MICROLAYER HISTORY. RUN E12.

Test liquid: ethanol

 $P = 33,3 \text{ kN/m}^2$ $q/A = 65,7 \text{ kW/m}^2$ $\Delta T_{\text{sub}} = -0,3 \text{ K}$ i.e. bulk saturated

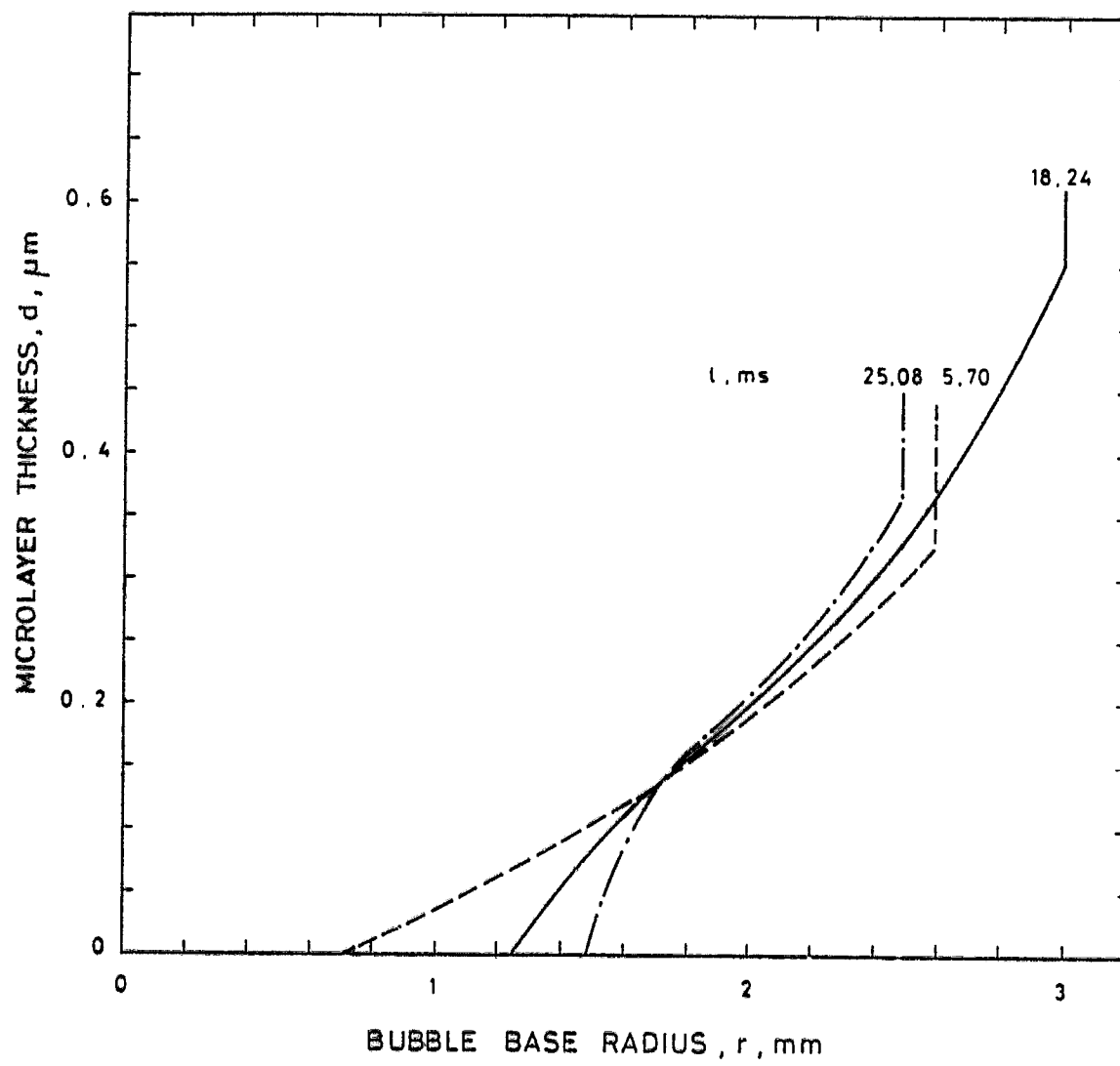


FIG. 8. 27 MICROLAYER PROFILES. RUN E12.

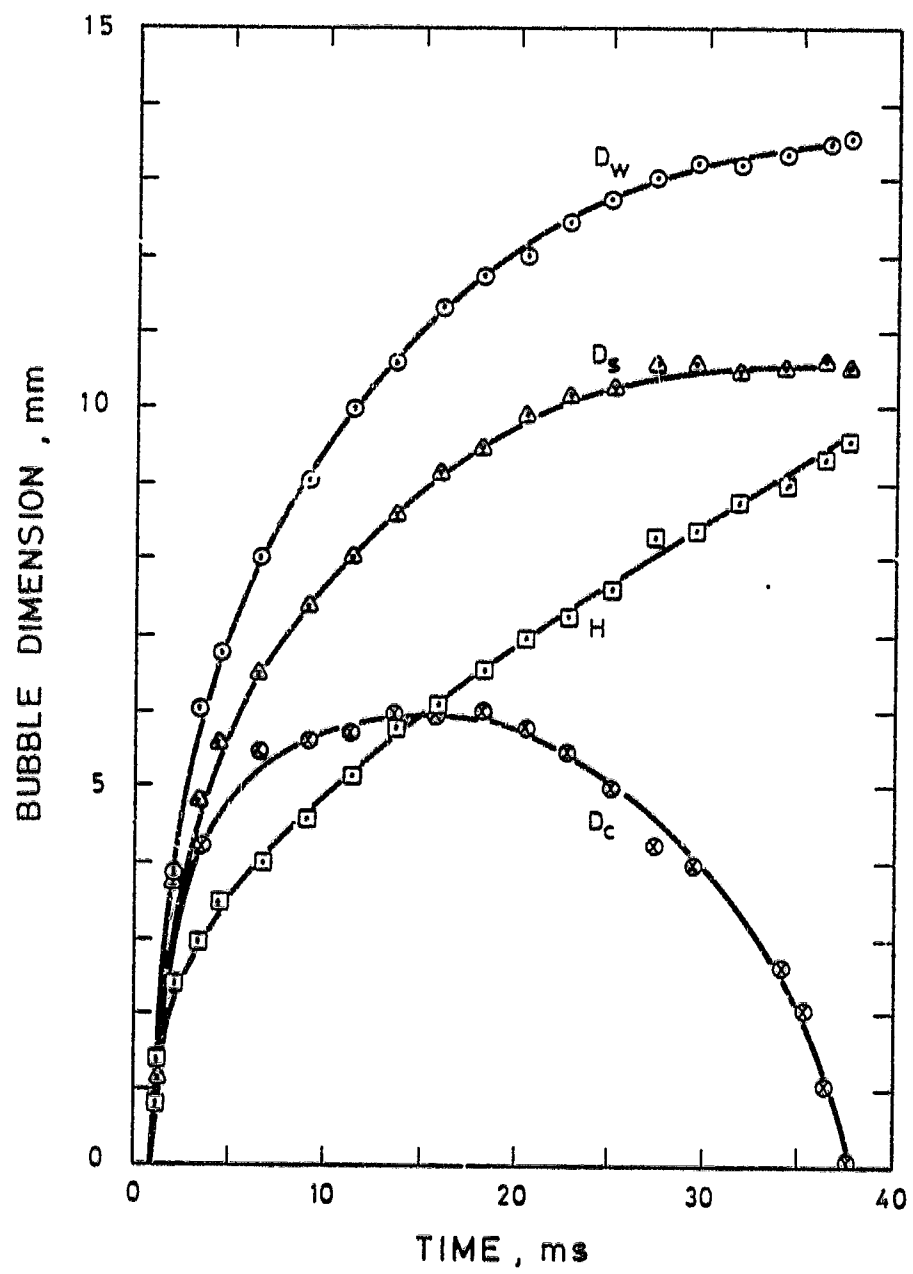


FIG. 8.28 BUBBLE DIMENSIONS. RUN E12.

Estimates of the apparent contribution of (visible) microlayer evaporation are, however, possible and are given below.

Visible evaporation (microlayer thinning) was in all runs restricted to the inner microlayer edge, generally to thicknesses lower than the second maximum of reflectance ($0,166 \mu\text{m}$ for methanol and $0,162 \mu\text{m}$ for ethanol). Beyond this inner edge, thicknesses remained constant or increased with time. Figure 8.29 (a) illustrates a typical case. The dimension d'' is the microlayer thickness given by the second maximum of reflectance. The profile shown at time t_1 is the first profile recorded in a run, the profile at time t_2 is the one for which the bubble base radius corresponding to the thickness d'' , is a maximum ($r_{d''} = r_{d'' \text{ max}}$), and the profile at time t_3 is the one for which the dry patch radius is a maximum. The shaded area, rotated about the vertical bubble axis, gives V_{LE}^* , the apparent or visible volume of liquid evaporated.

For purposes of obtaining a simple estimate of V_{LE}^* the profiles may be simplified as shown in Figure 8.29(b), that is, the cross-section of V_{LE}^* is taken as triangular, with height d'' and the base $r_{d'' \text{ max}}$. In general this results in an overestimate of V_{LE}^* . In Runs M10 and M20 (see Figures 8.5 and 8.19), however, the maximum dry patch radius was essentially equal to $r_{d'' \text{ max}}$ and the inner microlayer edge had become vertical up to a thickness of d'' ; thus the estimate is accurate. In Run E8 (see Figure 8.22) a very slight thinning is possibly detectable at a thickness of $0,162 \mu\text{m}$; this, however, is negligible.

For the simplified profiles of Figure 8.29(b) the apparent volume of liquid evaporated is given (by the difference between the volumes of a cylinder and an included cone) as follows:

$$V_{LE}^* = (2\pi/3)r_{d'' \text{ max}}^2 \cdot d'' \quad (8.1)$$

The thickness d'' may for simplicity be taken as $0,17 \mu\text{m}$ for

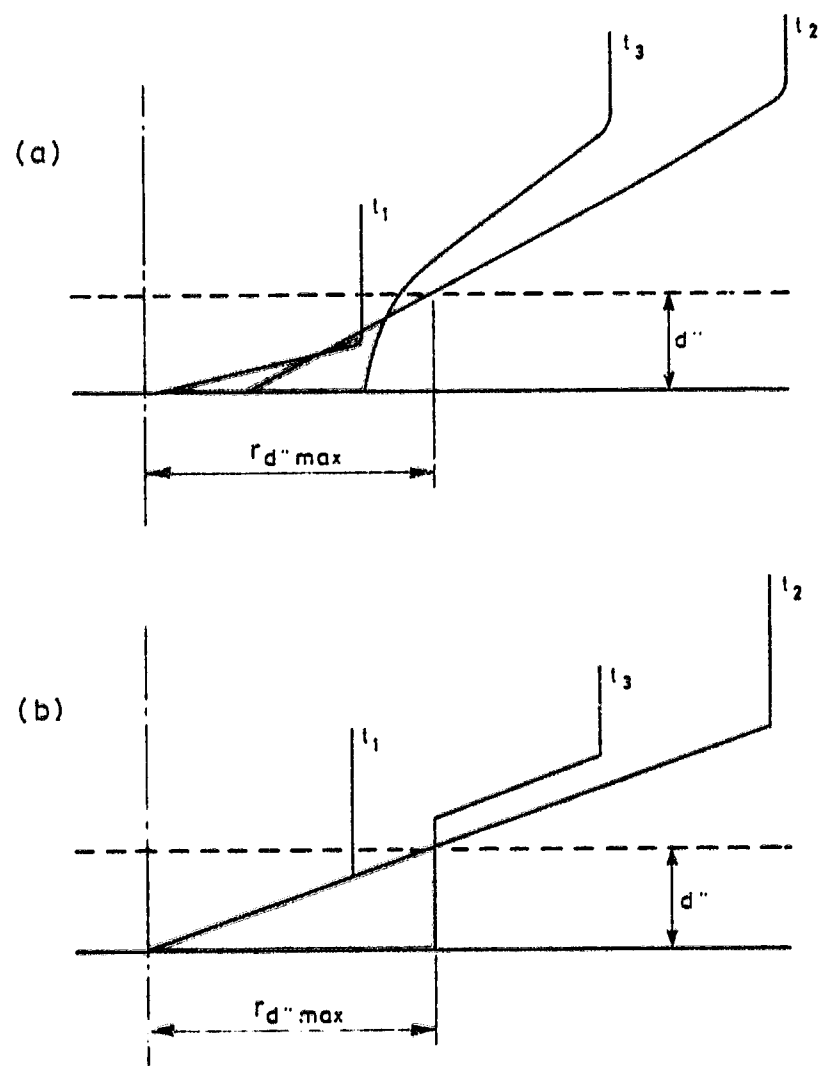


FIG. 8.29 MICROLAYER PROFILES

(a) Observed

(b) Simplified

both methanol and ethanol runs. Thus, in units of mm and mm³, equation 8.1 becomes

$$V_{LE}^* = 3,55 \times 10^{-4} r_{d''max}^2 \quad (8.2)$$

The volume of vapour due to the apparent microlayer evaporation is

$$V_{VE}^* = (\rho_L/\rho_V) V_{LE}^* \quad (8.3)$$

The calculation of V_{VE}^* and its contribution to the maximum bubble volume V_{max} , is given for all runs in Table 8.4. The values of $r_{d''max}$ and of V_{max} were read from the appropriate figures in Section 8.4. The density ratios ρ_L/ρ_V are given in Appendix 15.

TABLE 8.4

CONTRIBUTION OF VISIBLE MICROLAYER EVAPORATION TO
MAXIMUM BUBBLE VOLUME

Run	ΔT_{sub} K	$r_{d''max}$ mm	V_{LE}^* mm ³	ρ_L/ρ_V	V_{VE}^* mm ³	V_{max} mm ³	$\frac{V_{VE}^*}{V_{max}} \cdot 100$ per cent
M10	10,2	2,0	0,0014	2330	3,3	1640	0,20
M13	9,9	3,5	0,0044	2480	10,9	1410	0,77
M14	-0	3,0	0,0032	1810	5,9	1180	0,50
M19	20,2	1,8	0,0012	1810	2,2	165	1,33
M20	-0	2,1	0,0016	1430	2,3	625	0,37
E 8	2,6	1,8	0,0012	1700	2,0	293	0,68
E12	0	2,0	0,0014	1290	1,8	617	0,29

The visible microlayer contribution, V_{VE}^*/V_{max} , is in all cases extremely small. For the runs at bulk saturation, where V_{max} is

equal to the total volume of vapour released into the attached bubble, the contribution is less than or equal to 0,5 per cent. Thus q_{ML}^{i*}/q_{LH}^i , the apparent microlayer contribution to the total area-of-influence latent heat transport, is similarly 0,5 per cent or less. For the subcooled boiling runs, in view of bubble recondensation and of the values of V_{VE}^*/V_{max} , it may be safely concluded that the same result holds. Thus we have for all runs

$$q_{ML}^{i*}/q_{LH}^i \leq 0,005 \quad (8.4)$$

The net microlayer contribution, q_{ML}^i/q_{LH}^i , cannot be estimated without further information on microlayer liquid replenishment.

For purposes of later discussion of inflow, the vapour volume V_V^m , resulting from the complete evaporation of one stagnant microlayer of maximum size will now be estimated. The microlayer is assumed to have a linear profile, a base radius of r_{max} ($= \frac{1}{2} D_{cmax}$) and an edge thickness, d_{max} , equivalent to the highest-order fringe recorded. (In actual fact d_{max} is a little larger and may be obtained by extrapolation of the profile from the highest-order fringe to D_{cmax} ; the correction ranges from 1 to 14 per cent which, for the present order-of-magnitude estimate, is negligible.) By analogy with equations 8.1 to 8.3, the volume of vapour evolved is given by

$$V_V^m = (\pi/6)(\rho_L/\rho_V) D_{cmax}^2 \cdot d_{max} \quad (8.5)$$

The evaluation of V_V^m is given for all runs in Table 8.5. The values of D_{cmax} , d_{max} and V_{max} were again read from the appropriate figures in Section 8.4.

TABLE 8.5

CONTRIBUTION OF COMPLETE EVAPORATION OF ONE
STAGNANT MICROLAYER TO MAXIMUM BUBBLE VOLUME

Run	ΔT_{sub} K	D_{cmax} mm	d_{max} mm	V_V^m mm ³	V_{max} mm ³	$\frac{V_V^m}{V_{\text{max}}} \cdot 100$ per cent
M10	10,2	9,22	$0,664 \times 10^{-3}$	68,9	1640	4,2
M13	9,9	10,75	$0,498 \times 10^{-3}$	75,0	1410	5,3
M14	~0	8,40	$0,498 \times 10^{-3}$	33,4	1180	2,8
M19	20,2	5,02	$0,332 \times 10^{-3}$	7,9	165	4,8
M20	0	6,72	$0,498 \times 10^{-3}$	16,7	625	2,7
E 8	2,6	5,60	$0,324 \times 10^{-3}$	9,0	293	3,1
E12	~0	5,98	$0,486 \times 10^{-3}$	11,8	617	1,9

The ratio V_V^m/V_{max} , that is, the contribution of one complete stagnant microlayer to the maximum bubble volume ranges from 2 to 3 per cent for the saturated boiling runs, and from 3 to 5 per cent for the subcooled runs. These amazingly low values indicate that, if the microlayer contribution to the total bubble growth is to be predominant (as the microthermometry studies of Section 4.2 indicate) then the microlayer replenishment must be such that numerous equivalent liquid volumes pass through one microlayer during its lifetime. The possibility of such flows is discussed in the next chapter.

8.6 COMPARISON OF EXPERIMENTAL AND PREDICTED MICROLAYER THICKNESSES

It was shown in Section 4.2.6 that the theoretical equations predicting the microlayer thickness for hemispherical bubbles are all reducible to the form

$$d_o = \text{const.} (v_l t_g)^{\frac{1}{2}} \quad (8.6)$$

where d_o is the initial microlayer thickness, that is, the edge thickness undiminished by evaporation, and t_g is the corresponding bubble growth time. The constant was 0,8 for the equation of Cooper and Lloyd [100], 0,89 for that of Olander and Watts [147] and 1,27 for that of van Ouwert [110].

Equation 8.6, with the constant taken as 1,00, is tested below for two bubble growth times, firstly, for 1 ms, at which stage the bubbles were still roughly hemispherical, and secondly, for t^* , the growth time required to reach the maximum microlayer radius, at which stage all bubbles had departed considerably from the hemispherical shape. Results are shown in Table 8.6. The values of t^* were read from the figures of Section 8.4; the evaluation of the kinematic viscosity, ν , at temperatures of $(T_w + T_{\text{sat}})/2$ is given in Appendix 16.

The predicted thicknesses in all cases exceed the experimental thicknesses by factors of greater than 100. No explanation for the discrepancy can at this stage be advanced. The possibility of gross measurement errors in the experimental values is discussed in the next chapter.

TABLE 8.6

PREDICTED AND EXPERIMENTAL MICROLAYER THICKNESSES

Run	$v_L = \frac{\mu}{\rho}$ m ² /s	t^* (= t for D_{max}) ms	$d_o (t = 1 \text{ ms})$		$d_o (t = t^*)$	
			predicted (νt) ^{1/2}	exptl	predicted (νt) ^{1/2}	exptl (highest fringe)
			μm	μm	μm	μm
M10	$0,498 \times 10^{-6}$	19,0	22,5	0,08	97,5	0,664
M13	$0,512 \times 10^{-6}$	16,5	22,6	0,08	91,9	0,498
M14	$0,495 \times 10^{-6}$	15,5	22,2	0,08	87,4	0,498
M19	$0,485 \times 10^{-6}$	7,5	22,0	0,08	60,2	0,332
M20	$0,459 \times 10^{-6}$	11,0	21,1	0,08	80,2	0,498
E 8	$0,745 \times 10^{-6}$	8,0	27,5	0,08	77,1	0,324
E12	$0,682 \times 10^{-6}$	15,5	26,1	0,08	102,8	0,486

The results of the seven runs presented in the present chapter were checked, by examination of the projected ciné frames, for some 20 additional runs. All these confirmed the very low microlayer thickness, the very low apparent microlayer evaporation contribution and the presence of microlayer liquid replenishment.

CHAPTER 9DISCUSSION9.1 INTRODUCTION AND OUTLINE

The last chapter has shown that under the experimental conditions of the present study microlayers are of sub-micron thickness, that microlayer thinning (visible evaporation) is restricted to the inner layer edge with the thickness elsewhere remaining constant or increasing with time, and that the contribution of this visible evaporation to the total vapour flow into bubbles is negligible. These findings are in apparent conflict with the results of surface thermometry studies and, in particular, with those of Cooper and Lloyd [99, 100] who, for experimental conditions fairly close to those of the present study, calculated the microlayer evaporation to be essentially the sole contributor to the bubble vapour input. On the assumption of a stagnant microlayer, their calculated initial layer thicknesses exceeded by factors of the order of 100 the thicknesses measured in the present study. The results of the two studies can be reconciled, as has already been indicated, if microlayers are assumed to be continuously replenished, such that a liquid volume equivalent to numerous measured microlayer volumes passes through and is evaporated from the layer in one bubble lifetime. Evidence for microlayer replenishment, in the form of a demonstration of microlayer thickening, has already been given in the last chapter.

In order to investigate this replenishment further, Section 9.2 below provides a numerical analysis of the rate of evaporation of thin microlayers (as observed in this study),

but on the assumption of stagnant liquid. It is shown that the observed microlayers should, even on the most extreme of unfavourable assumptions, evaporate completely in less than 1 ms. Their persistence at constant thickness for some 30 ms thus suggests liquid replenishment, independently of previous evidence. Further, the analysis demonstrates that, in the presence of sufficient replenishment, a liquid volume equivalent to numerous microlayers could, under the conditions of our experiments, be vaporised within a bubble lifetime.

The mechanism of liquid flow into and within the microlayer is not examined. Section 9.3, however, provides photographs of lateral bubble interaction which show that the forces causing microlayer inflow appear to be of the same order of magnitude as those associated with bubble deformation and departure.

Section 9.4 examines such overall forces for the isolated bubbles recorded in this study. The forces of inertia, surface tension, buoyancy and viscosity are evaluated. During the first few milliseconds of bubble growth inertia forces predominate and surface tension forces are high. Throughout the remaining bubble life buoyancy forces predominate. It is the buoyancy forces which are considered to be responsible for the continuous lifting of the bubbles off their microlayers, or, in other words, for the microlayer in-flow.

Finally, Section 9.5 discusses the microlayer interferometry studies of Judd [150, 151] which were initiated in response to the published preliminary results of the present study [152]. In his first study [150], Judd used conventional interferometry illumination and obtained results in agreement with those here reported. In his second study [151], he employed laser illumination and obtained contradictory results. The possibility of an outright error either in the present study or in Judd's laser illumination study is discussed.

9.2 THEORETICAL RATE OF EVAPORATION OF STAGNANT MICROLAYER

The microlayer histories presented in Chapter 8 consistently show that over some portion of the layer, generally midway between the inner and outer edge, the thickness remained approximately constant with time. (The locus of the thickness 0,332 μm in Run M10 (Figure 8.5) serves as an example.) This behaviour was interpreted as the dynamic equilibrium between evaporation at the microlayer surface and replenishment by radially inflowing liquid. In order to obtain some idea of the magnitude of the replenishment, and, simultaneously, to dispel doubts on the sufficiency of the heater input and capacity, the rate of evaporation of a stagnant layer of such thickness is now analysed.

The problem to be solved is one of transient conduction in three regions in contact, the first, the glass substrate, being semi-infinite, the second, the heating film, having internal heat generation, and the third, the evaporating microlayer, having a moving boundary. Since a numerical method of solution is indicated, no significant simplifications, other than the assumption of one-dimensional conduction, are necessary. Before formally setting up the differential equations of the problem, a realistic boundary condition for the evaporating microlayer face must be defined.

9.2.1 Boundary Condition at Microlayer Liquid-Vapour Interface

In the analysis of bubble growth in superheated liquids the temperature T_{sat} has customarily been ascribed to the liquid-vapour interface [149, 185, 186]. The same boundary condition was applied by Torikai et al [105] in their exploratory estimates of conduction across the microlayer.

This treatment neglects the resistance to heat (or mass)

transfer at the liquid-vapour interface itself. The requirements of superheating at the surface of liquids evaporating into their saturated vapours had, of course, been known from kinetic theory for many years. For the case of non-metallic liquids evaporating at low rates and at atmospheric pressure, however, both the predicted and the measured surface temperatures were found to exceed T_{sat} by a fraction of 1°C [187 - 190]; thus the interface temperature drop was negligible compared with the overall film temperature difference. With rapidly evaporating bubble interfaces this is not necessarily the case and the application of the T_{sat} boundary condition can lead to serious overestimates of the evaporation rate.

Considerations of kinetic theory and various restrictive assumptions have led several early workers (for example [190, 191]) to the following equation for the net rate of interphase mass transfer:

$$w = n (M/2\pi RT_s)^{1/2} (P_s^0 - P) \quad (9.1)$$

Here w is the mass rate of evaporation per unit area, n is the evaporation coefficient (to be discussed below), the temperatures and pressures are indicated in Figure 9.1, and P_s^0 is the vapour pressure corresponding to T_s .

A more rigorous treatment, incorporating corrections for the bulk movement of vapour away from (or towards) the liquid surface was given by Schrage [192] and results, when applied to the present case (Figure 9.1), in:

$$w = \{n/(1 - \frac{1}{2}n)\} (M/2\pi R)^{1/2} \{P_s^0/T_s^{1/2} - P/T_{\text{sat}}^{1/2}\} \quad (9.2)$$

The identical expression has been deduced from Schrage's work in the condensation study of Mills and Seban [193].

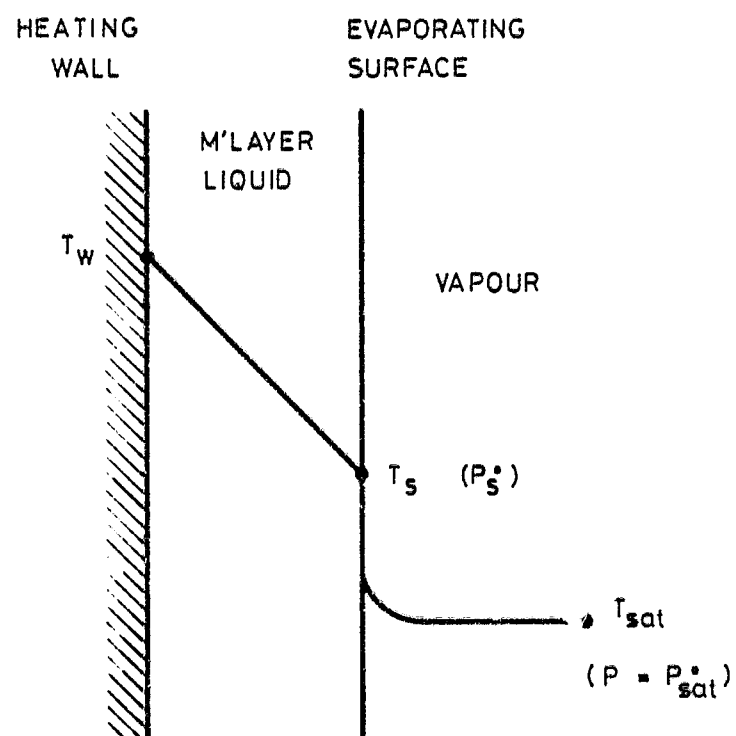


FIG. 9.1 TEMPERATURE DISTRIBUTION:
MICROLAYER AND VAPOUR

Equation 9.2, although itself not entirely rigorous and amenable to further refinement [194], is adopted in the present study.

The problem now remaining is the evaluation of n , the evaporation (or condensation) coefficient. In kinetic terms, the coefficient was originally seen as the fraction of the number of molecules which, under equilibrium conditions, are condensed upon striking the condensate surface. In practice, however, it has come to signify the ratio of the measured evaporation rate to the theoretical maximum evaporation rate and is thus dependent on the evaporation equation adopted. A voluminous and conflicting literature (reviewed in [15, 16, 192]) exists on the subject. It appears that in many early studies giving low fractional values of the coefficient, the measuring techniques were inadequate, or the experimental set up and the adopted evaporation equation (for example, equation 9.1) were incompatible. While some low values have been explained in terms of restraint against molecular rotation in the liquid [195], it is generally found that over the years and with increasing experimental sophistication, the evaporation coefficient has moved progressively closer to unity. A class of liquids does, however, exist, our test liquids methanol and ethanol included, which has given consistently low values.

Table 9.1 shows the available evaporation coefficients for methanol and ethanol.

TABLE 9.1

EVAPORATION COEFFICIENTS

Liquid	Source	T, °C	η
Methanol	Delaney et al [196]	7	0,017
		-10	0,022
		-27	0,030
Methanol	[196] from data of Baranaev	-3	0,017
	[196] from data of Harkins and Roberts	0	0,026
Ethanol	Buka [197]	12,4 to 15,9	0,023 to 0,029
	von Bogdandy et al [198]	0	0,036

In assessing the validity of these low values it should be noted that for n-butyl alcohol Slegers and Seban [199] have demonstrated an evaporation coefficient equal to or greater than 0,25, and further, have shown the older value of 0,025 (for sec-butyl alcohol) [200] to have been due to the presence of non-condensables.

While all evaporation coefficients lower than unity are open to doubt, values ranging from 0,01 to 0,1 will, for purposes of estimating a minimum microlayer evaporation rate, be considered in the present study.

9.2.2 Formal Definition of Problem and Method of Numerical Solution

The problem to be solved may now be formally defined as summarised in equations 9.3 to 9.13 below and in Figure 9.2.

Conduction

In region 1 (glass):

$$\partial T / \partial t = \alpha_1 \partial^2 T / \partial x^2, \quad 0 < x \leq b-0 \quad (9.3)$$

In region 2 (SnO₂ film):

$$\partial T / \partial t = \alpha_2 \partial^2 T / \partial x^2 + q^* / \rho C, \quad b+0 \leq x \leq c-0 \quad (9.4)$$

where q^* is the internal heat generation per unit time and volume.

In region 3 (liquid microlayer):

$$\partial T / \partial t = \alpha_3 \partial^2 T / \partial x^2, \quad c+0 \leq x \leq s \quad (9.5)$$

Boundary conditions

Continuity of temperature at $x=b$ and $x=c$:

$$T(b-0, t) = T(b+0, t) \quad (9.6)$$

$$T(c-0, t) = T(c+0, t) \quad (9.7)$$

Continuity of heat flux at $x=b$ and $x=c$:

$$k_1 \partial T(b-0, t) / \partial x = k_2 \partial T(b+0, t) / \partial x \quad (9.8)$$

$$k_2 \partial T(c-0, t) / \partial x = k_3 \partial T(c+0, t) / \partial x \quad (9.9)$$

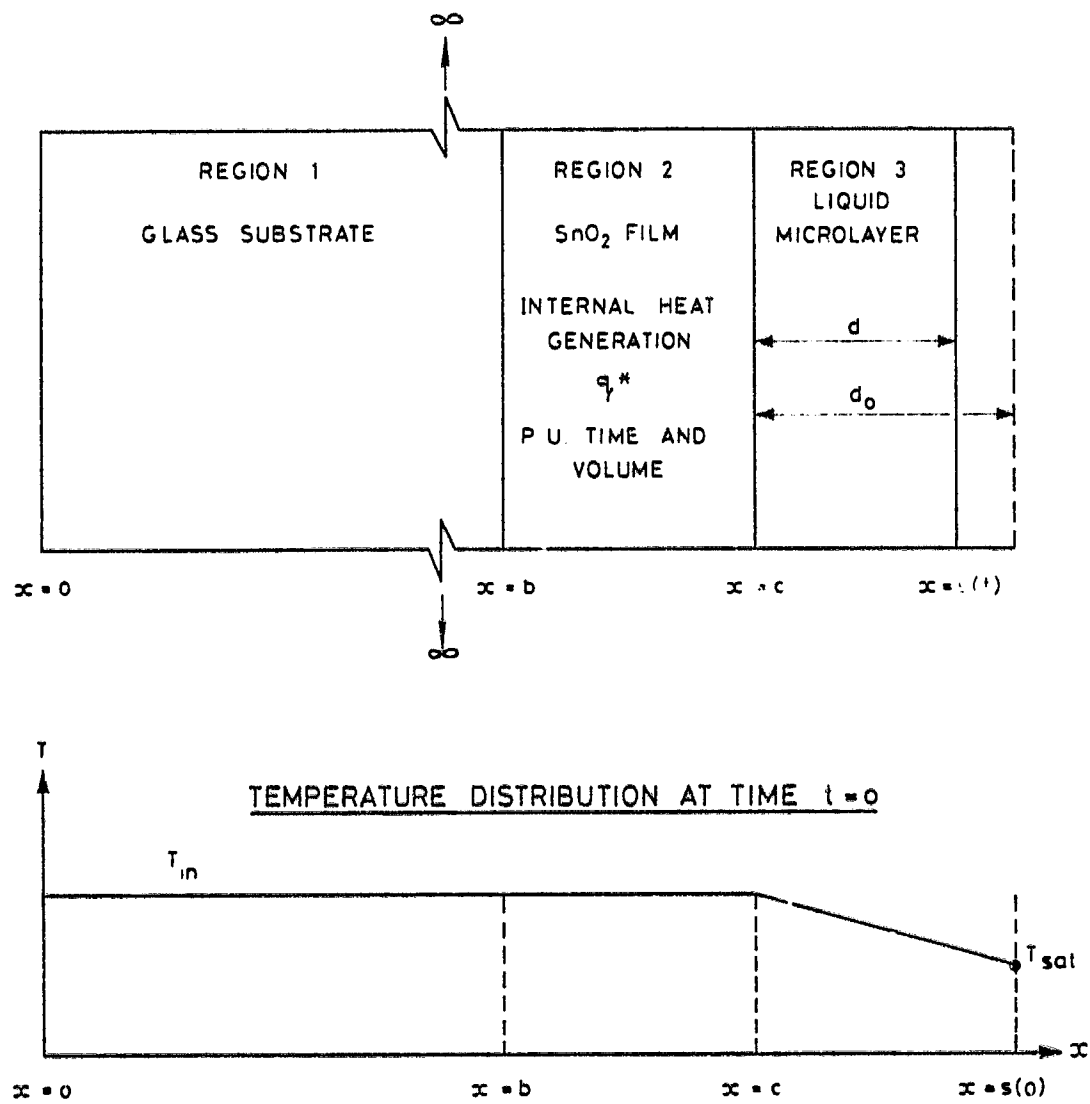


FIG. 9.2 DEFINITION OF MICROLAYER CONDUCTION PROBLEM

'Semi-infinite' extent of region 1:

$$T(0,t) = T_{in} \quad (9.10)$$

where $T_{in} = T(0,0)$ is the initial temperature, see Figure 9.2,

$$\partial T(0,t)/\partial x \approx 0 \quad (9.11)$$

that is, the temperature gradient at $x=0$ remains unchanging at its initial value of zero (irrespective of conduction and evaporation in region $x>0$); this condition is satisfied by making region 1 sufficiently thick.

Evaporation at $x=s$:

$$-k_3 \frac{\partial T(s,t)}{\partial x} = \left[\frac{n\lambda}{1-n} \right] \left[\frac{M}{2\pi R} \right]^{\frac{1}{2}} \left[\frac{p_s^0}{T_s} - \frac{p_{sat}}{T_{sat}} \right] \quad (9.12)$$

see equation 9.2 and Figure 9.1.

Moving boundary at $x=s$:

$$ds/dt = (k_3/\lambda \rho_L) \cdot \partial T(s,t)/\partial x \quad (9.13)$$

where the temperature gradient is given by equation 9.12.

Initial conditions

$$T(x,0) = T_{in} , \quad 0 \leq x \leq \infty \quad (9.14)$$

$$T(s,0) = T_{sat} \quad (9.15)$$

$$-\partial T(x,0)/\partial x = (T_{in} - T_{sat})/d_0 , \quad \infty \leq x \leq s \quad (9.16)$$

This is the initial temperature distribution shown in Figure 9.2. The method of solution of the problem was such that any

other initial distribution could have been handled. The present distribution was chosen for simplicity, and the value of T_{in} was varied to simulate various initial conditions.

Equations 9.3 to 9.5, subject to the conditions of equations 9.6 to 9.13, were solved by a modified Crank-Nicolson implicit finite-difference method. The standard Crank-Nicolson method [201, 202] involves the subdivision of the x and t regions into increments of fixed size (with the increment boundaries intersecting at so-called node points), the replacement of the differential equations at each node point with an approximating algebraic equation, these giving a set of simultaneous linear equations, and the forward propagation of the solution in a step-by-step fashion for successive time increments. A major modification of the method, applied to the successively thinner region 3, consisted of the re-evaluation, for each time step, of the size of the x increments, while keeping their numbers constant. Such variable-space networks have been employed, coupled with somewhat different numerical schemes, in freezing and melting problems [203, 204].

The temperature gradients at the internal boundaries (equations 9.8 and 9.9) and at the evaporating boundary (equation 9.12) were approximated by three-point difference equations having discretisation errors consistent with those of the Crank-Nicolson method. The derivation of the equations (given in Appendix 18) followed Saul'yev's method of fictitious nodes [205].

The difference equation for the evaporating boundary ($x=s$) was, in keeping with equation 9.12, non-linear in $T(s, k+1)$, where $k+1$ denotes the time step to be solved. (The equations at all other nodes were linear.) In order to maintain linearity, and thus to profit from the fast algorithms

available for the solution of simultaneous linear equations, the temperature $T(s,k+1)$ was determined by an iterative procedure using the known temperature $T(s,k)$ as a starting value. With $T(s,k+1)$ and all other temperatures known, the instantaneous evaporative heat flux and the incremental movement of the boundary for each time step could be calculated. The computation was terminated when the microlayer had thinned down to less than 5 per cent of its original thickness. Details of the entire procedure are given in Appendix 17.

In view of the several novel features of this scheme a formal stability analysis should ideally have been performed. This was not attempted; instead, stability was investigated by numerical experiments. These showed that for the 'reasonable' step sizes normally employed with the Crank-Nicolson method, the scheme was unstable. Stability was finally obtained with exceedingly small time steps, typically such that some 10 000 steps were necessary for the solution of one microlayer evaporation problem. This approach was suggested by Starfield [] who had experienced (and overcome) similar difficulties in the solution of a rock-cooling problem involving an evaporative boundary condition on a wet mine-stope face. Checks on the accuracy and general reliability of the numerical solutions are discussed in Appendix 18.

9.2.3 Results and Assessment of Microlayer Replenishment

Computational results are now given for the evaporation, under the conditions of Run M10, of a stagnant methanol microlayer of initial thickness 0,332 μm .

Figure 9.3 shows the layer thickness versus time for an initial heater temperature of 72 $^{\circ}\text{C}$ (that is, for the measured T_w of Run M10) and for three values of the evaporation

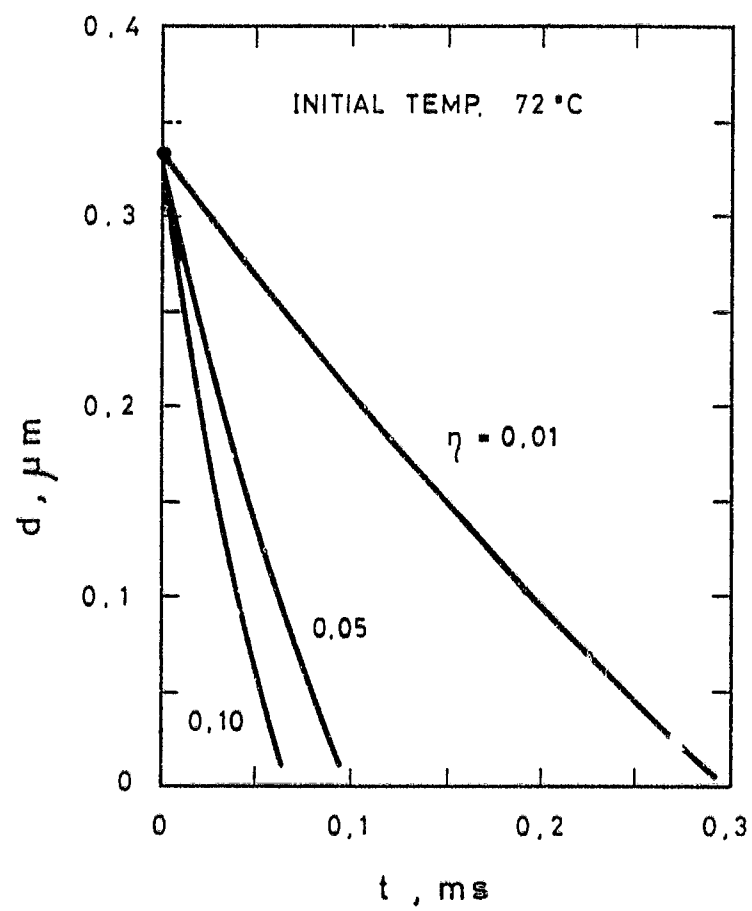


FIG. 9.3 RATE OF EVAPORATION OF STAGNANT
MICROLAYER; INFLUENCE OF EVAPORA-
TION COEFFICIENT
 Conditions of Run M10

coefficient. Even in the most unfavourable case ($\eta = 0,01$) the predicted time for the total evaporation of the layer is a mere 0,3 ms. In Run M10 the measured thickness of 0,332 μm persisted without significant change for 30 ms (Figure 8.5). Clearly this invariant thickness cannot be attributed to a heater of insufficient capacity and input acting on a stagnant layer; even in the absence of the evidence of layer thickening the notion of liquid replenishment would suggest itself.

Figure 9.4 shows the computed heater surface temperature, T_w , and the microlayer face temperature, T_s , for the evaporation processes given in Figure 9.3. The microlayer face temperature, initially at T_{sat} ($= 35^\circ\text{C}$) immediately jumps to a value quite close to the heater surface temperature, T_w , and, as the layer thins down, approaches T_w progressively more closely. (With exceedingly thin layers, of the order of 0,05 μm , computation errors become significant and T_w and T_s become indistinguishable.) Application of the simplified boundary condition $T_s = T_{\text{sat}}$, discussed in Section 9.2.1, would thus have led to severe overestimates of the evaporation rate.

It might, with reference to Figures 9.3 and 9.4, be noted that the evaporative heat flux varied about a mean of 900 kW/m^2 for the case of $\eta = 0,01$, and about a mean of 4000 kW/m^2 for $\eta = 0,10$. The heater input corresponded to a mean flux of 70 kW/m^2 . The input thus served primarily to raise the heater temperature during the waiting period and not to evaporate the microlayer directly.

Figure 9.5 shows the rate of evaporation of the same stagnant microlayer for initial heater temperatures, T_{in} , of 72°C , 62°C and 52°C and for an evaporation coefficient of 0,01 throughout. The two lower temperatures can be taken as representing errors of 10°C and 20°C in the measurement of T_w , or, alternatively, as being equivalent to severely

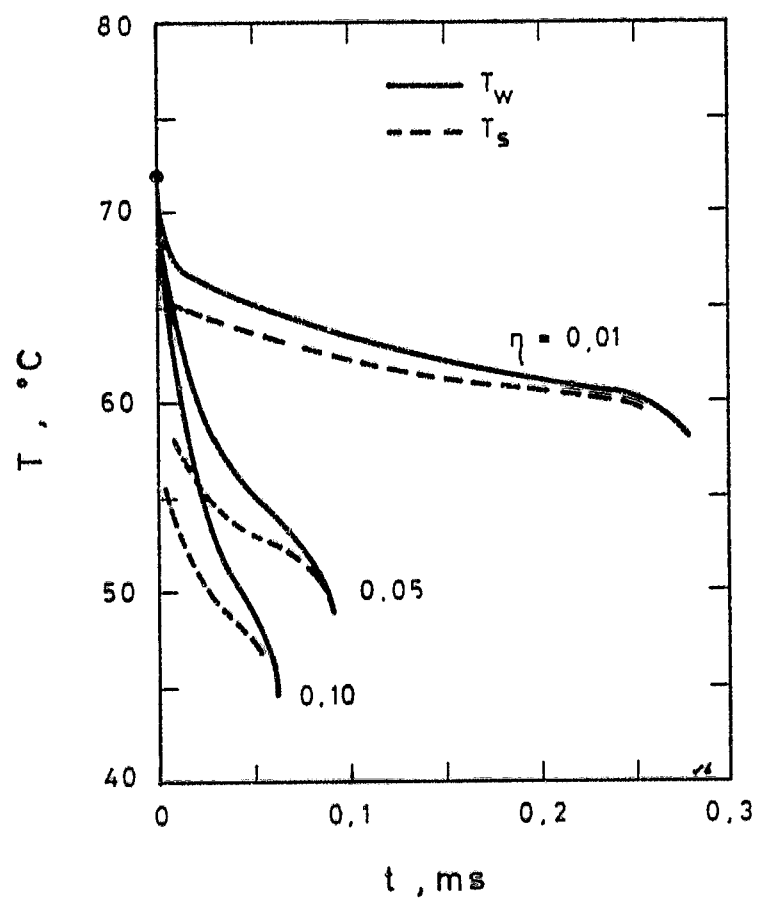


FIG. 9.4 HEATER SURFACE TEMPERATURE AND
LIQUID-VAPOUR INTERFACE TEMPERATURE;
EVAPORATION OF STAGNANT LAYER
 Conditions of Run M10

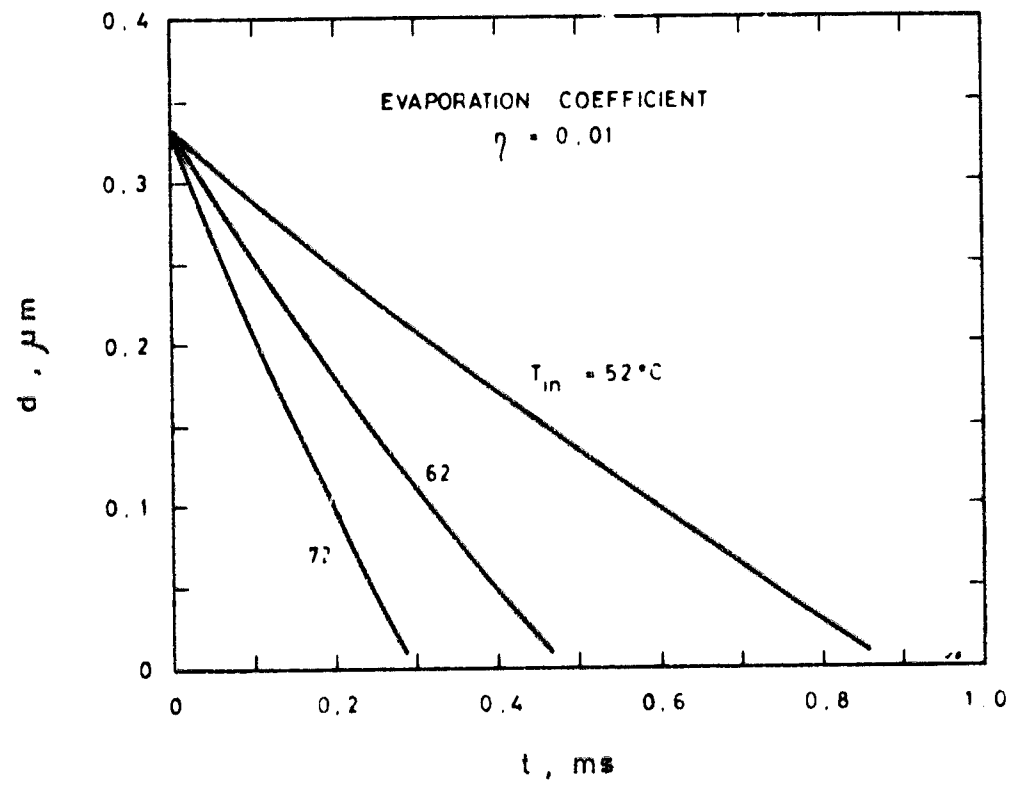


FIG. 9.5 RATE OF EVAPORATION OF STAGNANT MICROLAYER;
INFLUENCE OF INITIAL HEATER TEMPERATURE

lowered temperatures in the surface region of the heater only. In the extreme case of $T_{in} = 52^{\circ}\text{C}$ (and $\eta = 0,01$) the layer still evaporates to completion in less than 1 ms. Thus even under unrealistically adverse conditions a stagnant microlayer could not persist for 30 ms.

Figure 9.6 gives the temperature distribution within the heater at the time of completion of evaporation, for the three cases treated in Figure 9.5. Starting with the distribution resulting from the evaporation of one layer from a heater with a uniform initial temperature of 72°C , it is clear that several such evaporation processes would be necessary to produce a situation thermally equivalent to a uniform heater temperature of 62°C . At that stage, however, a stagnant layer would still evaporate completely in 0,5 ms. By the same argument, numerous evaporation processes would be necessary to produce the situation equivalent to a uniform $T_{in} = 52^{\circ}\text{C}$, at which stage a stagnant layer would still evaporate completely in less than 1 ms. It would appear then that a large number of stagnant microlayers could in immediate succession have been evaporated from the heater within 30 ms.

In view of this finding and of the previous argument demonstrating the necessity of microlayer replenishment, it would seem feasible that within one bubble lifetime a total liquid volume equivalent to numerous microlayers could have flowed through and have evaporated from each of our experimentally observed layers. Thus, despite the extremely low and near-constant microlayer thicknesses observed throughout this study, the total evaporation from each microlayer might well have been a significant contributor to the vapour input to a bubble. The thickness measurements of the present study and the evaporation heat flux measurements of Cooper and Lloyd [99, 100] thus appear to be reconciled.

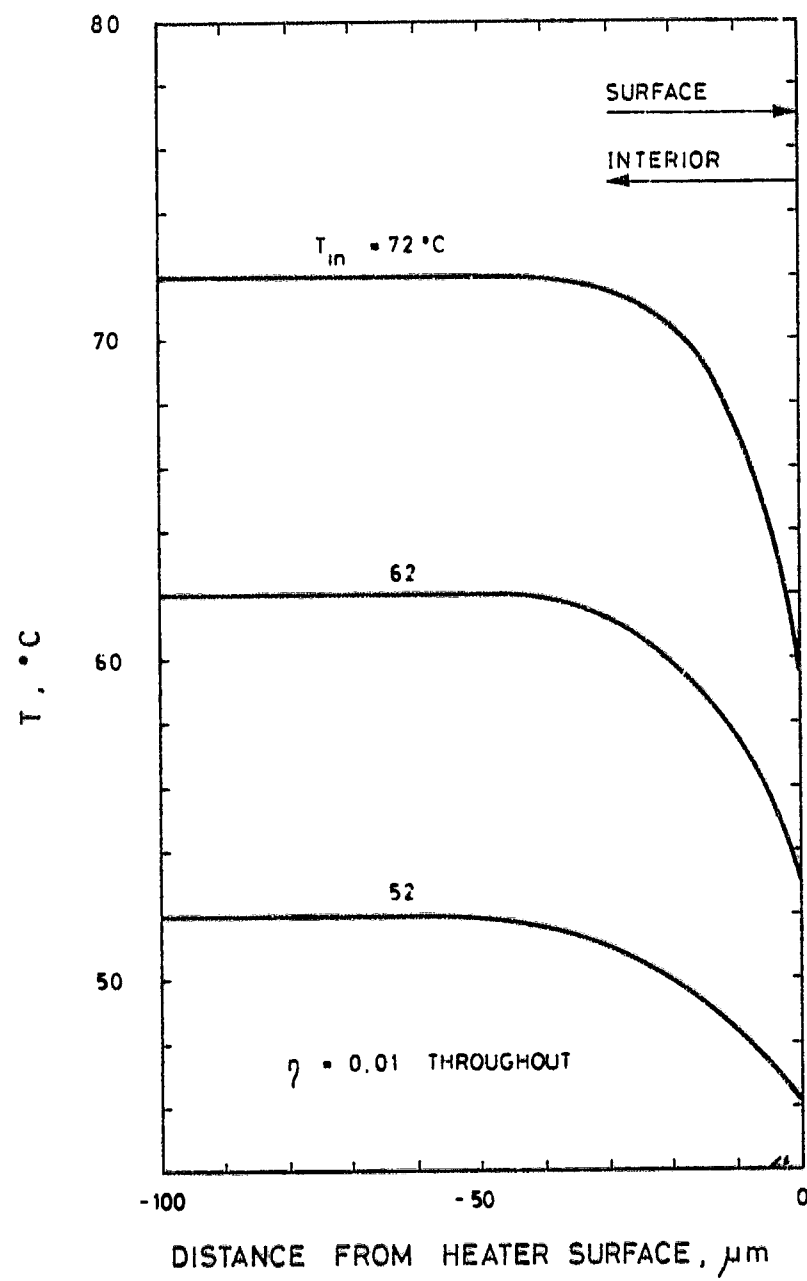


FIG. 9.6 HEATER TEMPERATURE DISTRIBUTION ON
COMPLETION OF EVAPORATION;
SEE FIG. 9.5

9.3 MICROLAYER FLOODING DURING LATERAL BUBBLE INTERACTION

While the mechanism of replenishment flow into and within the microlayer was not analysed in this work, it was felt intuitively that very high pressure gradients parallel to the surface would be required to cause any significant flow in such an extremely narrow space. Photographic records presented in this section, however, suggest that the required forces are no higher than those involved in bubble deformation and departure.

Figure 9.7 shows two bubbles, from naturally occurring nucleation sites, interacting laterally during subcooled boiling at elevated heat flux. (Test conditions are given in the figure caption.) The right-hand bubble in growing causes a radial outflow of liquid which slightly deforms the neighbouring bubble, floods its microlayer and dry patch, and lifts it completely off its base. The flooding, it should be noted, raises the microlayer thickness beyond the limit of fringe visibility, that is, to a value greater than $4\text{ }\mu\text{m}$ (see Section 6.3.4), or greater than 8 times the microlayer edge thickness, and, secondly, sweeps from the outer to the inner microlayer edge in some 6 ms. The volumetric flow rate would thus seem equivalent to at least 40 microlayer volumes in one bubble lifetime of 30 ms.

It would appear then that for isolated bubbles the microlayer replenishment postulated earlier could be due to forces of the same order of magnitude as the forces associated with bubble deformation and departure. In the next section such forces will be evaluated, and examined for compatibility with a microlayer inflow (or bubble lift-off) mechanism.



FIG. 9.7 MICROLAYER FLOODING DURING LATERAL
BUBBLE INTERACTION. RUN M11.
 Methanol, $q/A = 87,1 \text{ kW/m}^2$, $P = 25,6 \text{ kN/m}^2$,
 $\Delta T_{\text{sub}} = 7,9 \text{ K}$, interval between frames 2,15 ms.

9.4 FORCES ACTING ON BUBBLES

9.4.1 Definition and Significance of Forces

The development, given in this section, of a set of forces relating to bubble deformation and lift-off, closely follows the work of previous investigators [100, 139, 184] and in particular, the detailed treatment given by Cooper and Lloyd [100].

Consideration is given initially to a spherical bubble of radius R growing or collapsing in an infinite liquid. For such a bubble the difference between the pressure within the vapour (assumed uniform) and the pressure in the liquid at infinity is given by the modified Rayleigh equation as follows:

$$\Delta P = \rho_L \{ R(d^2R/dt^2) + \frac{3}{2} (dR/dt)^2 \} + 4\mu (dR/dt)/R + 2\sigma/R \quad (9.17)$$

The first term accounts for the contribution of the inertia forces per unit area to the total pressure difference. The second term is the contribution of the viscous stresses and the third term is the usual surface tension stress. The terms are labelled P_i , P_v and P_s .

$$P_i = \rho_L (R\ddot{R} + 1.5 \dot{R}^2) \quad (9.18)$$

$$P_v = 4\mu \dot{R}/R \quad (9.19)$$

$$P_s = 2\sigma/R \quad (9.20)$$

Further, the following buoyancy (or gravity) term may be defined:

$$P_g = (\rho_L - \rho_V)gR \quad (9.21)$$

This may be regarded as the buoyancy force on the bubble divided by a typical area (πR^2). The buoyancy term cannot be related to the other three terms in a single equation.

The four force terms were developed for spherical bubbles in an infinite liquid. In applying these to the oblate bubbles of this study the radius R is replaced, as in the work of Johnson et al [139], by the equivalent spherical radius, that is, by $D_s/2$ (see equation 7.2). The significance of the forces for bubbles in the vicinity of a surface has been discussed in detail by Cooper and Lloyd [100, 184] and, to some extent, tested experimentally by Johnson et al [139]. In summary:

The inertia term P_i relates to the bubble's tendency to assume a hemispherical shape. In the absence of all forces other than inertia a hemispherical bubble will remain attached to the surface even as growth ceases [100].

The viscous term P_v relates to the bubble's tendency to form a microlayer. Cooper and Lloyd [100] point out that the term $(\mu \dot{R}/R)$ is the viscous stress which would result if the liquid formed a boundary layer of thickness R and of linear velocity profile. If this stress is small compared with the inertia stress, the layer will be small compared with R .

The surface tension term P_s relates to the tendency of the bubble to assume a spherical shape. If, in the presence of a microlayer, the surface tension term is large compared with the inertia term, the layer will tend to curl away at the outer edge and the bubble will tend to pinch off. Ideally the P_s stress should be evaluated in terms of the radius of curvature at the outer layer edge. In the form of equation 9.20, P_s may be regarded as the lower limit of the stress tending to round off the bubble at the microlayer.

The buoyancy term P_g relates to the tendency of the bubble to lift off. In the presence of a microlayer, a high buoyancy to inertia ratio implies a tendency towards microlayer thickening, that is, towards microlayer in-flow.

9.4.2 Computation and Results

In order to evaluate the bubble forces with reasonable accuracy it was necessary, firstly, to correct the radius versus time data such that the time origin ($t=0$) corresponded at least approximately to the instant of nucleation of the bubble. (Throughout this section the term 'radius' and the symbol 'R' refer to the equivalent spherical radius). It will be recalled from Chapters 7 and 8 that the time $t=0$ was arbitrarily assigned to the ciné frame immediately preceding the frame in which a bubble first appeared. The true $t=0$ lay somewhere in the time interval between these first two frames. The original R versus t data were thus hand-smoothed, extrapolated to $R=0$ and shifted on the time axis such that $R=0$ and $t=0$ coincided.

Radius values at convenient values of time were read from the corrected hand-smoothed curves. The first and second derivatives, $\dot{R}$ and $\ddot{R}$, as functions of time, were established for these data.

Since the data appeared to follow no readily-recognisable function some numerical method of differentiation was called for. Various schemes [100, 139] were tested and the following finally adopted:

A non-linear equation of the form

$$R = at^h \quad (9.22)$$

was fitted by least-squares regression to overlapping segments of the curve. The first group of successive data points, odd

in number (3, 5, 7, 9 or 11), and starting with data point 1 was fed to a regression program (Appendix 19) and the fitted values of the coefficients a and b applied at the central point. For example, for a 5-point fit, R (equation 9.22) and $\dot{R}$ and $\ddot{R}$ (equations 9.23 and 9.24 below) were calculated for data point 3.

$$\dot{R} = abt^{(b-1)} \quad (9.23)$$

$$\ddot{R} = ab(b-1)t^{(b-2)} \quad (9.24)$$

The procedure was then repeated for the next group of points, this time starting with point 2 and, for a 5-point fit, giving R , $\dot{R}$ and $\ddot{R}$ at point 4. The procedure was successively applied to the whole range of data and the bubble forces (with physical properties evaluated at the saturation temperature) calculated at each point.

For each curve-fitting operation the regression program provided, in addition to the fitted parameters a and b , their variance-covariance matrix. From this, by the methods outlined in Appendix 19, the confidence limits on the bubble forces could be calculated. The use of this facility in trial computations established the most suitable number of points per fit. The result, on all data sets, was an 11-point fit for the inertia forces and a 3-point fit for the remaining forces. Further details of the fitting procedure are given in Appendix 1'. Physical properties entering into the bubble force equations are given in Appendices 15, 16 and 17.

The bubble forces thus evaluated are shown for Runs M10, M20 and E12 in Figures 9.8 to 9.10. Test conditions are summarised in the figure captions. In all cases a very similar pattern of forces is demonstrated:

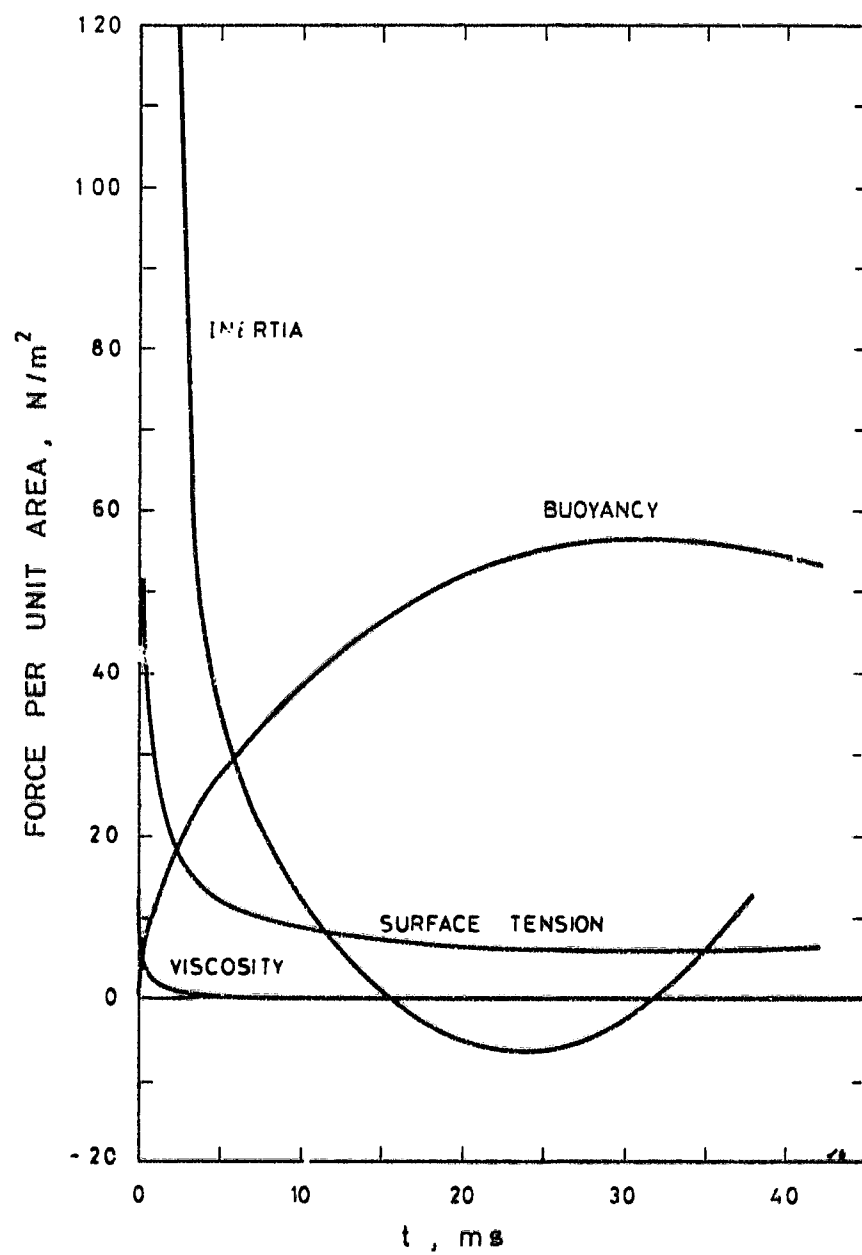


FIG. 9.8 BUBBLE FORCES, RUN M10

Methanol; $P = 25,6 \text{ kN/m}^2$, $q/A = 70 \text{ kW/m}^2$

$\Delta T_{\text{sub}} = 10,2 \text{ K}$, $T_w = 72,0^\circ\text{C}$

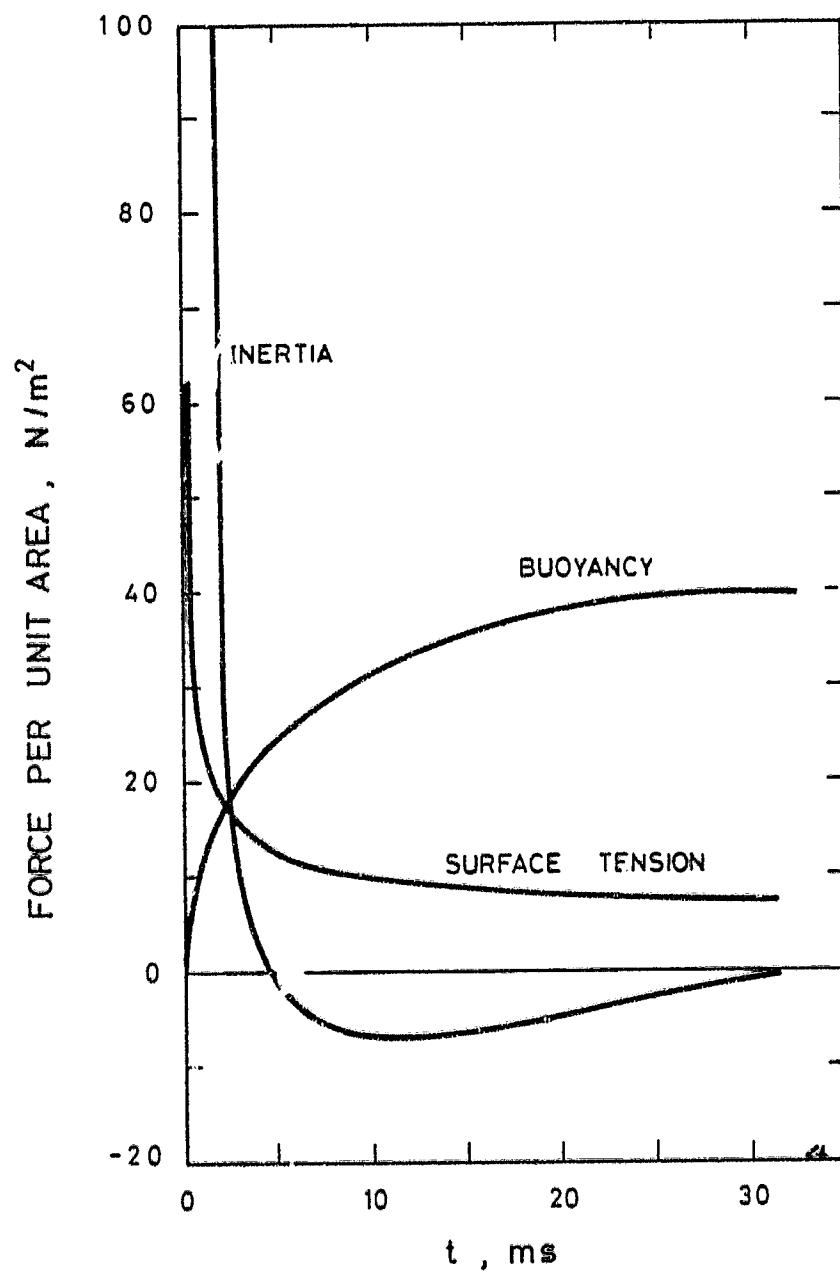


FIG. 9.9 BUBBLE FORCES, RUN M20
 Methanol; $P = 42,6 \text{ kN/m}^2$, $q/A = 61,9 \text{ kW/m}^2$,
 $\Delta T_{\text{sub}} \approx 0$, $T_w = 77,0 \text{ }^\circ\text{C}$

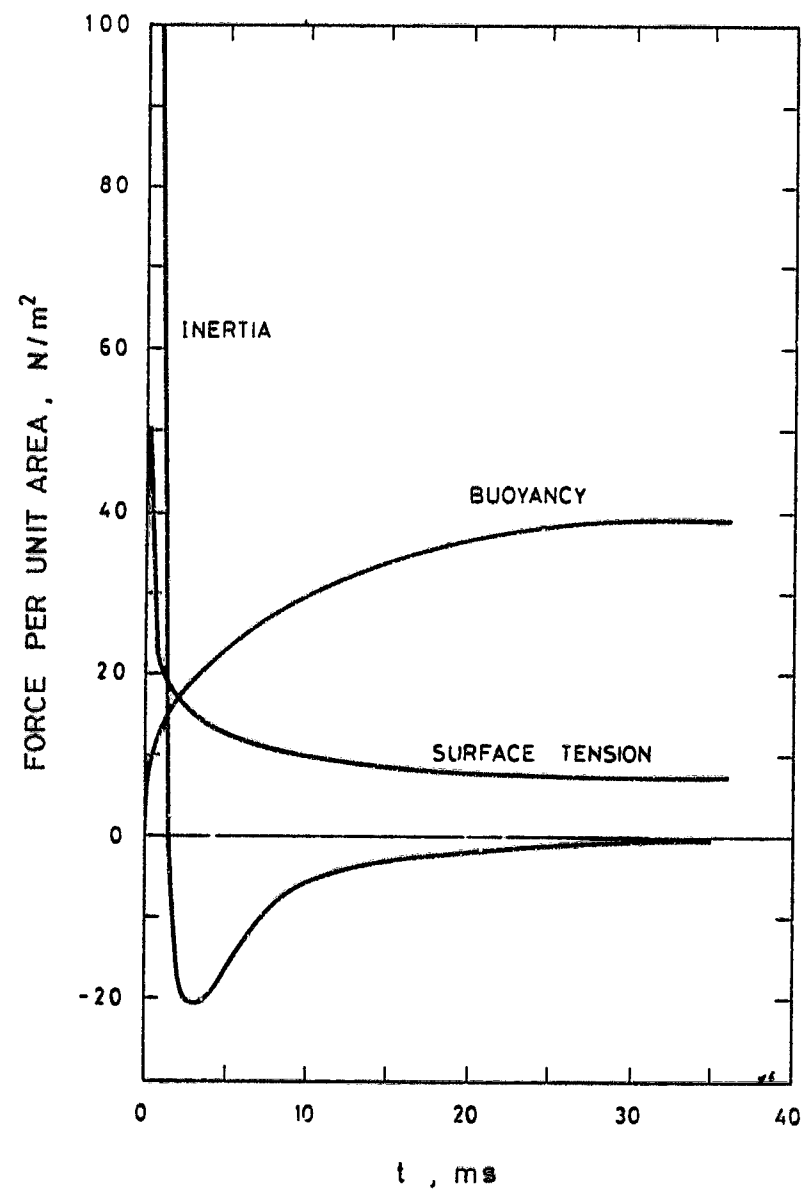


FIG. 9.10 BUBBLE FORCES, RUN E12
 Ethanol; $P = 33,3 \text{ kN/m}^2$, $q/A = 65,7 \text{ kW/m}^2$
 $\Delta T_{\text{sub}} \approx 0$, $T_w = 85,0 \text{ } ^\circ\text{C}$

The inertia forces are predominant in the early stages of growth and then fall away rapidly. Simultaneously the surface tension forces, falling more slowly, become significant compared with the inertia forces and remain so for the rest of the bubble life. This would suggest an initially hemispherical bubble later tending towards an increasingly spherical shape. This geometry, as Chapter 8 has shown, was indeed observed.

The viscous forces are shown for Run M10 only, but were similarly insignificant compared with the inertia forces for the remaining runs. The bubbles would thus be expected to have formed a microlayer, which was of course the case.

The buoyancy forces were, with the exception of the first few milliseconds of growth, the predominant forces on all three bubbles. Such bubbles would thus tend to lift off from their microlayers, that is, to experience microlayer in-flow, throughout the major portion of their period of attachment. It would appear, then, that forces compatible with the continuous microlayer replenishment mechanism have been found.

9.5 MICROLAYER INTERFEROMETRY OF JUDD

In 1969 a preliminary report on the present work was published [152]. The paper gave an outline of the optical theory and of the apparatus of microlayer interferometry, then presented abbreviated results for Run M13 and pointed out the discrepancy between these measured microlayer thicknesses and those deduced from surface microthermometry.

In 1970 R L Judd of McMaster University, in response to this paper, wrote to the author stating his intention to reproduce the interference microlayer measurements. A previous apparatus [88] furnished with a transparent oxide-coated heating surface was at his disposal. Subsequent

correspondence and accompanying photographs demonstrated that Judd had experienced considerable difficulty in obtaining fringes of sufficient visibility. By improvements in camera technique, and possibly also by following up some suggestions made by the author (these were specifically requested) Judd was able to remedy this situation so that by the end of 1972 he obtained analysable fringes and published a set of results [150].

Microlayer profiles from Judd's study [150] are shown in Figure 9.11. Test conditions, shown in the figure caption, differed not greatly from those of the present study; the test liquid, similarly, was an organic compound, dichloromethane. Microlayers are shown to have been of sub-micron thickness, to have thinned progressively during the early stages of the bubble life and to have thickened at the outer edge in the last reported microlayer profile. (This profile, at $t = 2.5$ ms, presumably occurred well before bubble departure). While Judd's study furnished no bubble profile views and thus no volumes, it is clear that the contribution of visible microlayer evaporation to the total vapour flow into bubbles was very low indeed. Judd's work thus confirmed the results of the present study in all respects. This confirmation served as an encouragement towards the examination, given in the foregoing sections, of the apparent conflict between the optically-determined microlayer results and the thermometry-based predictions of Cooper and Lloyd [99, 100].

The interference fringes in Judd's study [150] though analysable, still suffered from a disconcerting lack of sharpness and contrast. Attempts at improvement evidently continued, for in 1975 (during the writing-up of the present work) Voutsinos and Judd published a further microlayer study [151], experimentally similar to the previous, but with the conventional arc-lamp and line filter illumination replaced by laser illumination. The resulting fringes were sharper and, more significantly, greatly increased in number.

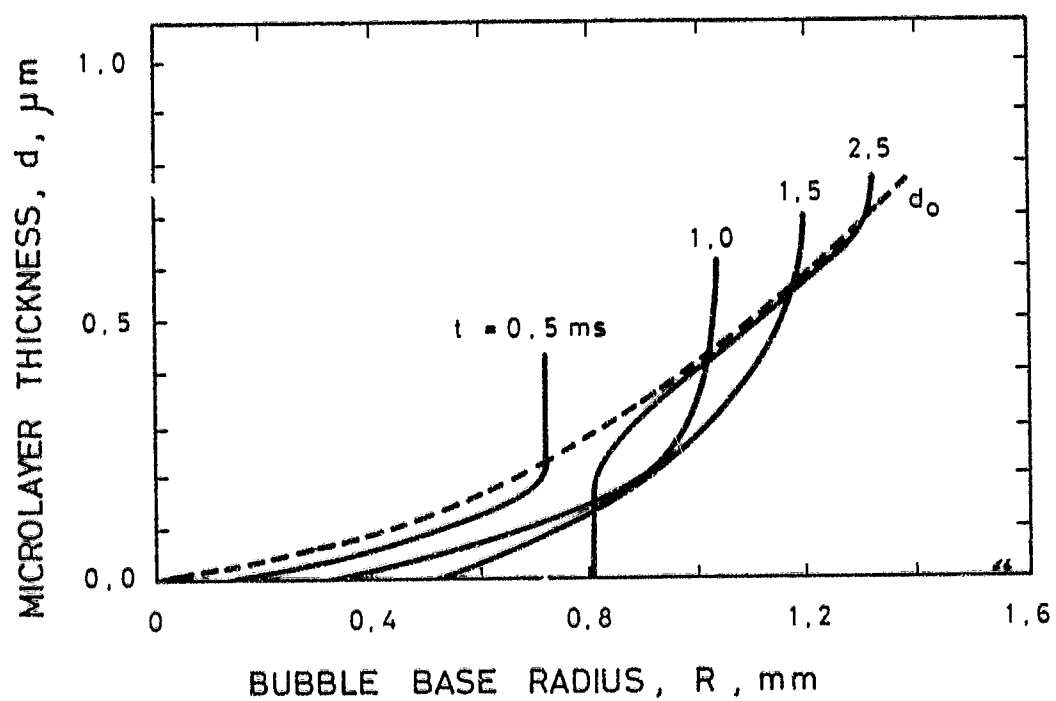


FIG. 9.11 MICROLAYER PROFILES. JUDD [150]
 Dichloromethane; $P = 46,0 \text{ kN/m}^2$
 $q/A = 70,2 \text{ kW/m}^2$, $\Delta T_{\text{sub}} = 0,5 \text{ K}$

Analysis of the fringes led to the microlayer profiles shown in Figure 9.12. The profiles are re-plotted on logarithmic co-ordinates in Figure 9.13 and compared with those of the previous study. The two sets of results are seen to have differed sharply. Firstly, the new microlayer thicknesses were higher by almost an order of magnitude, secondly, the thickening at the outer layer edge was no longer observed, and thirdly, the progressive thinning corresponded to a contribution of microlayer evaporation to the overall surface heat flux of 25 per cent.

Voutsinos and Judd state [151] that they consider the results of Judd's first study to be incorrect because 'failure of the optical filter to properly isolate the spectral lines from 0,5770 μm to 0,5791 μm from the remainder of the spectrum led to the formation of indistinct fringes which resulted in erroneous determination of the fringe orders.' Further, concerning the order-of-magnitude difference between the microlayer thicknesses in their own work and in the author's data for Run M13 [152], they suggest that this might be due to differences in test conditions, but that 'the possibility exists that Jawurek's results are in error for a reason similar to that which invalidated Judd's earlier work'.

The explanation of the divergent results in terms of differences in test conditions would seem to be ruled out: The comparison, in Figure 9.13, of the results of Voutsinos and Judd with those of Run M20 shows the persistence of the order-of-magnitude differences in thicknesses, even for very similar test conditions. The existence of an outright error in the present work, or alternatively, in that of Voutsinos and Judd, must thus be contemplated.

The laser results of Voutsinos and Judd agree broadly (and without the invocation of an additional replenishment

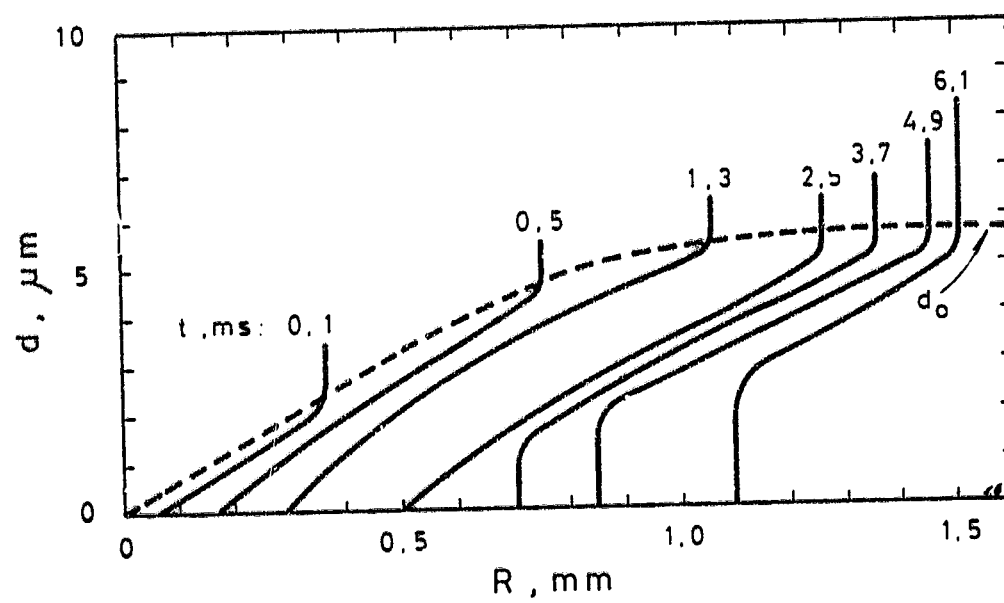


FIG. 9.12 MICROLAYER PROFILES. VOUTSINOS & JUDD [151]
 Dichloromethane ; $P = 48,0 \text{ kN/m}^2$
 $q/A = 60,9 \text{ kW/m}^2$, $\Delta T_{\text{sub}} = 5,6 \text{ K}$

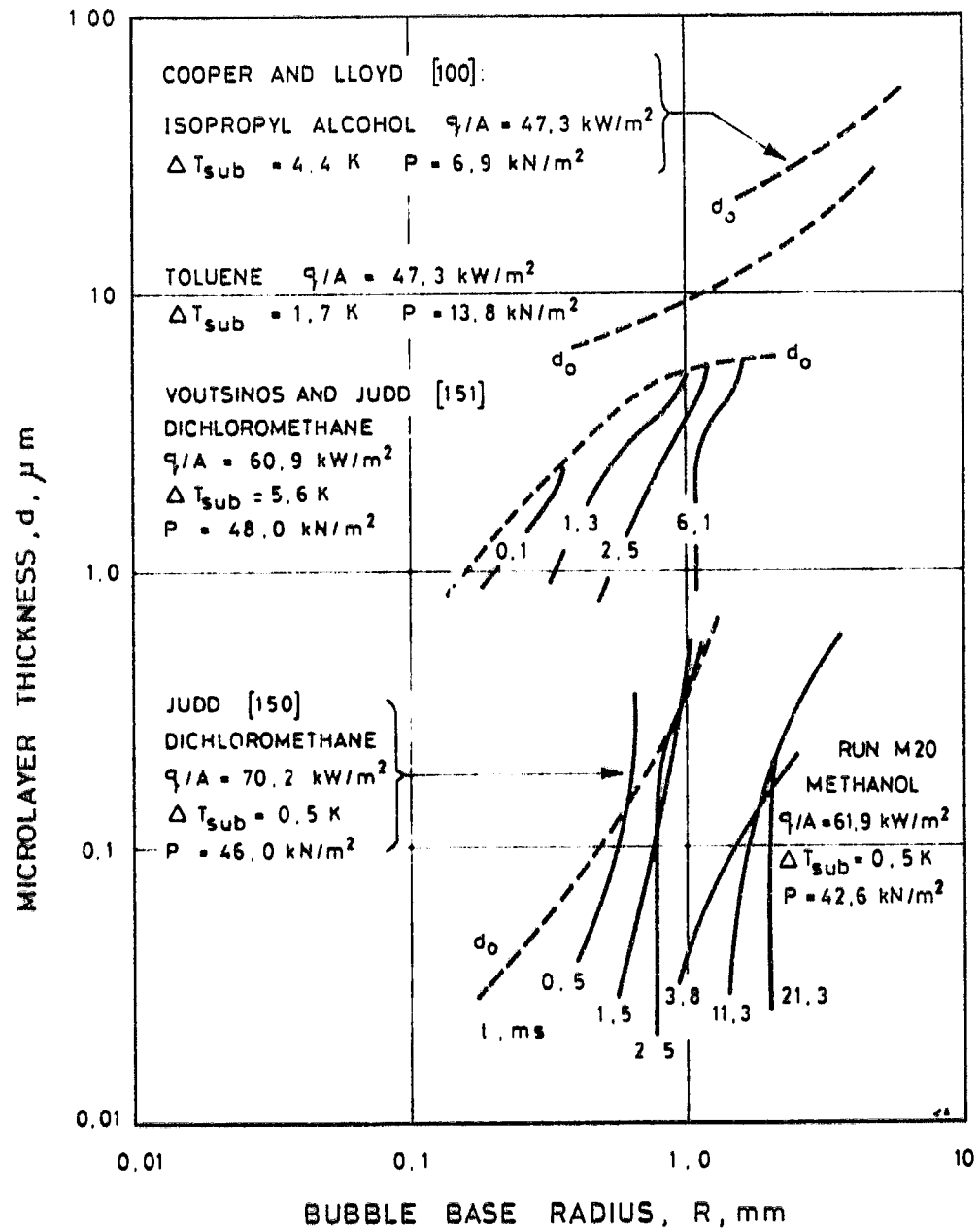


FIG. 9.13 COMPARISON OF MICROLAYER PROFILE RESULTS

Author Jawurek H H

Name of thesis Latent Heat Transport and microlayer Evaporation in Nucleate boiling 1977

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