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Two-phase problems in nuclear reactors[1]

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Abstract

The primary circuit fluid in water-cooled reactors serves two purposes, one being to provide cooling and the second to serve as neutron moderator. It is in performing these functions satisfactorily and reliably that the problems in two-phase flow in nuclear reactors reside.

In assessing the moderator properties of the fluid flowing through the core it is important to have accurate information on its steam quality. The determination of the steam quality presents difficulties due to sub-cooled boiling which occurs in all modern power reactors. Efficient cooling depends on an adequate heat transfer from the fuel elements surface to the fluid and on stable flow conditions in the core. Problems of critical heat flux in boiling and instabilities and pulsations in two-phase flow are discussed. The authors presentations are explained on the strength of his own test data.

Zusammenfassung

Zweiphasenprobleme im Kernreaktor

Bei wassergekühlten Reaktoren hat das Primärkreis-Medium zwei Aufgaben, nämlich die der Kühlung und die der Neutronenmoderierung. Aus der einwandfreien und zuverlässigen Erfüllung dieser Aufgaben erwachsen die Probleme der Zweiphasenströmung im Kernreaktor.

Für die Moderatoreigenschaften des das Core durchströmenden Fluids muß dessen Dampfgehalt genau bekannt sein. Seine Bestimmung bereitet insbesondere bei dem in jedem modernen Leistungsreaktor auftretenden unterkühlten Sieden Schwierigkeiten. Für die Kühlung müssen hinreichend guter Wärmeübergang von der Brennelementoberfläche an das Fluid sowie stabile Strömungsverhältnisse im Core gewährleistet sein. Es werden die Probleme der kritischen Heizflächenbelastung beim Sieden sowie der Instabilitäten und Pulsationen in Zweiphasenströmungen diskutiert. Die Ausführungen werden anhand eigener Messungen erläutert.

EURATOM KEYWORDS

TWO-PHASE FLOW	HEAT TRANSFER
WATER COOLED REACTORS	FUEL ELEMENTS
IN PILE LOOPS	SURFACES
MODERATORS	FLUIDS
REACTOR CORE	STABILITY
STEAM QUALITY	CRITICAL HEAT FLUX
SUBCOOLED BOILING	BOILING
POWER REACTORS	MEASUREMENT
COOLING	HOT CHANNELS

1. Introduction

The fluid flowing in the primary loop of a nuclear reactor serves two purposes: Firstly, it serves for heat transport, and secondly, in the case of water-cooled reactors, it also acts as neutron moderator. A full understanding of its thermodynamic and hydrodynamic behaviour under the conditions prevailing in reactors is therefore of great importance, especially as heat transport in the cooling channels of the reactor core represents a system of impressed heat flux and because under conditions of insufficient cooling, the temperature may rise to a level where destruction of the reactor core is liable to result.

While for the prediction of the moderator effect the factors of interest are primarily the coolant density and in boiling and two-phase flow, the steam quality in the reactor core, the other function, i. e. cooling, depends on two requirements being met:

Efficient heat transfer from the surface of the fuel elements to the coolant
and
stable flow conditions at adequately high velocities to ensure safe removal of the heat absorbed by the coolant due to heat transfer from the core to a heat sink.

Thus we have as the most important problems of two-phase flow to obtain information on, and achieve control of, fluid density or steam quality, respectively, and of the heat transfer in the core, and to ensure stable flow conditions.

2. Hydrodynamic and thermodynamic conditions in two-phase flow

As a rule, modern water-cooled power reactors are designed with sufficiently high power densities for local boiling to take place in the zones of maximum heat flux, i. e. in the so-called "hot channels" even during operation in the sub-cooled range. Steam bubbles are liable to form on a heat-emitting liquid-cooled wall when the surface temperature of the wall exceeds the saturation temperature of the liquid. Steam bubbles tend to originate from active boiling nuclei, i. e. cavities resulting from the natural roughness of the wall and grow very rapidly due to the supply of energy and fluid from the superheated liquid boundary layer directly adjoining the heating surface, this layer being in a thermodynamically meta-stable state. Evaporation takes place instantaneously where the heating surface offers active nuclei. As it forms and develops, the steam bubble derives more than 90% of the energy stored in it indirectly from the superheated boundary layer and only, a minor portion is supplied directly by the heated wall. In other words, the greater part of the heat transport between the wall and the coolant takes place via the liquid phase.

Experience has shown that in a cooling channel having liquid admitted in the sub-cooled state and a water/steam mixture leaving at the other end, the point where steam bubbles tend to form first is not the point where saturation temperature is reached but, on high heat flux heating surfaces rather at a point where the bulk of the liquid is still considerably sub-cooled. This phenomenon is referred to as "sub-cooled boiling".

The processes in a high heat flux channel with coolant entering considerably sub-cooled are shown schematically in Fig. 1.

There are four distinct zones. In zone I the high degree of sub-cooling of the fluid causes straight single-phase flow to prevail without any boiling.

Due to the heat supplied, the temperature of the coolant on its way through the channel rises steadily and, consequently, the heating surface temperature will eventually reach a value sufficient to provide initial nuclei for steam bubbles. Sub-cooling of the flow is, however, still high enough to prevent the steam bubbles from growing beyond the very thin bound-

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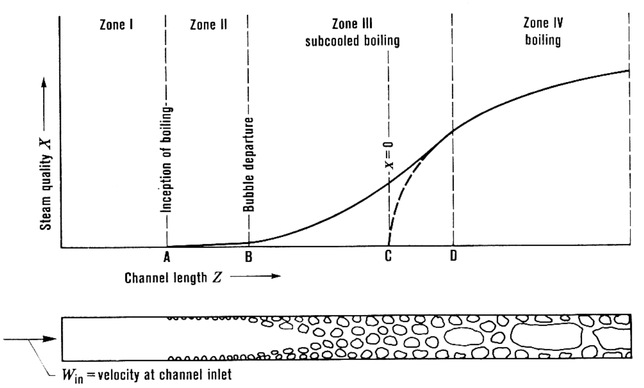

Fig. 1: Boiling zones in high flux channels receiving sub-cooled flow. X = true steam quality, x = steam quality according to thermodynamic equilibrium, —— actual variation of steam quality, ---- variation of steam quality if thermodynamic equilibrium existed

Fig. 1: Siedezonen in hochbelasteten Kanälen bei unterkühlter Zuströmung. X = wahrer Dampfgehalt, x = Dampfgehalt nach thermodynamischem Gleichgewicht, —— tatsächlicher Verlauf des Dampfgehaltes, ---- Verlauf des Dampfgehaltes, wenn thermodynamisches Gleichgewicht herrschte

ary layer because condensation takes place at the head of the bubble. Thus, a bed of steam bubbles closely packed on the heating surface will form in this second zone. Since, as the temperature of the coolant increases further along the channel, the rate of bubble growth will eventually reach a value which, in the region of the boundary layer, will outweigh the condensation process allowing the bubbles to reach a size where the forces due to flow and buoyancy cause them to detach from the surface. However, the bubbles now detaching from the surface in this third zone tend to condense relatively quickly after floating free in the sub-cooled core of the liquid flow. The border between the zones II and III has been referred to in the literature as the bubble separation point. The steam quality of the flow in this third zone undergoes a marked rise along the channel until eventually, when the liquid has reached saturation temperature, what is called "net steam production" takes place.

In contrast to this, an energy balance–assuming thermodynamic equilibrium–would not predict the start of boiling until at the point C as shown by the dotted line in Fig. 1. This constitutes a first concrete part problem, namely, how to predict the steam quality and the start of boiling in the cooling channels of the reactor core in which, because of the high heat flux densities, thermodynamic equilibrium cannot obtain. Mathematical models for the prediction of steam quality under conditions of sub-cooled boiling have been published by *Bowring* [1], *Lavinge* [2] and *Levy* [3]. In point of fact, the greatest uncertainty of these models is just in the exact prediction of the inception of boiling, i.e., the determination of that point along the channel length where boiling takes place for the first time. Experimentally, the start of boiling can be conveniently determined by spacing several thermocouples on the length of a fuel rod and recording the variation of the surface temperature as the heat flux is gradually increased. Since, when bubble formation starts, the heat transfer coefficient rises suddenly, a pronounced jump will be observed in an oscillograph recording of the temperature variations which marks the inception of nucleate boiling at this point. Fig. 2 shows typical oscillograph traces obtained in this manner. The object of the test was the nine-rod bundle shown in the lower part of the illustration in which the centre rod was fitted with several thermocouples spaced over the length and circumference of the rod. Plotted against time as the abscissa are the electric heating current I simulating the nuclear heat source as well as the temperatures at two points on the rod surface. In addition, the temperature difference of the fluid between the points 3 and 4 was measured at the outlet of the coolant from the bundle. Point 3 is located in a flow zone which compared to point 4 has a higher enthalpy increase because of the proportionately greater heated circumference. As the heat flux increases, this temperature difference rises initially, the curve showing a point of reversal on inception of sub-cooled boiling and the agitating effect and turbulence produced by bubbles eventually results in vigorous mixing leading to a noticeable temperature equalisation between the two sub-channels.

By means of such simple measurements, the start of boiling can relatively easily be determined by the discontinuity in the temperature/time curve of the rod surface. Comparing the test data with the results derived from the various mathematical models, considerable discrepancies will be found to exist. Generally, the models tend to predict a much later start of boiling than observed in tests. Obviously, this appreciably affects the steam quality to be expected in the channel under conditions of sub-cooled boiling. An attempt has therefore been made in Fig. 3 by the substitution of such test data for the start of boiling in the calculating model presented by *Levy* [3] to correct the predicted steam qualities. By way of comparison, the steam qualities obtained with *Levys* model so corrected have been plotted in this graph against the values from *Levys* original model as well as values predicted on the basis of the simple energy balance assuming thermodynamic equilibrium. If, according to the energy balance, the boiling limit were reached at a channel length Z = 90 cm, the initial bubble formation according to Levy would take place at Z = 65 cm. Tests have shown, however, that boiling starts as early as 6 cm downstream of the point where heating is started. As a result, the steam qualities found along the channel length differ considerably, too.

When boiling eventually has fully developed, a different flow profile will obtain in the cooling channel depending on the proportion of gaseous-phase fluid. With low steam

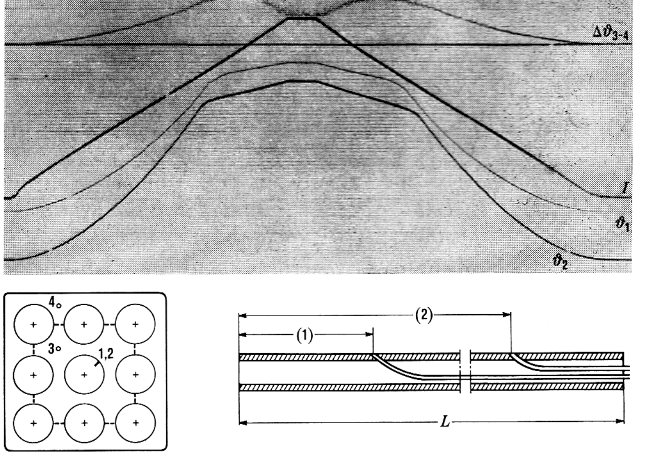

Fig. 2: Temperature pattern and measuring technique for determining inception of boiling during sub-cooled boiling.
ϑ_1, ϑ_2 = heating surface temperature, $\Delta\vartheta_{3-4}$ = temperature difference between cooling water flows in channels 3 and 4, I = el. heating current

Fig. 2: Temperaturverlauf und Meßmethode zur Bestimmung des Siedebeginns bei unterkühltem Sieden.
ϑ_1, ϑ_2 = Heizflächentemperatur, $\Delta\vartheta_{3-4}$ = Temperaturdifferenz der Kühlwasserströme in den Kanälen 3 und 4, I = el. Heizstrom

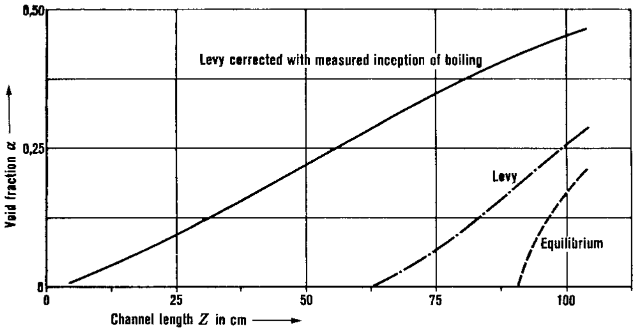

Fig. 3: Influence of boiling onset on steam quality.
$P = 100$ kgf/cm², $m = 100$ g/cm² s, $\Delta\vartheta_{in} = 40$ deg C, $q = 80$ W/cm²

Fig. 3: Einfluß des Siedebeginns auf den Dampfgehalt. Druck 100 at, Mengenstromdichte 100 g/cm² s, Eintrittsunterkühlung 40 grd, Heizflächenbelastung 80 W/cm²

qualities, individual steam bubbles are likely to be uniformly distributed in the fluid flow and this is referred to as "bubble flow".

As boiling increases, large individual bubbles tend to form in a transition range, especially with relatively small mass flow densities which are liable to disturb considerably the steady process of fluid transport through the cooling channel. This is referred to as "slug flow". As steam quality increases further, a point will be reached where so much liquid has evaporated that the liquid column which has been more or less coherent up to that point is broken up by the steam, leaving liquid only deposited and flowing up on the heating surface as a film while in the free flow area steam with entrained water droplets prevails. This flow regime is termed "annular flow".

Before discussing the equilibrium of forces and problems arising in this type of two-phase flow it appears desirable first of all to consider the process of straight heat transfer from the heating surface to the boundary layer of cooling fluid on it.

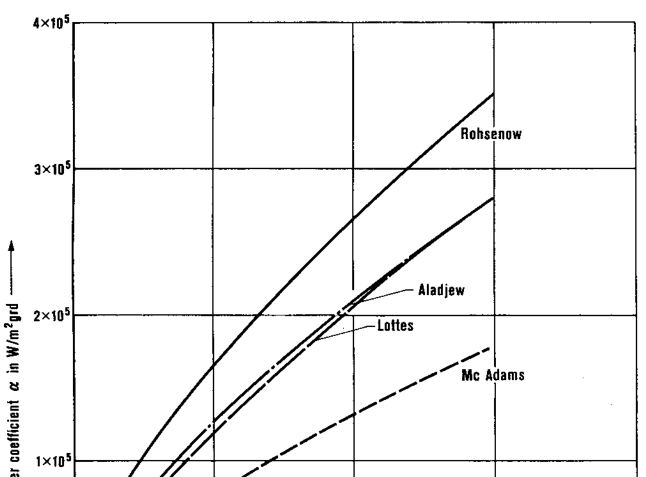

Fig. 4: Heat transfer coefficients during nucleate boiling $p = 70$ bar, $m = 200$ g/cm² s

Fig. 4: Wärmeübergangszahlen beim Blasensieden

3. Heat transport from wall to cooling fluid

Heat transfer in boiling generally permits very high heat flux levels because, firstly, the agitating effect of the steam bubbles causes intense turbulence on the heat emitting wall and, secondly, the steam bubbles are capable of carrying away large amounts of energy in the form of latent heat, i.e., heat of evaporation. As a typical example, Fig. 4 is for a mass flow of 200 g/cm²s and a pressure of 70 kgf/cm² and provides an idea of the heat transfer coefficients to be expected in boiling under conditions of forced convection. The values calculated according to the methods of 4 different authors [4 to 7] have been potted. It is known from numerous tests that during boiling the heat transfer coefficients depend to a large extent on the impressed heat flux and rise considerably as the rate of heat transfer is increased. Therefore, the heat flux has been selected as abscissa in Fig. 4. The proportion of the heat transfer due to forced convection is independent of the heat flux and, in the case of the example illustrated in Fig. 4, amounts to approximately 20 000 W/m² deg C. The graph shows that the heat transfer coefficients tend to improve considerably as the rate of heat transfer is increased and with heat flux levels of 300 W/cm² attain values as high as some 200 000 to 300 000 W/m² deg C. This trend of heat transport is most conducive to the safety of a water-cooled reactor because, as a result, when the heat flux is increased, the surface temperatures of the fuel elements rise but little.

The reason for the improvement in heat transport is to be found in the high rate of increase in the number of boiling nuclei and steam bubbles as the heat flux is increased. However, if the heat flux is increased beyond a preset value, the steam bubbles suddenly form a solid blanket which isolates the heating surface from the cooling fluid. The result is a sudden breakdown of heat transport leading to a marked temperature rise and eventually what is termed "burnout", destruction of the heating surface. The heat flux level at which heat transport from the initially high values of nucleate boiling suddenly decreases to low insufficient values of film boiling is termed the "critical heat flux".

Physically, two different processes can be distinguished in the formation of film boiling. They are associated with the previously mentioned flow regimes viz. bubble flow and annular flow.

During bubble flow, the bubbles accumulate mainly near the heated wall whereas water-phase fluid prevails in the core of the flow. Thus the maximum steam quality is to be observed close to the heating surface. During nucleate boiling, there is a steady and intense fluid exchange between the boundary layer adjacent to the heat surface and the core of the flow because the amount evaporated has to be replaced by liquid. As evaporation increases, a sudden breakdown of this mechanism of fluid exchange is observed. The bubble layer detaches from the wall, the liquid stagnates for a short period on the heating surface and, on account of the high heat flux, this leads to violent evaporation and, consequently, to a steam film forming on the heated surface. Thus, nucleate boiling has changed into film boiling, a process described in American literature as "Departure from Nucleate Boiling".

Conditions are different in the case of annular flow. Here a liquid film forms on the wall whereas the core of the flow is made up of steam. In contrast to bubble flow, the distribution of the steam quality has a maximum at the centre of the flow area. As heat flux is increased, the proportion evaporating from the water film rises, resulting in a reduction of the

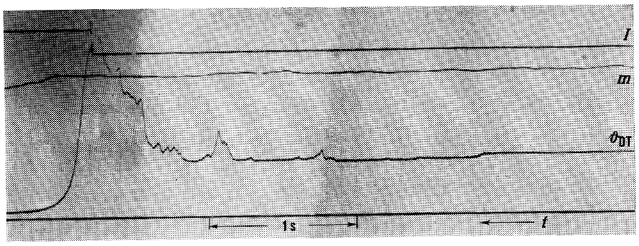

Fig. 5: Variation of temperature on heating surface during onset of film boiling.
$m = 351{,}9\ \mathrm{g/cm^2\,s}$, $p = 101$ bar, $x_E = 0{,}0265$, $x_A = 0{,}0889$, $q_{BO} = 336\ \mathrm{W/cm^2}$, I = power, m = mass flow density, ϑ_{DT} = temperature of the heating surface

Fig. 5: Temperaturverlauf an der Heizfläche bei Einsetzen des Filmsiedens.
$m = 351{,}9\ \mathrm{g/cm^2\,s}$, $p = 101$ bar, $x_E = 0{,}0265$, $x_A = 0{,}0889$, $q_{BO} = 336\ \mathrm{W/cm^2}$, I = Heizstrom, m = Mengenstromdichte, ϑ_{DT} = Temperatur der Heizfläche an der Burnoutstelle

thickness of the layer. When the water film eventually has become sufficiently thin, a dry spot may form on the heating surface causing a rise in temperature. This, again, marks the onset of film boiling. In this case, however, it is referred to as "dry-out".

The temperature rise on transition from nucleate to film boiling when the DNB point is reached takes place within a few seconds as can be seen from Fig. 5 in which an oscillograph tracing of the variation of the heating surface temperature in time has been reproduced at the moment nucleate boiling changes to film boiling. The time as abscissa runs in this trace from right to left. Furthermore, the step in the line shown at the top of the trace marks the time at which an electrically controlled safety device, the so-called burnout detector, shuts down the power by which the test channel is heated electrically. Looking at the temperature trace it can be clearly seen at the right hand side of the diagram how the wall temperature suddenly decreases by a small amount of approximately 5 to 10 deg C. Subsequently, brief statistical variations are seen to develop which, in an almost step-like rise, lead to film boiling. The slight drop in temperature on inception of the boiling crisis suggests an improvement of heat transfer by complete evaporation of the two-phase boundary layer existing up to that point. This supports the assumption that a stagnating liquid layer would evaporate suddenly giving rise to the state of film boiling.

In the literature published over the last 20 years are recorded more than a thousand tests in respect of the critical heat flux in boiling. A systematic evaluation of these tests affords in understanding of the hydrodynamic and thermodynamic parameters on which this phenomenon is dependent. It appears to be a practical approach in a systematic study of the burnout parameters to look upon the inception of film boiling as a function of the purely local conditions of flow regime and thermodynamic state when only the local values of steam quality and flow conditions enter into the burnout considerations themselves. True, this involves a prior study to define the history of the flow on its way to the burnout point. This includes such phenomena as mixing processes between the flow lines of widely different enthalpy rise and turbulence effects.

With a view to the practical design of water-cooled reactors, attempts have been made to cover the thermodynamic and hydrodynamic parameters in what are referred to as burnout equations. These are invariable of an empirical nature and their mathematical results have usually been matched to published test data by a comparison using coefficients. For power reactors of the pressurized water and boiling water type it is frequent practice today to apply the equation evolved by *Tong* [8], also referred to as W3 correlation, and the equation by *Janssen-Levy* [9].

The question now arises as to how accurate such burnout equations can predict the critical heat flux for any fuel element bundle in a pressurized water or boiling water reactor. *Tong* [10], for instance, used several thousand measuring points in developing his equations and he estimated the uncertainty in his W3 correlation at ± 23 % which is made up as follows:

1. Statistical variations are to be expected in the degree of turbulence in two-phase flow and in the surface roughness of the heating metal. This leads to an uncertainty of ± 3 %.
2. Both the reactor fuel element and the rod bundle in the test loop are subject to manufacturing tolerances. These are estimated by Tong to introduce an additional uncertainty of ± 5 %.
3. The equation being of a strictly empirical character with a limited number of empirical constants cannot provide a fully correct physical coverage of all parameters. In view of this, Tong holds there is a further uncertainty of ± 5 % resulting from the form of the equation.
4. The greater part of the sources of errors is attributed by Tong to the different measuring techniques employed and the different loop characteristics existing. Strictly an uncertainty residing in the measurement, this is estimated by Tong at ± 10 %.

It would be of interest now to make a test to study whether *Tongs* assumption of tolerances is too conservative or too optimistic.

At an annual forum of European two-phase experts a comparison was made of the different calculating methods employed for the prediction of the critical heat flux in boiling and a fuel element bundle was taken as a basis whose characteristics somewhat deviated from those usually encountered in power reactors. As shown in Fig. 6, it consisted of 9 rods each subjected to a different rate of heating. The variation of the heat flux over the length of each rod is shown schematically by the graph at the bottom right-hand side of Fig. 6. This bundle was operated in the boiling water range at a pressure of approximately 30 bar. The hatched area in the upper graph in Fig. 6 indicates the range of scatter resulting from the use of the different calculating methods which had been

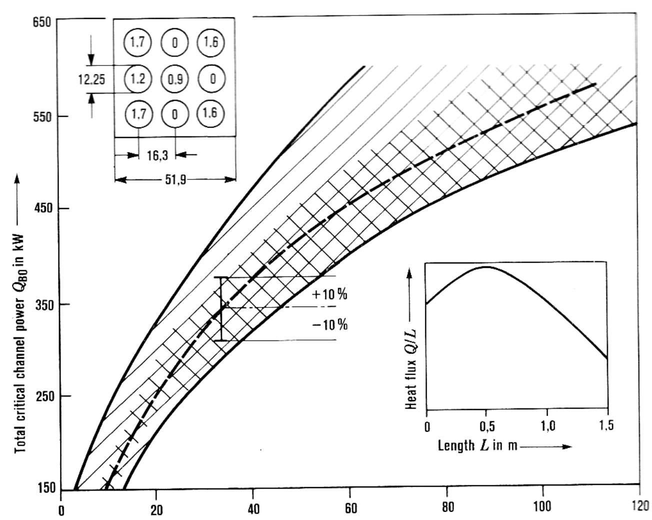

Fig. 6: Comparison of calculations to determin critical heat flux
---- mean values of experimental results

Fig. 6: Berechnungsvergleich zur kritischen Heizflächenbelastung.
---- Mittelwerte der experimentellen Ergebnisse

applied to some ten different points. The actual burnout values, i. e. those measured, are shown by the dotted line in the graph. In order to bring out even better the percentage difference existing between the individual predictions, a range of scatter of ± 10% about the measured burnout curve as the mean value has been shown by crosshatching. From this, the conclusion can be drawn that *Tongs* estimation of the uncertainty in his method is too conservative rather than too optimistic.

Special studies in connection with the burnout problem have been directed towards exploring the possibilities of raising the critical heat flux and determining the burnout levels to be expected under transient conditions of coolant flow and reactor power. An effective means of improving the critical heat flux is provided by simple swirl or turbulence-producing fittings in the fuel element bundle. If these are made in the form of spiral fins extending over the full length of the fuel element it is possible–as measurements have shown–to increase the critical heat flux by more than 100%. But even short spiral vanes with a length of 30 to 50 mm which can be fitted to existing spacers are apt–as other measurements have proven–to ensure an improvement in the critical heat flux of up to 40% without attracting a penalty in the form of, say, fretting corrosion or manufacturing difficulties as in the case of full length fins.

The critical heat flux during power and mass flow transients is an area that has seen little research up to now. Measurements of the burnout during rapid power increase have recently been made on 4-rod and 9-rod clusters [11]. These have shown that, depending on the geometry in the core as well as the duration and gradient of the power excursion, appreciably higher critical heat flux levels may in some cases be expected than in steady state operation. However, because of the small number of measuring points investigated, this improvement should not, for the time being, be taken advantage of in general design practice.

Power reactors are invariably designed with a high safety factor against burnout. Nevertheless, the question as to what heat transfer coefficients are to be expected under conditions of film boiling would be of interest in the case of reactor accidents, such as pump failure and coolant loss. As an example, the heat transfer coefficients during film boiling have been plotted in Fig. 7 for a mass flow of 70 g/cm²s against the system pressure for various heat flux levels. It can be seen that in film boiling–similar to what was found during nucleate boiling–the heat transfer coefficients tend to vary considerably with the heat flux applied. On the average, however, the heat transfer coefficients are well within a range where appreciable heat flux values are obtainable at heating surface temperatures near the safe limit, i.e. where no destruction of the fuel element does occur.

4. Heat transport from reactor core to heat sink

As mentioned initially, it is important that in addition to adequate heat transfer from the fuel element wall to the coolant satisfactory heat transport is ensured from the reactor core to the heat sink, be it a steam raising unit or a turbine connected into the primary loop. Stable flow conditions through the reactor core as are necessary to this end depend on equilibrium existing between the driving forces and the resisting forces. In a heated channel where evaporation occurs the thermodynamic and hydrodynamic conditions can be represented by the mass balance, energy balance and force balance. As each phase flows at a different velocity, it

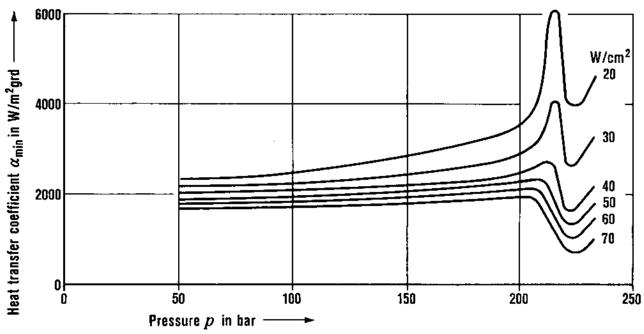

Fig. 7: Heat transfer coefficients during film boiling $m = 70$ g/cm² s

Fig. 7: Wärmeübergangszahlen beim Filmsieden, $m = 70$ g/cm² s

is necessary to consider the balances separately for each of the two phases.

Regarding the volume element $F \cdot dz$ of a heated channel, the mass balance is given by the continuity equation reproduced in equation (1) with the simple statement that the mass storage in time within the volume element is equal to the difference between the quantity entering and the quantity leaving.

$$\frac{\partial}{\partial t}\left[\varrho_W (1-\alpha) + \varrho_D \alpha\right] + \frac{\partial}{\partial z}\left[\varrho_W w_W (1-\alpha) + \varrho_D w_D \alpha\right] = 0 \quad (1)$$

where

α	the volume fraction of steam in the volume element under consideration
ϱ_W, ϱ_D	the density of the water or steam respectively
w_W, w_D	the flow velocity of the two phases

The coordinate of the channel length extending in the flow direction is designated z and the time t.

Similar as in the case of the mass balance, the change in time of the energy available in the volume element in the energy balance of equation (2), too, is obtained as the difference between the inflowing and outflowing energies increased, however, by the heat introduced into the volume element by the heating of the channel.

$$\frac{\partial}{\partial t}\left[\varrho_W h_W (1-\alpha) + \varrho_D h_D \alpha\right] + \\ + \frac{\partial}{\partial z}\left[\varrho_W h_W w_W (1-\alpha) + \varrho_D h_D w_D \alpha\right] = q \frac{U_b}{F} \quad (2)$$

In this equation, the specific enthalpy is designated h whereas U_b represents the heated circumference of the volume element and q the heat flux passing from the channel wall into the cooling fluid.

With regard to the force balance represented by equation (3), the condition to be satisfied is that the momentum change in time in the volume element has to be equal to the difference between the inflowing and outflowing kinetic energies plus the end forces acting on the element.

$$\frac{\partial}{\partial t}\left[\varrho_W w_W (1-\alpha) + \varrho_D w_D \alpha\right] + \frac{\partial}{\partial z}\left[\varrho_W w_W^2 (1-\alpha) + \varrho_D w_D^2 \alpha\right] = \\ = \frac{\partial p}{\partial z} - \tau_W \frac{U_W}{F} - g\left[\varrho_W (1-\alpha) + \varrho_D \alpha\right] \quad (3)$$

In view of the friction-affected flow and the buoyancy in the vertical channel this equation has the last two terms indicated added to it. In equation (3), U_W is the wetted circumference of the channel and τ the wall shear stress of the liquid phase. For a full description of the processes in the two-phase flow, however, these three equations are not sufficient and, as a rule, the two phases are linked in the literature by an empirical function for the velocity ratio between water and steam.

In deciding the stability of the flow and, consequently, the stability of heat transport from the heat source to the heat sink, it is necessary to have information on the characteristic behaviour of the pressure loss in a two-phase flow. If one considers a heated channel of constant heat flux to which water is admitted at a given degree of sub-cooling, the heat supplied to the water in the case of high mass flows will not be sufficient to produce an appreciable amount of steam. In this range, therefore, the resistance characteristic will substantially follow the well known parabolic pattern of single-phase liquid flow. If mass flow is decreased while heat flux is maintained, steam generation will increasingly take place and, in spite of decreasing flow, the two-phase flow then developing will result in the pressure loss through the channel rising again. At very small mass flows, nearly all water will eventually evaporate and steam flow will prevail. The resistance characteristic will approach the parabola for the friction losses in pure steam flow.

The state of flow in the cooling channel is a function of the equilibrium between the driving forces and the resisting forces and the operating points of the system are given by the intersections of the delivery and resistance characteristics. A simple stability criterion has been given by *Ledinegg* [12] which, while neglecting a number of parameters, states that these intersections represent stable operating points as long as the gradient of the resistance characteristic is greater or equal to the gradient of the delivery characteristic. *Ledinegg* does not take into account the fact that hydrodynamic processes in any twophase flow are invariably allied to thermodynamic processes such as energy storage in the form of evaporation heat in the steam bubbles, after-evaporation, recondensation and heat congestion in the fuel rod. In considering these strictly hydrodynamic and thermodynamic processes, the feedback of these oscillating processes via reactivity changes in the reactor core is not proposed to be discussed.

An exact determination of the stability conditions in two-phase flow should, on the one hand, include the system of equations mentioned earlier for mass energy and force balances and, on the other hand, an analytic assessment of all transfer and storage possibilities existing in heat exchangers, vessels and pipework upstream and downstream of the heated channel, in other words, provide a description of the whole coolant circuit. There are several references in the pertinent literature [13 to 16]. They can be classified into calculating models that achieve an analytical solution of the problem by linearizing the equation systems and models that evaluate the equation system by stepwise numerical intergration on a computer. It is not proposed to deal with these calculating models here. Instead, an experimental example for the occurrence of periodical instabilities in a twophase flow system is reproduced in Fig. 8. Plotted against time extending as abscissa from right to left are the mass flow and the pressure drop across the cooling channel. Drawing the mean of the widely oscillating trace of the pressure variations a mean pressure difference of approximately 0,2 bar is found at the point shortly before inception of the instability. However, this pressure difference shows wide fluctuations. On inception of the instability, i. e. after the initial decrease

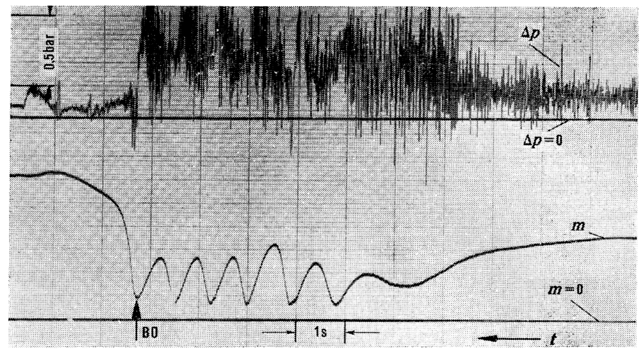

Fig. 8: Pressure and mass flow variations during periodic instability m = 250 g/cm² s, $\Delta\vartheta_E$ = 30 deg inlet sub-cooling, q = 160 W/cm²

Fig. 8: Druck- und Mengenstromschwankungen bei periodischer Instabilität. m = 250 g/cm² s, $\Delta\vartheta_E$ = 30 grd Eintrittsunterkühlung, q = 160 W/cm²

in mass flow, oscillating flow develops. Almost in phase with the oscillation of the mass flow are the fluctuations of the pressure drop across the channel with the pressure minimum being invariably associated with a mass flow maximum.

5. Conclusion

The purpose of this paper has been to give a survey only and no attempt has been made to deal in detail with selected problems of two-phase flow research. In order to have this comprehensive work evaluated and reduced to generally valid statements and mathematical relationships it is necessary to give priority to advancing and consolidating the similarity considerations which have recently been taken in hand and which, in the single-phase field, have long become common knowledge. Then it will be possible by generalization to enhance the value of the data obtained so far and to carry on future work in this field with many times less cost and improved usefulness.

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