The hydraulic characteristics of a naturally circulating boiling water system

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The Hydraulic Characteristics of a Naturally Circulating Boiling Water System *)

by Dr. Ir. C. L. Spigt

Summary

A review is given of the various types of flow oscillations which can occur in steam-water mixtures. After a description of the experimental set ups, results are given of a systematic experimental study on the steady-state characteristics and the onset and character of a common type of oscillations, which are caused by the interaction between the steam production and the recirculation rate. It is shown that there is a connection between the stability characteristics of the steady-state and the onset of flow oscillations in a two-phase mixture. Futhermore, the burn-out heat fluxes are presented obtained in transient conditions. The experimental results are compared with those obtained from a theoretical study.

1. Introduction

Nuclear reactors in which the fuel rods or plates are cooled by circulating water have shown performance characteristics that make them attractive as possible producers of heat, which can be converted into electrical power or used for propulsion purposes. Particularly those reactors in which naturally or forcedly circulating coolant boils with or without net steam production, seem to offer such potentialities. A demand for exploiting these characteristics is the ability to predict accurately the heat transfer and fluid flow characteristics in the applied coolant system. This holds even more particularly for those nuclear reactors in which the circulating water is also used as moderator, the density of which has a major effect on its potential to slow down the fission neutrons. In many cases the characteristics of the coolant impose a limit on the power output of a nuclear reactor.

Therefore, in many research establishments an extensive research programme is being carried out to obtain basic data on the heat transfer and fluid flow characteristics of naturally and forcedly circulating boiling water systems, especially under conditions prevailing in nuclear reactors.

One of the main items in these studies are the stability and transient characteristics of a boiling system. These characteristics are of special importance from an operational point of view. They determine the stability and the controllability, and the knowledge of them is necessary for the development and the evaluation of the safety aspects of a nuclear reactor, as well as for an accurate design of the control devices. On the other hand, these studies are related to the onset under particular conditions of spontaneous flow oscillations in naturally as well as in forcedly circulating boiling systems. These oscillations may be of different origin and nature and may have a great influence on the operating limits of a nuclear reactor. They may, for instance, be responsible for large power oscillations owing to neutronic feedback. Furthermore, the heat transfer characteristics may change considerably and an appreciable reduction in the power levels where burn-out occurs may be expected.

In the Laboratory for Heat Transfer and Reactor Engineering of the Mechanical Department of the

*) Deze publikatie beschrijft de voornaamste resultaten van een onderzoek dat is uitgevoerd in het Laboratorium voor Warmtetechniek en Reactorbouw van de Technische Hogeschool te Eindhoven onder contract met Euratom. Het artikel bevat de tekst van een voordracht gehouden tijdens de Atoomforumconferentie te Bunnik. Voor een uitvoerige beschrijving van de verkregen resultaten wordt verwezen naar het proefschrift van de auteur (1).

Technological University of Eindhoven a research programme is being carried out, comprising experimental as well as theoretical studies on the dynamic characteristics of a two-phase flow. This programme is of a fundamental nature and not directly related to a specific reactor design. The programme is sponsored by the Atomic Energy Commission (AEC) of the USA and Euratom. The work to be described in this publication is part of this programme. It deals with the onset of flow oscillations in a single fuel channel of a reactor, where the coolant and moderator are boiling water circulating by natural or forced convection, and which is designed for net steam production.

2. Types of flow oscillations

In connection with these reactors, concern has been expressed about different types of coupling effects and flow oscillations. A diagram in which two types of coupling characteristics are indicated is shown in Figure 1. The coupling effect between the steam-void volume or the density of the moderator and the reactor power has been mentioned. In Figure 1, this process is indicated by the feedback path outside the broken lines. The reactivity is defined as the extent, by which the neutron multiplication factor exceeds one. If the coupling between steam-void volume and nuclear power turns into regenerative feedback, divergent power oscillations may occur.

A second feedback path is indicated within the broken lines of Figure 1. A change in steam-void volume in a coolant channel causes a change in the pressure drop along the channel and thus in the coolant flow rate in that channel, which, in its turn, causes a change in the steam production and thus in the steam-void volume. If, owing to this feedback the system becomes unstable, heavy flow oscillations occur. In this type of flow oscillations the intercoupling of the boiling channel with the other parts of the system plays an important role. In the recent literature these flow oscillations are sometimes referred to as pressure drop oscillations and density wave oscillations. In boiling water reactors with forced circulation, incorporating a pump which keeps the inlet flow constant and independent of the change in pressure drop along the channel, the feedback mentioned is not present. Also, the intercoupling between parallel channels in a boiling water reactor may produce flow oscillations which are related to the second type just mentioned. Both are from a purely hydrodynamic origin.

Besides, other types of hydraulic flow instabilities may occur in a naturally or forcedly circulating coolant channel, for instance, nucleation instabilities (these instabilities are caused physically by a building up of a certain superheat followed by a sudden evaporization of the liquid phase with resultant rapid increase in specific volume and in pressure), flow-pattern instabilities (these are connected with the variety of possible geometric configurations into which the two phases can arrange themselves: bubbly, churn, slug, annular, etc.), acoustical or propagation waves (connected with the compressibility characteristics mainly of the gas phase in a two-phase mixture) and thermal oscillations (connected with the occurence of burn-out in the high quality region).

3. Description of the experimental set-ups

3.1. *Pressurized boiling water loop*

The hydraulic phenomena occurring in a single coolant channel have been studied outside the reactor in a pressurized boiling water loop where the nuclear fuel rod is simulated by an electrically heated tube of the same configuration and dimensions. Up till now the experimental part of the programme was restricted to the operation under conditions of natural convection.

A simplified flow scheme of the loop is given in Figure 2.

The test section is placed in the cylindrical part of the pressure vessel. The loop is filled with demineralized water up to a certain level. The channel formed by the heating element and the shroud is called the riser; the one formed by the shroud and the pressure vessel, the downcomer. Since the shroud is open at both ends, the two channels are in direct connection with each other. When the element is heated, the water in the riser starts to boil and vapour is formed. Owing to the resulting density difference between the fluid in the riser and in the downcomer the steam-water mixture in the riser flows upwards by natural convection. At the water surface steam and water are separated. The water returns to the inlet of the riser through the downcomer. The steam flows to the condenser and the condensed steam is returned to the downcomer.

Before returning to the riser, the water passes through a subcooler giving the possibility to subcool the water and to adjust the inlet temperature. The secondary circuit of the subcooler consists of four helical tubes through which an adjustable quantity of water flows. For a closer control of the inlet temperature a preheater has been installed which can be controlled automatically or manually. The subcooler circuit can be extended with a centrifugal pump and connecting tubes for carrying out forced circulation measurements. Inlet subcoolings up to 50 °C can be obtained.

The pressure vessel is made of stainless steel 316 and has a design pressure of 40 atmospheres. The cylindrical part has an inner diameter of 0.15 m and a length of 3 m. During operation the variation in water level, caused by thermal expansion and by the formation of steam is kept within certain limits

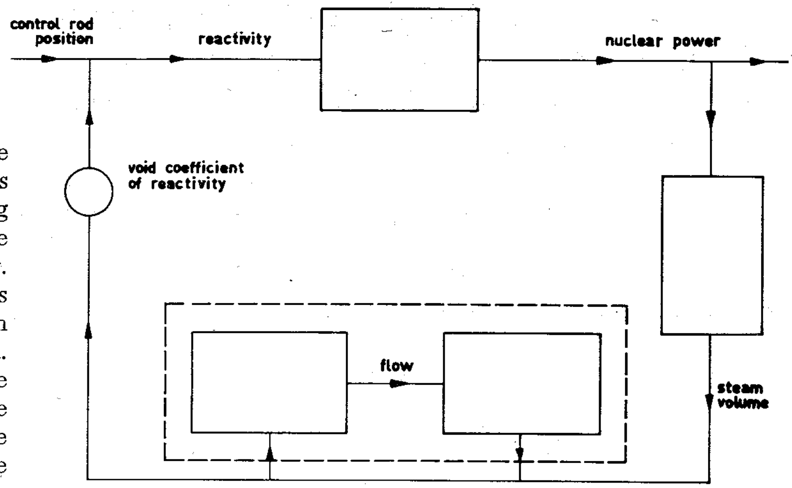


Fig. 1. Feedback paths in a nuclear boiling water reactor.

by means of a water drum connected with the pressure vessel. In the condenser the steam is condensed inside three cooled tubes by sprinkling cold water on the tubes. The coolant flow to the condenser is controlled automatically or manually. The two test sections used consist of a stainless steel tube placed centrally inside the shroud. On both ends red copper electrodes have been soldered. An asbestos graphite gland with spring pressure at the bottom electrode allows for expansion of the element. The top electrode is connected to a flange which is insulated from the pressure vessel as is the shroud. The bottom electrode is connected to the pressure vessel and both are earthed. The two test sections, denoted as Test Section I and II, differ only with respect to the inner diameter of the shroud, which is 50.0 and 58.8 mm respectively. The inner and outer diameters of the heating element are constant along the length giving a uniform heat load distribution. The hydraulic diameter of the test sections is 16.16 mm for Test Section I and 25.03 mm for Test Section II.
The maximum current and power which can be supplied is 14,000 A, DC and 1,000 kW.

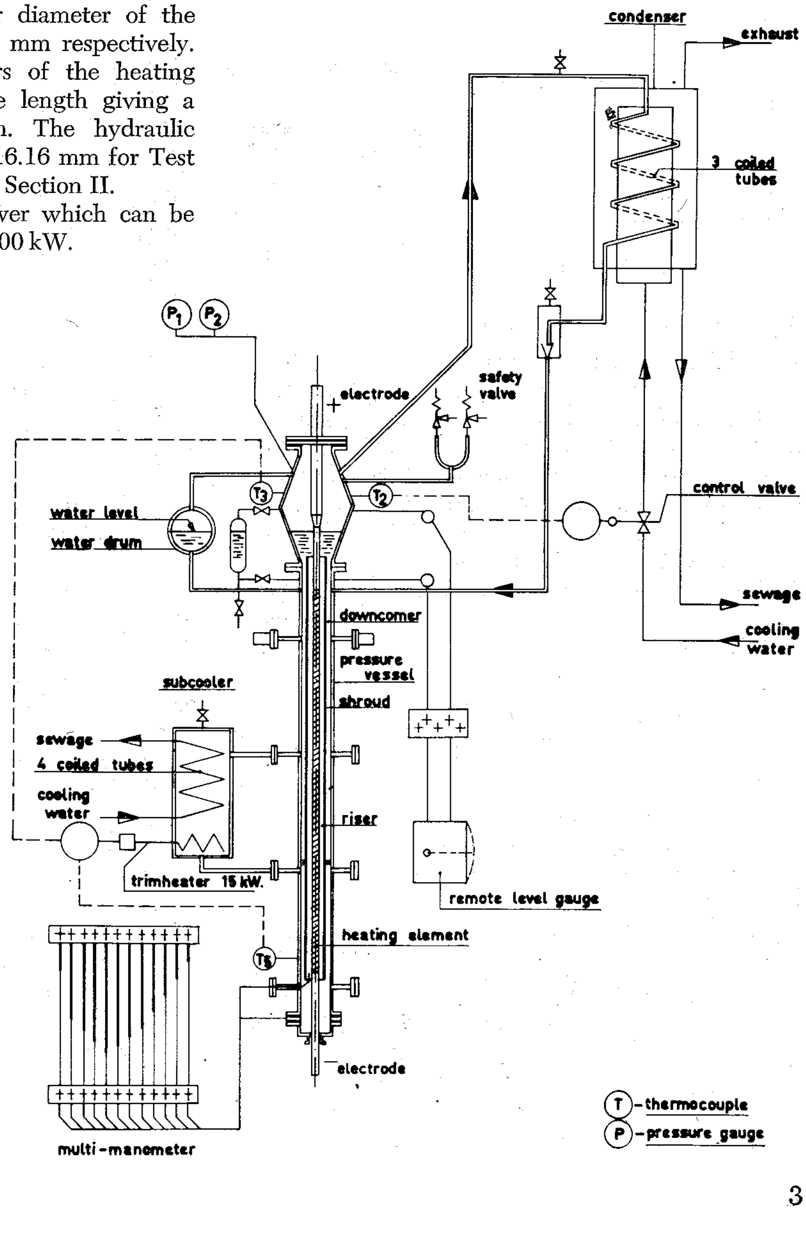
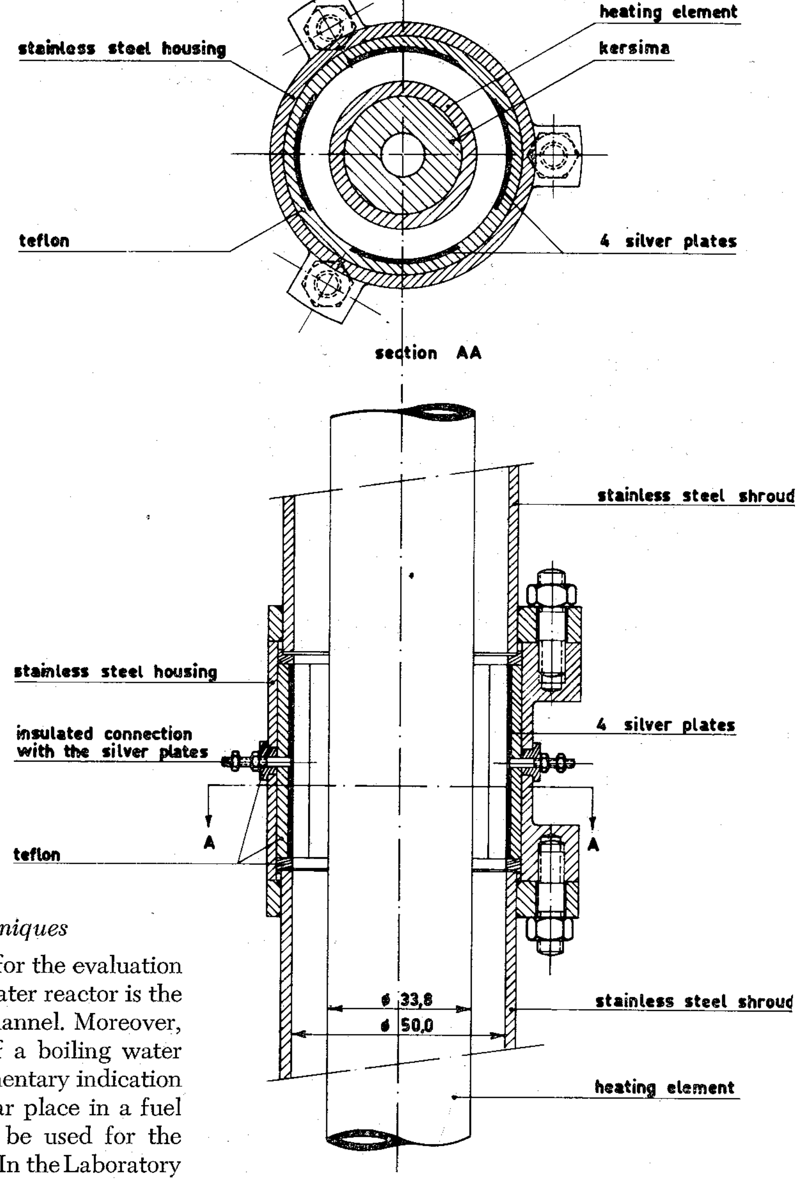


Fig. 2. Flow sheet of the pressurized boiling water loop.

Fig. 3. Impedance void gauge.



3.2. *Void fraction measuring techniques*

One of the important parameters for the evaluation of the performance of a boiling water reactor is the void distribution along the fuel channel. Moreover, for the study of the dynamics of a boiling water reactor it is desired to have a momentary indication of the void fraction at a particular place in a fuel channel. Several techniques can be used for the measurement of the void fraction. In the Laboratory for Heat Transfer and Reactor Engineering of the Technological University of Eindhoven three methods are in operation:

1. the γ-ray attenuation method,
2. the impedance method,
3. the acoustical method.

In this study the first two methods have been used. The γ-ray attenuation method using the pulse counting technique has been applied for void fraction measurements in steady-state conditions at one fixed location along the channel. This technique is based on the difference in absorption characteristics of γ-rays for water and steam. In the experimental programme reported here some special measures have been taken in connection with the construction of the loop and some others to improve the accuracy of the method (see Ref. 1). A Thulium-170 source of approximately 300 mCurie and a half-live of 127 days can be positioned inside the heating element. The γ-quanta are going through the two-phase mixture and, after passing a collimator, are detected by four scintillation counters grouped around the riser.

The technique for measuring the void fraction in a two-phase flow system using the impedance method has been developed for the test section geometry under discussion. The application of this measuring technique is based on the determination of the conductance in a two-phase mixture with respect to that in a one-phase liquid at the same temperature. The heating element is used as the first electrode. In the hull of the shroud four plates are placed at a given axial location (see Figure 3), insulated from the shroud but connected to each other, to act as a second electrode. Nine of these void gauges are located along the coolant channel. The theoretical basis of this technique was derived by Maxwell (Ref. 2). By assuming an

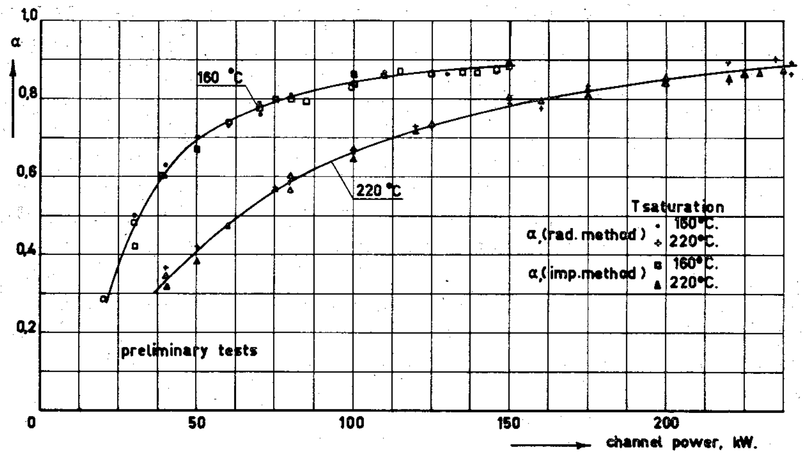


Fig. 4. Comparison of γ-ray and impedance void fraction measurements.

analogy between the electrical conductivity and the dielectric constant of a mixture it follows that for a bubbly or a mist flow

$$\frac{\varepsilon - \varepsilon_2}{\varepsilon + 2\,\varepsilon_2} = \alpha \frac{\varepsilon_1 - \varepsilon_2}{\varepsilon_1 + 2\,\varepsilon_2}$$

where ε is the conductance of the mixture,
ε_1 that of the discontinuous phase (steam),
ε_2 that of the continuous phase (water),
and α is the void fraction.

The calibration of the impedance void gauge has been carried out in a perspex loop, filled with water, in which air was blown. In the pressurized boiling water loop a void gauge has been placed at the location of the γ-ray source and the values obtained with both methods could be compared. Some results are given in Figure 4. The difference between the two methods is less than ± 2% void. On the whole the reproducibility of the impedance method is better than that of the γ-ray attenuation method and is generally within ± 1% void.

4. Experimental results

In the following some results will be given obtained in the experimental programme on the characteristics in steady-state conditions, the onset of flow oscillations, the stability characteristics and the burn-out heat fluxes in oscillating conditions.

4.1. *Steady-state*

A prerequisite to an understanding of the dynamic behaviour of a two-phase flow is an understanding of the characteristics of the steady-state.
For a given geometry, the hydraulic behaviour of a natural circulation loop is basically determined by its independent variables:

a. the total heat input, Q;
b. the system pressure or corresponding saturation, T_{sat};
c. the temperature at the inlet of the coolant channel T_{in} or the corresponding subcooling $\triangle T_{sub} = T_{sat} - T_{in}$.

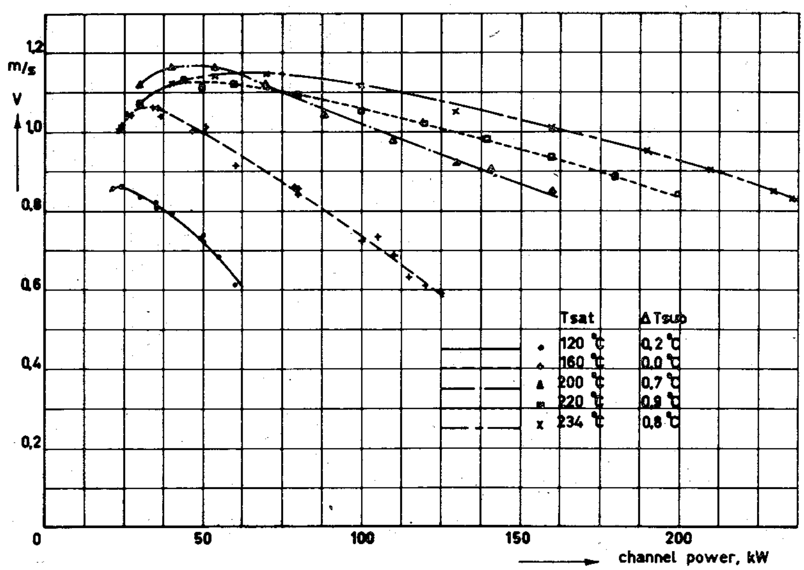


Fig. 5. The circulation rate as a function of channel power for various system pressures, Test Section I.

In Figure 5 results are given of the water circulation rate, measured at the inlet, as function of the channel power at various saturation temperatures for Test Section I. As is shown, there is a maximum in the circulation rate versus channel power curve. At low power the driving head increases as a result of the increase in void. This produces the rising part of the curve. At high channel power the two-phase friction pressure drop and acceleration-pressure drop become progressively more important (the two-phase friction-pressure drop increases roughly with $(1-\alpha)^{-2}$) resulting in a decrease in the circulation rate with increasing channel power. As can be concluded from Figure 5, there is a maximum in the circulation rate in dependence of system pressure at low channel powers too. At high saturation temperatures, i.e. high system pressures, the driving head decreases with increasing temperature owing to the decreasing void fraction. This results in a decreasing circulation rate. At low saturation temperatures, the subcooled region, defined as the region where the average bulk liquid temperature is below the local saturation temperature, increases with decreasing temperature, since the influence of the pressure difference between top and bottom of the coolant channel due to the hydrostatic head, becomes more significant. The driving head and hence the circulation rate will, therefore, decrease with decreasing pressure.

In Figure 4 the influence of the operating conditions on the void fraction α near the exit of the heated part of the coolant channel is given. It can be shown that in first approximation, $\alpha/(1-\alpha)$ is proportional to the channel power, which explains the decreasing slope of the void fraction versus power curves with increasing channel power.

An increase in pressure results in a decrease in void fraction, which is largely due to the increase in density per unit volume of steam. This effect is most significant in the low pressure range.

The ability to predict the steam volume fraction in a two-phase system as a function of the design and operating parameters is very important for a complete performance and stability evaluation of the system. The existing theoretical and empirical formulations correlate either the actual steam volume fraction or the phase velocity ratio V_s/V_l, usually called the slip ratio S, which is actually defined as: (see Ref. 1)

$$S = \frac{\int^{A_c} \alpha' V_s' \, dA}{\int^{A_c} (1-\alpha') V_l' \, dA} \cdot \frac{1-\alpha}{\alpha}$$

A_c is the cross-sectional area of the channel, and dA a small segment of it. The prime denotes local values with respect to the radius or dA. Therefore, the slip ratio is the ratio of the weighted averages of the phase velocities and not, as it is defined in some publications, the ratio of the averages of the phase velocities. From the equation it may be concluded that even where the local velocity of the steam phase equals that of the water phase (ic. $V_s' = V_l'$) the slip ratio may have an appreciable value. Furthermore, depending on the concentration and phase-velocity distribution, the slip ratio may also be smaller than unity. A slip ratio based on average values of the phase velocities is very difficult to establish and of no use for the designer.

To obtain more detailed information the measured void data have been plotted in the weighted mean steam velocity-average volumetric flux density plane, proposed by Zuber and Findlay (Ref. 3). In Figure 6 vertically is plotted the volumetric flow

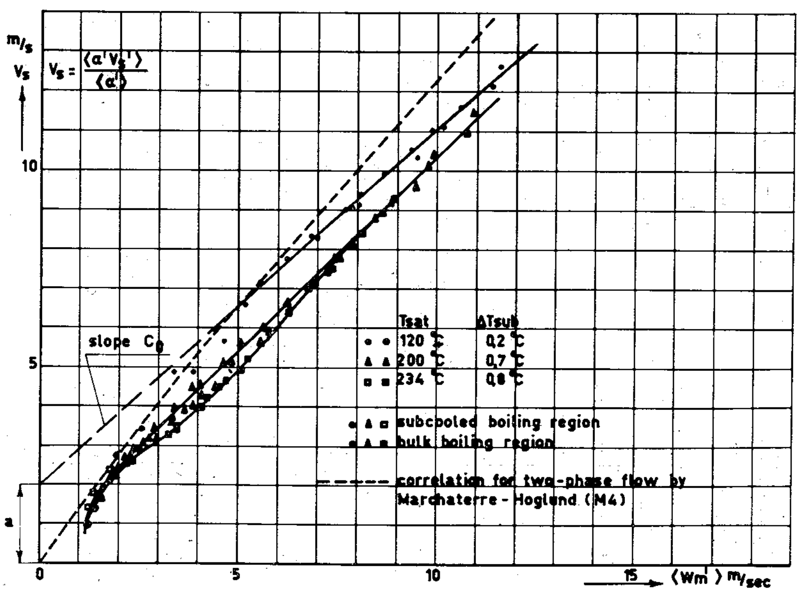


Fig. 6. Void fraction data plotted according to Zuber and Findlay (3) for three system pressures, Test Section I.

rate of steam per unit cross-sectional area divided by the void fraction and horizontally the volumetric flow per unit cross-sectional area. The prime denotes again local values with respect to the radius, and $< >$ denotes average values over the cross-section. According to Zuber and Findlay the relationship between the two quantities for a fully established flow profile and for a two-phase flow system in which a change of phase does not occur is a straight line. The slope of the line is called the distribution parameter C_0 which is determined by the flow profile over the cross-section and is less than unity when the void fraction near the wall is larger than in the center and above unity when the opposite is true. The intersection of the vertical axis, a, is determined by the local relative velocity between the two phases.

In Figure 6 the measured void fraction data are plotted. As is shown the linear relationship holds good even for an evaporative system and it may be concluded that fully developed flow conditions are present over the larger part of the coolant channel and throughout a wide range of operating conditions. Only at the lower volume flows a systematic deviation from the straight line is observed. Most of the data points in this region were obtained in lower parts of the boiling region, where the flow is probably developing. The slope of the lines is very near to unity which indicates that flat flow profiles for the velocity or concentration distribution and probably an established annular flow regime are present. The intersection with the vertical axis found by extrapolation of the straight line decreases with increasing system pressure being in agreement with the decreasing local slip between the two phases.

As is shown, the widely used Marchaterre - Hoglund correlation yields too high values of the weighted average steam velocity and this would result in too low values of the void fraction for a given volumetric flow of the mixture.

4.2. *The onset of flow oscillations*

In carrying out the steady-state measurements presented before, different types of flow oscillations appeared, although the independent quantities as channel power and condenser and subcooler heat removal were constant. It was possible to distinguish three types of oscillations having a frequency of roughly 0.03, 1.0 and 15.0 c.p.s. A systematic research has been carried out as regards the oscillations of the intermediate frequency only, which can be expected to occur and have been observed in the fuel channels of a boiling water reactor. Only these will presently be described.

In Figure 7 recordings are reproduced of the signal from the differential pressure gauge connected to the pitot-tube. These recordings have been made in preliminary tests with a test section similar to Test Section I. Three series of recordings for different system pressures are shown. In each series the channel power is increasing from the top to the bottom of the figure. For each single series, the sensitivity of the differential pressure gauge is constant, so that the recordings can directly be compared with each other. The output voltage of the pressure gauge has been translated into Newtons per square meter, using the manometer readings of the pitot-tube. By increasing the channel power, the average differential pressure decreases, which is in agreement with the character of the circulation rate versus channel power curve as given in Figure 5. By increasing the power from 87 to 90 kW at 120 °C saturation temperature, the onset of spontaneous flow oscillations can be observed. The frequency of these oscillations is about 1 c.p.s. while the amplitude is roughly 5 times larger than the average value at low power level. Furthermore, it can be concluded from Figure 7 that at this low system pressure flow reversal is present. At 200 °C saturation temperature the oscillations start at a higher power level, while they develop more gradually once the power is increased. Furthermore, the amplitude is smaller and the flow rate periodically drops to zero, but does not reverse. At even higher pressure (234 °C saturation temperature), the flow maintains a positive value.

The onset of severe hydraulic oscillations is clearly demonstrated in Figures 8 and 9. Vertically there is plotted the root mean square value of the fluctuating part of the signal from the differential pressure gauge indicating the variations in pressure loss and thus in mass flow rate across the inlet of the coolant channel. For comparison the steady-state value of the differential pressure across the inlet at the maximum in the relevant circulation rate versus channel power curve is also given.

The influence of system pressure on the onset of hydraulic oscillations is given in Figure 8. As is shown, the fluctuations in the pressure drop across the inlet at a saturation temperature of 120 °C are as large as 6 times the value ever obtained in steady-state conditions. At this temperature it is not difficult to determine the channel power at which hydraulic instabilities start. At a channel power of 65 kW, the fluctuations in pressure drop increase sharply. The system pressure has a stabilizing effect in that sense that, at higher pressures, severe oscillations start at higher channel power. Moreover, at higher system pressures, the onset is not sharply defined. There is hardly any abrupt change from stable into unstable operation as a result of a single incremental power change, but a steadily growing lack of stability is observed over a range of power inputs. The designer of nuclear reactors must decide what fluctuations should be tolerated and up to how far in the region of flow oscillations, operation is permissible.

At low subcoolings, the effect of increased subcooling upon the onset of severe hydraulic oscil-

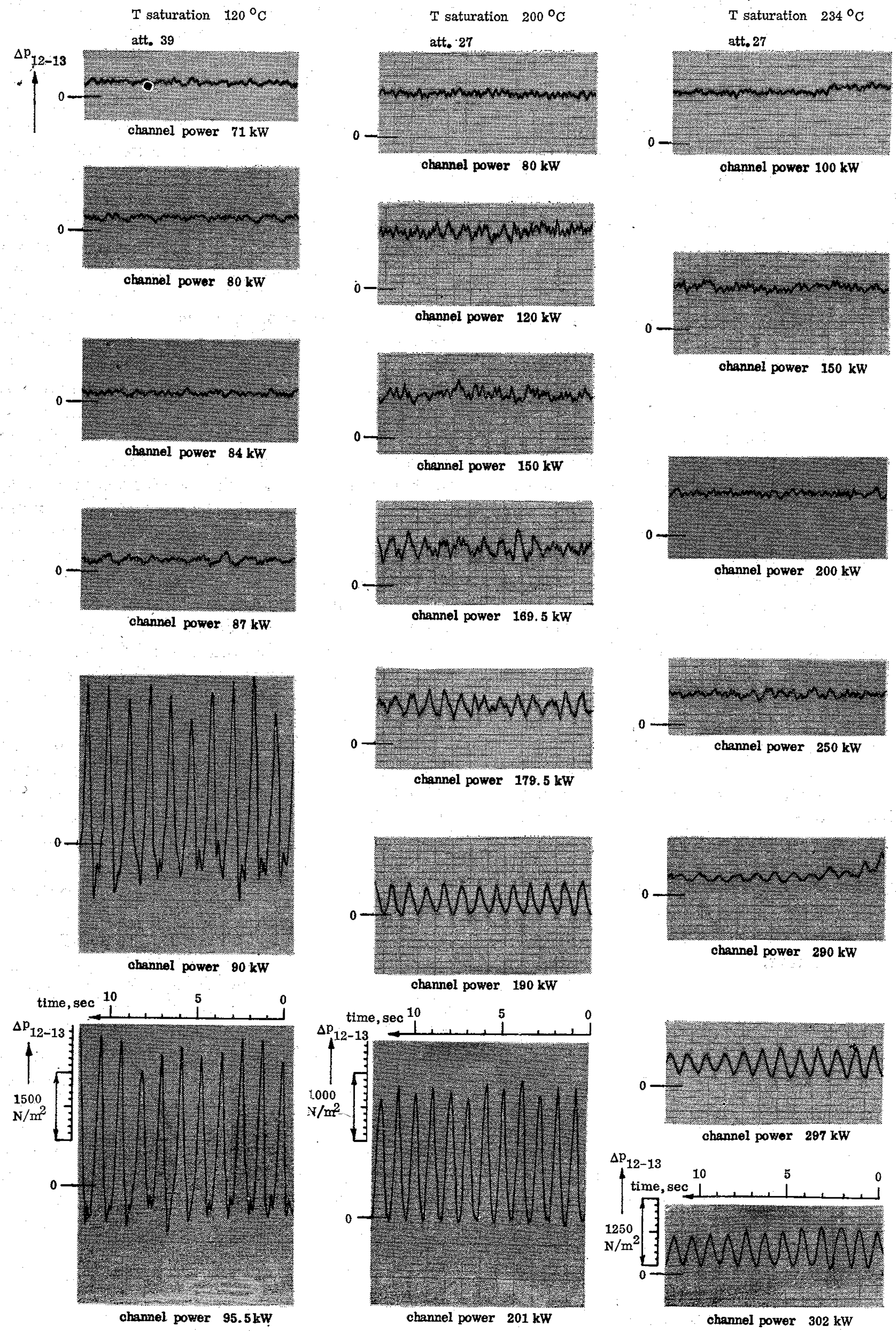


Fig. 7. Recordings of the signal from the pitot-tube.

lations is opposed to that at high subcoolings, as is shown in Figure 9. At low subcoolings, an increase in subcooling precipitates the onset of large flow oscillations. At high subcoolings an increase in subcooling postpones the onset of large flow oscillations. For instance, at 43 °C subcooling for Test Section I, severe hydraulic oscillations occur at about 250 kW channel power (Figure 9) compared with these oscillations starting at about 160 kW for 0.7 °C subcooling (Figure 8, in either case at 200 °C saturation temperature). On the contrary, at a subcooling of 11.0 °C the oscillations start at about 145 kW channel power. As a first approximation, it can be stated that the observed hydraulic oscillations occur only at a high value of the void fraction at the exit of the channel. Any increase in pressure and subcooling, or both, therefore, have a stabilizing effect. The destabilizing effect of subcooling is caused by the increase of the subcooled region, which will result in longer transport times and phase lags in the circuit. This destabilizing effect is predominant at low subcoolings.

The increase in hydraulic diameter has a stabilizing effect. The occurrence of severe hydraulic oscillations in Test Section II at a particular operating condition of pressure and subcooling has been shifted to higher channel powers compared with Test Section I. With Test Section II, the inverse effect of increased subcooling at low and high subcoolings is even more clearly demonstrated, see Figure 9.

As a rule, the inlet flow oscillations are measured primarily in experiments on hydrodynamic instability. The void fraction oscillations are of importance as well, since these oscillations produce oscillations in the nuclear power of a reactor. The behaviour of the steam-void in the boiling channel of Test Section I is shown in Figure 10 at a saturation temperature of 200 °C and a channel power of 160 kW. Recordings of the signal from the various impedance gauges for three different subcoolings, but at constant channel power and system pressure are given together with a recording from the differential pressure gauge connected to the pitot-tube. Though the mass flow at the inlet (i.e. the differential pressure from the pitot-tube) at a subcooling of 0.5 °C exhibits oscillations of a modulated character, the void is oscillating only in the lower part of the channel; only the lowest void detector gives evidence of any void oscillations. As mentioned before, the stability of the system will deteriorate as subcooling is increased. The modulation of the mass flow disappears at a subcooling of 3 °C and the void shows oscillations with a longer period over a substantial part of the boiling channel. The void oscillations are now maximal at void gauge 6, and not at the bottom of the channel. This location corresponds approximately with that where saturated boiling, as calculated from a heat balance, starts. By further increasing the subcooling to 9.5 °C, the maximum in void oscillations shifts to the location of void gauge 5. The void near the exit of the channel is now likewise oscillating with a small amplitude.

When one compares the signals under the last operating condition, a phase shift of roughly 180° can be observed between the oscillations in void at the bottom and the top. Futhermore, there is a difference of phase of 180° between the oscillations in mass flow and the steam-void at the bottom. These results have been confirmed by theoretical calculations.

From the foregoing it may be clear, that steady-state criteria for the onset of flow oscillations as for instance "the exit void fraction may not exceed a certain value" or "the power level must be less

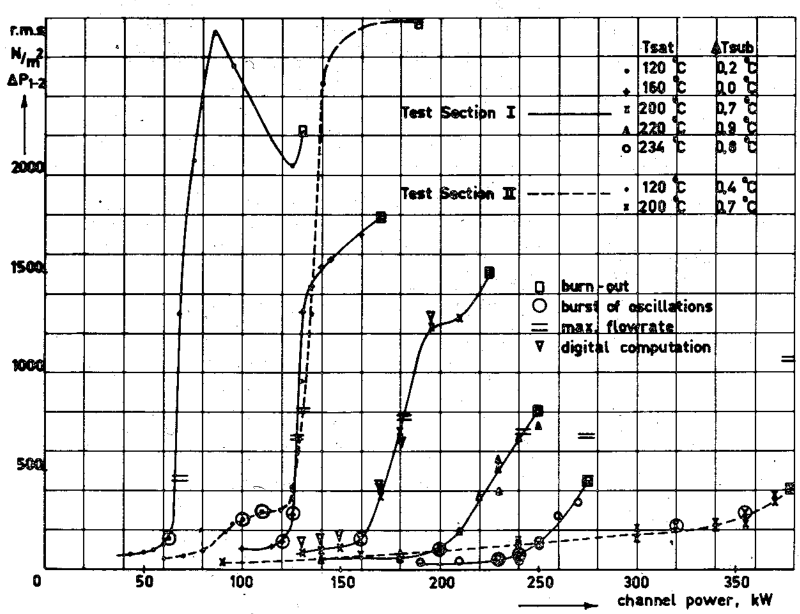


Fig. 8. The influence of system pressure on the onset of hydraulic instabilities, Test Section I and II.

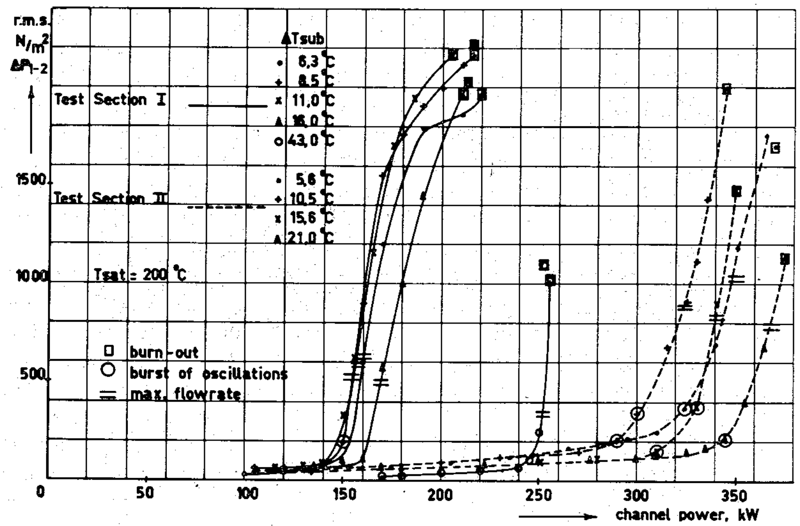


Fig. 9. The influence of subcooling on the onset of hydraulic instabilities, Test Section I and II.

than that corresponding to the maximum in the circulation rate versus channel power curve" (Figure 5), as proposed by some authors, are not meaningful.

4.3. *Stability measurements*

In order to obtain information on the stability and response characteristics of a steady-state condition of a boiling system, frequency response measurements were carried out in the region of 0.01 to 2 c.p.s. In these measurements the heating power to the boiling loop was oscillated with a sine of small amplitude and varying frequency and the resulting time-dependent variation in the physical quantities, i.e. the amplitude and phase relationship between input and output, was observed. In the following this relationship is called the transfer function and it will be shown that transfer functions and the occurrence of severe hydraullic oscillations are closely related subjects.

In Figure 11 the transfer function is given from channel power to the differential pressure from the pitot-tube (i.e. the inlet mass flow rate) for Test Section I, for three values of the channel power at a saturation temperature of 200 °C. The measured value of the amplitude of the responding signal per kW channel power oscillation was made dimensionless by dividing it by the corresponding value at zero frequency obtained from steady-state measurements.

From the amplification characteristics it can be concluded that a resonance peak is present which increases in magnitude with channel power, while also the peak shifts to a higher frequency. Comparison of these results in this frequency region with those presented by St. Pierre (Ref. 4), for a forcedly circulating boiling system, which are showing no resonance peak, leads to the conclusion that these resonance peaks are characteristic for a naturally circulating system. In contrast to a forcedly circulating system, in a naturally circulating system the inlet mass flow and the steam-void are strongly intercoupled. The resonance peak, therefore, is caused by this intercoupling effect. The amplitude ratios together with the behaviour of the phase shifts show that the system is only weakly damped. By increasing the channel power, the damping forces become relatively smaller and the system becomes less stable. At the highest channel power of 145 kW, which was only about 20 kW lower than the channel power where spontaneous severe hydraulic oscillations start, a sharp falling off in the phase shift is observed, indicating that the system approaches an unstable condition. It can be claimed that then the resonance peak has been transformed into spontaneous flow oscillations owing to the fact that the dynamic intercoupling between the steam-void in the channel and the inlet mass flow has become unstable and not because the flow is responding to a present boiling instability, or flow-pattern instability. A study of the onset of flow oscillations, therefore, cannot be made from the equations describing the steady-state, but must start from equations incorporating dynamic effects. The Ledinegg instability (Ref. 5), cannot be the explanation for the flow oscillations under discussion. For determining the onset of flow oscillations, the describing equations may be linearized. For determining the characteristics of the flow oscillations non-linear effects must be introduced into the equations.

In Figure 12, the influence of subcooling on the stability of a steady-state is shown. Results of transfer functions are given for Test Section I at a saturation temperature of 200 °C and a channel power of 113 kW. The transfer function from channel power to the differential pressure from the pitot-tube has been plotted for three values of subcooling. Comparing the curves of 1.5 °C and 10.8 °C subcooling at the corresponding resonance frequency shows that the amplitude ratio at 10.8 °C subcooling is greater than at 1.5 °C subcooling at

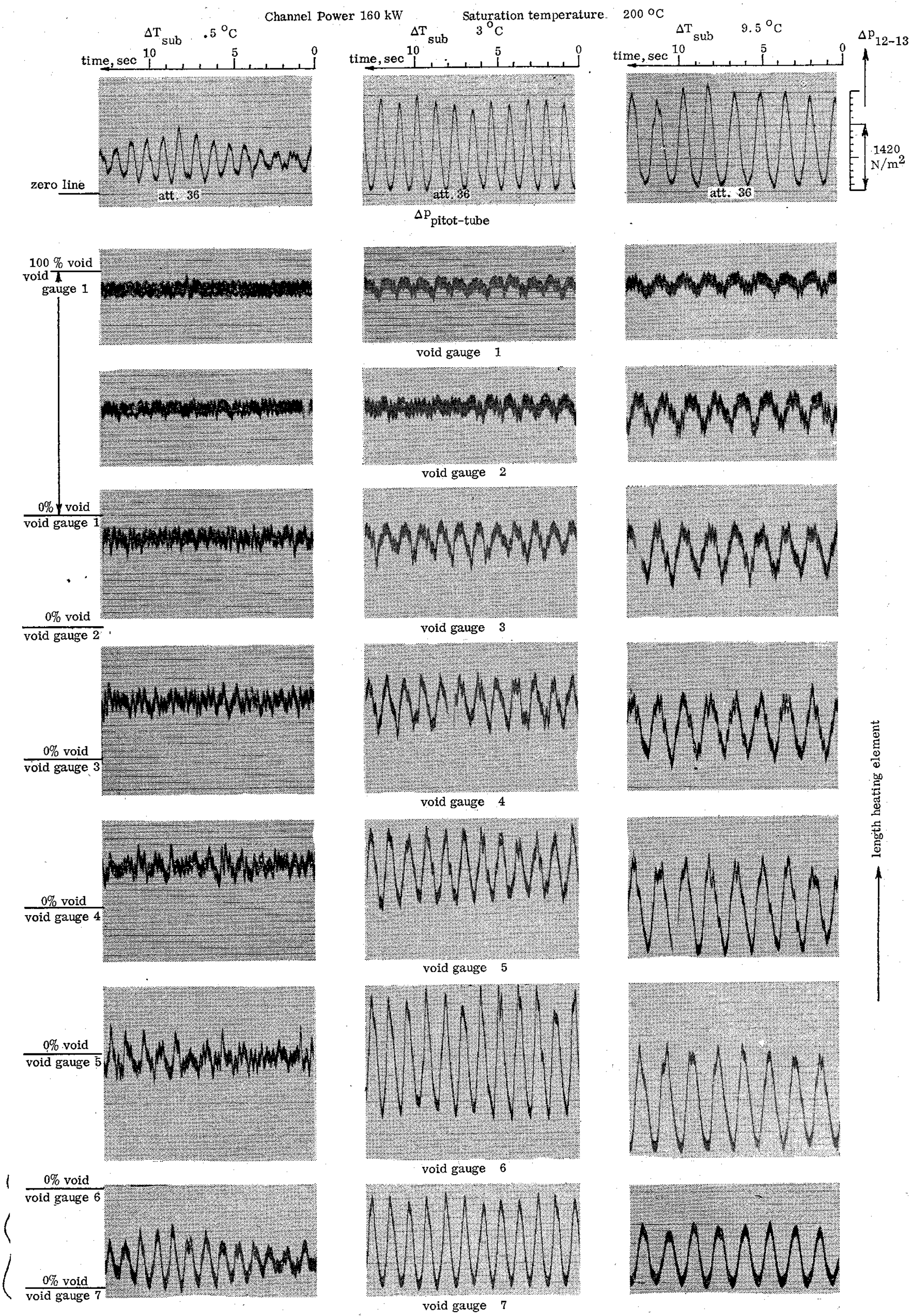


Fig. 10. Recordings of the signals from the various void gauges and the pitot-tube, Test Section I.

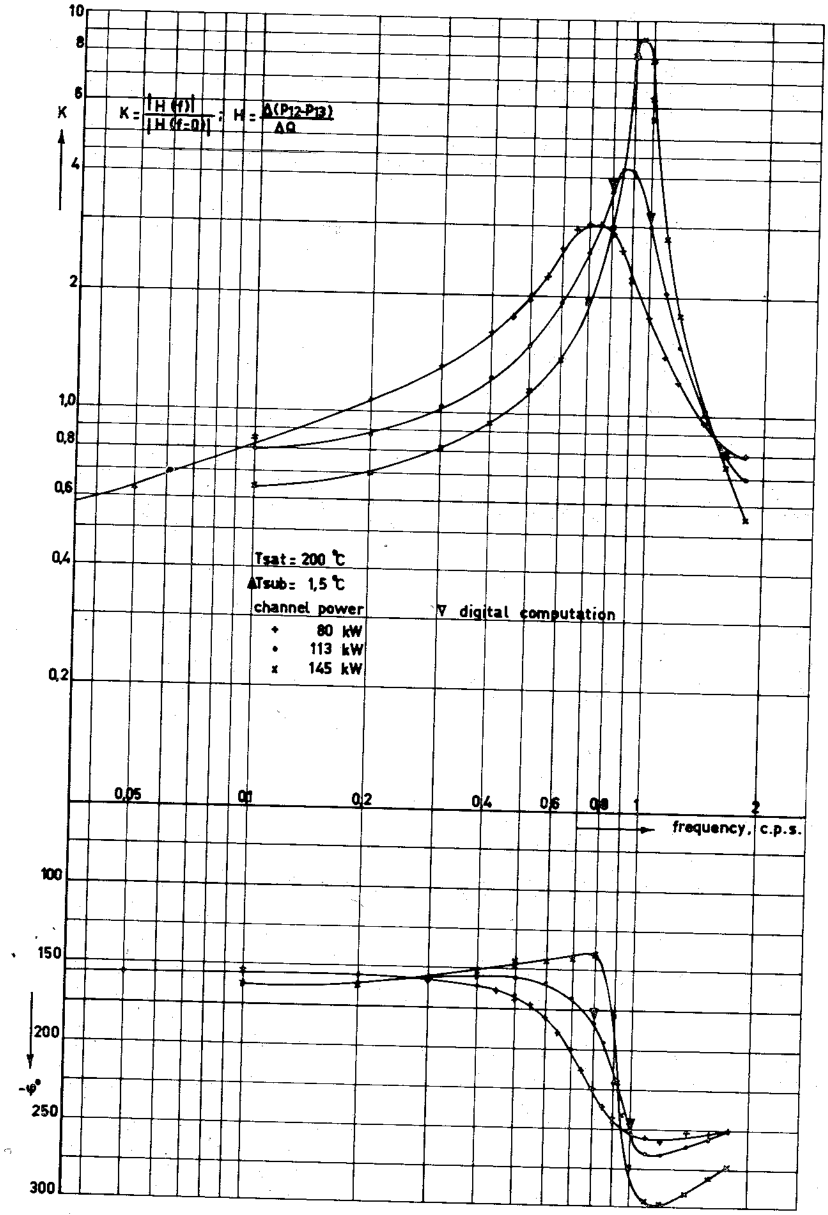


Fig. 11. Transfer functions from channel power to inlet mass flow for various channel powers, Test Section I.

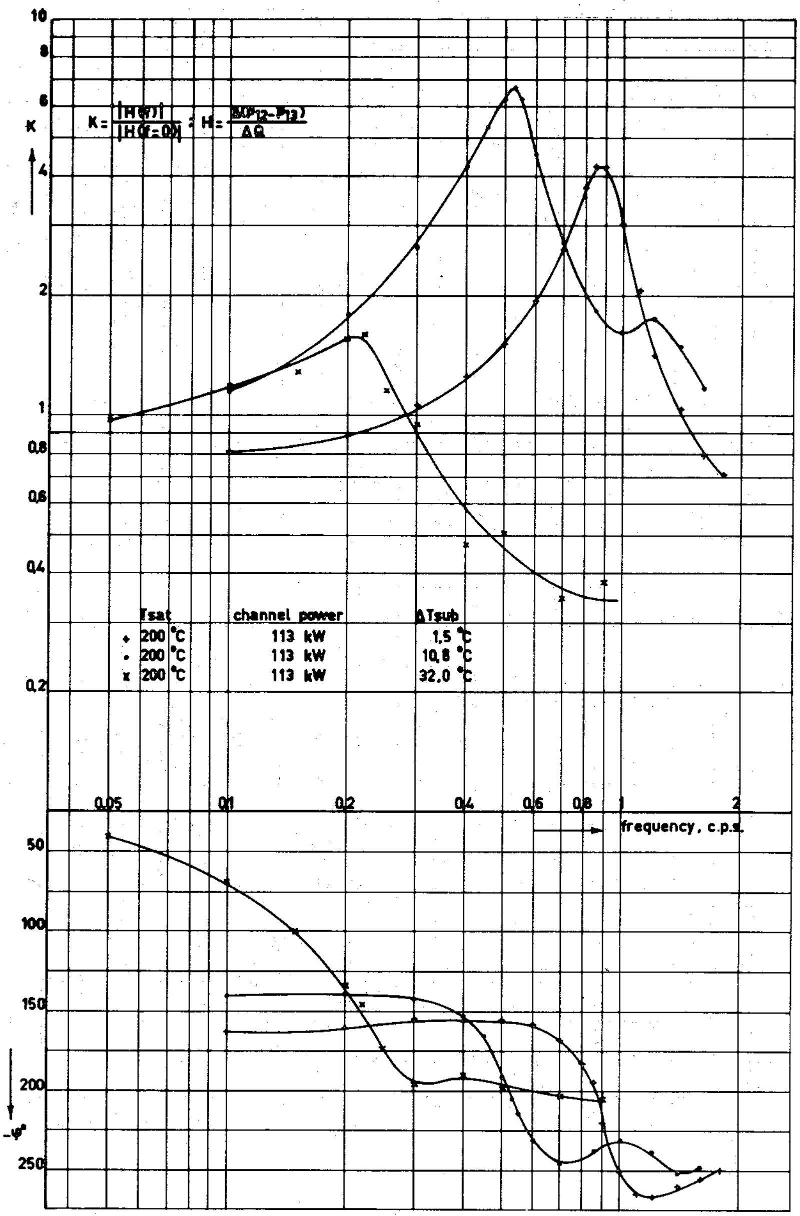
about the same phase shift. This indicates that increased subcooling in the lower subcooling range does indeed impair the stability of the steady-state. At high subcooling rates, the opposite is true. Increasing subcooling from 10.8 °C to 32.0 °C results in lowering the resonance peak, whilst also the phase shift is reduced. This indicates that at high subcooling rates, any further increase in subcooling has a stabilizing effect. The opposite effects of increasing subcooling in the low and high subcooling ranges are similar to the effects of subcooling observed in the experiments carried out to establish the onset of hydraulic oscillations.

4.4. *Burn-out*

All the experimental series for measuring the steady-state characteristics and the onset of hydraulic oscillations have been carried out by increasing the channel power at a constant system pressure and a constant inlet-subcooling. In all series the channel power was increased in small steps until the burn-out trip was reached.

The burn-out heat fluxes (e.g. the channel power at which burn-out trip occurred divided by the area of the heated surface) are plotted as functions of the saturation temperature and subcooling temperature in Figures 13 and 14 respectively. In these figures also the heat fluxes at the instability threshold have been plotted. As can be concluded from Figure 13 the burn-out heat flux increases with saturation temperature and thus with system pressure. This is a result similar to that obtained

Fig. 12. Transfer functions from channel power to inlet mass flow at various subcoolings, Test Section I.



in forced-circulation experiments under stable flow conditions and at low system pressures. It is observed that at higher saturation temperatures the curve of the threshold of instability approaches the burn-out curve. With Test Section II and at a system pressure of 30.7 ata (saturation temperature of 234 °C) burn-out was obtained without the flow passing through the oscillation region.

The curve, representing the burn-out heat flux as a function of subcooling passes through a minimum. The trend of the burn-out heat flux decreasing with increased subcooling for low subcooling rates is the opposite to what is normally observed in burn-out experiments with forced circulation, in which the burn-out heat flux generally decreases with increasing steam quality at the outlet. One explanation may be that unstable natural convection burn-out for low subcooling rates resembles that with employing a "soft inlet" (i.e. steam quality at the inlet), when at a low flow rate and at low pressure the trend of the burn-out heat flux increasing with increasing quality at the burn-out point has been observed at low quality. At high subcooling temperatures the burn-out heat flux increases again with increased subcooling similar to the behaviour of the instability threshold. Then, the flow oscillations become smaller and the subcooled region larger. Because burn-out occurred almost only under unstable conditions, the measured values can be expected to be low compared with data obtained under stable conditions with natural or forced

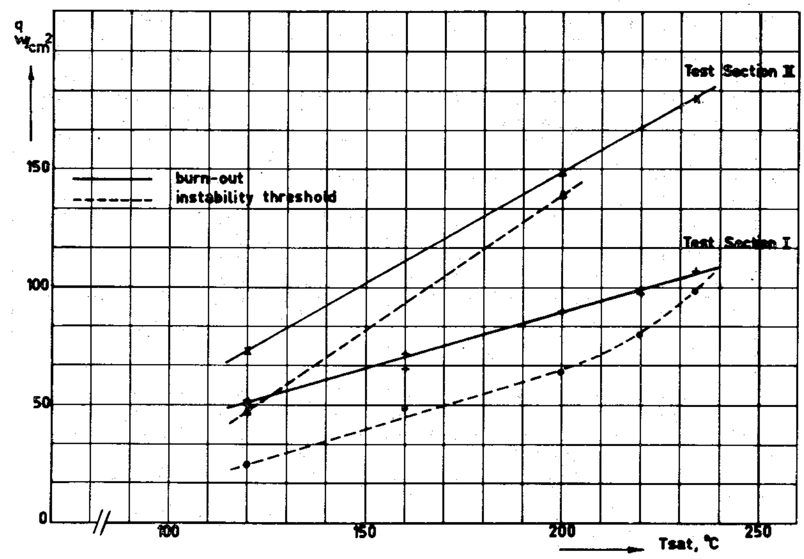


Fig. 13. Heat fluxes at burn-out and instability threshold as functions of saturation temperature, Test Section I and II.

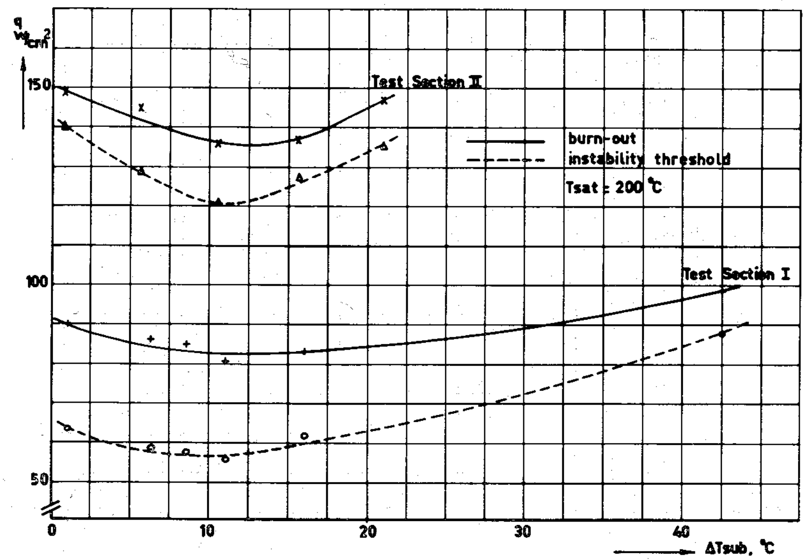


Fig. 14. Heat fluxes at burn-out and instability threshold as functions of subcooling temperature, Test Section I and II.

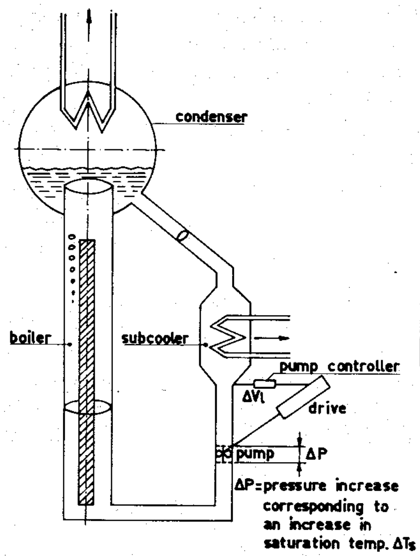
circulation conditions. Evidence of this is provided by the measurements reported in Ref. 6.

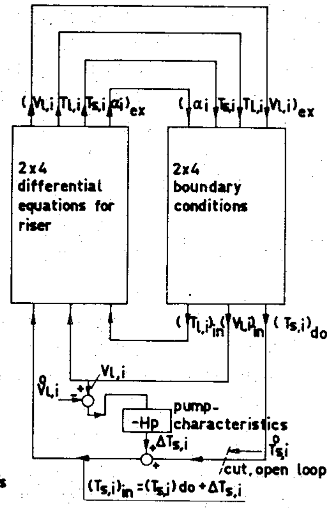
5. Theoretical studies

For making an analysis of the characteristics of a boiling system the four equations have been formulated, describing the steady-state and transient behaviour of a two-phase flow system (Ref. 7). These four equations are the laws of conservation of mass, momentum and energy for the mixture and an equation describing the transport of steam in axial and radial direction, determined by the process of bubble diffusion, interaction effects between bubbles etc. Owing to the complex nature of this last equation and the difficulties involved in using this equation in a performance calculation of a two-phase system another equation has been formulated for determining the quantity of steam. For that purpose, the energy equation is split up into two separate equations, one equation governing the warming up of the liquid phase of the flow and a second representing the heat supply to the vapour part (Ref. 1). In the four equations formulated the independent variables are the time and the space coordinate; the four dependent variables are the void fraction α, the temperature of the liquid and vapour phase T_l, and T_s (local saturation temperature) and the velocity of the liquid phase V_l. The other variables present in the equations are determined by the equations of state and by the correlation functions for the slip ratio, the two-phase friction losses, and the ratio of the heat supplied to the steam phase and the total heat input. The boundary conditions for the boiling channel and thus the describing equations are formed by the

Fig. 15. Natural circulation boiler (left).

Fig. 16. Block diagram of natural circulation boiler (right).



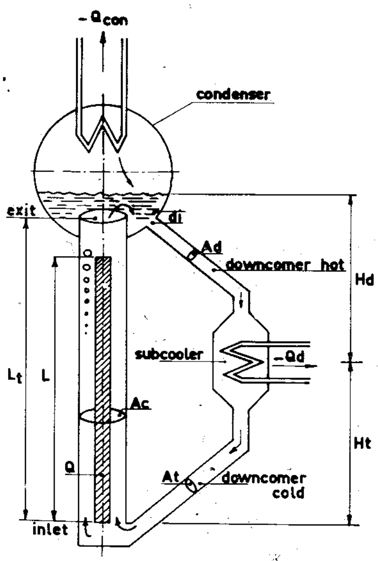


condenser, downcomer, subcooler etc. Also these have been formulated in terms of the conservation laws. All these equations have been linearized while it is assumed that the variations of the variables with time are harmonic. The harmonic variations are complex quantities and have to be expressed in their real and imaginary constituents. The ultimate result of this is the doubling of the number of equations and variables in the unsteady case. The procedure is shown schematically in Figures 15 and 16. The equations have been programmed for a digital computer (Ref. 8).

The hydraulic instabilities that are of interest here are those that are typical for a natural circulation system. In a forced circulation boiler with steep head-flow characteristics, no hydraulic instabilities, such as considered here, have been found. This suggests considering a forced circulation boiler, as shown in Figure 17. A pump is present in the downcomer, which pump generates a pressure rise, corresponding with a rise in saturation temperature $\triangle T_s$. The pump measures the fluctuations in mass flow $V_{l,i}$ and translates these variations into a rise in saturation temperature of $\triangle T_{s,i}$. Now it is assumed that the system is brought into excitation by controlling the pump with a sinusoidal signal corresponding to a desired fluctuation in mass flow $\mathring{V}_{l,i}$. The magnitude of $\triangle T_{s,i}$ (see Figure 18) is dependent upon the difference between $\mathring{V}_{l,i}$ and $V_{l,i}$. H_p is a measure of the pump characteristics. The transfer functions for the open-loop without a pump and with a cut, as indicated by the broken line in Figure 18, are defined as:

$$G_1 = \left\{ \frac{(T_{s,i})_{do}}{(\mathring{T}_{s,i})_{in}} \right\}_{H_p = 0}, \text{ and}$$

$$G_2 = \left\{ \frac{V_{l,i}}{(\mathring{T}_{s,i})_{in}} \right\}_{H_p = 0}$$



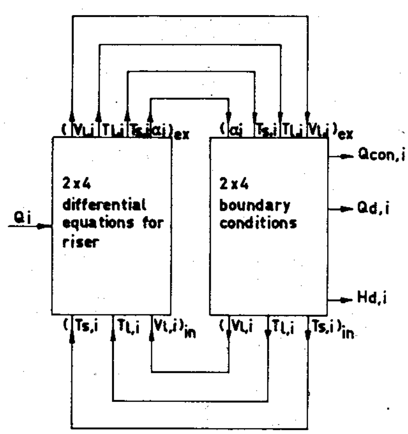


Fig. 17. Forced circulation boiler (left).

Fig. 18. Block diagram of forced circulation boiler (right)

Here G_1 and G_2 determine the change in saturation temperature (pressure) and mass flow rate at the outlet of the downcomer upon a variation in saturation temperature (pressure) at the inlet.
The transfer function for the closed-loop from imposed modulation in mass flow to the saturation temperature at the inlet of the channel can then be written as:

$$\frac{T_{s,i}}{H_p \mathring{V}_{l,i}} = \frac{H_p}{1 - G_1 + H_p G_2}$$

An instability condition is obtained when this transfer function approaches infinity. There are two situations where this is indeed the case:

a. For large values of H_p, the system will always be stable, as long as G_2 does not approach zero. From the definition of G_2, it may be concluded that G_2 will normally never approach zero, unless resonance conditions appear within the boiling channel.

b. For sufficiently low values of H_p, an instability condition is obtained when G_1 becomes $+1$ for some frequency.

It appears to be advantageous, therefore, to calculate the open-loop transfer functions G_1 and G_2 and see wether the modulus of G_1 approaches a condition of $+1$, and at the same time whether the phase angle has reached a value of about 0° or 360° for $H_p = 0$, or whether the modulus of G_2

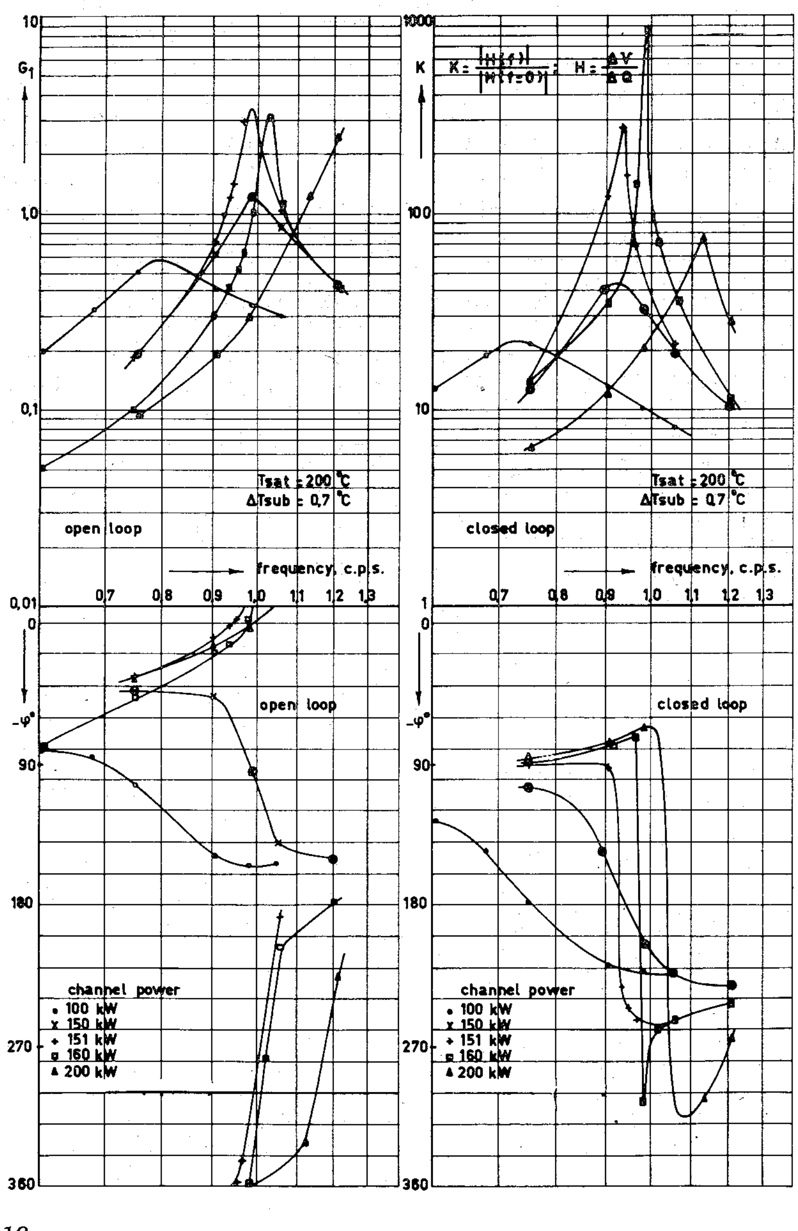


Fig. 19. Open and closed-loop transfer functions in the intermediate frequency range. Test Section I.

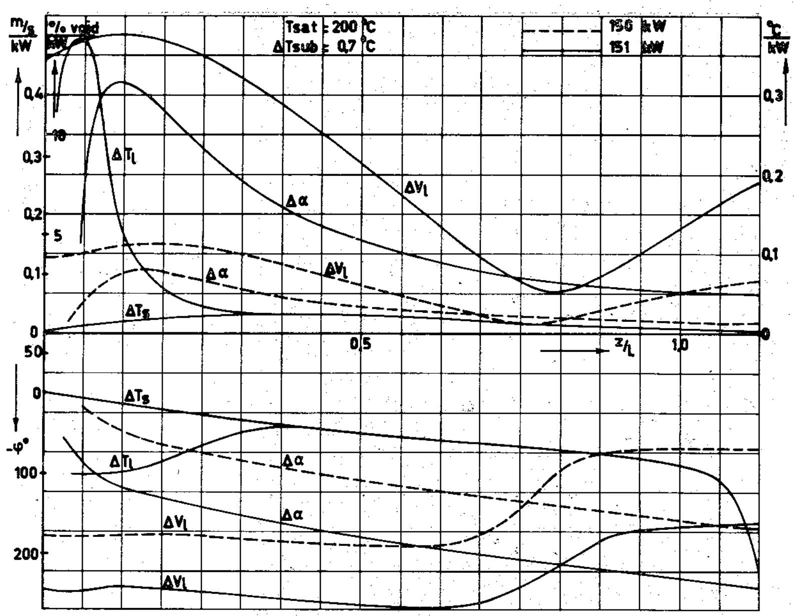


Fig. 20. Calculated longitudinal distributions, 0.947 c.p.s., Test Section I.

has become sufficiently small, for large values of H_p.

In Figure 19, open and closed-loop calculated characteristics have been plotted for a saturation temperature of 200 °C in the frequency range of 0.6 to 1.4 c.p.s. for different channel powers. In the plot of the open-loop characteristics, G_1, it is shown that an unstable condition is passed when progressing from 150 to 151 kW channel power. At 150 kW the modulus of G_1 becomes larger than unity but the phase angle does not approach the value of 0 or 360°. At 151 kW the modulus of G_1 is larger than unity and the phase angle becomes zero, which indicates that the system is unstable. A comparison between the open and closed-loop results at 150 and 151 kW leads to the conclusion that large amplitudes in inlet mass flow are present. The predicted instability threshold of 151 kW at a frequency of 0.947 c.p.s. is in fairly good agreement with the experimentally determined values of 162 kW and a frequency of 0.93 c.p.s. (Figure 8).

In Figure 20, the calculated distributions along the channel of the variation in the saturation temperature $\triangle T_s$, in liquid temperature $\triangle T_l$, in the liquid velocity $\triangle V_l$ and in the void fraction $\triangle \alpha$ upon a sinusoidal variation in channel power are shown for a frequency of 0.947 c.p.s. and at 151 kW channel power, just after the predicted onset of hydraulic instabilities. At 150 kW channel power, just before the predicted onset, curves are only given for the void fraction and the liquid velocity. As is shown, when the system is unstable (151 kW), there is a difference in phase of about 180° between the void fractions at the inlet and exit of the channel and between the void fraction at the inlet and the inlet mass flow rate. The oscillations in void fraction are largest at the bottom of the channel. These results correspond to the observations made in the recorded signals during hydraulic oscillations, see for instance Figure 10.

The oscillations in liquid velocity at a channel power of 151 kW are largest at the inlet and outlet of the channel and smallest at about 0.8 L from the bottom of the channel. At this minimum, a large variation in phase shift of about 120° occurs. The fluctuations at the inlet and outlet are, therefore, about opposed in phase and resemble more or less a standing wave, i.e. it represents approximately a half-wave length oscillation in the velocity distribution.

Besides the 1 c.p.s. oscillations, also the other observed flow oscillations, mentioned in the beginning of section 4.2 could theoretically be determined. The low frequency oscillations are related to the Ledinegg instability (Ref. 5) and the high frequency oscillations to the acoustical characteristics of the channel, connected with the compressibility characteristics of the gas-phase.

6. Concluding remarks

In the foregoing some results of an experimental and theoretical study have been reported on the steady-state and stability characteristics of a naturally circulating boiler. The experimental research programme is continued by studying the influence of the time response of the heating element and the incorporation of a pump in the downcomer on the two-phase flow characteristics. In the theory, further analysis work will be carried out regarding the phenomena govering the onset and character of the hydraulic instabilities and concerning the choice of parameters which are of less significance and may hence be safely disregarded.

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The Steam Generating Heavy Water Reactor

by H. Cartwright

Introduction

The Steam Generating Heavy Water Reactor (SGHWR) is a direct-cycle, boiling light-water reactor of pressure-tube design, moderated partly by heavy water and partly by the light-water coolant. The first power station of this type has been designed and is being constructed by the United Kingdom Atomic Energy Authority (UKAEA) at its Winfrith site in Dorset, England, with the co-operation of British industry.

The pressure-tube design permits extrapolation of the present 100 megawatts (electrical) (MW (e)) design to larger sizes, without introducing new technical problems, simply by increasing the number of tubes while retaining the same fuel element.

Selection of a nuclear power station for utility use must be based primarily on economic factors, backed by adequate built-in safety. While the cost per unit of electricity produced from any nuclear power station depends on the site selected, the customer's requirements and other variables, certain features of reactor systems can be recognised as fundamentally desirable and likely to lead to low costs with a high degree of safety. For the SGHWR these are:

(a) simplicity of a direct cycle with no external boilers;

(b) absence of a pressure vessel;

(c) factory fabrication and initial assembly of the main components, leading to high standards of integrity;

(d) ease of refuelling (based on a single fuel element per channel) either on-load or off-load, leading to high availability;

(e) good neutron economy of the system, which demands relatively low enrichment of fuel (SGHWR's using natural uranium are a future possibility);

(f) separation of civil work and plant erection phases.

Economically Attractive

The SGHWR is economically attractive and can readily be incorporated into a customer's system. The reactor core consists of a bank of pressure tubes which pass through vertical tubes in a calandria or tank containing the heavy-water moderator. Each pressure tube and calandria tube is separated by a gas gap. This arrangement gives good thermal efficiency and neutron economy, and has safety advantages over designs in which the pressure tube is cooled by the moderator.

In the Winfrith Heath prototype, there are 112 pressure tubes fabricated from zirconium alloy.