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Instability and unproportional pressure variations near the thermodynamic critical point in a closed thermosyphon

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Abstract—Unproportional pressure variations are observed in a closed thermosyphon during the transition from subcritical to supercritical states and vice versa. Such a stepwise pressure variation is encountered at lower pressures and with a larger step when the heat flow rate is large and the tube is steep. The unsteady behaviour is explained by an instability occurring in the two phase counterflow of vapour and liquid: at high subcritical pressures, the densities of liquid and vapour become equal and reduce buoyancy and separation of streams. This brings forth the collapse of the internal circulation flow. Only when this flow is re-established in the one phase supercritical region—with a preceding unproportional pressure increase—steady-state operating conditions are obtained again.

1. INTRODUCTION

A CLOSED thermosyphon—sometimes called a 'wickless heat pipe'—is a device for heat transmission. It consists of a straight or curved tube, filled with a fluid and closed on both ends. The tube is used in an inclined or vertical position; heat is supplied to the lower part (heating zone) and it is removed from the upper part (cooling zone). In between these zones there is an adiabatic transport zone. Temperature differences inside the thermosyphon cause density differences in the fluid and a natural circulation flow. Natural circulation requires a body force field, which is provided by gravity in most cases, and causes an upward heat transport. In rotating machinery centrifugal acceleration may be used for a radial heat transport. Fluid particles start to move in the direction of the body force field, as soon as heat is removed from the cooling zone.

This buoyancy-driven circulation is coupled with the transport of heat from the heating zone to the cooling zone, bringing forth a rather high effective thermal conductivity if the tube is considered as a conducting rod. The fluid may be of single phase (liquid or vapour) or of two phase (liquid and vapour), which occurs at subcritical pressures only. In such a 'two-phase thermosyphon', evaporation of liquid takes place in the heating zone and condensation in the cooling zone, respectively.

Natural convection with or without a change of phase is caused by a simultaneous transfer of heat and momentum. Besides a number of other influences—like intensity of the body force field, geometry of the tube, heat flow rate—the thermophysical properties of the fluid are of special importance. These depend on the kind and the state of the fluid. It is well known that heat transfer in a fluid near its thermodynamic critical state is highly effective as many of the properties approach extreme values.

Schmidt *et al.* [1] first suggested and used this type of heat transport in a thermosyphon in 1939. In their experiments, a closed loop filled with NH_3 was used. Encouraged by the results, several further investigations grew out of this work. The closed loop arrangement was pursued in some papers (e.g. Tanger *et al.* [2]), and recently new importance arose in applications for nuclear reactor cooling. The straight tube closed thermosyphon—which is a simpler and inexpensive version of the loop—has been studied by a great number of authors in the last years, but only Schmidt [3] (with NH_3 and CO_2 as working fluids), Hahne [4] (with CO_2), Stoyanov [5] (with NH_3), and Groß [6–8] (with R115) investigated the behaviour in near-critical states. A review is given in [7].

Most of the papers, published in the last few years, deal with the 'maximum heat flow rate', as the limitation to the operating range. This limitation is reached when the mechanism of fluid-flow changes in some part of the thermosyphon bringing forth a drastic degradation in heat transfer. Many authors investigated the most important limitations:

Excess or lack of liquid fluid. The volume fractions of the gaseous and liquid part of the fluid inside the tube depend on the pressure and the amount of fluid filled in: at high pressures there may be a 'dryout' if the amount of fluid is too small [9–16]. If this amount is too large, the entire fluid becomes liquid [9, 10]. In any case there occurs a transition from a 'two-phase thermosyphon' to a 'single-phase thermosyphon' with a change of transport mechanism inside the tube.

Departure from nucleate boiling (*DNB*). DNB is observed on the inner heating surface, when intensive bubble nucleation impedes a sufficient onflow of liquid to the heating surface. The result is a rather large wall superheating [9, 13, 14, 17, 18].

Entrainment, flooding. Entrainment and, in con-

NOMENCLATURE

A	area
d	diameter
Δh_v	enthalpy of evaporation
L	length
m	mass
$\dot{m}$	mass flow rate
p	pressure
$\dot{Q}$	heat flow rate
s	thickness of a wall
U	voltage
V	volume
$\dot{V}$	volume flow rate
v	specific volume.

Greek symbols

α	heat transfer coefficient
Δ	difference
ϑ	temperature
λ	thermal conductivity
ρ	density
φ	angle of inclination
ψ	volume fraction of liquid.

Subscripts

c	critical
CW	cooling water
CZ	cooling zone
HZ	heating zone
i	inner
ref	reference
s	saturation
st	steel
TZ	transport zone.

Superscripts

$'$	saturated liquid
$''$	saturated vapour
$\bar{}$	mean value.

sequence, flooding are caused by an instability of the counterflow of vapour and liquid. Upstreaming vapour exerts a shear stress upon the liquid surface near the wall. Entrainment is established when particles of liquid are drawn into the upwards directed gas stream. As long as the carry-over is small, heat transfer is somewhat impeded but not basically changed [13, 19, 20]. If the amount of liquid, extracted by this shear stress action, exceeds some limit, the performance of the thermosyphon becomes periodic with an occasional 'dryout' in the heating zone and flooding in the cooling zone [14, 18, 21–30].

Pressure variations in the near critical state. Hahne [4] observed an unproportional pressure increase at the transition from subcritical to supercritical states: With stepwise increased heat flow rates, a proportional increase in fluid temperature and pressure and a steady-state heat transmission could be observed up to a certain subcritical pressure close to the critical. Exceeding this by a slight increase of the heat flow rate, temperatures and pressure started to rise rather strongly, and a steady state could only be achieved far above the critical point. This rise in pressure and temperature was extremely large for small inclination angles of the tube (steep tube). At inclination angles $\varphi < 40°$ the pressure became dangerously high before steady-state conditions could be obtained again, and the power supply had to be switched off. With a thin metal sheet inserted lengthwise into the tube, the unproportional pressure increase could be avoided. Steady-state measurements in the vicinity of the critical state now became possible even for the vertical tube.

So far, there was no explanation for this phenomenon. Based on new, more detailed experiments, an attempt is now made to present possible explanations.

2. EXPERIMENTS

2.1 *Experimental apparatus*

Figure 1 shows the experimental apparatus with its principal item, the thermosyphon, made of a straight circular steel tube (RSt 37), closed on both ends. In its lower part (heating zone) the steel tube is shrouded by a copper cylinder ($s = 15$ mm) with 12 electric heating elements imbedded in the axial direction. These are equally spaced around the perimeter and supplied by direct current in parallel. In the upper part (cooling zone) the steel tube, again shrouded by a copper cylinder, extends into an annulus in which water flows upwards in axial direction with a maximum temperature rise of 0.3 K. The adiabatic part between the heating and cooling zones is called the transport zone. The whole thermosyphon is well insulated and swivel-mounted in a frame. The angle of inclination with respect to the vertical may be varied from $\varphi = 0°$ (vertical) to $\varphi = 90°$ (horizontal).

2.2 *Working fluid*

As working fluid, refrigerant R115 (pentafluoromonochloroethane, C_2ClF_5) was used, with the critical data:

$$p_c = 31.258 \text{ bar} \quad \vartheta_c = 80.0\ °\text{C}$$

$$\rho_c = 591.4 \text{ kg m}^{-3}.$$

Two phases coexist in the subcritical state: the lower

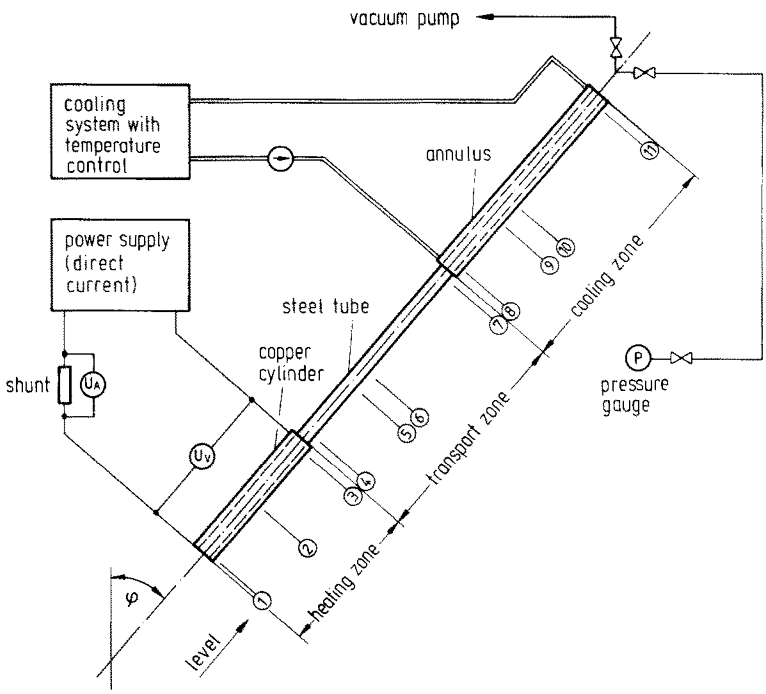

FIG. 1. Experimental apparatus.

part of the tube is filled with liquid, the upper part contains vapour.

The amount of fluid to be added is chosen to correspond exactly to the critical density ($\bar{\rho}/\rho_c = 1.0$). In this case, the volume fraction of liquid in the thermosyphon is somewhat less than 50% and almost independent of pressure when $p/p_c > 0.5$ (see Appendix). This holds for many pure fluids as shown by Van Poolen [31].

2.3. *Measurements*

Measurements were performed under steady-state conditions. The aim was to quantify the internal heat transfer coefficients. To this end, electric power, temperatures of the tube wall, pressure of the fluid, and ambient pressure and temperature have to be measured.

The electric power is obtained from the direct current and voltage. The voltage U_V is measured directly at the heating elements, the current can be taken from the voltage drop U_A at the shunt (Fig. 1). The calculated electric power has to be corrected by two factors, considering electric losses in the connections to the power source and heat losses though the isolation to the environment.

Outside wall temperatures are measured at 66 locations (Fig. 1): In 11 levels respectively, six thermocouples are welded at equal intervals around the outer surface to the steel tube, levels 1–3 are in the heating zone, levels 8–11 in the cooling zone. Measurements from all these thermocouples are taken 10 times within 10 min. Temperatures at the inner wall surface ϑ_{HZ} and ϑ_{CZ} are calculated from these data taking temporal mean values.

The fluid pressure is measured by a pressure gauge, mounted midway between the heating and cooling zone and connected with the thermosyphon at its upper end by a thin duct without isolation. The system

Table 1. Data of the experimental apparatus

Length of the heating zone	L_{HZ}	$= 501$ mm
Length of the transport zone	L_{TZ}	$= 746$ mm
Length of the cooling zone	L_{CZ}	$= 765$ mm
Inner diameter of the tube	d_i	$= 40$ mm
Thickness of the wall	s	$= 2$ mm
Inner volume	V	$= 2.514 \times 10^{-3}$ m^3
Area of the heating zone ($\pi d_i L_{HZ}$)	A_{HZ}	$= 0.06296$ m^2
Area of the cooling zone ($\pi d_i L_{CZ}$)	A_{CZ}	$= 0.09613$ m^2
Thermal conductivity of the steel tube	λ_{st}	$= 50$ W K^{-1} m^{-1}

pressure at the upper end of the thermosyphon is calculated from the pressure gauge reading, ambient pressure, and hydrostatic corrections.

2.4. *Parameters*

The following parameters were varied systematically in order to investigate the unproportional behaviour of the pressure near the critical state:

—angle of inclination ($\varphi = 0°, 10°, 20°, 40°, 60°$)
—heat flow rate ($\dot{Q} = 175, 375, 575, 775, 975, 1475, 1975$ W)
—cooling water temperature ($\vartheta_{CW} = 48.9$–81.5 °C).

At a constant tube inclination and a constant heat flow rate, the cooling water temperature was changed in steps, thus creating new thermodynamic states within the tube. A 'test point' is characterized by a certain combination of fluid pressure, heat flow rate, and inclination angle. A 'test series' consists of a number of test points, taken at constant heat flow rate and constant inclination angle, but at increasing and decreasing values of the fluid pressure.

3. RESULTS

3.1. *Relation between temperatures and fluid pressure*

Figure 2(a) shows the relation between fluid pressure and inside wall temperatures for one selected test series ($\dot{Q} = 775$ W, $\varphi = 60°$) in a pressure–temperature diagram. The diagram contains the boiling line, which gives the pressure p and the corresponding saturation temperature $\vartheta_s(p)$. The thermodynamic states of saturated vapour and saturated liquid are found on this curve for every subcritical pressure. In the cooling zone the pressure is nearly the same as in the heating zone. A difference is due to the hydrostatic head of the fluid. This amounts to less than 0.5% of the total pressure considered. The temperature of the inside wall is below the saturation temperature ϑ_s in the cooling zone, and it is above ϑ_s in the heating zone. The data points are given at the left and the right of the saturation line in Fig. 2(a) respectively.

In Fig. 2(b) a similar diagram is shown for the cooling zone, only drawn on a larger scale; for data points 1–3 also the cooling water temperatures are indicated. Test points 1–11 are taken for an increasing cooling water temperature, points 12–16 for decreasing water temperatures.

The stepwise increase in cooling water temperatures ϑ_{CW} and, in consequence, in the inside wall temperatures ϑ_{CZ} cause a stepwise rise of the pressure up to the state 10 with

$$\Delta\vartheta = \vartheta_{CW,10} - \vartheta_{CW,9} = 0.16 \text{ K};$$
$$\Delta p = p_{10} - p_9 = 0.04 \text{ bar}.$$

A further increase in water temperature causes an unproportionally larger increase in pressure:

$$\Delta\vartheta = \vartheta_{CW,11} - \vartheta_{CW,10} = 0.31 \text{ K};$$
$$\Delta p = p_{11} - p_{10} = 1.89 \text{ bar}.$$

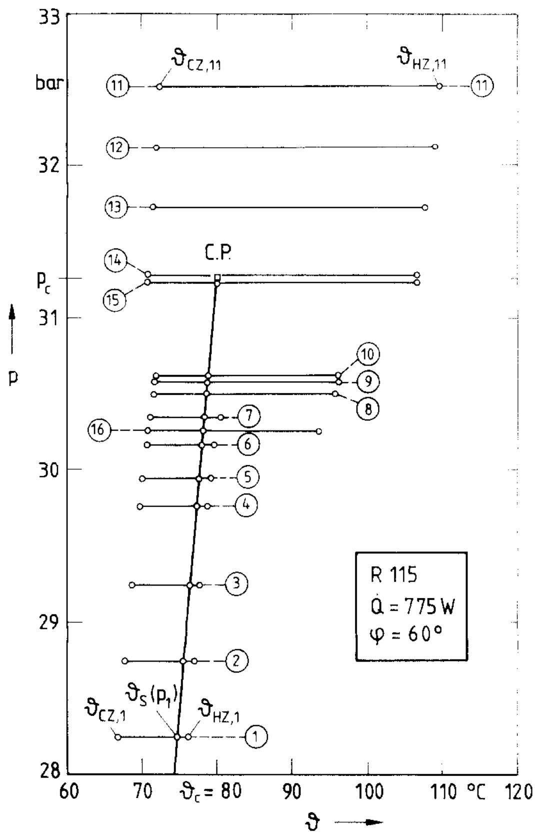

FIG 2(a). Pressure–temperature diagram for R115 and relation between fluid pressures and mean inside wall temperatures ϑ_{CZ} and ϑ_{HZ}.

Steady-state conditions are only obtained again at $p_{11} = 32.51$ bar. Thus the thermodynamic state within the tube has changed from subcritical to supercritical.

With a stepwise decrease in cooling water temperature a state 14 is reached slightly above the critical pressure. A subsequent temperature decrease brings forth:

$$\Delta\vartheta = \vartheta_{CW,14} - \vartheta_{CW,15} = 0.09 \text{ K};$$
$$\Delta p = p_{14} - p_{15} = 0.05 \text{ bar},$$

yielding a state closely below the critical. A further decrease in water temperature results in an unproportionally large pressure drop:

$$\Delta\vartheta = \vartheta_{CW,15} - \vartheta_{CW,16} = 0.04 \text{ K};$$
$$\Delta p = p_{15} - p_{16} = 0.97 \text{ bar},$$

to reach steady-state conditions. This brings the near-critical fluid well into the subcritical state.

3.2. *Effect of heat flow rate and inclination angle on the unproportional pressure variations*

Unproportional pressure variations as described in the previous section have been obtained in all test series, which could be extended into the near critical pressure region. In some cases—with small inclinations and large heat flow rates—the wall temperatures became

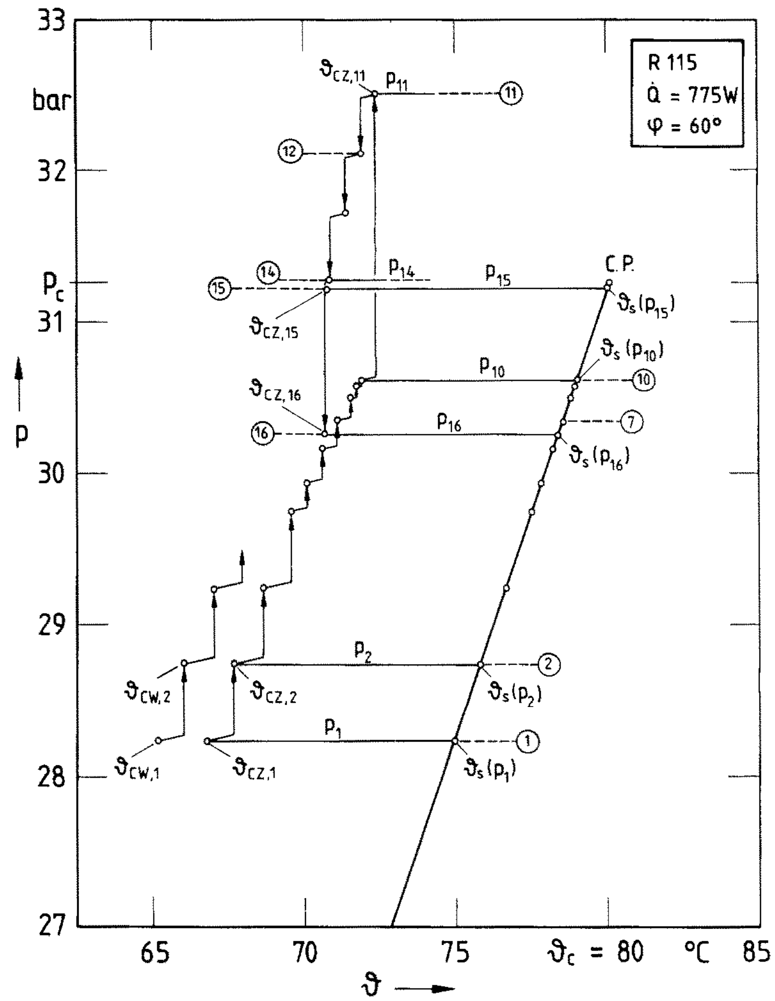

FIG. 2(b). Relation between fluid pressures and mean inside wall temperatures in the cooling zone ϑ_{CZ} for test points 1–16 of one test series ($\dot{Q}$ = 775 W, φ = 60°).

too large, and the power supply had to be switched off before near critical pressures could be obtained. Figure 3 presents those pressure values, which indicate the last steady state before (filled-in symbols) and the consecutive steady state after (hollow symbols) the unproportional pressure variation occurred. These depend on the heat flow rate and the inclination angle. The pressures are given in Fig. 3(a) for an increasing and in Fig. 3(b) for a decreasing pressure. For a better orientation the corresponding numbers of the test points from Fig. 2 are marked in Fig. 3 : unproportional variations can be seen in Fig. 3 from 10 to 11 and from 15 to 16.

The pressure values exhibit a hysteresis : in Fig. 3(a) the increasing pressures reach high subcritical values (filled-in symbols), before they rise beyond (hollow symbols) the critical pressure. In Fig. 3(b) the decreasing pressures also reach subcritical values (filled-in symbols) slightly below the critical pressure, before they decrease unproportionally (hollow symbols). There is a distinct influence of heat flow rate and inclination angle on the last pressure before the unproportional variation occurs for data in Fig. 3(a), but negligible influence for data in Fig. 3(b).

3.3. *Heat transfer in the heating and cooling zone*

The heat transfer between the tube walls and the fluid inside the thermosyphon is determined by various mechanisms. These are different in the heating and in the cooling zone and depend on the fluid pressure. Figure 4 shows the effect of the pressure on the heat transfer coefficients α_{HZ} and α_{CZ} in the heating and cooling zones, respectively, for the test series of Fig. 2 ($\dot{Q}$ = 775 W, φ = 60°). The coefficients α_{HZ} and α_{CZ} are evaluated according to the given equations, with the heat flow rate $\dot{Q}$, the heat transfer areas A_{HZ} and A_{CZ}, and the mean temperatures ϑ_{HZ} and ϑ_{CZ} of the inner surfaces in the heating zone and the cooling zone, respectively. The reference temperature ϑ_{ref} is chosen as the mean temperature in level 6. This level is in the adiabatic region midway between heating and cooling zone and far enough away from either zone to avoid superheating of the boiling liquid or subcooling of the condensate. The heat transfer inside a thermosyphon tube is

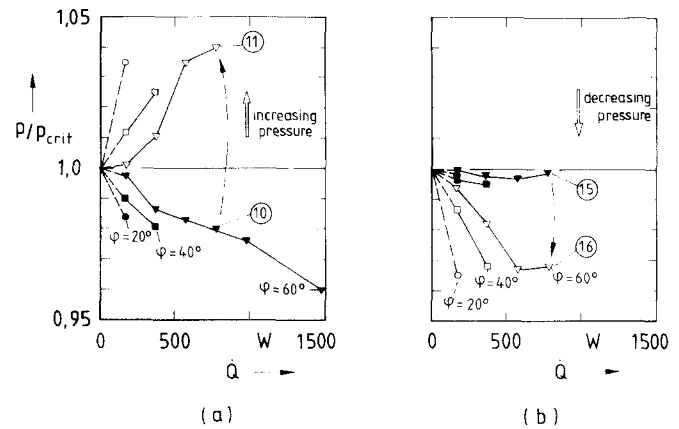

FIG. 3. Unproportional pressure variations connected with the transition between subcritical and supercritical states.

described in [7]. At subcritical pressures, up to about $p/p_c = 0.97$, nucleate boiling is found in the heating zone with an increasing coefficient α_{HZ}, when the pressure is increased. In total, the heat transfer is enhanced by the void fraction in the two phase mixture. Beyond $p/p_c \approx 0.9$ the flow rate of vapour $\dot{V}''$ increases, however, with increasing pressure (see Appendix and Fig. A2) and starts to impede the heat transfer in the upper part of the heating zone. This yields a slight decrease of α_{HZ} for $p/p_c > 0.93$. At about $p/p_c = 0.97$, DNB takes place and α_{HZ} exhibits a distinct decrease by an order of magnitude and maintains this low value until the onset of the unproportional pressure increase at about $p/p_c = 0.98$.

The heat transfer coefficient α_{CZ} for the cooling zone is principally smaller than α_{HZ}, with only a slight increase in the near critical region. This may be explained by condensation heat transfer on a laminar wavy film [7, 8].

In the supercritical region the heat transfer coefficients in both the heating and cooling zone decrease with α_{CZ} being always larger than α_{HZ}. Heat transfer coefficients, obtained at different heat flow rates and inclinations, exhibit a similar characteristic as those shown in Fig. 4 (see [7, 8]).

4. DISCUSSION

4.1. *Development of the thermodynamic states inside the thermosyphon*

The development of the thermodynamic states at a constant heat flow rate and inclination ($\dot{Q} = 775$ W, $\varphi = 60°$) will be discussed according to Fig. 2 for increasing and decreasing pressures. It is important to remember that there are different boundary conditions in the heating and cooling zone of the tube: constant heat flux in heating and constant wall temperature in cooling.

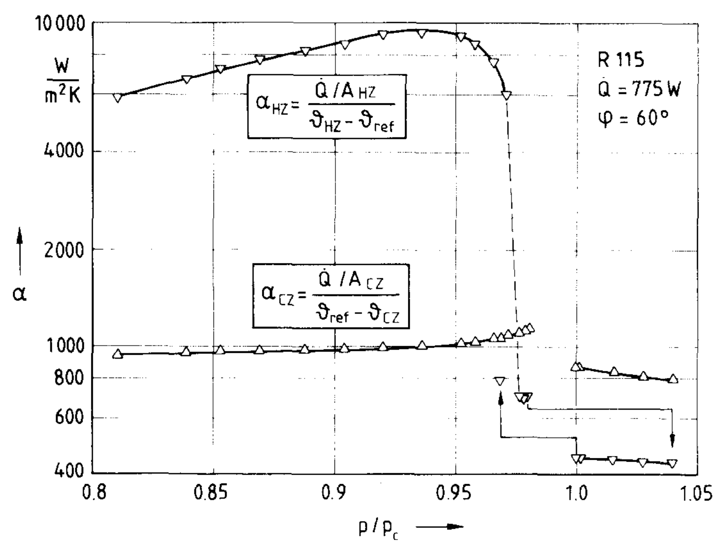

FIG. 4. Effect of the pressure on the mean heat transfer coefficients in the heating and cooling zone ($\dot{Q} = 775$ W, $\varphi = 60°$).

4.1.1. *Increasing pressure.* State 1 is marked by the pressure $p_1 = 28.24$ bar, taken at the upper end of the thermosyphon. Neglecting the hydrostatic head, all the fluid states are found on an isobaric line [Fig. 2(a)]. The fluid temperatures extend from the saturation temperature $\vartheta_s(p_1)$ (at the phase boundary between liquid and vapour) to the inner cooling wall temperature $\vartheta_{CZ,1}$ and to the inner heating wall temperature $\vartheta_{HZ,1}$. With a constant heating rate, state 1 is controlled by the cooling water temperature $\vartheta_{CW,1}$ which is somewhat smaller than $\vartheta_{CZ,1}$ [Fig. 2(b)]. When the cooling water temperatures are raised, steady conditions are obtained at higher pressures giving thermodynamic states 2, etc. until the last subcritical state 10 is reached with a pressure of $p_{10} = 30.62$ bar ($p/p_c = 0.980$). For all these changes of state, the pressure increase appears to be proportional to the increase in water temperature. For the explanation of the unproportional pressure increase, the heat transfer mechanisms occurring in the heating zone and the mass transport in the transport zone have to be considered: the distinct decline of the heat transfer coefficient in the heating zone (Fig. 4) at $p/p_c = 0.971$ ($p_7 = 30.35$ bar) indicates the occurrence of DNB. With a further increase of the cooling water temperature and thus an increase of the pressure in the tube, the volume flow rate of either vapour and liquid stream increases (see Fig. A2) as the heat of evaporation becomes smaller. Both streams are in counterflow. The buoyancy forces and the separation of the two phases decrease as the density difference $\rho' - \rho''$ becomes smaller with increasing temperature. It is assumed that increasing mutual flow disturbances finally cause the breakdown of the internal transport mechanisms (e.g. at state 10 when ϑ_{CW} is further increased). When the internal convection is hampered, vapour may accumulate in the heating zone and force the liquid column upward into the cooling zone. Here, under increasing pressure and a temperature well below that of saturation, only liquid will exist so that instead of condensation heat transfer only single-phase natural convection heat transfer can occur.

With insufficient cooling at the upper end, while the heat input remains constant at the lower end, temperature and pressure in the tube are bound to rise.

A steady-state operation mode is only obtained again, when the mass transport mechanism in the tube is re-established. This could be by a periodic flooding of the heating and cooling section respectively or by the re-formation of a circulatory flow inside the tube. Since no periodic temperature and pressure fluctuations have been observed, the second possibility is considered: when a natural convection circulatory flow becomes destabilized by increasing the cooling water temperature at subcritical conditions, it can only be stabilized again (provided the water temperature is kept constant) at supercritical conditions, when the difference between bulk and wall temperature is large enough for steady-state heat transfer by single-phase natural convection.

Thus, a slight pressure increase in the subcritical state which is proportional to an increase in wall temperature, finally yields to an unproportional pressure increase which brings the fluid inside the tube into a supercritical state in order to obtain steady-state working conditions again.

4.1.2. *Decreasing pressure.* For the conditions underlying Fig. 2 a decrease of cooling water temperature at state 11 brings forth a decrease of pressure to state 12 which is also in the supercritical region: with the heat flow rate being constant, the fluid temperature in the cooling zone can proportionally decrease with the cooling water temperature and consequently the pressure will decrease proportionally [see Fig. 2(b)].

A natural convection flow may be maintained down to the critical point or even slightly below as for state 15 ($p_{15} = 31.32$ bar, $p/p_c = 0.999$). In the heating zone the pressure may be critical or still supercritical (taking instrumental uncertainties into account). The steady-state equilibrium in such cases is rather unstable: the pressures measured are only quasi-steady in many cases and show oscillations of ± 0.05 bar within a second.

A further decrease in cooling water temperature brings forth the separation of the fluid into two phases. Liquid is generated at locations of low temperature, and it requires about one half of the entire tube volume. Thus the cooling zone becomes flooded spontaneously. The circulatory flow remains stable with high temperature vapour in the heating zone and low temperature liquid in the cooling zone. Heat is transferred from the superheated upstreaming vapour to the subcooled downstreaming liquid across the phase boundary. Maybe, the heat transfer is connected with evaporation and condensation (one large convection roll is created in this case) or there is no change of phases in which case two separate rolls are then established in the heating and cooling zone. Heat transfer coefficients are rather low in both sections (Fig. 4). Such a kind of circulatory flow is able to exist as long as the cooling zone remains flooded. For a slightly decreased wall temperature, the circulation is destabilized when the buoyancy forces—increasing with the density difference $\rho' - \rho''$—exceed the shear forces between the upstreaming vapour and the downstreaming liquid. Liquid is then removed from the cooling section and film condensation is re-established with rather large heat transfer coefficients. The difference between wall and saturation temperatures becomes much smaller than before, and thus the pressure decreases unproportionally until state 16 is reached ($p_{16} = 30.26$ bar, $p/p_c = 0.968$).

4.2. *Effect of heat flow rate and inclination on the unproportional pressure variations*

Figure 3 shows the pressures just before and after the unproportional variations occur for different heat flow rates and inclination angles.

4.2.1. *Increasing pressure.* For larger heat flow rates, the unproportional pressure increase sets in at a lower pressure ratio; for a smaller inclination angle (e.g. a steeper tube) the unproportional pressure increase is also found at lower pressure ratios. This behaviour agrees with results in [4] and fits well into the assumption made above. An increase of the heat flow rate results in an increase of the volume flow rates of both liquid and vapour. Thus counterflow disturbances may become effective already at lower pressures yielding to a collapse of the inside transport mechanisms and the consequences mentioned before.

With smaller inclination angles, the separation of phases with different densities is less pronounced and counterflow disturbances are more likely to occur. The unproportional increase in pressure is larger when the heat pipe is steeper and it is also larger with larger heat flow rates. This would indicate that a flow which becomes easily unstable, i.e. at a low subcritical pressure ratio, is also difficult to be stabilized again, i.e. at a high supercritical pressure ratio. In some cases, e.g. high heat flow rate in vertical tubes, the re-stabilization could not be reached within the allowed temperature region in the heating zone.

4.2.2. *Decreasing pressure.* The effect of heat flow rate and inclination angle on the lowest steady pressure before the unproportional drop occurs is small and somewhat unsystematic.

All these lowest steady pressure values have in common that they are very close or even a little below the critical pressure [see Fig. 3(b)].

5. CONCLUDING REMARKS

In ref. [4] it is reported that unproportional pressure variations can be suppressed by inserting a thin metal sheet axially into the transport zone. Such a sheet divides the inner cross area of the inclined tube into an upper and lower half and helps to separate the counterflow phases. It has been shown that such a sheet acts stabilizing even at very small inclinations ($\varphi \approx 0°$).

Another possibility of avoiding the unproportional pressure variations is to take a smaller amount of fluid to be filled into the thermosyphon. If the amount of fluid is less then the critical for the given tube volume, the ration $\bar{\rho}/\rho_c$ which equals 1 in the critical case will become $\bar{\rho}/\rho_c < 1$. From Fig. A3 (Appendix) it can be observed that the volume fraction of liquid decreases with increasing pressure at higher p/p_c. If a proper amount of fluid is chosen, the rise of the liquid column is small enough to remain below the cooling zone. As shown in [6] it is possible to avoid any unproportional pressure variation, if the relative density is taken as $\bar{\rho}/\rho_c = 0.933$ at a heat flow rate of $\dot{Q} = 600$ W in the same thermosyphon tube as described here. However, if the amount of filling is too small, dryout problems will arise.

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APPENDIX

The effect of variable properties on fluid flow rate and on liquid fraction inside the tube

For the understanding of the effect of fluid pressure on fluid flow and heat transfer it is useful to evaluate the flow rates of liquid and vapour circulating inside the tube. For this purpose, Fig. A1 gives the effect of pressure on some selected properties (values are taken from [32]):

—Specific volumes of a saturated liquid (v') and vapour (v''): when p/p_c increases v' shows a steady increase and v'' a steady decrease. They become equal for $p/p_c = 1.0$.

—Difference of the densities of saturated liquid and vapour ($\rho' - \rho''$): when p/p_c increases near the critical point, ($\rho' - \rho''$) decreases exponentially to a value of zero for $p/p_c = 1.0$.

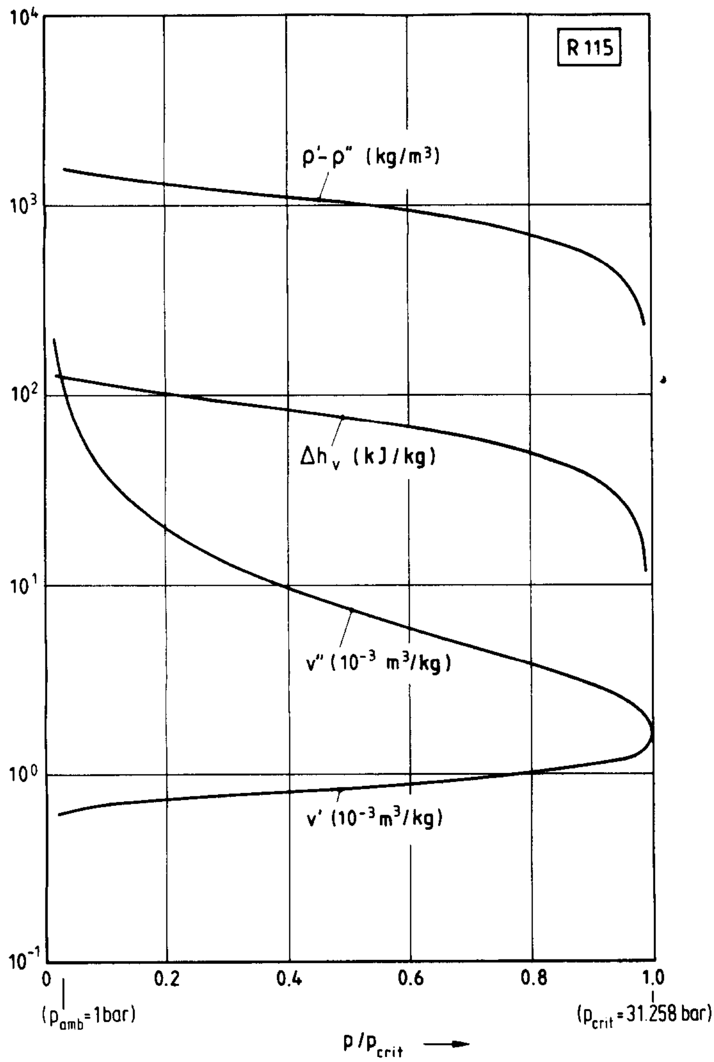

FIG. A1. Effect of the pressure on various properties of refrigerant R115.

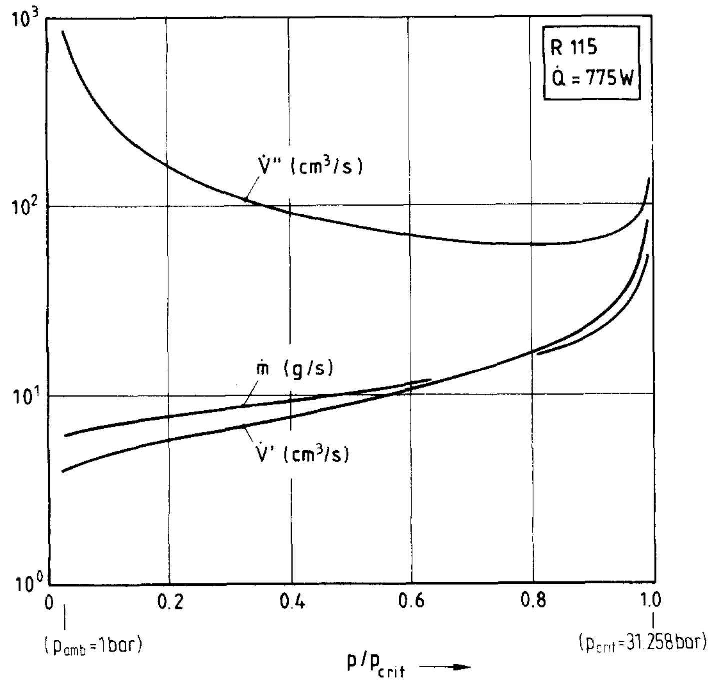

FIG. A2. Effect of the pressure on the flow rates of mass and volume at a constant heat flow rate ($\dot{Q}$ = 775 W).

—Enthalpy of evaporation (Δh_v): Δh_v exhibits a similar behaviour as $(\rho' - \rho'')$ and becomes also zero for $p/p_c = 1.0$.

Using these properties the mass flow rate $\dot{m}$ and the volume flow rates of liquid $\dot{V}'$ and vapour $\dot{V}''$ can be calculated for any constant heat flow rate, e.g. $\dot{Q}$ = 775 W (test series shown in Fig. 2):

$$\dot{m} = \frac{\dot{Q}}{\Delta h_v} \quad \dot{V}' = \dot{m} \cdot v' \quad \dot{V}'' = \dot{m} \cdot v''.$$

The effect of fluid pressure on these flow rates is presented in Fig. A2. The mass flow rate, necessary to transport $\dot{Q}$, increases monotonously when the pressure is raised. It becomes infinite for $p/p_c = 1.0$ if subcooling and superheating of the fluid are neglected. The same behaviour is shown by the volume flow rate $\dot{V}'$ of the liquid running down the wall after condensation. The volume flow rate $\dot{V}''$ of the vapour shows a partly different behaviour. When the pressure increases $\dot{V}''$ decreases until $p/p_c = 0.8$ is exceeded; then it also increases up

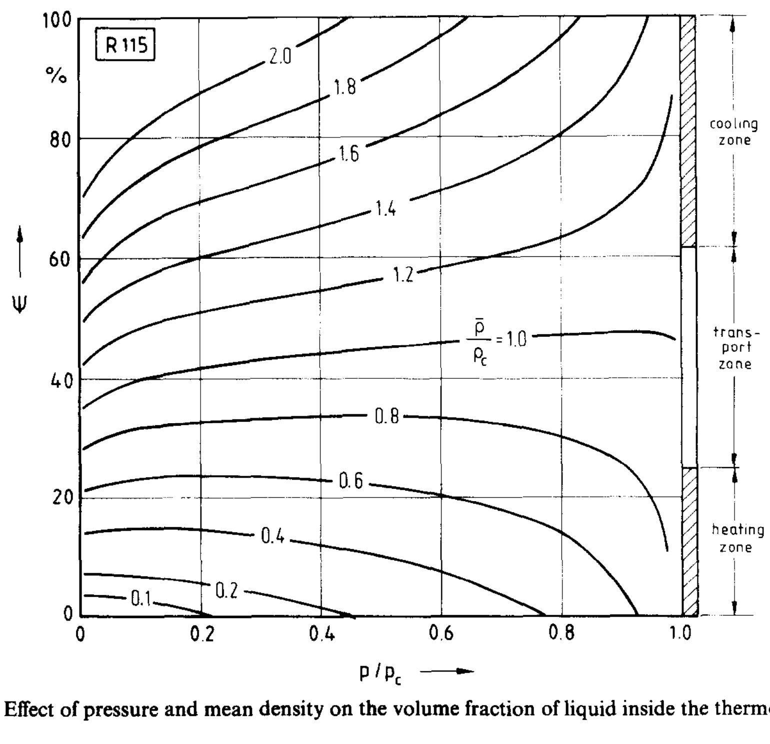

FIG. A3. Effect of pressure and mean density on the volume fraction of liquid inside the thermosyphon.

to infinity. For $p/p_c \to 1.0$, the volume flow rates of liquid and vapour become very large approaching $\dot{V}'/\dot{V}'' = 1.0$. Thus the counterflow of liquid and vapour inside the tube becomes more and more intense when the pressure is raised above $p/p_c = 0.8$.

The volume fraction of liquid ψ depends on the amount of fluid filled into the tube and on the pressure. The following relation holds for $\rho' \geqslant \bar{\rho} \geqslant \rho''$ (two-phase region):

$$\psi = \frac{\bar{\rho}-\rho''}{\rho'-\rho''}$$

where $\bar{\rho}$ is the mean density ($\bar{\rho} = m/V$). The fraction ψ is plotted in Fig. 7 vs the relative pressure p/p_c depending on the relative density $\bar{\rho}/\rho_c$. On the right axis, the lengths of heating, transport, and cooling zones are given. The theoretical location of the meniscus between liquid and vapour (indicated by the $\bar{\rho}/\rho_c$ curves) can be taken from Fig. A3 for equilibrium conditions. When power is supplied, the circulation of fluid disturbs this location: Some of the liquid runs down the walls as a condensate film, some of the vapour rises up in the heating zone forming a two-phase mixture. These two effects tend to compensate each other, but nevertheless there will be a difference between theoretical and actual filling heights.

For a smaller than critical amount of fluid being added ($\bar{\rho}/\rho_c < 1.0$), the liquid fraction decreases for an increasing pressure, approaching zero for high subcritical pressures. This 'dryout' in the heating zone is connected with fairly high temperatures. For a larger than the critical amount of fluid ($\bar{\rho}/\rho_c > 1.0$), the liquid fraction increases until the entire tube is filled with liquid even at subcritical pressures. A small temperature increase will bring out a large, maybe dangerous, pressure rise.

INSTABILITE ET VARIATIONS DE PRESSION NON PROPORTIONNELLES

Résumé—On observe des variations de pression non proportionnelles dans un thermosiphon pendant la transition entre les états sous-critique et supercritique. Une telle variation de pression en échelon est rencontrée aux faibles pressions et avec un large saut quand le flux thermique est élevé et le tube profond. Ce comportement instable est expliqué par une instabilité qui apparait dans le contrecourant à deux phases de vapeur et de liquide : aux pressions sous-critiques élevées, les densités de liquide et de vapeur deviennent égales et elles réduisent l'effet de pesanteur et la séparation des courants. Ceci conduit à la disparition de l'écoulement de circulation interne. Seulement lorsque l'écoulement est rétabli dans la région supercritique monophasique—avec un accroissement de pression non proportionnel—les conditions opératoires permanentes sont à nouveau obtenues.

INSTABILITÄT UND UNPROPORTIONALE DRUCKÄNDERUNG NAHE DEM THERMODYNAMISCH KRITISCHEN PUNKT IN EINEM GESCHLOSSENEN THERMOSYPHON

Zusammenfassung—In einem geschlossenen Thermosyphon werden beim Übergang zwischen unter- und überkritischen Zuständen unproportional große Druck- und Temperaturänderungen beobachtet. Diese sprunghaften Änderungen setzen bei umso kleineren Drücken ein und sind umso stärker ausgeprägt, je größer der Wärmestrom und je kleiner die Neigung des Thermosyphons gegenüber der Senkrechten ist. Die Unstetigkeit ist damit zu erklären, daß die im Rohr auftretende zweiphasige gegenläufige Strömung von Dampf und Flüssigkeit bei sehr hohen Drücken instabil wird, da mit Annäherung an den kritischen Punkt durch die Angleichung der Dichten Auftrieb und Trennwirkung stark abnehmen. Erst nach Wiederherstellung einer stabilen Zirkulationsströmung im überkritischen Einphasengebiet—mit vorausgegangenem unproportionalem Druckanstieg—sind stationäre Betriebszustände zu erreichen.

НЕУСТОЙЧИВОСТЬ И СКАЧКИ ДАВЛЕНИЯ В ЗАМКНУТОМ ТЕРМОСИФОНЕ В ОКРЕСТНОСТИ КРИТИЧЕСКОЙ ТОЧКИ

Аннотация—Скачкообразные изменения давления наблюдаются в замкнутом термосифоне при переходе от докритического к сверхкритическому состоянию и наоборот. Такие скачки наблюдаются при низких давлениях, причем они больше, когда изменение теплового потока и перепад диаметра трубы велики. Нестационарное поведение объясняется неустойчивостью при двухфазном противотоке пара и жидкости: при больших докритических давлениях плотности пара и жидкости сравниваются и подъемная сила и разделение потоков уменьшаются. Это приводит к разрушению внутреннего циркуляционного течения. Только в случае восстановления этого потока в однофазной докритической зоне (с предшествующим скачком давления) вновь достигаются стационарные рабочие условия.