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Hysteresis effect in a double channel natural circulation loop

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Abstract

This study investigates experimentally the effect of power changing procedure, i.e., heating or cooling, on thermal hydraulic behavior of a double channel natural circulation loop. The data obtained at an inlet temperature of 60°C and loop pressure of 1.5 bar ($N_{sub} = 107$) demonstrate clearly the existence of hysteresis effect at the onset of two-phase flows. During the heating procedure single-phase flow prevails in both channels while the heating power is increased gradually from 1.0 kW ($N_{pch} = 1.25$) to 4.5 kW ($N_{pch} = 5.63$). The loop flow rate, i.e., the sum of flow rate through each channel, increases with increase in power with an exponent of 0.336, which is consistent with theoretical prediction. The onset of significant two-phase flow takes place when the power for both channels reaches 5.0 kW ($N_{pch} = 6.26$), which agrees well with Saha and Zuber's model prediction on the net vapor generation. Moreover, the flow is found to be unstable. Nearly out-of-phase, large amplitude oscillations with reversed flow display in both channels. The two-phase flow and nearly out-of-phase oscillations persist while the power is gradually decreased from 5.0 kW ($N_{pch} = 6.26$) to 2.0 kW ($N_{pch} = 2.50$). The magnitude of oscillation is relatively independent of power, while the period is increased significantly as power is reduced. Single-phase flow is restored when the power is reduced to 1.5 kW ($N_{pch} = 1.88$) and the corresponding flow rate is close to that during the heating process. The hysteresis effect suggests that the loop state be a path function. This is of significant interest for the operation of a natural circulation loop. © 2000 Elsevier Science Ltd. All rights reserved.

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1. Introduction

Two-phase natural circulation loops have many industrial applications, including next generation light water nuclear reactors (Aritomi et al., 1992), because of its simplicity, high heat transfer capability and passive nature. The fluid flow in a natural circulation loop is driven by the density gradient in the loop. The loop flow rate results from the delicate balance between gravity head available due to downcomer and various pressure losses in the loop. Consequently, the flow rate in a natural circulation loop depends on the heating power. Under the conditions of low heating power, single-phase flow prevails in the loop and, as is well-known, the mass flow rate is proportional to the one-third power of heating power. Once two-phase flow is initiated in the loop after the heating power reaches some value, the flow rate strongly depends on the void fraction in the loop and unstable situation could be arisen (Fukuba and Kobori, 1979; Lin and Pan, 1994). Recently, it has been shown theoretically (Jeng and Pan, 1994; Knaani and Zvirin, 1993) that multiple flow rates may be present at a given heating power, especially under low pressures and high inlet subcooling conditions (Jeng and Pan, 1994). This suggests that a hysteresis effect may take place in a natural circulation loop. That is to say, the loop mass flow rate at a given heating power may be different during heating and cooling procedures.

Hysteresis effects are not uncommon in thermal fluid systems. For examples, Bankoff and Lee (1987) presented experimental data on flooding transitions in nearly-horizontal countercurrent stratified steam–water flow and the mechanism of hysteresis effects owing to condensation was examined; Celata et al. (1991) also studied the hysteresis phenomena on flooding; recently, Shoukri et al. (1994) reported the existence of significant hysteresis effects in countercurrent gas–liquid flow limitations in vertical tube; Yamane and Orita (1994) revealed hysteresis for flows in a collapsible tube.

It is well established that boiling heat transfer is subject to two kinds of hysteresis: at the onset of nucleate boiling and between minimum film boiling heat flux (MFB) and critical heat flux (CHF). For the later case, during the heating procedure nucleate boiling prevails until CHF is exceeded, at which departure form nucleate boiling takes place and film boiling occurs. If the heating power is subsequently reduced, film boiling will persist until the heat flux is reduced to MFB, at which nucleate boiling is restored. As for the hysteresis at the onset of nucleate boiling, boiling is initiated at a relatively high wall superheat during a heating procedure. Once boiling begins, the surface temperature drops sharply and the heat flux versus wall superheat relation follows the nucleate boiling curve if the heat flux keeps increasing. If the heating power is decreased subsequently, boiling persists even though the heat flux is reduced below the incipient value. This kind of hysteresis at the onset of nucleate boiling is particularly significant for well-wetting fluid. The small contact angle of well-wetting fluids leads to the surface cavities to be flooded, except for the very small ones. Consequently, a very high wall superheat is needed to initiate boiling. Once boiling begins, the sliding of bubbles on the surface activates other cavities and intensive boiling causes the wall superheat to drop sharply (Hino and Ueda, 1985). However, once boiling prevails on the surface, reducing heat flux (or wall superheat) does not cease boiling immediately. Many cavities will remain active and boiling persists, even though the heat flux is reduced below the incipient one. A hysteresis effect thus displays at the onset of nucleate boiling. Bar-Cohen (1992) and Bräuer and

Mayinger (1992) provided extensive reviews on such kind of hysteresis effect. Since the two-phase flow in the present natural circulation loop is produced by boiling in the heated region, the hysteresis phenomena at the onset of nucleate boiling is believed to have some effect on the thermal hydraulic behavior of a natural circulation loop. The objective of the present study is to explore experimentally the possible hysteresis effect on a low-pressure double channel natural circulation loop under conditions of low heating powers at a high inlet subcooling.

2. Experimental apparatus and procedure

The experimental apparatus employed in this study is a modified loop, previously used in our laboratory (Hsieh et al., 1997). The new setup allows heating the two parallel channels with equal or unequal power using a single DC power supply. Moreover, the flow rate in each channel could be measured independently.

2.1. Natural circulation loop

The loop employed in this experiment is illustrated in Fig. 1 with dimensions. The loop essentially consists of two parallel channels, each having a heated section and a long riser, a condenser, a downcomer, and upper and lower horizontal sections. One of the two parallel channels is a stainless steel tube and the other is a transparent Pyrex one allowing flow visualization. The heated section in both channels is an annular design with the electric heater in the center. The outer diameters of the annular for heated channels are 20.2 mm. The outer diameters of the electric heated rods in the stainless steel and Pyrex tubes are 10.6 mm and 11.2 mm, respectively. The heated length of the heated rod is 1.1 m, but only the top 0.92 m is actually heated. The inner diameter of the riser in both channels is 20.2 mm.

To enable independent flow measurement, both channels have their own lower horizontal section, in which an orifice flowmeter is installed. The flow-meters were calibrated for both forward and reversed flows. The uncertainties in measurements in the low flow rate region for forward and reversed flows are ± 8.4 and ± 4.9 kg/h, respectively, in the glass channel and ± 4.5 and ± 7.3 kg/h, respectively, in the stainless steel channel. Both sections are made of stainless steel with an inlet diameter of 20.2 mm. Due to space limitation, the lower horizontal section for the steel channel is at a plane perpendicular to the loop. The two horizontal sections are connected with a common plenum at the bottom of the downcomer. The plenum is also connected with a pressurizer, which helps to adjust the loop pressure to a desired level. An upper plenum is also provided at the top end of both channels to ensure both channels having common ends. No modification was made for the upper horizontal sections, condenser, downcomer, and the secondary cooling sections. These sections have been described in a previous paper (Hsieh et al., 1997).

The entire loop, except for most part of the glass channel, is insulated with glass fibers to minimize heat loss. To heat both channels with equal or unequal power using a single power supply, an extra heater with a variable resistance and a circuit is designed in parallel with the heater in the stainless steel channel to control the power dissipated in each channel. This additional heater is placed in a constant level water tank to dissipate heat, if any. The current

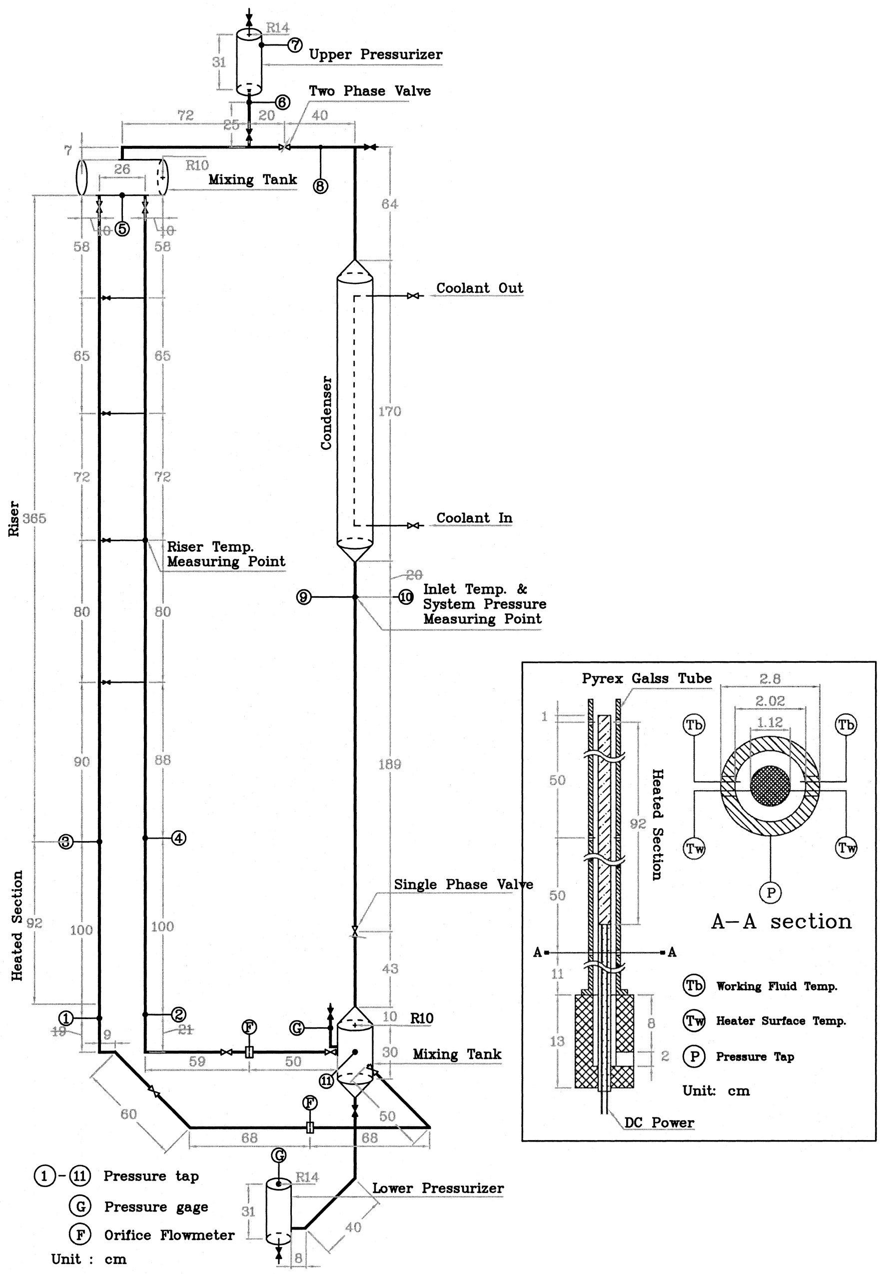

Fig. 1. The double channel natural circulation loop.

in the circuit and voltage drop through each heater in the two channels are measured to determine the heating power in each channel. The uncertainty for power measurement is 1.2% based on the uncertainties in the current and voltage measurements estimated from meter scales.

The uncertainties in the measurements of pressure/differential pressures are $\pm 0.2\%$ F.S. based on the calibration given by the supplier. The uncertainties in temperature measurements is $\pm 0.87°C$.

2.2. Data acquisition system

The outputs of thermocouples, flowmeters and differential pressure transducer are recorded and analyzed by two data acquisition systems: HP3852 and AR1100A. Both are connected with a personal computer. The former provides 96 channels for recording the data with a sampling rate of 0.5 Hz, while the later has four channels only and can have a much higher sampling rate. The AR1100A was used during parts of transient and quasi-steady state intervals which are of significant interest for detail analysis for the measurement of the mass flow rates in each of the two channels, riser fluid temperature in the glass tube and voltage outputs of the LEDs. The signals of LEDs are used to synchronize the thermal hydraulic data with the flow pattern image data from two cameras in the risers (Hsieh et al., 1997).

2.3. Experimental procedure

The purpose of this study is to explore the effect of power changing procedure, i.e., heating or cooling, on loop thermal hydraulic behavior. Thus, the temperature of working fluid, i.e., distilled water, at the exit on the primary side of the condenser, is maintained at $60 \pm 1°C$ by adjusting the flow rate and inlet temperature of the secondary cooling water. This temperature is not much different from that at the inlet of both channels during steady flow and is called the inlet fluid temperature. Under oscillating conditions, the fluid temperature at the inlet of both heated sections is also oscillating due to possible reversed flow. Therefore, the inlet temperature at the inlet of heated section cannot be treated as a controllable variable. The nitrogen gas pressurizer at the upper horizontal section regulates the system pressure at the exit of condenser (see Fig. 1) such that the gauge pressure there is at 3 ± 0.5 m H_2O. Under such conditions, the average pressure in the heated section is approximately 1.5 atm. Both channels were heated with approximately the same power in the present study. The steel channel is heated with a power slightly (about 2%) higher than the glass channel.

The heating power is gradually increased to a power somewhat higher than 1.0 kW to heat up the working fluid such that the inlet temperature can reach the desired level, i.e., 60°C. Subsequently, the power was reduced to and maintained at 1.0 kW. During the heating experiment, care was exercised to maneouver the operation such that steady state is obtained for each power level with the inlet temperature and loop pressure at the desired levels. At that stage, the lower pressurizer is disconnected with the loop. After the steady state is maintained for about 1 h, AR1100A was turned on to record the data for 52 min. Thereafter, the power is increased to the next level. Again, care was exercised to approach and maintain the steady (or quasi-steady if flow oscillations appear) state. The top gas relief valve may need to open from time to time during the adjusting period to keep the loop pressure at the designated value. This similar procedure was repeated until the power level reaches 5.0 kW. This series of experiments took about 48 h. Thereafter, the cooling experiments were conducted at different intervals.

As for cooling experiments, the heating power was first gradually increased to and kept at 5.0 kW at the desired inlet fluid temperature and loop pressure. Again, data were recorded for 52 min using AR1100A when the quasi-steady state has been reached for about 1 h. Thereafter, the power was decreased to the next designated level. Care was also exercised to reach the steady or quasi-steady state at the desired conditions for data recording. This similar procedure was repeated until the power level at each channel was reduced to 1.0 kW. To examine the repeatability of the cooling experiment, another series of cooling experiments were conducted about 4 weeks after the first one.

3. Results

3.1. Hysteresis effect and nondimensionalization

The effect of power changing procedure, i.e., heating or cooling, on mean flow rate in each channel and total loop flow rate is illustrated in Fig. 2. To generalize the results of this study, Fig. 2, and later Fig. 3, are presented in nondimensional forms. The representative scales of mass flow rate and heating power have been derived by Jeng and Pan (1994) based on the peak flow rate in the well-known relationship between mass flow rate and heating power in a two-phase natural circulation loop with saturated liquid at the inlet. For the peak flow rate, both gravitational and frictional pressure drops are important. According to this principle, Jeng and Pan (1994) were able to obtain the following expressions for the reference loop mass flow rate and power, respectively, as (Wu et al., 1996)

$$W_s = \rho_f A_H u_s \tag{1}$$

$$Q_s = W_s i_{fg} \frac{v_f}{v_{fg}} \tag{2}$$

where A_H is the cross-sectional area of the heated section (in the steel channel), ρ_f and v_f are the density and specific volume, respectively, of saturated liquid, v_{fg} is the difference in specific volume of saturated vapor and liquid, i_{fg} is the latent heat of evaporation, and the characteristic velocity u_s is given as

$$u_s = \left(\frac{gD_H}{f_{2\phi}}\right)^{1/2} \tag{3}$$

where D_H is the hydraulic diameter of the heated section (in the steel channel) and g is the gravitational acceleration. Assuming that the two-phase frictional factor, $f_{2\phi}$, (not multiplier) is twice the single-phase factor, $f_{1\phi}$, and using a single-phase frictional factor from a closed-square natural circulation loop (Su et al., 1991), one obtains

$$f_{2\phi} = 2f_{1\phi} = \frac{0.426}{Re^{0.241}} \tag{4}$$

Substituting Eq. (4) into Eq. (3) results in the following equation for the characteristic velocity (Lin and Pan, 1994)

$$u_s = \frac{1.62 g^{0.569} D_H^{0.705}}{\nu_f^{0.127}} \tag{5}$$

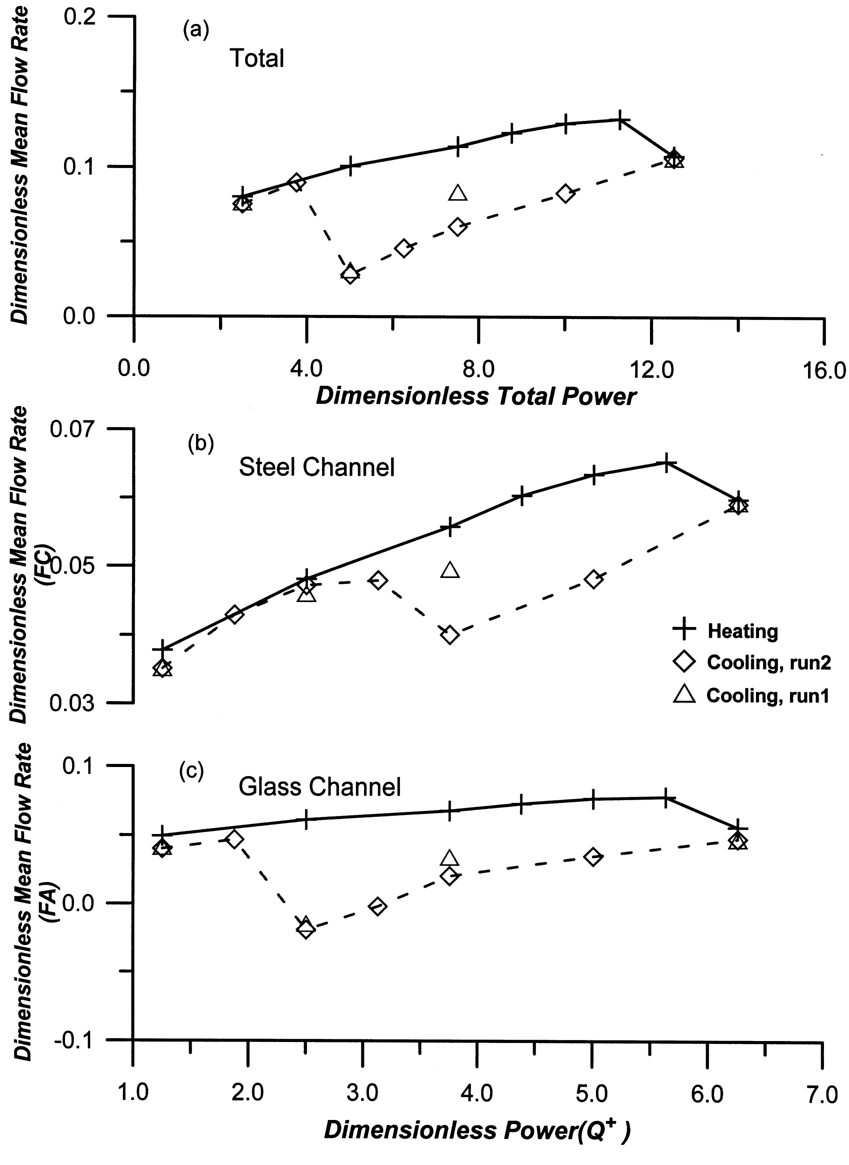

Fig. 2. Effect of loading procedures on: (a) loop flow rate, (b) flow rate in the steel channel (FC), (c) flow rate in the glass channel (FA).

where ν_f is the kinematic viscosity. The nondimensional mass flow rate and heating power can thus be expressed as

$$W^{+} = \frac{W}{W_s} \tag{6}$$

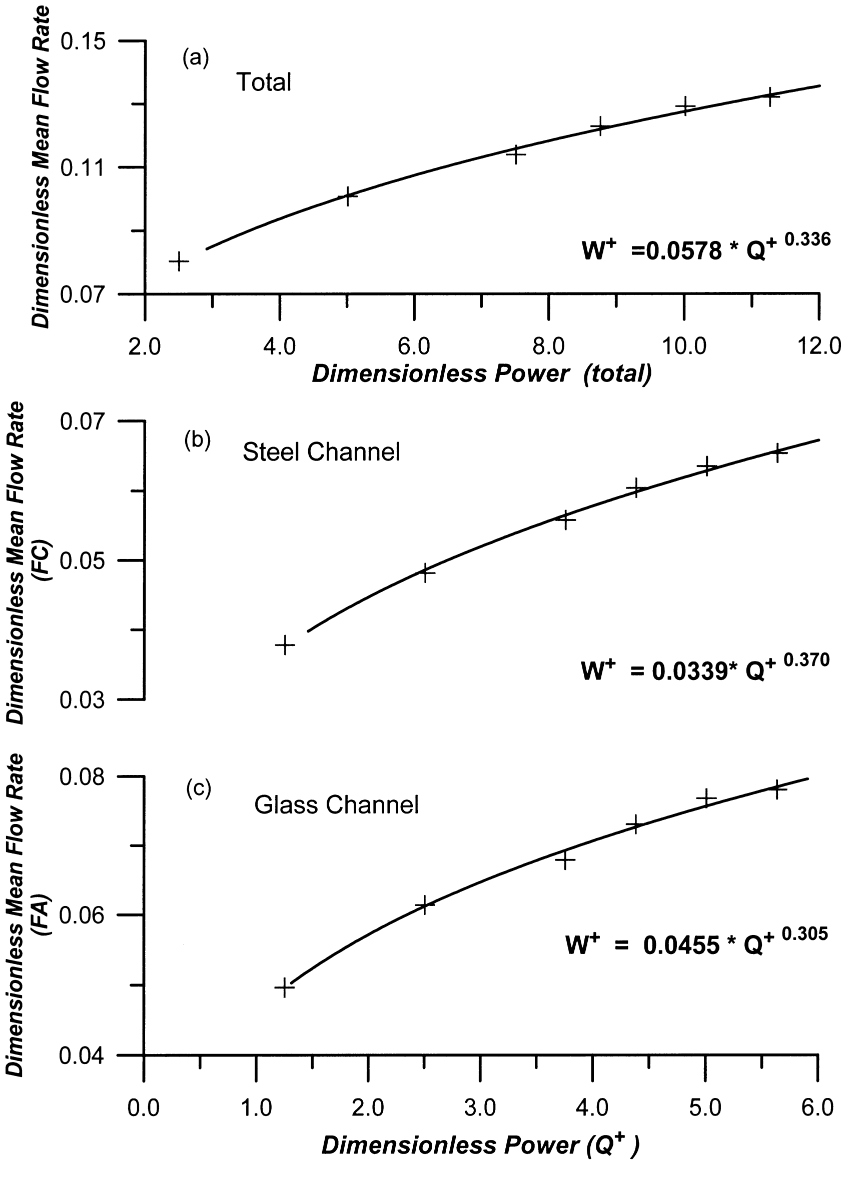

Fig. 3. Best fit of single-phase flow rate vs. power during the heating procedure: (a) loop flow rate, (b) flow rate in the steel channel (FC), (c) flow rate in the glass channel (FA).

$$Q^{+} = \frac{Q}{Q_s} = \frac{Q v_{fg}}{W_s i_{ig} v_f} = N_{pch} \tag{7}$$

The nondimensional power is actually the well-known phase change number, which plays a very important role in determining the stability of a two-phase flow system.

The inlet subcooling is usually normalized as the nondimensional subcooling number in the literature of two-phase flow stability. It is also adopted in this present study.

$$N_{sub} = \frac{i_f - i_i}{i_{fg}} \frac{v_{fg}}{v_f} \tag{8}$$

where i_i is the inlet subcooling and i_f is the saturated liquid enthalpy. For the present experimental condition, i.e., $P = 1.5$ atm, inlet temperature of 60°C, the corresponding subcooling number is $N_{sub} = 107$. Using the saturated properties at the system pressure of 1.5 atm, the numerical values of u_s, W_s and Q_s are 1.7 m s^{-1}, 1420 kg h^{-1} and 799 W, respectively. The characteristic mass flow rate is based on the cross-sectional area of the steel channel. As will be seen later, this characteristic mass flow rate is about twice the peak flow rate. This indicates that the characteristic velocity chosen in the present study is of the right order of magnitude and is representative. Considering the highly subcooled state at the inlet for the present study, the peak flow rate lower than the characteristic value is reasonable. If the flow area is based on the hydraulic diameter, the characteristic mass flow rate will be 443 kg h^{-1} (Wu et al., 1996).

The data shown in Fig. 2 clearly demonstrate the existence of hysteresis effect. For heating powers larger than 2.0 kW ($Q^{+} = 2.50$) and less than 5.0 kW ($Q^{+} = 6.26$), both the average flow rates in each channel and the loop flow rate during the heating process are significantly higher than that during the cooling procedure. Moreover, good repeatability is illustrated for various heating power, except for 3.0 kW ($Q^{+} = 3.75$), during the cooling process. Good repeatability is also illustrated for the three independent runs at 5.0 kW ($Q^{+} = 6.26$). Detailed evaluation of experimental data reveals that the poor repeatability under a power input of 3.0 kW ($Q^{+} = 3.75$) may be due to a slight difference in loop pressure between two runs. There is about 0.5 m H_2O shift in the range of pressure oscillation. In addition, a slight deviation from the quasi-steady state in pressure oscillation is present. It should also be noted that the relatively low mean flow rates are due to cancellation of forward and reversed flows. The contribution of low instantaneous flows, being with a relatively large measurement uncertainty as noted previously, has an insignificant effect on the mean flow rate due to their very short time duration, as shown in Fig. 4.

For the heating procedure, the mass flow rate increases with increase in heating power from 1.0 kW ($Q^{+} = 1.25$) to 4.5 kW ($Q^{+} = 5.63$). However, it is decreased while the heating power is increased from 4.5 kW ($Q^{+} = 5.63$) to 5.0 kW ($Q^{+} = 6.26$). From 1.0 kW ($Q^{+} = 1.25$) to 4.5 kW ($Q^{+} = 5.63$) single phase flow prevails in the loop. The regression of data (Fig. 3) indicates that the flow rates increases with increase in power with an exponent of 0.305 and 0.370, respectively, for flow rate in each channel and 0.336 for the loop flow rate, which is the sum of flow rate through each channel, as shown in Fig. 3. This is consistent with the theoretical result of 0.33 in the literature (Zvirin, 1981) and demonstrates the reliability of the present data.

Significant two-phase flow is initiated while the heating power is increased from 4.5 kW ($Q^+ = 5.63$) to 5.0 kW ($Q^+ = 6.26$). Using the best-fitted power-flow relation in the glass channel, the channel flow rate at 5.0 kW ($Q^+ = 6.26$) would be 98.9 and 97.0 kg/h in the glass and steel channel, respectively, if single-phase flow is assumed. Applying the channel flow rates and

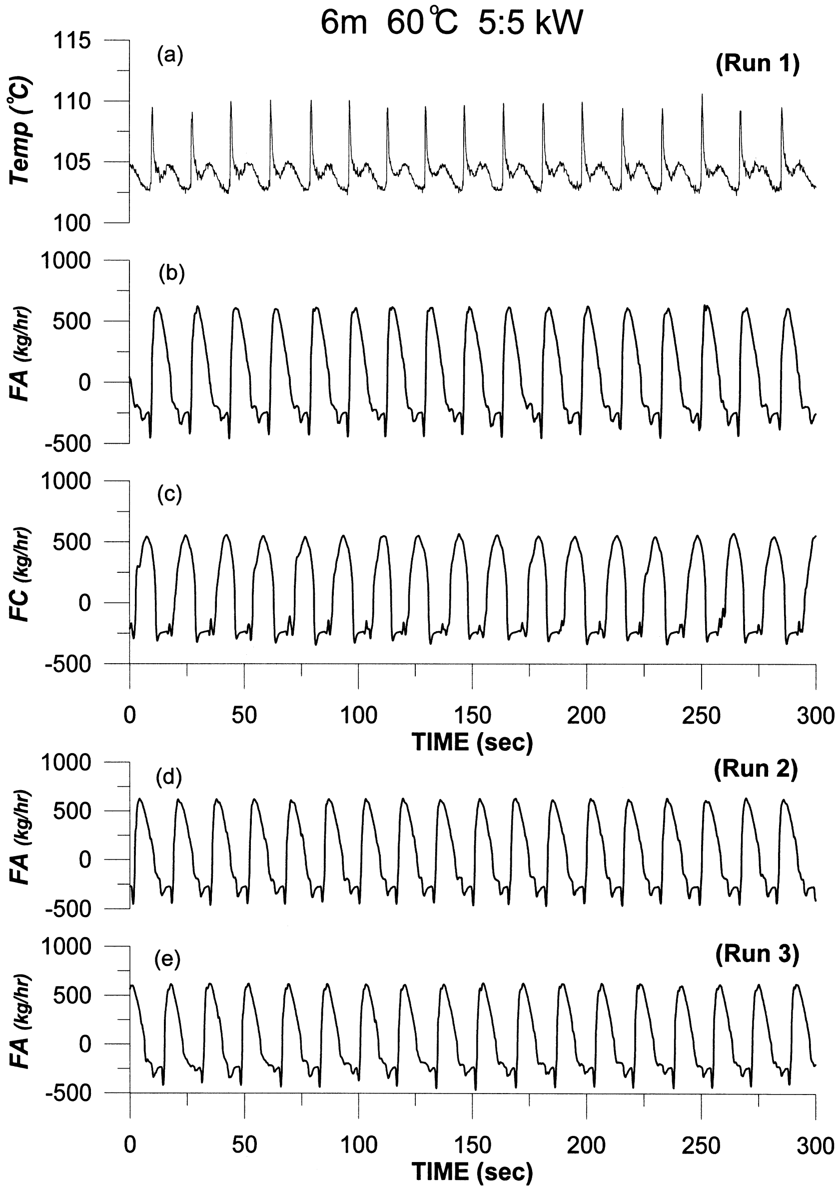

Fig. 4. Time evolution of thermal hydraulic properties at 5.0 kW: (a) riser fluid temperature in the glass channel, (b) flow rate in the glass channel (FA), (c) flow rate in the steel channel (FC), (d) flow rate in the glass channel for run 2, (e) flow rate in the glass channel for run 3.

Saha and Zuber's model (Saha and Zuber, 1974 see Appendix A), the bubble departure points are predicted at 1.08 L_H and 1.04 L_H for the glass and steel channel, respectively, under the conditions of inlet temperature of 60°C and average pressure of 1.50 bar. Here L_H is the heated length. The bubble departure point is the location of net vapor generation. If it is coincident with the top end of the heated region, the heated power can be treated as the incipient power of significant two-phase flow in the loop. Considering the uncertainty of 25% of the Saha–Zuber's model, significant two-phase flow initiation at 5.0 kW ($Q^+ = 6.26$) under the conditions of the present study agrees quite well with the model prediction, i.e., 5.4 kW based on the conditions in the steel channel.

3.2. Parallel channel oscillation

As shown in Fig. 4, the flow is no longer steady but oscillates with a large amplitude. The period of the oscillation is 16.4 s. Moreover, the oscillation in the steel channel is nearly out-of-phase with the glass channel. The flow rates in the glass channel from two other runs are also displayed to illustrate the repeatability of the experiments. The peak forward flow rate is as high as 600 kg/h, much higher than the mean flow rate, 85 kg/h in steel tube and 68 kg/h in glass channel. The mean flow rate in the glass channel is smaller than the steel channel because its hydraulic diameter in the heated section is smaller. It is heated with a somewhat lower power and has no adequate insulation. Therefore, the glass channel allows a larger heat loss and its effective heating power is smaller. The data also shows that the peak forward flow drops sharply and reversed flow follows. Direct flow visualization at the glass channel indicates that the peak forward flow is associated with the formation of slug and/or churn flow in the riser. The very existence of such flow patterns suggests that the void fraction in the channel is high and the buoyancy force is significant. Thus, a high flow rate is resulted. Such a high flow rate, however, suppresses boiling in the heated section and the flow rate is thus sharply decreased (Wu et al., 1996). On the other hand, the steel channel is now at the stage of high flow rate and presumably slug and churn flows prevail. Such a large flow in steel channel will push some of the hot fluid into the glass channel and enhance the reversed flow there. The entering of warm fluid leads to boiling in the top portion of the heated region. Boiling quickly spreads over the lower heated region. The wall bubbles were found to coalescence very quickly to form a vapor blanket in the annulus. Such a vapor blanket expands immediately and expels the liquid both upward and downward. As a result, a second peak in reversed flow is recorded in the glass channel. In fact, such a reversed flow spike is so high that it exceeds the calibration range of the flowmeter for the reversed flow. Since the time duration for the spike is short, it is believed that the cut-off of the spike has a negligible effect on the determination of mean flow rate. This expanded large vapor blanket provides a very large buoyancy force and makes the fluid flow forward again and rises sharply to its positive peak. At this stage, the flow patterns in the riser is typically slug or churn flow. Therefore, the thermocouple in the lower riser will sense the vapor temperature and a temperature spike is recorded as shown in Fig. 4a.

The mean flow rate at 5.0 kW ($Q^+ = 6.26$) being lower than that at 4.5 kW ($Q^+ = 5.63$) may be explained as follows. Although the forward peak flow rate at 5.0 kW ($Q^+ = 6.26$) is much higher than the steady single-phase flow rate at 4.5 kW ($Q^+ = 5.63$), revealing the large natural circulation force due to void, the presence of reverse flow with a relatively long duration time levels off the mean flow. In fact, the mean flow rate in each channel at 5.0 kW ($Q^+ = 6.26$) is lower than that of 4.5 kW ($Q^+ = 5.63$). The mean loop flow rate also demonstrates the same trend. On the other hand, the nearly out-of-phase oscillation in the two channels suggests that an inner flow loop is formed by the two channels in addition to the main circulation loop. The flow direction in the inner loop changes its direction periodically. Such an inner flow loop with alternative flow direction will certainly dissipate much of the energy provided by the heater. Consequently, the loop mass flow rate is not as high as it

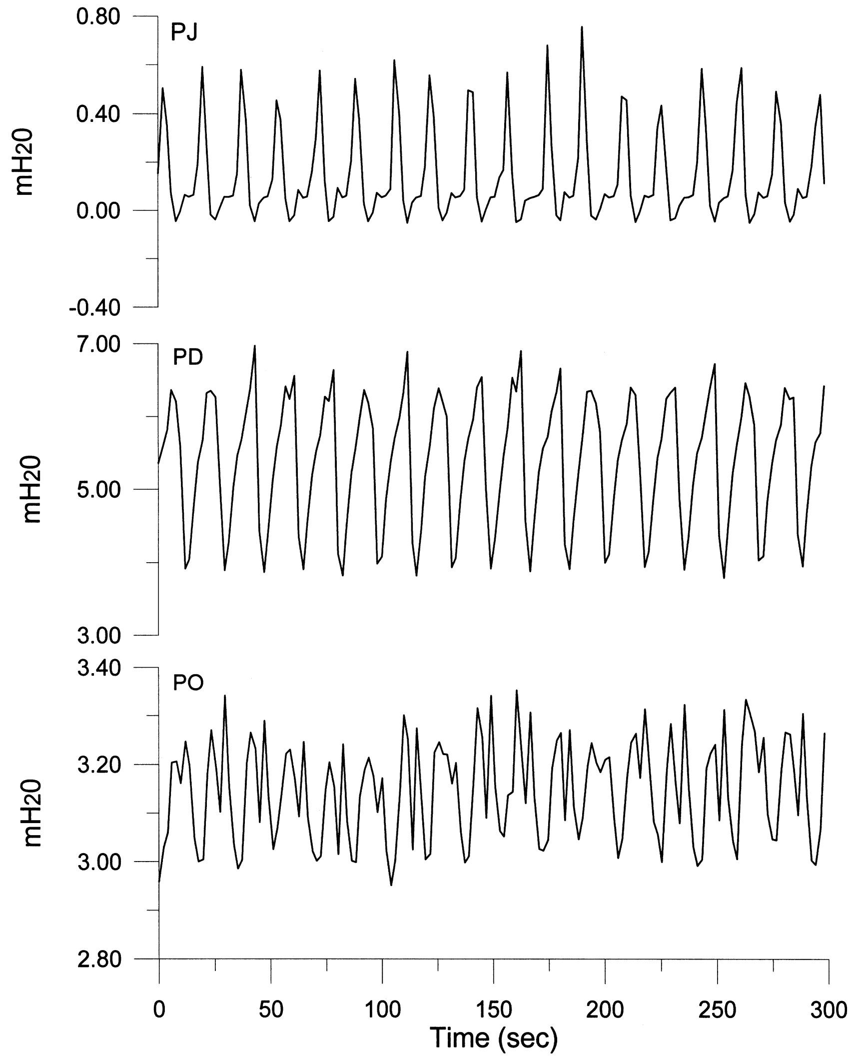

Fig. 5. Gauge pressures and differential pressures at various locations. PJ: differential pressure in the heated section of glass channel; PD: gauge pressure at the inlet of heated section of the glass section; PO: gauge pressure at condenser outlet (run 1).

should be. And, it becomes possible that the mean loop flow rate at 5 kW ($Q^+ = 6.26$) is actually lower than that at 4.5 kW ($Q^+ = 5.63$), which has a lower driving potential.

It will be interesting to examine the pressure variations during the oscillation. Fig. 5 illustrates the gauge pressures at the inlet of the heated section of glass channel (PD) and at the outlet of condenser (PO) and the pressure drop in the heated section of the glass channel (PJ) for both channels heated at 5 kW ($Q^+ = 6.26$). As indicated earlier, the pressure at the condenser outlet is considered as the system pressure in the present study. The pressure data was recorded by HP3852 with a sampling rate of 0.5 Hz. This sampling rate is somewhat low; therefore, the pressure data shown are not as smooth as the data shown in Fig. 4.

It is clear that the gauge pressure at the inlet (PD) and differential pressure in the heated section of the heated section (PJ) oscillate with the same frequency as the flow rate (see Fig. 4, run 1). The pressure at the inlet of the heated section oscillates almost out-of-phase with the flow rate. The low pressure corresponds to the peak flow rate, at which slug and churn flows are expected to be in the heated section and the riser. The presence of these large bubbles will reduce significantly the hydraulic head in the channel and leads to a low pressure at the inlet. Conversely, during the reversed flow period, the channel is filled with mostly liquid resulting in a high pressure at the inlet.

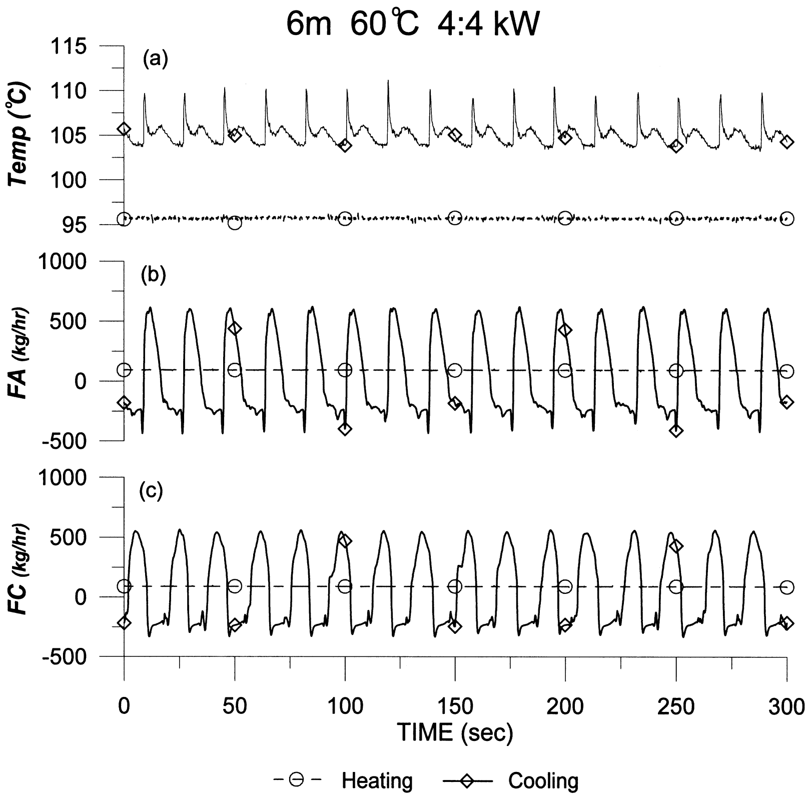

Fig. 6. Time evolutions of thermal hydraulic properties at 4.0 kW: (a) riser fluid temperature, (b) flow rate in the glass channel (FA) (c) flow rate in the steel channel (FC).

For the pressure drop in the heated sections, the major pressure peaks are much steeper than those of gauge pressure at the inlet. The pressure drop rises quickly while the flow rate begins to fall down. It reaches the peak value at the middle of the reversed flow period, at which boiling begins near the top of the heated section. A small but negative differential pressure peak appears while the channel flow rate reach its second reversed flow peak. Subsequently, the pressure drop rises with the loop flow rate. A secondary peak appears corresponding to the peak flow rate. Such a variation pattern suggests that the pressure drop in the heated section is dominated by the gravitational pressure drop under the conditions of this study. The largest pressure drop begins to appear after the passage of slug or churn flow pattern and the void fraction in the heated section is zero or little. The low sampling rate may fail to record the highest pressure drop corresponding to the liquid height. The larger flow rate and void fraction can only result in a minor peak in the pressure drop variation.

Unlike the heated section, the pressure variation at the condenser outlet is of double frequencies. One of them, i.e., the second peak of each major pressure oscillation is coincident with the oscillatory loop flow rate in the glass channel (FA). In fact, the high pressure at the condenser outlet and low pressure at the inlet of heated section will result in a high flow rate in the glass channel as shown. The first peak in each major pressure oscillation is coincident with

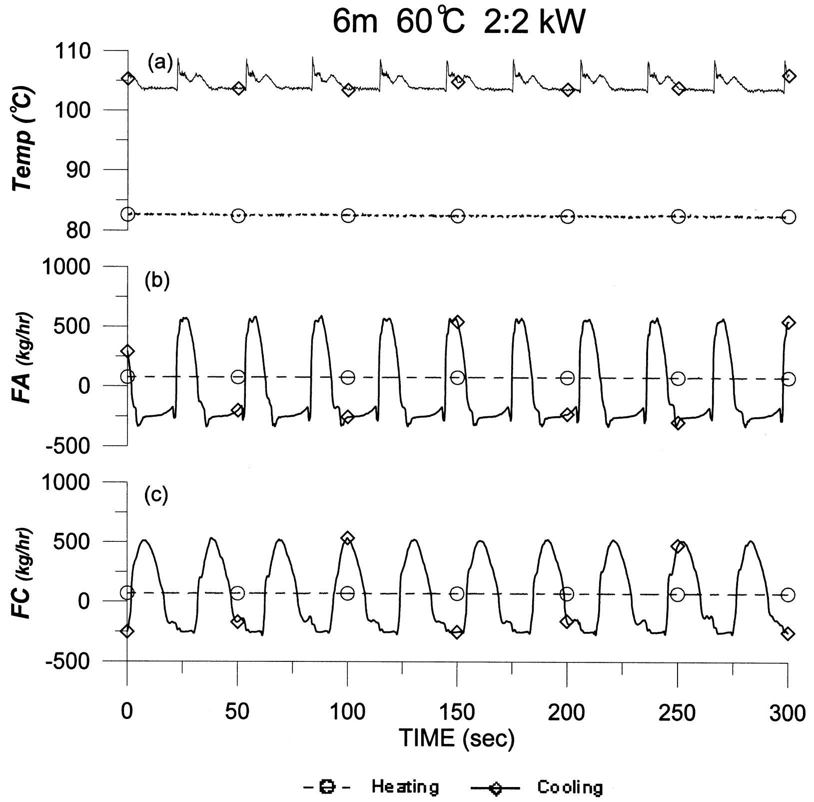

Fig. 7. Time evolutions of thermal hydraulic properties at 2.0 kW: (a) riser fluid temperature, (b) flow rate in the glass channel (FA) (c) flow rate in the steel channel (FC).

the peak flow rate in the steel channel (FC). Again, the first peak would explain the high flow rate in the steel channel.

3.3. Comparison of time-domain data under both heating and cooling conditions

The nearly out-of-phase oscillating two-phase flow persists while the heating power is reduced step by step to 2.0 kW ($Q^+ = 2.50$). Single-phase flow is restored when the power is further reduced to 1.5 kW ($Q^+ = 1.88$), at which the mean flow rate during cooling is close to that during heating. Figs. 6 and 7 exhibit the time evolutions of flow rate in each channel and riser temperature in the glass channel at 4.0 kW ($Q^+ = 5.01$) and 2.0 kW ($Q^+ = 2.50$), respectively. The data recorded during the heating procedure are also illustrated for comparison. It is clear that steady single-phase flow prevails in the loop under the conditions of heating procedure. Comparison among Figs. 4, 6 and 7 indicates that during the cooling procedure the oscillation magnitude is not markedly changed as the power is decreased from 5.0 kW ($Q^+ = 6.26$) to 2.0 kW ($Q^+ = 2.50$). However, the oscillation period is significantly increased. In particular, the period of reversed flow is extended remarkably. This is because that at lower powers, a longer incubation time is needed to heat up the fluid to initiate boiling (Aritomi et al., 1992). Once boiling is started and formation and spreading of bubble begins, coalescence and expansion of vapor blanket follows, and the forward flow is restored. Such a long period of reversed flow at 2.0 kW ($Q^+ = 2.50$) shown in Fig. 7b explains the negative mean flow rate shown in Fig. 2 for the glass channel. Such a peculiar phenomenon is of good repeatability as shown in Fig. 2. A positive mean flow in the steel channel and negative mean flow in the glass channel suggests that, on the average, the glass channel serves as another downcomer for the steel tube. This is because the glass channel is heated with a slightly lower power and has no proper insulation as the steel tube. The maximum heat loss due to natural air convection is estimated to be 0.18 kW, which is significant compared to the heating power of 2.0 kW.

4. Discussion

The data obtained in this study clearly illustrate the existence of the hysteresis effect in the present natural circulation loop. Evidently, once significant two-phase flow is initiated at some power, that may be referred to as the incipient power, it persists even though the power is reduced significantly below the incipient value. Such a hysteresis phenomenon is quite similar to that at the onset of nucleate boiling (Bar-Cohen, 1992). Boiling and subsequently bubble departure and significant two-phase flow can only be initiated in the loop at a relatively high power during a heating procedure. It may be speculated that subcooled water floods most of cavities on the heating surface in the present study; therefore, boiling is prohibited at lower powers. Once boiling takes place, many of the cavities are activated and remain active or contain residue vapor and can be activated at relatively lower powers. Moreover, the out-of-phase oscillation during two-phase flow on the two channels also provides hot fluids from a channel to the other channel from the top during its reverse flow period. This also enables boiling incipience at a lower power. Indeed, visual observation indicates that boiling begins at the top heated section and spreads downward during reversed flow period.

The hysteresis effect is of significant interest for the operation of a natural circulation loop. Under the situation of loop oscillation, an operator may attempt to reduce the heating power to suppress the instability. However, the hysteresis effect would hinder the immediate suppression of instability, as the oscillation may persist till a very low heating power.

5. Conclusions

This study investigates experimentally the effect of power changing procedure, i.e., heating or cooling, on a double channel natural circulation loop at the onset of two-phase flow. The data obtained clearly demonstrate the existence of hysteresis phenomena: at a given inlet subcooling and loop pressure, single-phase flow prevails in the loop during the heating procedure until the incipient power for two-phase flow is reached; at that power, however, the flow is found to be unstable; large magnitude, nearly out-of-phase oscillations with reversed flow appear in the two channels; such an unstable two-phase flow persists while the power is gradually reduced till a relatively low power, at which single-phase flow is restored. The hysteresis effect in the loop suggests that the loop state be a path function. It depends not only on the operating conditions but also on the process approaching it. This is of significant interest for the operation of a natural circulation loop.

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Appendix A. Saha and Zuber's (1974) model of net vapor generation point in a subcooled boiling channel

$$\Delta T_{\mathrm{sub}}(z)_{,\mathrm{nvp}} = \begin{cases} 0.0022\frac{q''D_{\mathrm{H}}}{k_{\mathrm{L}}} & \frac{GD_{\mathrm{H}}Cp_{\mathrm{L}}}{k_{\mathrm{L}}} < 70{,}000 \\ 153.8\frac{q''}{GCp_{\mathrm{L}}} & \frac{GD_{\mathrm{H}}Cp_{\mathrm{L}}}{k_{\mathrm{L}}} > 70{,}000 \end{cases} \tag{A1}$$

and

$$WCp_{\mathrm{L}}[\Delta T_{\mathrm{sub,\ in}} - \Delta T_{\mathrm{sub}}(z)_{,\mathrm{nvp}}] = q''Pz_{\mathrm{nvp}} \tag{A2}$$

where $\Delta T_{\mathrm{sub}}(z)_{,\mathrm{nvp}}$ is the liquid subcooling at the net vapor generation point, $\Delta T_{\mathrm{sub,\ in}}$ is the inlet subcooling, q'' is the surface heat flux, k_{L} is the liquid thermal conductivity, G is the mass flux, Cp_{L} is the specific heat of liquid and W is the loop mass flow rate.

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