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Pressure drop oscillation of steam–water two-phase flow in a helically coiled tube

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Abstract

Systematic investigations of pressure drop type oscillations and their thresholds were conducted for steam–water two-phase flow in a uniformly heated, helically coiled tube. Experiments were conducted to obtain the critical conditions for the occurrence of pressure drop type dynamic oscillation in a closed-circulation helically coiled tubing steam generator. The study showed that different locations of the tank which provided a compressible gas volume (called compressible volume for short) corresponded to different oscillation initial boundaries as well as different oscillating amplitudes and time periods. Moving compressible volume from the inlet of the steam generator upstream could dramatically suppress the occurrence of pressure drop oscillation. Also, non-uniform heat flux distribution along the evaporating channel could change oscillating boundaries significantly, especially when higher heat flux was applied to the higher mass quality region. A study of pressure drop oscillation in various steam generator inclinations showed that gravity has little influence upon the oscillating boundaries. Based on the experimental observations, a set of new methods was first proposed to eliminate the occurrence of pressure drop oscillation. © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: Pressure drop type oscillations; Steam–water two-phase flow; Closed-circulation helically coiled tubing steam generator

1. Introduction

Helically coiled tubes are utilized extensively in oil pipeline systems, heat exchanges, steam generators, chemical plants, etc., because of the practical importance of their high efficiency in heat transfer, compactness in structure, ease of manufacture and arrangement. They are also favored in space, navigation and other special techniques. Pressure for intensification has driven an increased demand for improved understanding of the key mechanisms responsible for fluid flow and heat/mass transfer encountered in these devices [1,2].

Many investigations have been performed on the internal flow of curved pipes and theoretical and experimental results reported. However, most of them were concerned with steady flows [3–5]. On the other hand, studies have become more important on unsteady or pulsating flows at the start–stop or undesirable accidents of pumps or other equipment. An increasing attention to problems of unsteady fluid mechanics and heat mass transfer has been seen in the last thirty years or so. Some important problems that have been successfully tackled include rocket combustion instabilities and unsteady atmospheric re-entry, while continuing challenges include the possibility of improving the efficiency of heat exchangers [1,6–10]. In some instances, it may even be beneficial to induce pulsation in the flow system if enhanced performance will ensue. However, very little research has been conducted on unsteady pipe flow, especially on unsteady flow in curved tubes. Only a few results concerning its complicated phenomena can be found in literature.

Guo et al. [11] reviewed previous studies of single-phase pulsation flow. They also conducted some experimental work on the oscillatory heat transfer

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Nomenclature

a	excitation parameter for oscillatory flow in [13]
A_p	oscillatory amplitude, kg/m^2
C_p	specific heat, kJ/m^3 °C
d	inner diameter of tube $d = 2r$, m
D	coil diameter $D = 2R$, m
f	frequency of oscillation, 1/s
G	mass flow rate, kg/m^2
h	heat transfer coefficient, kJ/m^2 °C
K	thermal conductivity, W m/°C
Nu	$h \times d/K$, local Nusselt number
P	pressure, Pa
q	heat flux, kW/m^2
Re	Reynold number
r	the radial coordinate, m
R	radius of coils, m
t	time, s
T	temperature, °C, K
S	interior heat source, W/m^3
Y	measurement value of wall temperature, °C

Greek symbols

μ	fluid dynamic viscosity, kg/m^2 s
ρ	fluid density, kg/m^3
ϕ	sensitivity coefficient
θ	the coordinate for the peripheral angle

Subscripts

c	coil average
f	fluid
I	inner
l	liquid phase
osc	oscillatory
p	peripheral point of cross section
s	cross-sectional average
slo	superficial for total liquid flow

characteristics of turbulent sub-cooled fluid flow in the ranges of $Re = 25\,000$–$125\,000$ and oscillation frequency $f = 0.05$–0.003, which correspond to the thermal-hydrodynamic oscillations encountered in a helically coiled tube boiling channel.

Flow instability induced by boiling-heat transfer appears widely in two-phase flow systems including reactors, steam generators, boilers and various heat exchangers. The oscillation of flow rate, system pressure and other associated parameters is generally undesirable as they can cause mechanical vibrations, high transient temperatures, control difficulty and even burnout of the surface. Therefore, finding out effective methods to avoid or control oscillation is one of the vital problems in practical engineering applications. Stenning [12,13] first identified three types of dynamic instabilities, namely, density waves oscillation (DWO), pressure drop oscillation (PDO) and thermal oscillation (THO). Among them, PDO and DWO are pure instability types while THO is considered a secondary phenomenon [14]. DWO is characterized by its period, which has the same scale of amount as fluid transit time (i.e., the time of fluid particles flowing through the evaporating region) and its amplitude, which is lower than that of PDO, so that it can sometimes be endured by equipment without causing serious loss. PDO, however, exhibits a long oscillatory period and is always accompanied by high oscillation amplitude and even dramatic wall-temperature jumping. Therefore, eliminating the occurrence of PDO in steam generating facilities is most important for the safety of equipment. The occurrence of PDO is closely related to the compressibility of fluid in the circulation loop. A compressible volume of gas can be provided by a surge tank, which is often installed to absorb the vibration induced by the pump, or a large amount of steam in the downstream of the evaporator, which may induce PDO as well.

During the past decades, numerous experimental and theoretical papers have been published clarifying PDO, and some important results have been obtained. Especially, an effective method of adding a throttle device upstream of the evaporator has been suggested, also confirmed by various industrial applications, for eliminating the occurrence of PDO. Most of the previous work has been conducted concentrating on the measurement or numerical simulation of oscillation boundaries of a parallel channel system. There is a shortcoming where the boundary condition of the evaporator is considered as a constant type of pressure drop, and the influence of the position of compressible volume in the loop may be neglected. However, in a closed-circulation loop, the boundary condition becomes a variable pressure drop, i.e., determined by pump dynamics. In this case, the hydrodynamic characteristics of the circulation loop and the compressible volume in the loop influence the oscillation boundaries. Thus, compressible volume positions should be considered, which has not been recognized so far.

Most of the modern thermal-propelled marine reactor steam generators are designed as close-circulation ones, in which the working fluid is operated by a pump or by natural circulation, while the power is provided by nuclear reaction or high-energy-density chemical fuel. In this case, non-uniformity of heat flux distribution is unavoidable. Knowledge of the influence of non-uniform heat flux distribution on the instability boundary is rare. Meanwhile, a marine facility or navigator may be operated at various inclinations during the rotating movement. The effect of gravity is induced upon the fluid flow. This is what the present study addresses. The present experiments are aimed at investigating the characteristics of PDO. Emphasis is placed on the influence of the installing location of the tank that provides a compressible gas volume within the circulation loop, non-uniform heat flux distributions along the evaporator and steam generator inclination upon the characteristics and boundary of PDO in a helically coiled tubing steam generator in a closed-circulation loop. As a result, a set of new methods is proposed to eliminate the occurrence of PDO.

2. Experimental apparatus and procedure

2.1. Experimental apparatus

The experiment was conducted in a closed steam–water two-phase flow test loop schematically illustrated in Fig. 1. It consists of a centrifugal pump to supply power for fluid flow, a pressurized nitrogen tank to maintain and control the system pressure, a series of orifice meters to measure water mass flow rate, a pre-heater to heat the fluid and control the inlet temperature, a test section, a water-cooled condenser and a water tank. The resistance of the tube wall of both the test section and the pre-heater was used to heat the working fluid with total power of 200 kW.

The test section was made of a 6448 mm long tube of $\varnothing$ 15×2 mm^2, with a helix angle of 4.27°, a coil diameter of 256 mm and a pitch of 60 mm, as shown in Fig. 1. It was thermally insulated with fiberglass. The mass flow rate of the working fluid was measured using three orifice meters of different ranges (in-house construction with standard specification) appended to three differential pressure transducers (Model 1151 DP) with response times of 0.1 s. Two manometers were used to measure the pressure at the inlet and outlet of the test section. The pressure drops of the test section were also measured using a differential pressure transducer. Three armoured thermocouples (made of NiCr and NiSi wires of 1 mm diameter) were installed in the core of the tube to measure the fluid temperatures at the inlet, central station and outlet of the test section. 102 thermocouples (made of NiCr and NiSi wires of 0.3 mm diameter) were installed on the outer surface of the heated tube wall to measure the temperature of the wall, eight ones were arranged uniformly along the periphery of the outer surface of the tube wall at each cross-section of every quarter turn of the first, second and third turns of coils, and all of them were attached to the tube wall and electrically insulated so that the heating effect of electrical current on it was avoided. All of the instantaneous signals of parameters and input powers of heat to test section and pre-heater were monitored and stored in an IBM PC computer via six isolated measurement pods, and were also recorded and monitored by an AR-5000 cassette tape recorder so that further statistical analysis could be done in some runs. The system for parameter measurement and collection is shown in Fig. 2.

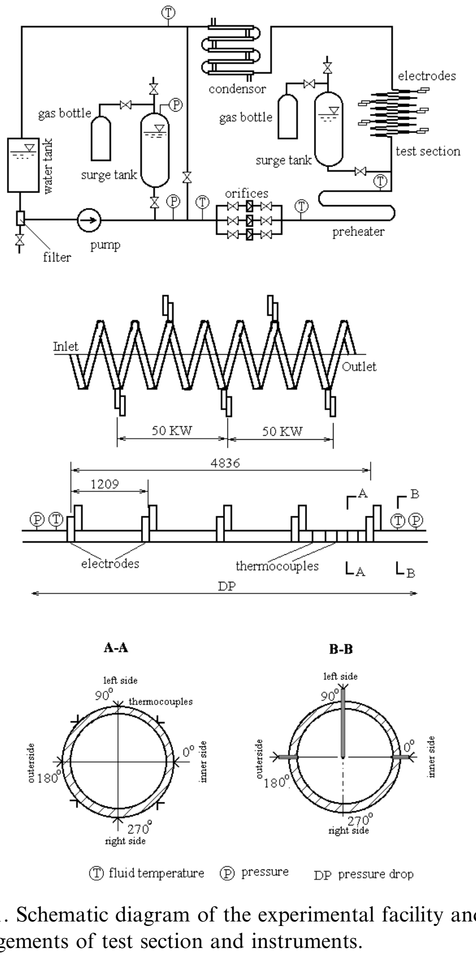


Fig. 1. Schematic diagram of the experimental facility and the arrangements of test section and instruments.

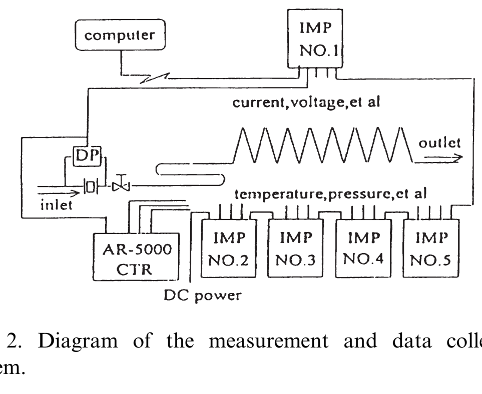


Fig. 2. Diagram of the measurement and data collection system.

2.2. Experimental procedure

In the present investigation, two locations of tank that provide a compressible gas volume were tested, one located at the inlet of the evaporator and another located at the outlet of the circulation pump. In order to examine the influence of non-uniform heat flux distributions along the evaporator, the helically coiled tube test section was divided into two parts and was supplied with several heat flux ratios. The influence of steam generator inclination was tested by positioning the test section at four different angles relative to the horizontal position. Detailed measurements of unstable boundaries were carried out under various mass flow rate, system pressure, inlet sub-cooling and compressible volume conditions.

Preliminary experiments of steady single-phase flow and heat transfer were performed to verify the reliability of the experimental system. The experimental results showed good agreement with the pressure drop correlation of Ito et al. [15] and with the average heat transfer coefficient correlation of Seben and Mclaughlin [16].

Steady state characteristics of the system were obtained in terms of pressure drop versus mass flow rate and used to locate instability boundaries and determine the steady and unsteady regions for following range of parameters: system pressure $P = 0.5$–3.5 MPa; mass flow rate $G = 150$–2500 kg/m^2 s; heat flux $q = 0$–540 kW/m^2.

The tests were conducted using following procedure:
1. establish operation parameters including the mass flow rate, compressible volume and inlet temperature and confirm their good stability;
2. increase the heat flux of the test section and await establishment of thermal equilibrium or; if unstable, await sustained oscillation, recording the average value of the system parameters and oscillatory records over two or more periods;
3. repeat the above procedure after adjusting the operating parameters.

2.3. Data reduction

Here, the numerical solutions developed by the present authors to calculate the temperature and heat transfer coefficient on the inner surface of the helically coiled tube [17,18] were employed in processing the data. The bulk temperature of the fluid was determined assuming thermal equilibrium and linear pressure differential gradient along the flow direction in the single-phase region.

3. Experimental results and analysis

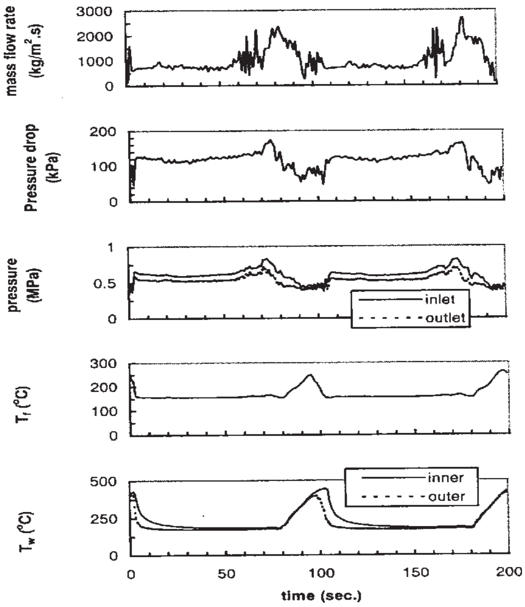
3.1. Pressure drop type oscillations and their thresholds

As indicated by Stenning [13] and also confirmed by other researchers ([19–21], etc.), the occurrence of PDO is determined by the compressible volume of gas of the two-phase flow system. Compressible volume of gas can be provided by a surge tank, which is often installed to absorb the vibration induced by the pump, or a large amount of steam downstream of the evaporator which may induce PDO as well. For these reasons, we first tested the PDO under two conditions: with and without a surge tank. The results demonstrated that PDO occurs only when there is a surge tank in the circulation loop, confirming the importance of compressible gas volume to the occurrence of PDO. As mentioned above, the experiments (as also theoretical works) carried out by previous researchers were usually conducted with a fixed surge tank position. None of them considered the influence of compressible gas volume position on PDO characteristics. To meet this requirement, two surge tanks of the same size were installed in our circulation loop as shown in Fig. 1(a). When the surge tank was located at the outlet of the pump, a large tube length and many throttle devices (such as valves) were present between the surge tank and the inlet of the test section. When the surge tank was located at the inlet of the test section (similar to Stenning's experiment loop) only a very short tube length was present between the surge tank and the inlet of the test section. Remarkably, we found that the PDOs obtained at these two surges tank positions are significantly different not only in their occurrence boundaries, but also oscillation amplitudes and time periods. Fig. 3 shows the typical PDO curves obtained, respectively, for the two above-mentioned cases.

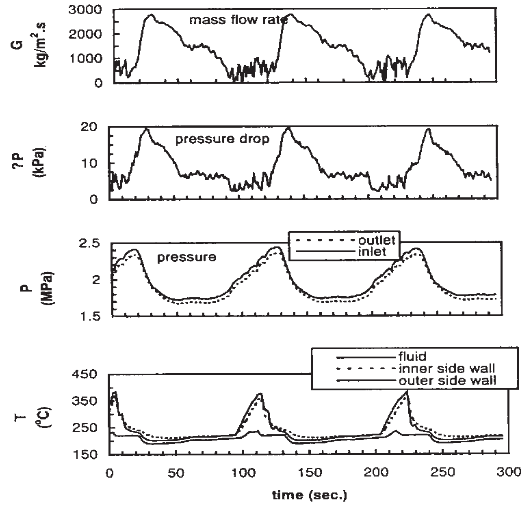
Fig. 4 shows their unsteady boundaries on the ΔP–G map. It can be seen that PDO occurred in the case of high exit mass quality with the first installation position of compressible gas volume, while PDO occurred in the starting process of evaporation and even at sub-cooled boiling conditions of outlet cross-section with the second position. Meanwhile, it was also found that the amplitude and time period of PDOs under the two various surge tank positions are significantly different. In the following sections, detailed features of the PDOs obtained from each compressible volume position will be documented.

3.2. Pressure drop oscillation when compressible gas volume is located at the outlet of the pump (1st PDO)

It can be seen from Fig. 5 that the initial mass quality increases with system pressure. This result is similar to the results obtained by other researchers [13,21]. In our experimental ranges, no oscillation appears when the system pressure is greater than 3.0 MPa. This may be due to the increase in system pressure causing a decrease in the density difference between the steam and liquid phases. The mass flow rate disturbance resulting from the pressure disturbance decreases compared with that under the low-pressure conditions. Fig. 5 also demon-

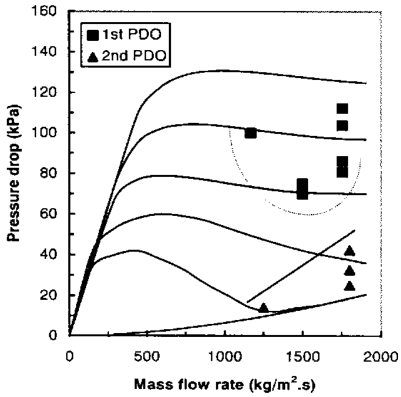


(a) 1st PDO



(b) 2nd PDO

Fig. 3. Typical oscillation curves when the compressible gas volume of system is oriented (a) in front of a pre-heater and (b) at the inlet of test section.



(1), $q_w=0$, (2), $q_w=100\ \mathrm{kW/m^2}$, (3), $q_w=200\ \mathrm{kW/m^2}$, (4), $q_w=300\ \mathrm{kW/m^2}$,

(5), $q_w=400\ \mathrm{kW/m^2}$, (6), $q_w=500\ \mathrm{kW/m^2}$

Initial boundaries of 1st PDO

Disappear boundaries of 2nd PDO

Fig. 4. Occurrence regions of two kinds of PDO on the hydrodynamic curve.

strates that the initial heat flux or initial mass quality of the 1st PDO decreases monotonically with increasing mass flow rate; it is different from that of a parallel channel system [14] or an open-circulation loop [13] and is considered the result of boundary conditions. In a closed-circulation loop, the pressure head provided by the pump is almost constant. With increase of mass flow rate, the pressure drop occupied by the single-phase flow part of the circulation loop increases and, therefore, pressure drop provided to the evaporator decreases. On the other hand, two-phase pressure drop is larger under the higher mass flow rate conditions. The sum of pressure drops of the evaporator and circulation loop can easily approach the pressure head provided by pump, so that the initial heat fluxes of PDO are lower in the higher mass flow rate conditions.

The influence of inlet subcooling on the 1st PDO is shown in Fig. 6, where the initial heat flux increases with inlet sub-cooling, especially in higher system pressure conditions. This is understandable since, in higher inlet subcooling conditions, higher heat flux is required to approach the same two-phase pressure drop as that of the lower inlet sub-cooling conditions. The initial boundaries of PDO are also affected by the size of compressible volume of gas. From experimental data it is found that the 1st PDO can be obtained only when this compressible volume reaches a certain value. For example, under the condition of $P = 0.5$ MPa, $\Delta T_{\mathrm{sub}} = 30$ K and $G = 1160$ kg/m^2 s, only when the compressible volume of gas $V_{\mathrm{st}} \geqslant 0.048$ m^3 can the 1st PDO be generated. The initial boundaries of the 1st PDO in helically coiled tubes with various helix axis inclinations are also compared. However, no significant difference is obtained among them. This may be due to the short length of the helix axis, and one can expect a pronounced difference when a long tube is used as the evaporator.

The present experiments show that oscillation amplitude depends on system parameters. Relatively, mass flow rate and pressure drop of the test section have

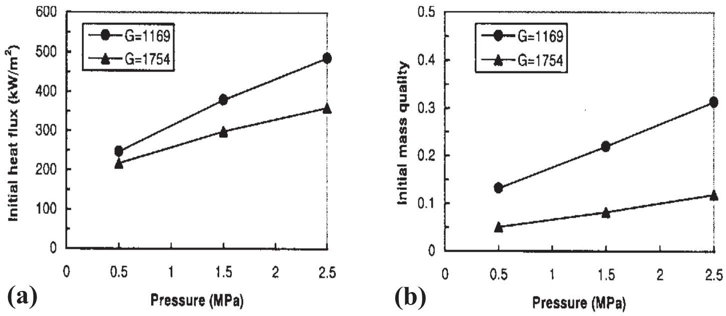


Fig. 5. Initial heat flux and mass quality of the 1st PDO under the compressible gas volume of system $V_{st} = 0.036$ m^3 and inlet subcooling $\Delta T_{sub} = 30$ K.

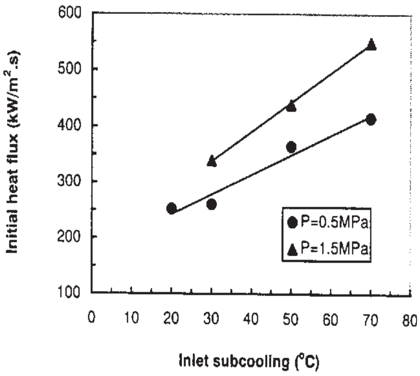


Fig. 6. Influence of inlet subcooling on initial boundary curves of the 1st PDO.

larger amplitudes compared with other parameters. For the most of the test runs, the two parameters can exceed 200% that of the stable value, and even reverse flow was sometimes induced. The jumping of wall and bulk temperatures can be observed at low peak values of mass flow rate, and, under these conditions, the tube wall of the evaporator may easily be deformed or even broken. No matter in what axis position the test section is placed and no matter how much the wall heat flux is, wall temperature jumping first occurs at the outlet cross-section of the evaporator and then quickly spreads upstream. This process usually occurs within a few seconds and therefore causes great internal stress in the tube wall. In our tests, torsion and deformation of the helically coiled tube test section can be observed in almost every oscillatory run.

Due to the influence of centrifugal force due to curvature of the coils, great difference exists among the wall temperatures at various points on the circumference of the tube. Fig. 3(a) shows the oscillation curves of the inner and outer wall temperatures of the outlet cross-section, where T_{wi} is the inner wall temperature while T_{wo} is the outer wall temperature. The highest wall temperature value obtained in the present experiments is located in the inner side of the tube and sometimes is even greater than 700°C and the corresponding fluid temperature jumping exceeds 400°C. At the peak value of wall temperature oscillation, the difference between inner and outer wall temperatures even approaches 200°C.

Different from the density wave oscillation, the time period of the 1st PDO is more than the transit time required for fluid particles to travel through the test section. The largest period obtained in the present tests is 350 s, while the lowest value is 20 s; however, most of the runs are concentrated within the range of 50–150 s. Also, the time period of the 1st PDO is greatly influenced by system parameters. The increase in compressible gas volume increases the time required for the surge tank to store and release energy and, therefore, increases the oscillation period. The increase in wall heat flux increases the time period and even induces the disappearance of the 1st PDO. Fig. 7 shows the dependence of the time period on mass flow rate and system pressure. It can be seen that time period increased with mass flow rate but decreased with system pressure.

Providing a universally accurate correlation for the prediction of pressure drop oscillation is very difficult,

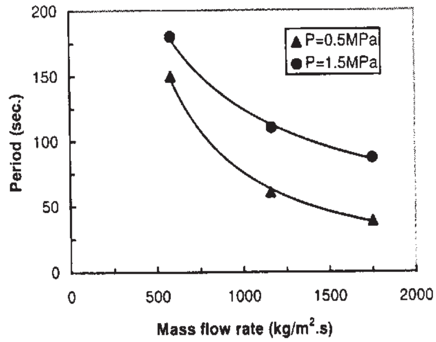


Fig. 7. Oscillation period of the 1st PDO at $\Delta T_{sub} = 20$ K, $V_{st} = 0.036$ m^3.

since the initial boundaries of oscillation are significantly varied with experimental facilities. Based on the non-dimensional analysis of one-dimensional two-fluid model equations [22,23], the following non-dimensional parameters, i.e., phase change number N_{pch}, sub-cooling number N_{sub}, density number N_d, Froude number Fr and Dean number Dn, are introduced to describe the occurrence boundaries of first pressure drop oscillation in the following expression:

$$f(N_{pch}, N_{sub}, N_d, Fr, Dn) = 0. \tag{1}$$

For helically coiled tubes, the above-mentioned parameters can be identified as

$$N_{pch} = q_w/(Gh_{fg}), \tag{2}$$

$$N_{sub} = c_{p1}\Delta T_{sub}/h_{fg}, \tag{3}$$

$$N_d = \frac{\rho_1}{\rho_g}, \tag{4}$$

$$Fr = GA/(\rho g H), \tag{5}$$

$$Dn = Re(d/D)^{0.5}. \tag{6}$$

In terms of these groups, the experimental data indicate the following correlation for the onset of first pressure drop oscillation:

$$N_{pch} = 2.13 Fr^{1.48}\, N_{sub}^{-0.89} N_d^{-0.16}\, Dn^{-0.43}. \tag{7}$$

Comparison between the experimental data and the calculated result shows that more than 95% of the data fall within a 10% error range. However, it should be noticed again that this correlation is effective only in the facilities and test conditions described in this paper, and one should be very careful when trying to introduce it to other conditions.

3.3. Pressure drop oscillation when the compressible volume is located at the inlet of the evaporator (2nd PDO)

In fact, the original definition of PDO is obtained by locating a tank which provides a gas-compressible volume for the system at the evaporator inlet, where the PDO occurred in the initial process of boiling [12]. General features of the 2nd PDO have been well reported by many researchers and, therefore, our attention will be focussed on the difference between the two types of PDO observed.

Since the 2nd PDO occurs at the beginning of the boiling process at the tube outlet cross-section, its initial boundaries cannot be characterized by means of initial mass quality and initial heat flux but, rather, by means of critical gas-compressible volumes. Our experimental results reveal that, under a certain parameter range, if the 2nd PDO does not appear, the 1st PDO will not appear either. Fig. 8 shows the critical compressible gas volumes required for the occurrence of 1st and 2nd PDOs. It can be seen that the increase in system pressure and mass flow rate linearly increases the critical compressible gas volume. However, the compressible volume required for the occurrence of the 1st PDO is nearly 2–4 times that required for the 2nd PDO. In other words, moving the compressible gas volume of the circulation system upstream of the test section can significantly eliminate the occurrence of PDO. On the other hand, the initial condition of the 2nd PDO is also affected by mass flow rate and inlet sub-cooling. The increase in mass flow rate and inlet sub-cooling increases the critical compressible volume almost linearly. This is because the loop pressure drop between surge tank and evaporator is negligible so that the evaporator is almost provided with constant boundary conditions for pressure drop.

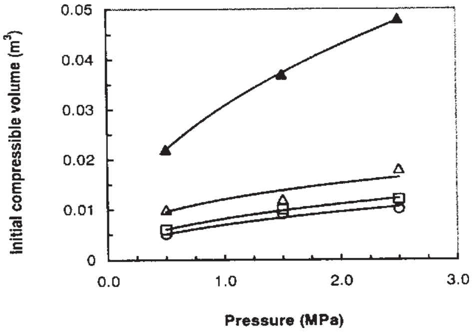


Fig. 8. Critical system compressible volume required for the occurrence of PDO: ○ $G = 580$, 2nd PDO; □ $G = 1140$, 2nd PDO; △ $G = 1760$, 2nd PDO; ▲ $G = 1140$, 1st PDO.

The 2nd PDO starts at the beginning of the evaporation process at the outlet of the test section. The amplitude peak of its pressure drop is different from that of the 1st PDO and just equal to the pressure drop of single phase corresponding to the peak amplitude of mass flow rate oscillation. The inlet mass flow rate has a greater oscillatory amplitude than the 1st PDO. However, the oscillation of mass flow rate is confined only downstream of the surge tank. Fig. 9 shows the oscillation curves of mass flow rate measured upstream and downstream of the surge tank, where the flow rate measured upstream of the surge tank remains almost without oscillation. Also, the oscillating amplitude of the 2nd PDO is quite different from that of the 1st PDO, where temperature jumping occurs frequently. The wall temperature amplitudes obtained during the 2nd PDO are lesser than that of the steady-state values by 15%, and the amplitude of fluid temperature is even lesser than the saturated value by 10%.

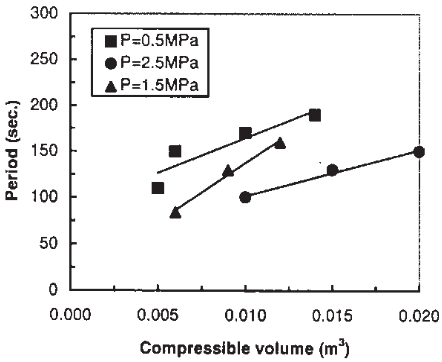


Fig. 9. Mass flow rate measured upstream and downstream of surge tank during the 2nd PDO.

The period of the 2nd PDO is of the same order as that of the 1st PDO owing to the same mechanisms. But the time period of the 2nd PDO almost linearly decreases with increase of compressible gas volume, while the 1st PDO has a parabolic relation with this volume. Within the oscillating process of the 2nd PDO, fluid fills in and releases out of the surge tank periodically, so that its period is closely related to the size of compressible gas volume. Large compressible gas volume corresponds to a long time period for the fluid filling in and overflowing from the surge tank, as shown in Fig. 10. This figure also demonstrates that the increase in system pressure decreases the oscillation period of the 2nd PDO. In addition, the time period of the 2nd PDO is also decreased by wall heat flux and mass flow rate, and these results are very similar to those obtained by Stenning [13].

Similarly, here we try again to provide a correlation to predict the occurrence boundaries of the 2nd PDO.

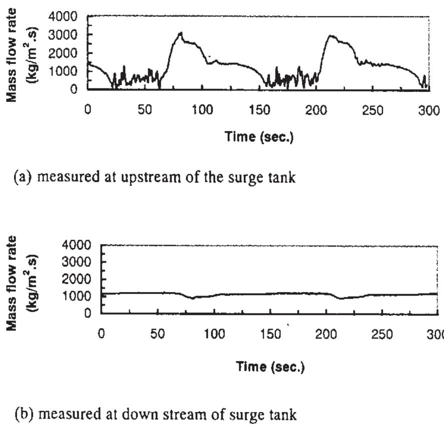


Fig. 10. Oscillation period of 2nd PDO at $\Delta T_{sub} = 30$ K and $G = 1160$ kg/m^2 s.

As the occurrence of the 2nd PDO is mainly controlled by mass flow rate, system pressure and inlet subcooling, it cannot be embodied by means of initial heat flux [24]. Therefore, our correlation is suggested by means of initial compressible volume

$$V_{st} = 1.117 N_d^{0.01} N_{sub}^{0.02}. \tag{8}$$

The error of this correlation in our experimental data is less than 12%.

3.4. Pressure drop oscillation under non-uniform heat flux conditions

Most of the heat exchangers are performed under non-uniform heat flux conditions along the flow direction. Available literature provides little information on this special area. In this context, a series of experiments was conducted to examine the influence of non-uniform heat flux distribution. The test section was divided into two equal parts and separately heated with two electrical power transformers, as shown in Fig. 2. The ratios of power supplied to the two sections are 1:1, 1:2 and 2:1, respectively. Totally, 25 runs were carried out and analyzed.

The results demonstrate that the non-uniform heat flux distribution has no significant effect on the 2nd PDO; however, it influences the initial boundaries of the 1st PDO remarkably, as shown in Fig. 11. Here, q_{w1} is the heat flux input to the first part, and q_{wt} is the average heat flux of the two parts. The initial heat flux under this condition is defined as the average heat flux of two parts. It can be seen that the initial heat flux and initial mass quality of non-uniform heat flux conditions are obviously lesser than that of the uniform conditions, especially when the heat flux is lower.

The influence of mass flow rate under various heat flux ratios is also shown in Fig. 11. It reveals that the initial mass quality decreases with increasing mass flow rate, and the influence of non-uniform heat flux distribution is pronounced even in low-mass flow rate conditions.

Another important result obtained is that the peak values of wall temperature under conditions of $q_{wl}/q_{wt} > 0.5$ are even greater than those of uniform heat-flux conditions. Therefore, in the design or operation of industrial equipment, higher heat flux should be avoided from the high-mass quality region, especially to those performed with low system pressures.

3.5. Discussion of the methods used to eliminate the pressure drop oscillation

An effective method to eliminate the pressure drop oscillation is to install a throttle device (such as throttle valve, throttle orifice and so on) at the inlet of the

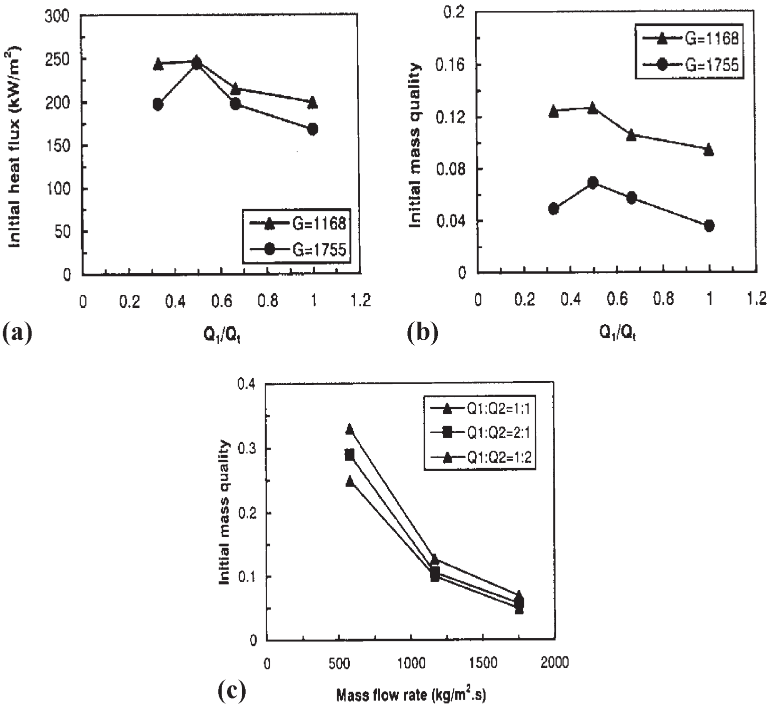


Fig. 11. Influences of non-uniform heat flux distribution and system mass flow rate on the initial heat flux and initial mass quality ((a), (b) under $P = 0.5$ MPa, $\Delta T_{sub} = 20$ K, for 1st PDO; (c) for 2nd PDO; Q1, Q2 are the heat flux for the first- and second-heated sections of test tube respectively).

evaporator, as indicated by Stenning [12]. This method has been confirmed by other researchers and has also been proved by industrial practice. However, it should be mentioned that the installation of a throttle device also increases the circulating resistance, which is not expected. Moreover, in our experimental facilities, we installed a throttle orifice of small diameter ($d_o/d = 5/11$, where d_o is the orifice diameter and d, the tube inner diameter), but even then could not suppress the occurrence of PDO. That means that this method is efficient only when the pressure drops in the evaporator are low and have the same magnitude as that of the throttle device.

Therefore, moving the compressible gas volume far away from the inlet of the evaporator is an effective method. The advantage of this method is that we need no additional throttle device, and the resistance of the loop between the compressible gas volume and the two-phase region exhibits a similar function as a throttle device. Certainly, eliminating the compressible gas volume from the two-phase flow facilities is the most effective method, if it is possible.

In addition, the PDO also can be suppressed by arranging the heat flux distribution uniformly. If non-uniform distribution cannot be avoided, then arrange more heat flux in the low-mass quality region as far as possible.

4. Conclusions

Pressure drop oscillations (PDO) in a closed-circulation helically coiled tubing steam generator were studied by employing water as the working fluid. Detailed information of initial boundaries as well as oscillating amplitudes and periods were obtained and discussed. Emphasis was placed on the influence of compressible gas volume positions, non-uniform heat flux distributions and various helix-axis inclinations on the PDO characteristics. A set of methods was first suggested to eliminate the occurrence of PDO in the closed-circulation system.

1. The positions of compressible gas volume in two-phase flow loop affect the occurrence boundaries of PDO remarkably. Different compressible gas volume positions correspond to different initial boundaries, oscillatory amplitudes and periods of PDO. The results obtained from two positions of compressible gas volume are reported, and their boundaries are correlated by means of non-dimensional parameters as Eqs. (7) and (8).
2. Non-uniform heat flux distribution seriously decreases the initial boundaries of PDOs, especially of those occupying higher heat flux values at higher mass quality regions. Helix-axis directions have no significant influence on PDO characteristics.

3. New methods are suggested to eliminate the occurrence of PDO in a closed circulation loop. It is found that moving the compressible gas volume from the inlet of the evaporator to upstream of the circulation loop can dramatically suppress the occurrence of PDO. In addition, the PDO also can be suppressed by arranging the heat flux distribution uniformly. If non-uniform distribution cannot be avoided, then arrange more heat flux in the low mass quality region as far as possible.

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