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Transient convective heat transfer of steam–water two-phase flow in a helical tube under pressure drop type oscillations

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Abstract

Experiments for subcooled water flow and for steam–water two-phase flow were conducted to investigate the effects of pulsation upon transient heat transfer characteristics in a closed-circulation helical-coiled tube steam generator. The non-uniform property of local heat transfer with steady flow was examined. The secondary flow and the effect of interaction between the flow oscillation and secondary flow were analyzed on basis of the experimental data. Some new phenomena were observed and explained. Correlations were proposed for average and local heat transfer coefficients both under steady and oscillatory flow conditions. The results showed that there exist considerable variations in local and peripherally time-averaged Nusselt numbers for pulsating flow. Investigations of pressure drop type oscillations and their thresholds for steam–water two-phase flow in a uniformly heated helical tube were also reported. © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: Transient heat transfer; Pressure drop type oscillation; Heated helical tube

1. Introduction

Helically coiled tubes are utilized extensively for piping system of oil pipeline, heat exchanges, steam generators, and chemical plants, because of their practical importance of high efficiency in heat transfer, compactness in structure, ease of manufacture and arrangement. Horizontal direction of tube axis may be an attractive orientation due to its lower gravitational center and higher efficiency both in heat transfer and steam generation. It is also favored for space, navigation and other special applications. Pressure for intensification has driven increased demand for improved understanding of the key mechanisms responsible for fluid flow and heat/mass transfer encountered in these devices [1,2].

For internal flow of curved pipes, many investigations have been performed in practical use and many theoretical and experimental results have been 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 the undesirable accident of pump or other power equipment. An increasing amount of attention to problems of unsteady fluid mechanics and heat mass transfer has been seen in the last thirty years. Some important examples such as rocket combustion instabilities and unsteady atmospheric re-entry were tackled successfully due to improving the efficiency of heat exchangers [1,6–10]. In some instances it may even be beneficial to induce pulsation in flow system if enhanced performance will apply. However, few reports have been published on unsteady flow in curved tubes. Guo et al. [11] reviewed previous studies of single-phase pulsation flow and also reported some experimental work on the oscillatory heat transfer of turbulent subcooled flow in the ranges of $Re = 25,000$–$125,000$, oscillations frequency $f = 0.05$–0.003, which corresponded to the thermal-hydrodynamic oscillations encountered in a boiling helical tube.

Flow instabilities induced by boiling appear widely in two-phase flow systems. Oscillations of flow rate, system pressure and other associated parameters are generally

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Nomenclature

a	excitation parameter for oscillatory flow in [13]
A_p	oscillatory amplitude
$\overline{Bo}$	time-averaged Boiling number, $\bar{q}/(\overline{G_0 h_{fg}})$
C_p	specific heat, kJ/m^3 °C
d	inner diameter of tube, $d = 2r$ (m)
D	oil diameter, D (m)
$\overline{Dn}$	time-averaged Dean number, $Re(d/D)^{0.5}$
f	frequency of oscillation (1/s)
G	mass flux (kg/m^2 s)
h	heat transfer coefficient (W/m^2 °C)
k	thermal conductivity (W/m °C)
Nu	local Nusselt number, hd/k
P	pressure, (Pa)
q	heat flux (kW/m^2)
Re	Reynolds number
r	the radial coordinate (m)
R	radius of coils (m)
t	time (s)
T	temperature; period of oscillation for time averaged (s)

Greek symbols

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

Subscript

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
tp	two phase flow
w	tube wall

undesirable as they can cause mechanical vibrations, high transient temperatures, control problems, and even burnout of the heat transfer surface. Therefore, finding the effective methods to avoid or to control oscillation is one of the vital problems for engineering practice [12–14]. Numerous experimental and theoretical researches have been published to clarify the pressure drop oscillation, from which adding throttle device into upstream of the evaporator can eliminate the occurrence of pressure drop oscillation and has been especially suggested and confirmed by industrial applications as an effective method [13]. Most of previous work is conducted to concentrate on the measurement or numerical simulation of oscillation boundaries of a parallel channel system. There exists a shortcoming where the evaporator boundary condition is a constant pressure drop type and the influence of positions of compressible volume in the loop may be neglected. However, in a closed-circulation loop, the boundary condition is changed to a variable pressure drop type, determined by the pump dynamic. In this case, the hydrodynamic characteristics of circulation loop and compressible volume influence the oscillation boundaries. Thus compressible volume positions should be considered which have not been well studied.

Most of the modern thermal-propelled marine reactor steam generators are designed by means of close-circulation, 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. As the non-uniformity of the heat flux distribution would be unavoidable. The knowledge of how the non-uniform heat flux distribution influence the instability boundaries is rarely addressed. Meanwhile, a marine facility or navigator may be operated in various inclinations during the rotation movement. The difference of effective angles of gravity upon fluid flow is induced. This is also aim of the present study is addressed.

In the present paper the experiments for subcooled water flow and steam–water two-phase flow were conducted to investigate the effects of pulsation upon transient heat transfer characteristics in a closed-circulation helical coiled tubing steam generator. The secondary flow mechanism and the effect of interaction between the flow oscillation and secondary flow were analyzed on the basis of experimental data. A series of correlations were proposed for the average and local heat transfer coefficients. The results showed that there exist considerable variations in local and peripherally time-average Nusselt numbers for pulsating flow in a wide range of parameters.

2. Experiments

Experiment was conducted in a closed-cycle test loop of steam–water two-phase flow, which is schematically illustrated in Fig. 1. It consists of the following components: a centrifugal pump to supply power for the 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 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 test section and pre-heater was used to heat the working fluid with alternating electrical current

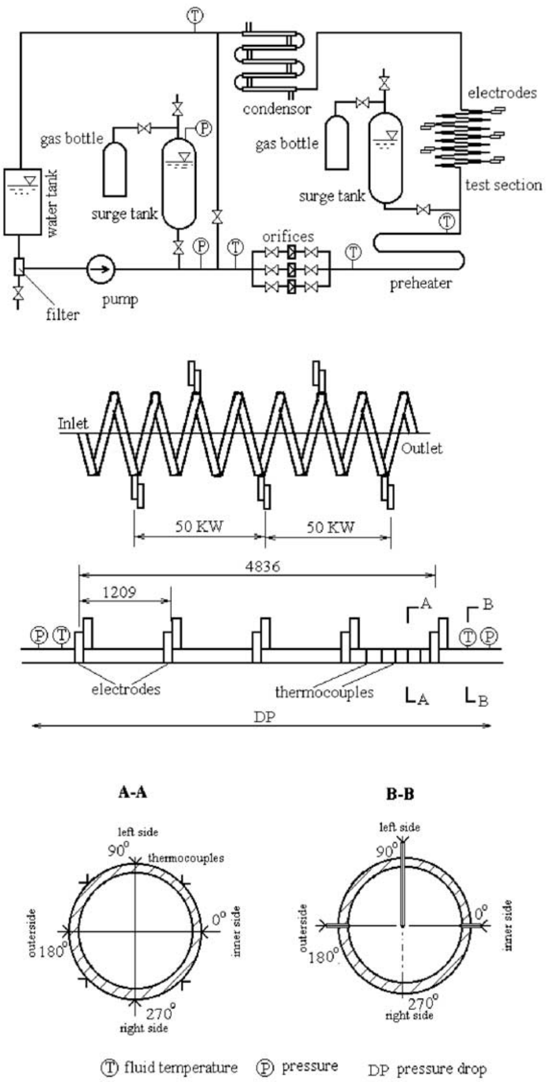


Fig. 1. Schematic diagram of experimental facility and the arrangement of test section and instruments.

delivering total power of 200 kW. The test section was made of a 6448 mm long tube of $\varnothing 15 \times 2$ mm, with helix angle of 4.27°, coil diameter of 256 mm and a pitch of 60 mm. It was thermally insulated and wrapped with fiberglass.

The mass flux of working fluid through the test section was measured by using three orifice meters in different ranges (in-house construction with standard specification) appending to three 1151-DP differential pressure transducers with response time of 0.1 s. Two manometers were used to measure the pressure at inlet and outlet of test section. The pressure drops of test section were also measured by a differential pressure transducer. Three armoured thermocouples (made of NiCr and NiSi wires with 1 mm diameter) were installed into the core of tube to measure the fluid temperature of inlet, central station, and outlet of test section. One hundred and two thermocouples (made of NiCr and NiSi wires with 0.3 mm diameter) were installed on the outer surface of heated tube wall to measure the temperature of wall, which were arranged uniformly along the periphery of outer surface of tube wall, eight for each cross-section at every position of quarter turn among the first, second and third turns of coils. All the thermocouples were attached to the tube wall and electrically insulated so that the effect of heating electrical current on it was avoided. All of the instantaneous signals of parameters, input powers of heat to test section and preheater were monitored and stored by a IBM PC computer via six Isolated Measurement Pods, and 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.

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 by Ito [15], and with the average heat transfer coefficient correlation by Seban 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 flux $G = 150$–$2500\ \mathrm{kg/m^2\,s}$; heat flux $q = 0$–$540\ \mathrm{kW/m^2}$.

As indicated by Stenning [13] and confirmed by other researchers [17–19], the compressible volume in two-phase system determines the occurrence of pressure drop oscillation. We first tested the pressure drop oscillation under two conditions with/without a surge tank. The results demonstrated that pressure drop oscillation

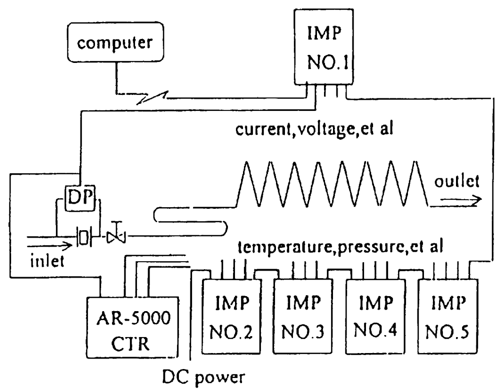


Fig. 2. Diagram of measurement and data collection system.

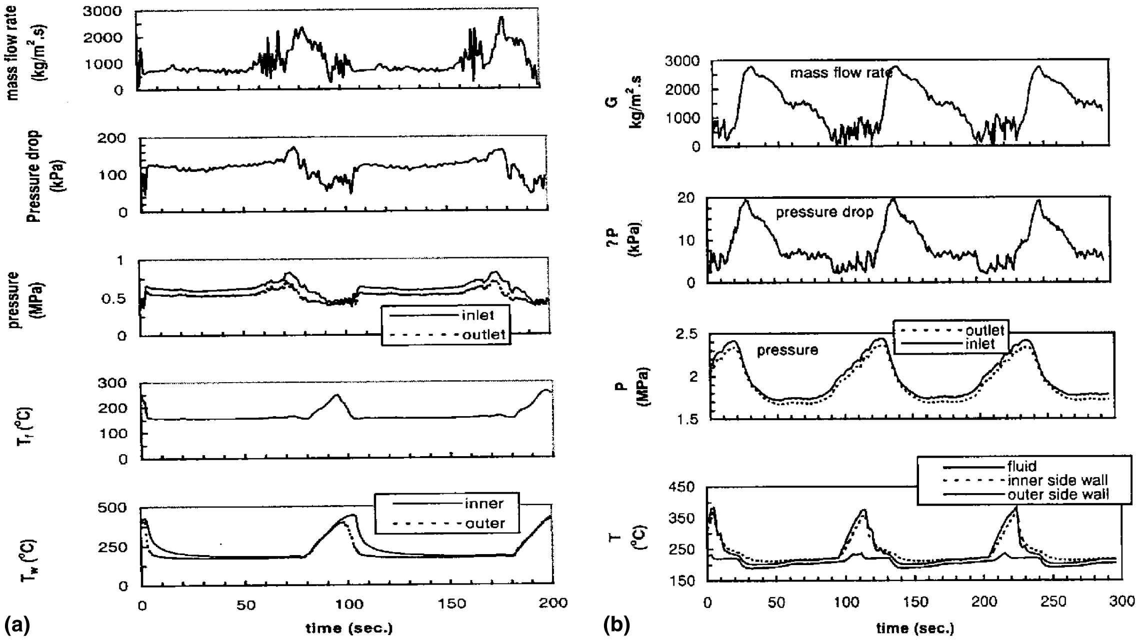
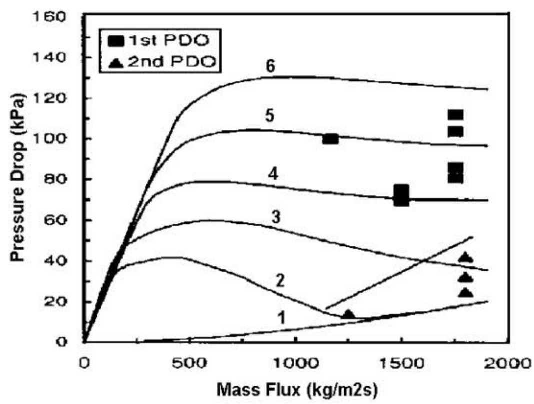


Fig. 3. Typical oscillation curves when the system compressible volume is oriented (a) in the front of pre-heater (1st PDO) and (b) at the inlet of test section (2nd PDO).

occurs only in those cases of having a surge tank in circulation loop. To consider the influence of compressible volume positions on the pressure drop oscillation characteristics, we installed two same-sized surge tanks at two different positions in the test circulation loop as shown in Fig. 1, at outlet of pump or at inlet of test section (similar to Stenning's test loop). We found that pressure drop oscillations obtained by these two surge tank positions are significantly different, not only of their occurrence boundaries, but also of oscillation amplitudes and time periods. Fig. 3 shows the typical oscillation curves obtained, respectively, in these two cases of compressible volume installation positions. Fig. 4 shows their unsteady boundaries on the map of pressure drop vs mass flux, from which pressure drop oscillation occurs in the case of high exit mass quality in test section with the first compressible volume installation position, while pressure drop oscillation occurs in the starting process of evaporation and even in subcooled boiling conditions of outlet cross-section with the second position. The amplitude and time period of pressure drop oscillations are quite different for surge tank positions.



(1), q_w=0, (2), q_w=100 kW/m^2, (3), q_w=200 kW/m^2, (4), q_w=300 kW/m^2, (5), q_w=400 kW/m^2, (6), q_w=500 kW/m^2

Initial boundaries of 1st PDO

Disappear boundaries of 2nd PDO

Fig. 4. Occurrence regions of two kinds of pressure drop oscillation on the hydrodynamic curve.

The tests were conducted using the following procedure: (1) establish operating parameters including the mass flux, compressible volume, and inlet temperature, and confirm their good stability; (2) increase the heat flux of 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 above procedure after adjusting the operating parameters.

The numerical solutions developed to calculate the temperature and heat transfer coefficient on inner surface of helically coiled tube by [20,21] were employed in data reduction. The bulk temperature of fluid is determined using thermal equilibrium and linear pressure gradient assumption along the flow direction. The results will be discussed in detail below.

3. Turbulent heat transfer for steady single phase water flow

3.1. Average heat transfer coefficient

As mentioned previously in [11], the average heat transfer coefficient in developed turbulent flow in helical tube can be calculated by the correlation of Seban and Mclaughlin [16] when $Re < 60{,}000$. When $Re > 60{,}000$, as shown in Fig. 5, our experimental data were found to be located between the predicted curve by correlation of Seban and Mclaughlin and correlation by Dittus–Boelter for straight tubes. A new correlation proposed for a wider range of Reynolds number as follows:

$$\overline{Nu}_c = \frac{\bar{h}_c d}{k} = 0.328 Re^{0.58} Pr^{0.4},$$
$$6000 < Re < 180{,}000, \qquad (1)$$

where the characteristic temperature is defined as the bulk temperature of the working fluid. A maximum derivation of $\pm 9.2\%$ was reached between the calculated value and experimental data.

It is seen from Fig. 5 that, with increasing Re to higher level, the average heat transfer coefficient for coils tend to coincide with the correlation for straight tubes. This was not reported in previous investigations. This trend implies that the effect of secondary flow in helical tubes on the enhancement of heat transfer becomes less significant due to thinning of the thermal boundary layer at much higher Reynolds number.

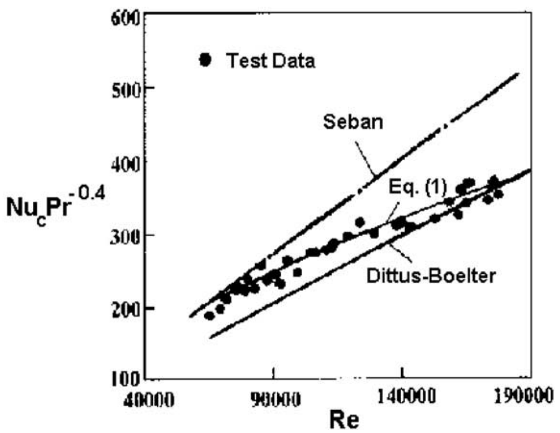


Fig. 5. Test results of the average turbulent heat transfer.

3.2. Longitudinal cross-sectional average heat transfer coefficient

As the coil is oriented horizontally, the angle between the gravity force and flowing direction changes continuously and periodically, the flow velocity distribution in radial direction on each cross-section varies also periodically along longitudinal axial direction of coils and the heat transfer rates do similarly. Fig. 6 illustrates the results of measurements for distribution of wall temperature along the helical axial direction. There exists a periodical feature and the lower value at the cross-section on which fluid flows upward in tube. The corresponding distribution of average heat transfer coefficient is shown in Fig. 7, which also has a periodic feature.

The average heat transfer coefficients in upward flow region of coils are higher than those in downward flow region and 120–130% of the average value of total coils. The reason may be that, the gravity force acts as a drag force for the upward flow and lead boundary layer near wall of tube thinner and cause the distribution of flow

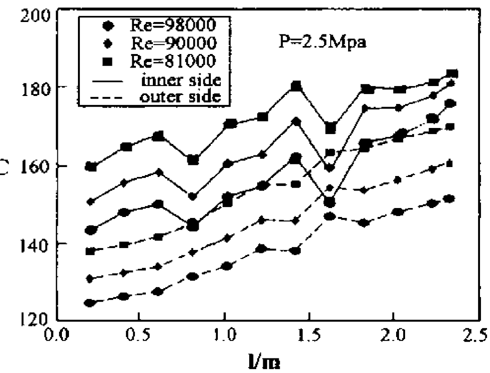


Fig. 6. Local wall temperature along tube axis.

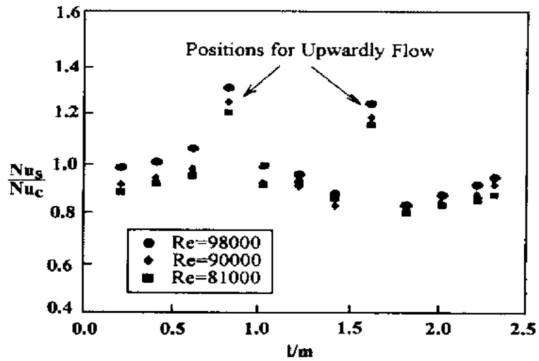


Fig. 7. Peripheral local turbulent heat transfer along the tube axis.

velocity in radial direction to be more uniform. Hence, it is unreasonable to use the cross-sectional average heat transfer coefficient at the outlet as the average value of total coils, as frequently used before.

3.3. Peripheral local heat transfer coefficients

The experimental data of the distribution of local heat transfer coefficient along the peripheral angle θ are shown in Fig. 8. The peripheral local heat transfer coefficient takes its largest value of about 1.6–2.2 times the cross-sectional average value, and takes its smallest about one half of the average value. With increasing Re and Pr, Nu_p/Nu_s becomes greater on the outside, but keep constant on the inner side. This was similar to that observed by Seban et al. [16], and can be correlated by

$$\frac{Nu_p}{Nu_s} = 0.22\left(\frac{RePr}{10^4}\right)^{0.45}(0.5 + 0.1\theta + 0.2\theta^2). \qquad (2)$$

This implies that the secondary flow pattern in coils is nearly symmetric circulation in a steady single-phase turbulent flow.

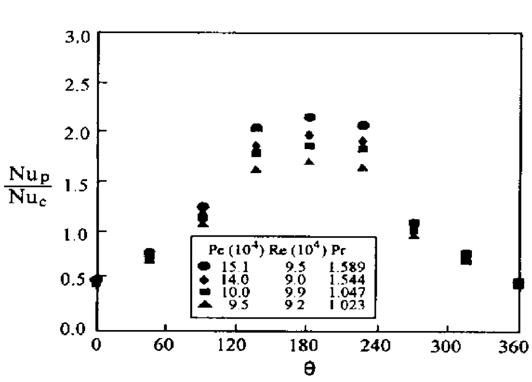


Fig. 8. Local turbulent heat transfer coefficient on the outlet cross-section.

4. Unsteady oscillatory heat transfer in single-phase water flow region

4.1. Time-averaged heat transfer coefficient

Stability prediction of the threshold by the analytical model requires more accurate understanding and knowledge of radial and axial heat conduction, diffusion effects in the single-phase region of the flow channel and thermal non-equilibrium effect in subcooled boiling. The first effect influences the axial propagation of enthalpy perturbation. There is promising, but somewhat limited, evidence that this results in a better analytical prediction of experimental system stability limits when the above effects are incorporated in the dynamic model of a boiling system [22–24]. No systematic study of these effects exists in the literature. The experimental work reported here is to study the characteristics of heat transfer with oscillatory turbulent subcooled flow corresponding to thermal hydrodynamic oscillations encountered in boiling channels.

Fig. 9 shows the experimental results of the time and cross-section averaged heat transfer coefficients for oscillatory single-phase turbulent flow. Fig. 9(a) compares the experimental data to some correlations such as Seban's [16] and Owhadi's [25] for coils and that for straight tube in steady flow. Figs. 9(b) and (c) show the variation characteristics of oscillatory heat transfer coefficient with the oscillatory frequency number $\overline{Wo}$ and the ratio of oscillatory amplitude $\bar{A}_p$, respectively.

The data of oscillatory heat transfer coefficients are totally higher than that of steady flow with corresponding conditions, obviously, oscillations enhance the single phase turbulent heat transfer, the enhancement effects of oscillations increase with increasing time-averaged Reynolds number Re, as shown in Fig. 10(a). Another important feature of oscillation heat transfer from Figs. 10(b) and (c) is that, there exists a strong dependence on oscillatory frequency and amplitude. We

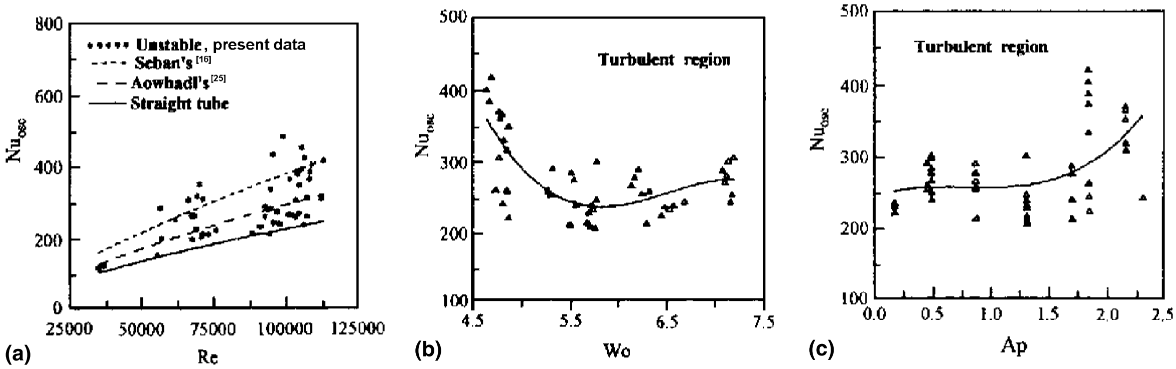


Fig. 9. Experimental results of averaged heat transfer coefficients for oscillatory single-phase turbulent flow.

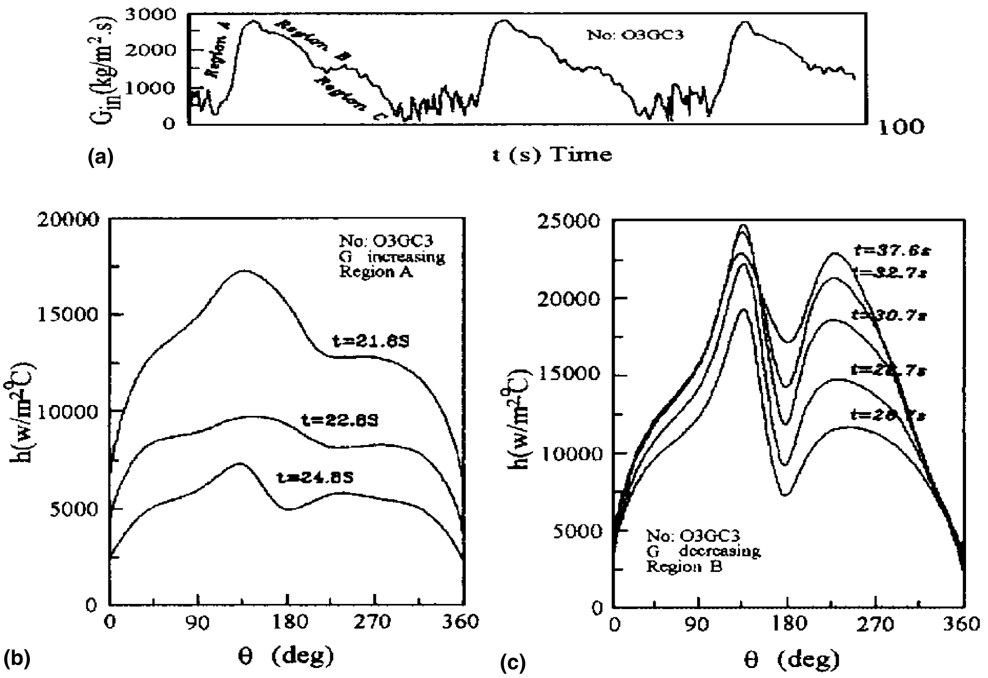


Fig. 10. Transient local heat transfer coefficient variation corresponding to the flow rate oscillation for single-phase turbulent flow.

introduce two non-dimensional parameters, the oscillatory frequency number, $\overline{Wo}$, and the ratio of oscillatory amplitude, $\bar{A}_{\mathrm{p}}$, defined, respectively, as follows:

$$\overline{Wo} = d\sqrt{2\pi f/\mu_1}, \tag{3}$$

$$\bar{A}_{\mathrm{p}} = -\frac{(G_{\max} - G_{\min})}{2\bar{G}_0}, \tag{4}$$

where $\bar{G}_0$ is the time-averaged mass flow rate; and $G_{\max}$ and $G_{\min}$ are the maximum and minimum value of mass flux in a oscillatory period, respectively; f is the frequency of oscillation. We find From Figs. 1(b) and (c) that, $\overline{Nu}_{\mathrm{osc}}$ decreases sharply with increasing $\overline{Wo}$ for $\overline{Wo} \leqslant 5.5$, and then increase very little for $\overline{Wo} > 5.5$. This means that the oscillation in a low frequency is more beneficial for heat transfer enhancement than that in high frequency.

The influence of $\bar{A}_{\mathrm{p}}$ can also be divided into two regions: $\overline{Nu}_{\mathrm{osc}}$ is nearly a constant when $\bar{A}_{\mathrm{p}} < 1.5$; $\overline{Nu}_{\mathrm{osc}}$ increases gradually with increasing $\bar{A}_{\mathrm{p}}$ when $\bar{A}_{\mathrm{p}} \geqslant 1.5$, indicating that the larger amplitude of oscillation will enhances heat transfer.

Based on the experimental data, the time-averaged heat transfer coefficient of helical coils with oscillatory single-phase turbulent flow can be correlated in following form:

$$\overline{Nu}_{\mathrm{osc}} = 0.147\overline{Wo}^{-0.31}Pr^{-4.4}\left(\frac{\overline{Dn}}{1000}\right)^{0.82}, \tag{5}$$

where $\overline{Dn}$ is the time averaged Dean number, defined as $Re(d/D)^{0.5}$. Eq. (5) is applicable for the range of oscillation $f = 0.05$–0.003, $\overline{Re} = 25{,}000$–$125{,}000$, and a reasonable agreement is reached within 15% derivation between the calculated value and experimental data.

4.2. Transient local heat transfer coefficients and their phase lag

Fig. 10 shows the flow rate oscillation and corresponding variations of transient local heat transfer coefficient in a time period. Here, $t = 21.8$–24.8 s corresponds to the increasing flow rate part of period, and $t = 26.7$–37.6 s is the decreasing flow rate part of period. The local heat transfer coefficient h_{osc} has a reversal phase angle characteristics relative to the flow rate oscillation, that is, h_{osc} decreased with increasing flow rate G, h_{osc} is the minimum when G is the maximum, and vice versa.

The distribution of transient local heat transfer coefficients in the peripheral direction θ at the outlet cross-section of coil becomes non-uniform and asymmetric; It is minimum in value at $\theta = 0°$ (see Fig. 1) and maximum approximately at $\theta = 135°$ (see in Fig. 1). With increasing flow rate, the h_{osc} tends to be uniform in the one cross-section; there exists one peak during the process of flow rate increasing, but two peaks during the process of flow rate decreasing as shown in Figs. 10(b) and (c). Hence, the actions and flow patterns of secondary flow in helical coils with oscillatory flow are obviously different from that with steady flow. The secondary flow becomes more complicated. There may exist more than two of circulation in a cross-section and even occurs reverse secondary flow-pattern. The interaction between the flow oscillations and the secondary flow is a very interesting and complicated problem to be further studied.

4.3. Transient local and time-averaged heat transfer coefficient for unsteady oscillatory steam–water two-phase flow under PDO processes

The experimental results demonstrated that the transient oscillatory heat transfer characteristics of various regions during a time period of pressure drop oscillation are quite different. The time-averaged heat transfer coefficient under pressure drop oscillation is much lower than that of stable conditions.

Fig. 11 shows the variation of local heat transfer coefficients in three different regions with time at outlet cross-section during the first PDO process. The time average heat transfer coefficients under first pressure drop oscillation were correlated as

$$\frac{h_{\mathrm{osc}}}{\bar{h}_{\mathrm{slo}}} = 21.4\frac{(1000\overline{Bo})^{0.748}\bar{N}_{\mathrm{d}}^{0.125}}{\overline{Wo}^{0.97}\bar{A}_{\mathrm{p}}^{1.4}\bar{x}^{0.42}}, \tag{6}$$

where

$$\bar{h}_{\mathrm{slo}} = 0.023\frac{\lambda}{d}\overline{Re}^{0.8}\overline{Pr}_{\mathrm{l}}^{0.4}\left[\overline{Re}_{\mathrm{l}}^{0.05}\left(\frac{d}{D}\right)^{0.1}\right]. \tag{7}$$

Fig. 12 shows the variation of local heat transfer coefficients in three different regions with time at outlet section during the second PDO process. A correlation can be proposed for the time-averaged heat transfer coefficients under second pressure drop oscillation as

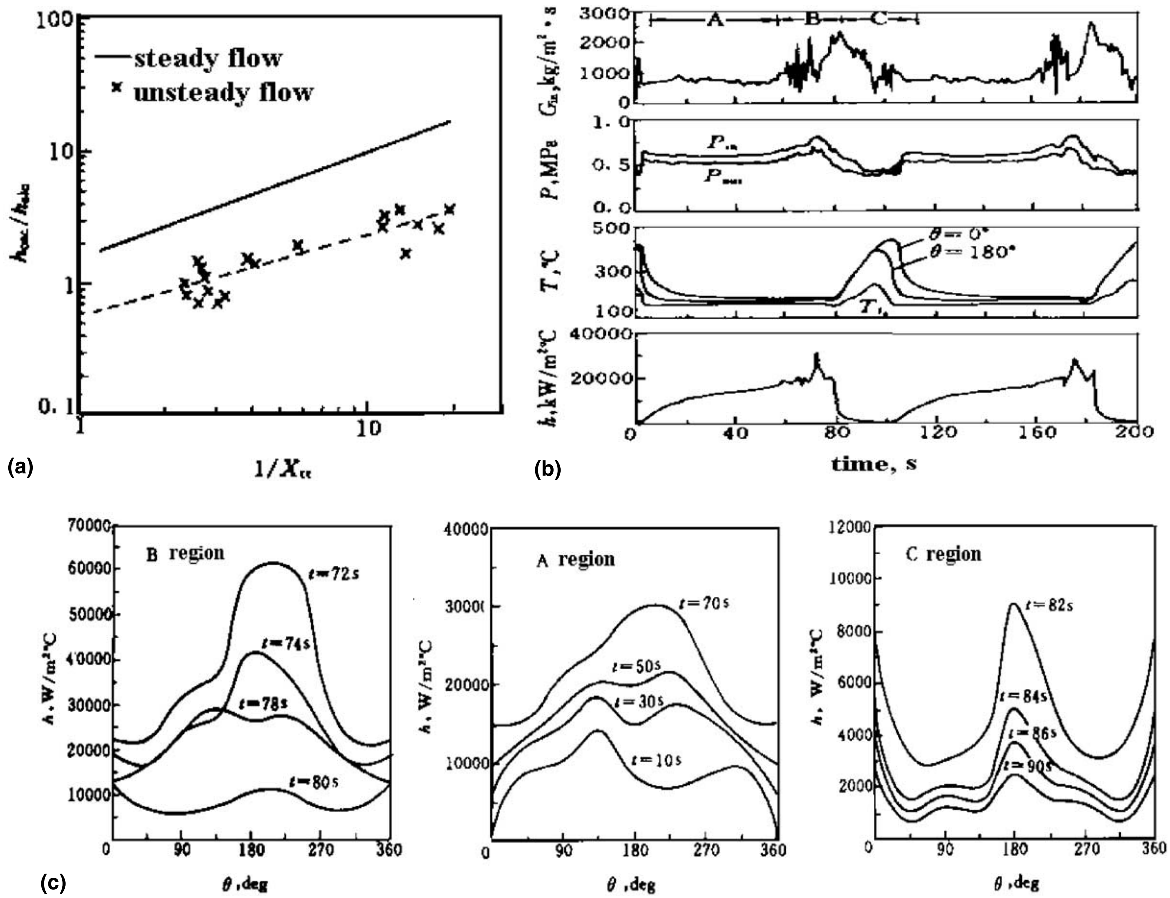


Fig. 11. The transient characteristics of local heat transfer and main parameter during first PDO process. (a) Comparison of the averaged heat transfer coefficient for unsteady flow to that for steady flow. (b) Variations of main parameters with time t. (c) Local heat transfer coefficient distrubution along peripheral angle θ.

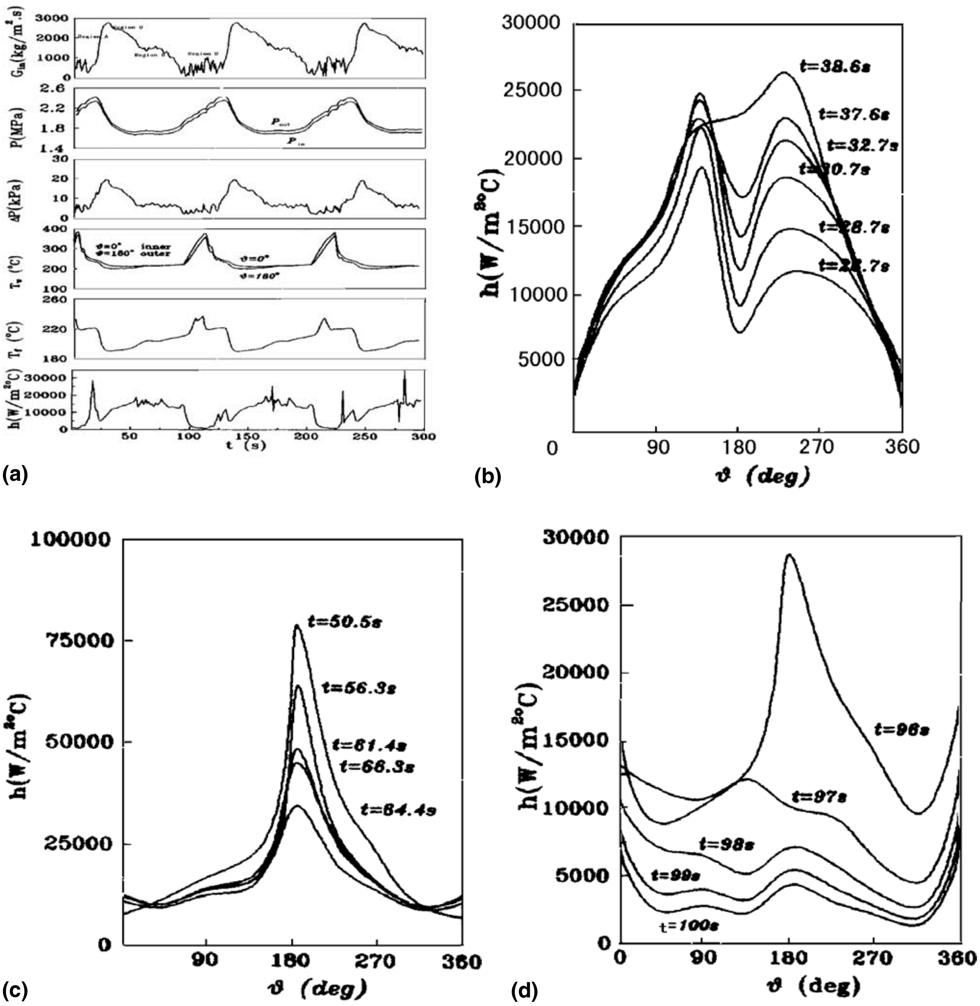


Fig. 12. The transient characteristics of local heat transfer and main parameter during second PDO process. (a) Variations of main parameters with time t. (b) Local heat transfer coefficient distrubution along peripheral angle *theta*. (c,d) Local transfer coefficient distribution along peripheral angle θ.

$$\frac{\bar{h}_{\mathrm{osc}}}{\bar{h}_{\mathrm{slo}}} = 0.051\overline{Wo}^{0.16} \cdot \bar{Bo}^{0.075} \cdot \bar{Dn}^{0.37}. \tag{8}$$

This equation fits 95% of data within $\pm 15\%$ deviation, with possible range of parameters: oscillatory frequency, $f = 0.05$–0.03 Hz; system pressure, $P = 0.4$–3.5 MPa; mass flux, $G = 200$–2200 kg/m^2 s.

5. Conclusions

Pressure drop oscillations in a closed-circulation helical coiled tubing steam generator were studied with water as working fluid. Experimental work was performed to investigate the oscillatory heat transfer characteristics of single-phase and two-phase flow and their phase lag relative to the phase feature of flow rate oscillations in the range corresponding to the PDO thermal-hydrodynamic oscillations encountered in a helical-tube boiling channel.

The cross-sectional average heat transfer at much higher Reynolds numbers and the non-uniform effects of local heat transfer along the longitudinal and peripheral directions were experimentally investigated in steady flow simultaneously. The correlation for the calculations of average and local heat transfer coefficients was proposed as Eqs. (1) and (2), respectively.

The phenomena observed and analyzed indicate that, in contrast to that of turbulent disturbances, the effects of secondary flow in curved tubes on heat transfer enhancement weaken in the range of much higher Reynolds numbers.

The time-averaged heat transfer coefficients of single-phase turbulent flow and two-phase flow under unsteady or oscillatory PDO conditions can be correlated by Eqs. (5)–(7), respectively. The main factors affecting oscillatory heat transfer include Reynolds number Re, Prandtl number Pr, oscillatory frequency and amplitude, analyzed by introducing two new non-dimensional parameters, i.e. W_o and A_p defined by Eqs. (3) and (4), respectively.

The transient local heat transfer coefficients oscillate with a reverse phase characteristics in contrast to the flow rate oscillations, and show an asymmetrical and non-uniform feature.

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