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The effect of internal surface modification on flow instabilities in forced convection boiling in a horizontal tube

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Abstract

This paper presents the experimental result of a study on the effects of heat transfer enhancement on two-phase flow instabilities in a horizontal in-tube flow boiling system. Five different heat transfer surface configurations and five different inlet temperatures are used to observe the effect of heat transfer enhancement and inlet subcooling. All experiments are carried out at constant heat input, system pressure and exit restriction. Dynamic instabilities, namely pressure-drop type, density-wave type and thermal oscillations are found to occur for all the investigated temperatures and enhancement configurations, and the boundaries for the appearance of these oscillations are found. The effect of the enhancement configurations on the characteristics of the boiling flow dynamic instabilities is studied in detail. The comparison between the bare tube and the enhanced tube configurations are made on the basis of boiling flow instabilities. Differences among the enhanced configurations are also determined to observe which of them is the most stable and unstable one. The amplitudes and periods of pressure-drop type oscillations and density-wave type oscillations for tubes with enhanced surfaces are found to be higher than those of the bare tube. The bare tube is found to be the most stable configuration, while tube with internal springs having bigger pitch is found to be the most unstable one among the tested tubes. It is found that system stability increases with decreasing equivalent diameter for the same type heater tube configurations; however, on the basis of effective diameter there is no single result such as stability increase/decrease with increasing/decreasing effective diameter.

Keywords: Two phase; Boiling flow; Instabilities; Dynamic instabilities; Pressure-drop type instabilities; Density-wave type instabilities; Heat transfer enhancement; Enhanced surfaces; Stability boundary

1. Introduction

There are a number of techniques to augment or enhance heat transfer. These techniques may be conveniently divided into three classes: passive, active and compound techniques. Passive techniques require no direct application of external power, whereas active techniques need an external activator/power supply to bring about the enhancement. Compound techniques involve a combination of techniques, namely two or more of the active or passive techniques may be utilized simultaneously to produce an enhancement that is bigger than the techniques operating separately. Table 1 summarizes active and passive enhancement techniques. Some of active or passive techniques are widely used for enhancing heat transfer in two-phase flow, whereas some of them are somewhat significant, not significant or not to believed to be relevant for two-phase flows (Bergles and Webb, 1985; Reay, 1991; Ohadi et al., 1996).

Considerable efforts have been made to augment heat transfer in two-phase flows by various enhancement techniques although most of the extensive research in heat transfer enhancement has been devoted to single-phase heat transfer. Enhancement techniques to augment heat transfer in boiling two-phase flows inside tubes can be broadly classified into two types, depending on whether the nucleate boiling mechanism or the convective mechanism is augmented. Some techniques such as specially designed structured and porous surfaces and the application of electrohydrodynamic fields enhance

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Table 1
Summary of passive and active enhancement techniques (Bergles and Webb, 1985; Reay, 1991; Ohadi et al., 1996)

Passive techniques	Single phase	Two phase	Active techniques	Single phase	Two phase
Treated surfaces	C	A	Mechanical aids	B	C
Rough surfaces	A	A	Surface vibration	B	B
Extended surfaces	A	B	Fluid vibration	C	B
Displaced enhancement devices	A	A	Electrostatic/electrohydrodynamic	B	A
Swirl flow devices	A	A	Suction /injection	C	N/A
Surface tension devices	N/A	A	Jet impingement	B	N/A
Additives	B	B	Rotation	C	A
			Induced flow	B	N/A

A: Most significant; B: significant; C: somewhat significant; D: not significant; N/A: not believed to be relevant.

the nucleate boiling component, while others such as internal fins, twisted tapes, helical wire inserts, microfins, and corrugations are mainly employed to enhance the convective component (Kandlikar et al., 1999). In this study in addition to smooth tube four enhanced tube configurations are used, which are effective on the enhancement of convective component.

Several types of instabilities induced by boiling two-phase flows are of interest for the design and operation of many industrial systems such as rocket nozzles, electron tubes, nuclear reactors, boilers, heat exchangers, steam generators, chemical reactors, distillation plants etc. Many types of instabilities have been observed in two-phase systems and these instabilities have been generally classified into two groups: static instabilities and dynamic instabilities. When the flow conditions change a small step from the original steady state and another steady state is not possible in the vicinity of the original state, then the flow is subjected to static instabilities. Static instabilities can lead either to a different steady-state condition or to a periodic behavior. Among the static instabilities are flow excursion, boiling crisis, bumping, geysering or chugging etc. In dynamic instabilities, the inertia and the other feedback effects have an essential part on the process and the system behaves like a servomechanism. Four types of dynamic instabilities have been observed and defined in the literature. These are "pressure-drop type oscillations", "density-wave type oscillations", "thermal oscillations", and "acoustic oscillations". Boiling flow dynamic instabilities are undesirable as they may cause forced mechanical vibration of components or system control problems. Instabilities can also affect the local heat transfer characteristics, possibly resulting in oscillatory wall temperatures, or may induce boiling crisis (critical heat flux, DNB, burnout, dryout) (Bergles, 1976). Knowing the characteristics of the oscillations and boundaries, important conclusions can be drawn for the design and safe operation of two-phase systems.

A great deal of efforts to enhance heat transfer in boiling two-phase flows has been devoted to developing apparatus and performing experiments to define the conditions under which an enhancement technique will improve heat transfer and pressure-drop characteristics for boiling two-phase flow. The heat transfer and friction factor become higher in two-phase and two-phase systems with augmentation. Although the heat transfer coefficient is higher in boiling two phase and boiling two phase with augmentation, it is much important to consider dynamic instabilities, because boiling two-phase flow systems are very sensitive to boiling instabilities. Therefore, it is necessary to determine the influence of the several system parameters such as geometry, pressure, flow rate, inlet temperature, inlet and exit restriction, and augmentation techniques etc. on the boiling two-phase flow system behavior. The main aim is to design a system with higher heat transfer coefficient, lower friction factor, and stable operation characteristics. For these reasons, the knowledge of the dynamic behavior of thermal systems which operate in forced convection flow boiling is important for the prediction and understanding of their local and global stability.

Several investigations have been carried out to investigate boiling two-phase flow instabilities. Among these are publications by Gouse and Andrysiak (1963), Cumo et al. (1981), Aritomi et al. (1983), Doğan et al. (1983), Bar-Cohen et al. (1987), Wang et al. (1989), Kakaç et al. (1990), Padki et al. (1991) and Ding et al. (1993). However, investigations dealing with the effect of heat transfer enhancement on boiling two-phase flow instabilities have been scant in the literature.

Lin et al. (1982) investigated experimentally boiling heat transfer in oscillating two-phase flows and effect of heater surface conditions in vertical tubes with six different surface conditions. They used Freon-11 as working fluid, obtained high-speed photographic observations and correlated the results for predicting the heat transfer coefficient.

Mentes et al. (1983a,b) carried out an experimental work to study the effect of different heater surface configurations on two-phase flow instabilities in a single channel, forced convection, up-flow system. They used Freon-11 as the working fluid and tested six different heater tubes with various inside surface configurations. They showed experimental results on system pressure-drop versus mass flow rate curves, and indicated stability boundaries on these curves. They made comparison of different heater tubes by the use of stability boundary maps and the plots of inlet throttling necessary to stabilize the system versus mass flow rate. They found that tubes with internal springs were to be more stable than the other tubes.

Mentes et al. (1983a,b) performed experimental and theoretical study to investigate the effects of heat transfer augmentation on two-phase flow instabilities in a single channel system with six different heater surfaces. The dependence of two-phase flow instabilities on the mass flow rate, heat input and inlet subcooling were studied. They evaluated experimental data using the steady-state pressure-drop versus mass flow rate curves along with the curves of additional inlet pressure drop required to stabilize the system during the oscillations, and generated tables using the experimental data. In their theoretical analysis, they used homogeneous equilibrium flow model and finite differences. They linearized the dynamic equations of the overall system to found characteristic equation for the system and analyzed this characteristic equation to determine the oscillation thresholds.

Kakaç et al. (1995) studied experimentally two-phase flow instabilities with augmented surfaces in a horizontal in-tube boiling system. In their experiments they used three different surface configurations. One of the heater tubes used had smooth surface and the other two had internal springs with different wire diameter and pitches. They plotted steady-state characteristic curves and studied the effect of surface augmentation on the steady-state characteristics, stability boundaries, dynamic instabilities such as pressure-drop type, density-wave type and thermal oscillations. The results of experimental and theoretical studies of pressure-drop type oscillations in a horizontal single channel flow by the use of the homogeneous model were given.

Widmann et al. (1995) studied experimentally the effect of augmented surfaces on two-phase flow instabilities in a horizontal system. They used three tubes made of different inner surfaces: the first tube was smooth tube, and the augmented surfaces were created by inserting springs with different pitches in the tube. Springs with pitches of 5 mm for second tube and pitches of 15 mm for third tube were used. The diameter of internal springs was 1.0 mm for both augmented surfaces. All experiments were conducted at constant heat input, system pressure, and exit restriction. Steady-state characteristics of the system were plotted in pressure-drop versus mass flow rate diagrams for each configuration. Pressure-drop type, thermal and density-wave type oscillations were found to occur and boundaries for the appearance of these oscillations were determined. Comparisons were made between the bare tube and the augmented tubes on the basis of characteristics of pressure-drop, density-wave and thermal oscillations.

Widmann et al. (1994) presented the result of an experimental investigation on the effect of inlet subcooling on two-phase flow instabilities in a horizontal in-tube flow boiling system. Three different surface configurations were used. These configurations were the same configurations mentioned in the above paragraph. However, in this study the effect of inlet subcooling on pressure-drop type, density-wave type and thermal oscillations were emphasized.

Kakaç and Cao (1999) presented the result of experimental study on the effect of different heater surface configurations on two-phase flow instabilities in single channel, forced convection, up-flow and horizontal system. They used Freon-11 as working fluid, and six different heater surface configurations for vertical system, and three different surface configurations for horizontal system. They studied the dependence of the characteristics of dynamic instabilities such as pressure-drop type, density-wave type instabilities and thermal oscillations on the augmented surfaces and made comparisons between the bare tube and the augmented tubes.

In this study, five different tube configurations were used to investigate the effect of heat transfer enhancement on boiling flow instabilities. The following five heat transfer surface configurations were used (Table 2): tube-1: smooth tube; tube-2 and tube-3: tube with internal springs; tube-4: tube with equilateral square ring; tube-5: tube with equilateral triangle ring. Heater tube surface configurations used in two-phase flow instabilities by several investigator are shown in Table 2 for comparison purposes. Lin et al. (1982) and Mentes et al. (1983a,b) used six different configurations in single vertical channel. Three different surface configurations were used by Widmann et al. (1994, 1995) and Kakaç et al. (1995) and experiments were conducted in a horizontal system. Although the second and third configurations were obtained by inserting internal springs in the horizontal tube, different wire diameter and pitch were selected in the present study. Especially the fourth and fifth configurations had mainly different configurations, which had equilateral square ring and equilateral triangle ring, respectively. Another distinguishing characteristic of the present experimental setup was that the length of heater tube was 3.50 m, which was longer than that of other studies with augmented surfaces (Table 2). The purpose of this study was to investigate the effect of these different augmented surface configurations on the two-phase flow dynamic instabilities and to compare these configurations to find out which configurations are advantageous in terms of stable operational characteristics.

Table 2
Studies of heat transfer enhancement on two-phase flow instabilities

Investigator	Configuration	Length of test tube (cm)	Enhanced surfaces	Wire diameter (mm)	Wire pitch (mm)	Equivalent diameter (d_e)
Lin et al. (1982) and Mentes et al. (1983a,b)	Single tube Vertical	60.5	1. Bare tube		–	7.493
			2. Threaded, 7.938 mm—16 threads per 25.4 mm	–	–	7.619
			3. Internal spring	0.794	19.05	7.446
			4. Internal spring	0.432	3.175	7.401
			5. Internal spring	1.191	6.350	7.192
			6. Coated with: Union Carbide Linde High Heat Flux Coating	–	–	7.073
Widmann et al. (1994, 1995) and Kakaç et al. (1995)	Single tube Horizontal	106	1. Bare tube	–	–	10.90
			2. Internal spring	1.0	5	10.50
			3. Internal spring	1.0	15	10.77
Present study	Single tube Horizontal	350	1. Bare tube	–	–	13.00
			2. Internal spring	1.800	3.600	9.56
			3. Internal spring	1.800	11.00	12.53
			4. Equilateral square ring	1.800	11.00	12.56
			5. Equilateral triangle ring	1.800	11.00	12.66

2. Experimental system and procedure

2.1. Experimental system

The experimental system used in the present research is shown schematically in Fig. 1. The setup is designed so that it is possible to generate three main types of two-phase flow oscillations: pressure-drop type, density-wave type and thermal oscillations. The system is mainly composed of three sections: fluid supply section, test section and fluid recovery section. The single-phase liquid from the fluid supply section enters the test section, turns into a two-phase mixture, as a result of heating, and, at the outlet, nearly all of the working fluid is in vapor phase. After leaving the test section, the working fluid enters the fluid recovery section, thus closing the loop. The working fluid used is Refrigerant-11.

The fluid supply section consists of a main tank (1), a flow control valve (2), a rotameter (3), and a subcooler (4). In the main tank (1), the working fluid is pressurized with nitrogen gas. The main tank is cylindrical vertical tank and manufactured to be able to withstand 30 bar operational pressure and made of stainless steel. A manometer displaying the pressure of the main tank is installed on the main tank. The flow rate of the working fluid is measured by a rotameter (3). The inlet temperature of the working fluid has the important effect on the two-phase flow characteristics and the subcooler (4) is used the working fluid to enter into the test tube at desired temperature. The subcooler is mainly a shell-and-tube heat exchanger, at which the working fluid passes through the tube side, while the cooling water passes through the shell side. Using this subcooler, the inlet temperature of the working fluid is set to the desired value. The temperature of the working fluid is measured with a thermocouple at the inlet of the test tube.

The test section where the oscillations are generated under controlled test conditions contains a surge tank (5), an inlet fluid control valve (6), a test plenum (7), a test tube (8), a DC power supply (9), a sight glass (10), and an exit restriction (11). The surge tank which is an important dynamic component of the system provides the compressible volume required for studying two-phase flow characteristics, and has a volume of 0.05 m^3. In order to observe the change of level of the volume of the working fluid and the level of the compressible volume in the surge tank, a transparent glass tube is installed at the side of the surge tank. The surge tank pressure is measured with a pressure gauge. During the two-phase flow oscillation experiments, the surge tank is partially filled with nitrogen gas and provides the necessary compressibility to the system and acts as a capacitance. A differential pressure transducer, a Bourdon type digital manometer, and a pressure transducer are used to measure the oscillations of the inlet mass flux of the working fluid, the inlet pressure of the working fluid, and the oscillations of the pressure of the working fluid, respectively. The control valve controls the flow rate into the system.

A stainless steel pipe of 0.018 m outer diameter, 3.4 mm wall thickness, and 3.5 m length is heated by a uniform electrical heat flux from a DC generator (22 kW power supply). To measure the oscillations of the wall temperature, 28 copper–constantan thermocouples are fixed on the outer surface of the test tube. Half of these thermocouples, i.e. 14 thermocouples are fixed along the top of the test tube, the other half of the thermocouples

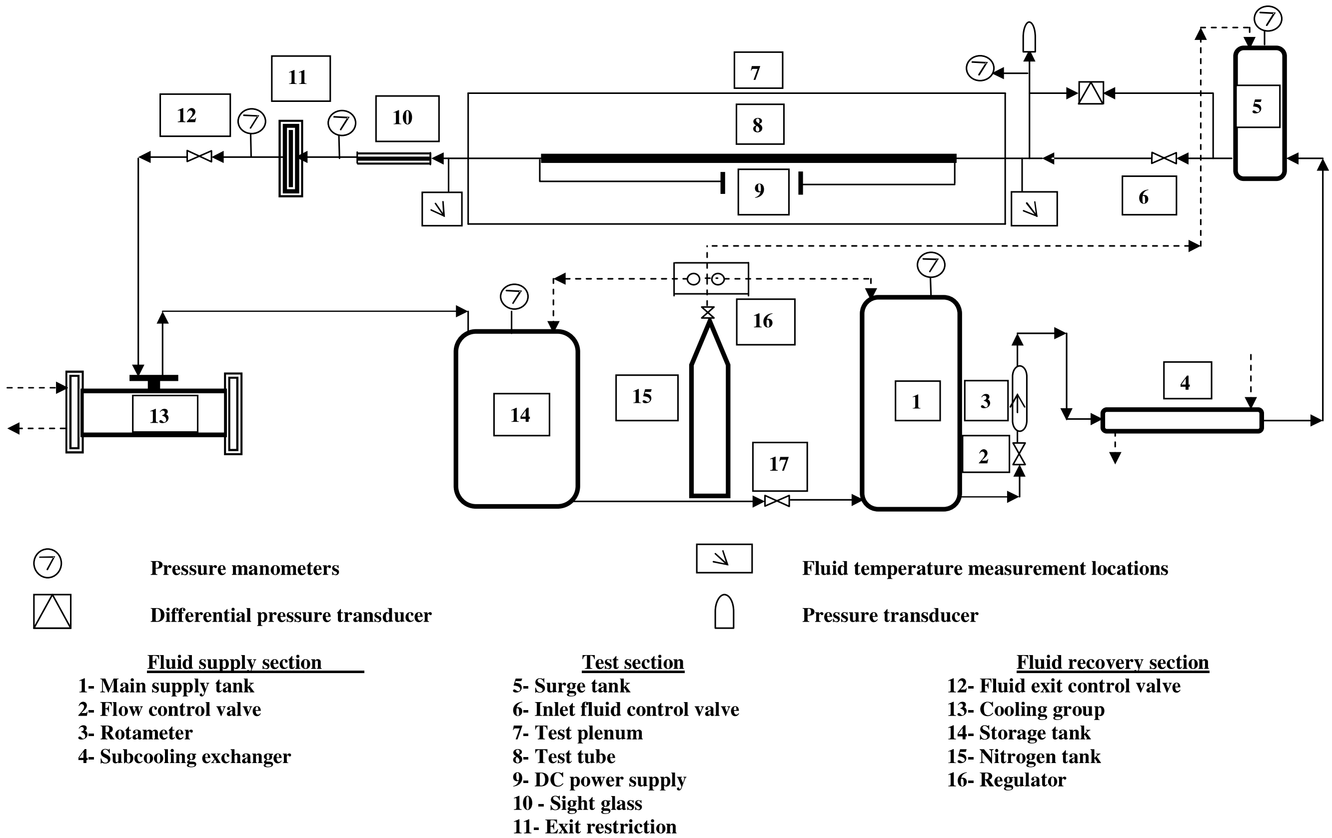

Fig. 1. Schematic diagram of experimental system.

are fixed along the bottom of the test tube. The fluid outlet bulk temperature is measured placing another thermocouple at the midstream inside the tube immediately after the test section. The exit restriction (11) creates the necessary pressure drop. In this system, an orifice plate is used as exit restriction and the diameter ratio of the exit restriction which is equal to 0.448 is used. Diameter ratio is defined as the ratio of the inner diameter of the orifice plate to the inner diameter of the tube ($\beta = d/d_i$). Downstream of the exit restriction, the exit pressure is measured with a pressure gauge. The test tube is uniformly electrically heated and insulated with glass wool insulation which can be withstand 1000 °C temperature. The heat input is determined by measuring the current and the voltage drop across the heated section. To avoid floating voltage effects, the thermocouple bead is insulated from the electrically heated tube wall surface by a dab of electrically non-conductive paste. The test tube is followed by a sight glass for visual inspection of the flow.

Five different tube configurations used in these experiments are shown in Fig. 2. The description of the configuration of the tubes is shown in Table 2. As seen, tubes are classified according to the their effective diameter and effective diameter is defined as (Kakaç, 1994):

$$D_e = (4U/\pi L)^{1/2}$$

U is the net inside volume and L is the length of the test tube.

Downstream of the test section the working fluid is sent to a recovery section which includes a fluid exit control valve (12), a cooling group (13), a storage tank (14), a nitrogen tank (15) and a regulator (16). Nearly all of the working fluid leaving the test section is in vapor phase and is condensed in the cooling group. The condensed fluid is sent to and collected in a storage tank (14) in liquid phase. Sending the working fluid in liquid phase from the storage tank into the main tank is achieved using high pressure nitrogen gas.

2.2. Experimental procedure

In the present investigation, two kinds of experiments were conducted, one to find out the steady-state characteristic of the system under each working condition, the other one to investigate the sustained oscillations of boiling two-phase flow. Steady-state characteristics of the system are 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 parameters:

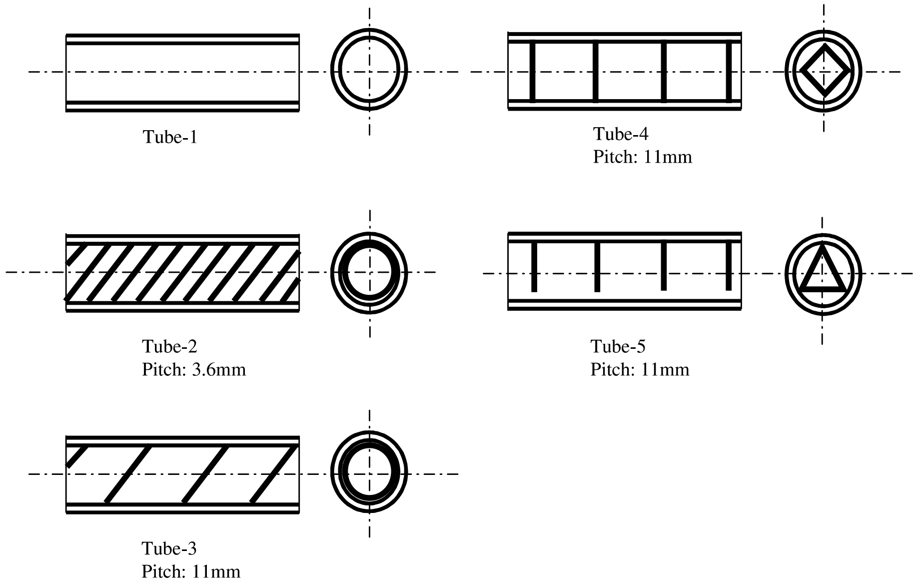

Fig. 2. Longitudinal cross-section of the test tubes used in the experiments.

System pressure: 7.5 bar
Mass flow rate: 9–75 g/s
Inlet subcooling: 16, 19, 22, 24 and 28 °C
Exit restriction diameter ratio: 0.448
Heat input: 16 kW

The tests are conducted using following procedure:

(1) The storage tank is filled with the working fluid, and the working fluid is transferred to the main supply tank.
(2) The main tank is pressurized to 7.5 bar using nitrogen gas by adjusting the pressure regulation valve.
(3) The inlet control valve is then opened, allowing the test liquid to fill the test loop.
(4) The subcooler is turned on and set at a desired temperature.
(5) The flow rate is then increased gradually to the desired starting point by slowly adjusting the inlet control valve.
(6) The rectifier is opened and the heat input is increased gradually to 16 kW.
(7) The system is allowed to become steady. Steady-state conditions are determined by observing the system pressure, the temperatures, and the flow rate.

In order to determine the steady-state characteristics, the experiments are started with a high mass flow rate, so that the system works in the single liquid phase region. After following the above seven items, the mass flow rate is measured by the rotameter. The pressure drop between the surge tank and exit pressures is measured by pressure gauges. Then the mass flow rate is decreased in certain steps to reach the two-phase region and finally the single vapor region (7.5 g/s for higher mass flow rate range, 2.5 g/s for lower mass flow rate range, as seen from Figs. 4–8). Thus, it is possible to plot the steady-state characteristic curve in a pressure-drop versus mass flow rate diagram. This procedure is repeated for the other four inlet temperatures.

The most important condition to reach the instabilities is a compressible volume in the system. This compressible volume is provided with a surge tank mounted before the test section. The surge tank acts as a capacitance, and provides the compressible volume required for studying the sustained oscillations. Therefore, the surge tank is an important dynamic component of the boiling two-phase system. For the experiments to study different type of oscillations, the surge tank is partly filled with the working fluid and pressurized to 7.5 bar by nitrogen gas. The control valve is opened, and the system works in the single liquid region, where no oscillations are expected, as explained in the above items (1)–(7). In this region the system is stable. Then the mass flow rate is reduced by a small amount using the inlet control valve and the system behavior is observed. After reaching the unstable region, the mass flow rate is first increased into the stable region and then decreased very slowly to locate the instability boundary. When reaching the instability boundaries, oscillations of inlet pressure, top and bottom wall temperatures and mass flow rate are recorded. All those parameters are also observed on a computer, where all data is processed using a data acquisition card. Mass flow rate reduction is continued to successively 9.18 g/s, covering the whole unstable region.

2.3. Experimental measurements and uncertainties

All temperature measurements are made by copper–constantan thermocouples with 0.25 mm diameter. Twenty-eight copper–constantan thermocouples are fixed evenly on the outer surface of the test tube. Half of these thermocouples, i.e. 14 thermocouples are fixed along the top of the test tube, the other half of the thermocouples are fixed along the bottom of the test tube. Fluid temperatures are measured with same type of thermocouples by placing them in the midstream of the fluid channel. The inlet temperature of the working fluid is measured with a thermocouple at the inlet of the test tube. The fluid outlet bulk temperature is measured placing another thermocouple at the midstream inside the tube immediately after the test section. The fluid inlet temperature is controlled within ±0.5 K, over the experimental range. The uncertainty of the temperature measurement is ±0.5 K.

Flow rate is measured at several locations using several types of flow rate measurement devices as shown in Fig. 1. The inlet mass flow rate is measured by a rotameter placed between the main tank and surge tank. The rotameter can withstand maximum 150 psig pressure and maximum 180 °F temperature. During the experiments, mass flow rate is changed between 9 and 75 g/s. The rotameter has a full-scale accuracy of ±2.5% and repeatability of ±0.5% of reading value.

Pressure is measured at several locations using several types of pressure measurement devices as shown in Fig. 1. Pressure measurements are made by both pressure gauges and pressure transducers. A differential pressure transducer placed before the test tube is used to measure the oscillations of the inlet mass flux of the working fluid. The response time is 0.1 ms for the differential pressure transducer, a full-scale accuracy is ±1%, and the total uncertainty is ±0.05%. A Bourdon type digital manometer is used to measure the inlet pressure of the working fluid. Digital manometer has a full-scale accuracy of ±0.25%. The oscillations of the pressure of the working fluid at the inlet of the test tube is measured by a pressure transducer. The response time is 0.1 ms for the pressure transducer, a full-scale accuracy is 0.25%, and the total uncertainty is ±0.1%. The pressure gauges had an accuracy of ±0.1% full-scale, and the total uncertainty is ±0.1%. Downstream of the exit restriction, the exit pressure is measured with a pressure gauge. Confidence levels for all of the measurements are 95%.

The test tube is uniformly electrically heated and the heat input is determined by measuring the current and the voltage drop across the heated section. The heat input is calculated as the product of the current supplied by DC power supply and the voltage drop across the heater tube. The overall uncertainty for heat input is about ±2%.

3. Results and discussion

3.1. The effect of the heat transfer enhancement on the steady-state characteristics

Steady-state characteristics of two-phase flows are generally presented by plotting pressure-drop versus mass flow rate diagram. The pressure drop of the system forms ordinate of the diagram and mass flow rate forms abscissa of the diagram. Pressure drop from the surge tank to system exit is defined as the system pressure drop (Mentes et al., 1983a,b; Kakaç, 1994).

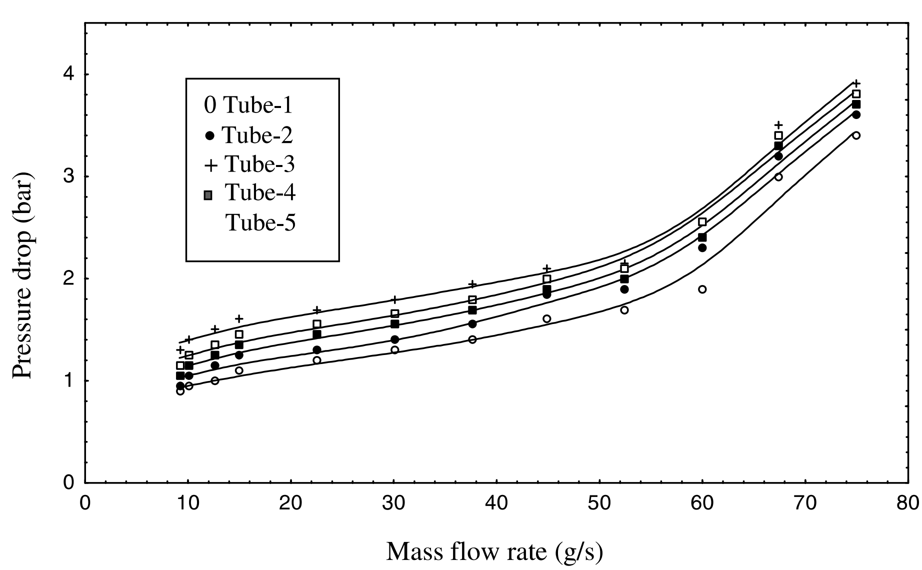

Fig. 3. Single-phase characteristic curves for different tubes.

Table 3
The comparison of enhanced surfaces

	Tube configurations				
	Tube-1	Tube-2	Tube-3	Tube-4	Tube-5
Equivalent diameter (d_e)[a]	****	*	**	***	****
Single phase pressure drop[b]	*	**	*****	***	****
Stability boundaries[c]	* (the most stable)	****	***** (the most unstable)	**	***
Period of pressure-drop type oscillations[d]	*	**	*****	***	****
Amplitude of pressure-drop type oscillations[d]	*	**	*****	***	****
Period of density-wave type oscillations[d]	*	**	*****	****	***
Amplitude of density-wave type oscillations[d]	*	**	*****	****	***

[a] Effective diameter increase as the number of "*" increase.
[b] Single-phase pressure drop increase as the number of " * " increase.
[c] Distance between the beginning and end of pressure-drop type oscillations increase as the number of " * " increase.
[d] Amplitude and period of the oscillations increase as the number of " * " increase.

Fig. 3 shows single-phase characteristic curves for five different heat transfer surface configurations. Tubes with enhanced surfaces create more pressure drop than the bare tube. Maximum pressure drop is obtained in the tube-3, whereas minimum pressure drop is obtained in the tube-1, i.e. in the bare tube. The reason of why enhanced surfaces have additional pressure drop is that heat transfer with augmentation is generally accompanied by an increase in pressure drop. Tubes are ordered from maximum pressure drop to minimum pressure drop as: tube-3; tube-5; tube-4; tube-2; and tube-1 (Table 3).

Steady-state characteristic curves of five different heater tubes are shown in Figs. 4–8. All curves show an inverted "S" shape for all investigated inlet temperatures and tube configurations. These curves generally do not intersect each other over the experimental range. At right hand side portion of the steady-state characteristic curve corresponding to higher mass flow rate the flow is single liquid phase. At this region the slope of the curve is positive. The first bubbles are observed and the two-phase region starts at the local minimum of this curve. From this local minimum point to local maximum point of characteristic curve, the slope of the curve is negative, and the pressure drop increases as the mass flow rate is decreased. Beyond the local maximum point of characteristic curve, the flow is at the single-phase vapor region and the pressure drop starts to decrease with decreasing mass flow rate. From these figures it may be seen that the pressure drop decrease with decreasing inlet temperature for all heater tube configurations.

In order to compare heater surface configurations on the basis of heat transfer enhancement steady-state characteristic curves for five different tubes with inlet temperature of 22 °C are presented in Fig. 9. Minimum pressure drop occurs in the bare tube, namely the pressure-drop in the tubes with enhanced surfaces is

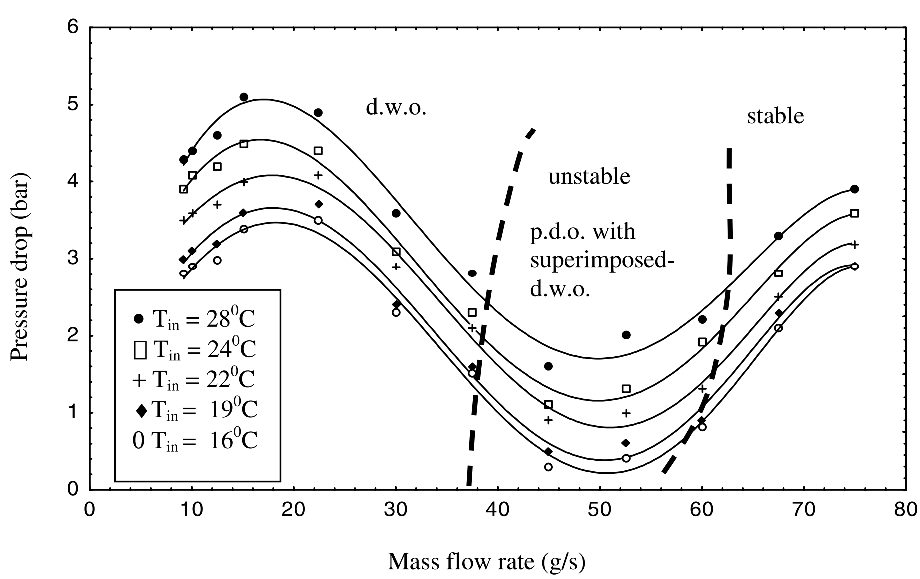

Fig. 4. Steady-state characteristic curves, tube-1.

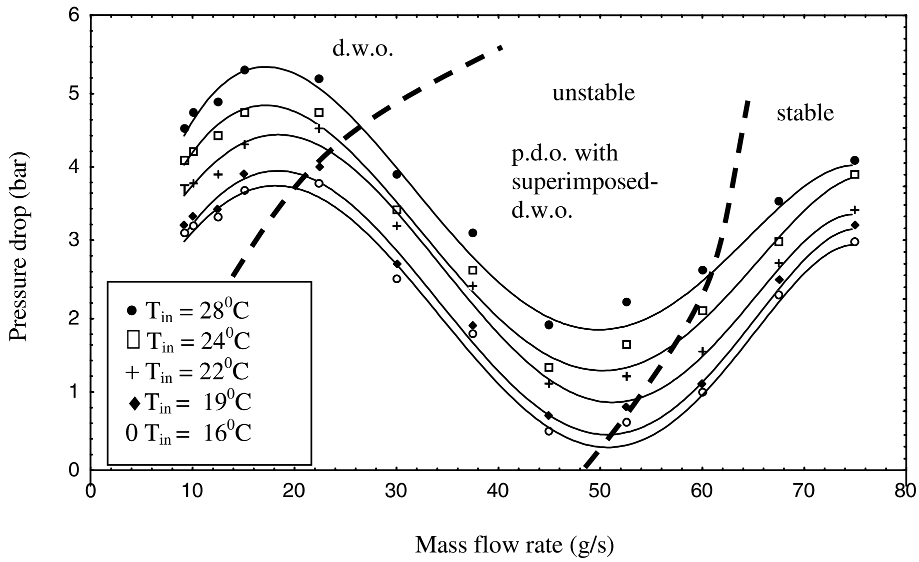

Fig. 5. Steady-state characteristic curves, tube-2.

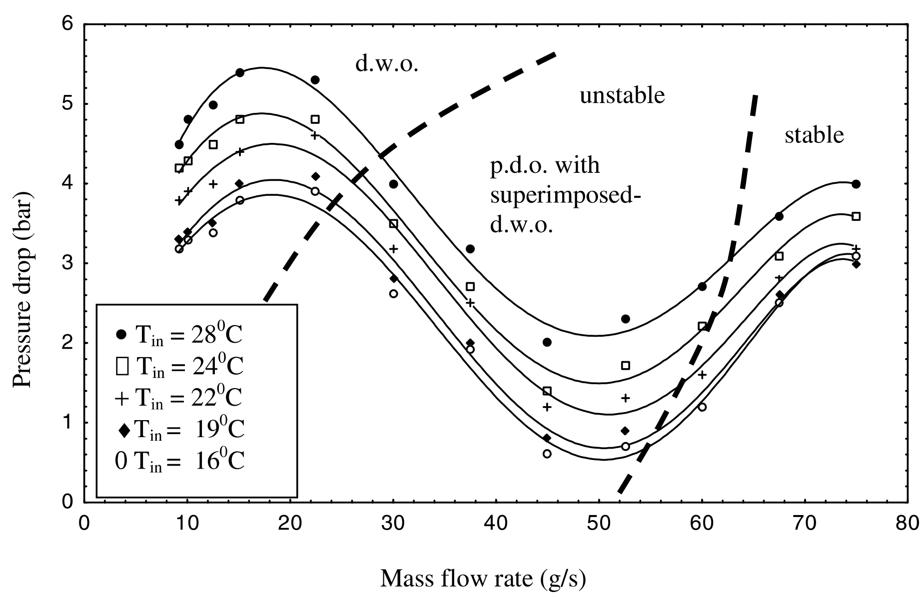

Fig. 6. Steady-state characteristic curves, tube-3.

higher than the bare tube. Pressure drop for tubes with enhanced surfaces is higher in comparison with the bare tube especially at lower mass flow rate. Stratification may be the reason of this situation, because frictional pressure drop is accompanied by additional pressure drop caused by higher vapor density if stratification occurs. Widmann et al. (1994, 1995) found similar results; namely it was found that stratification occurred at lower mass flow rates especially near the local maximum point of the steady-state characteristic curves for horizontal tubes with augmented surfaces and the pressure drop was found to be relatively higher for tubes with augmented surfaces in comparison with the bare tube.

3.2. The effect of the heat transfer enhancement on the stability boundaries

Stenning and Veziroğlu (1965) first identified three types of dynamic instabilities: pressure-drop type, density-wave type and thermal oscillations. Among them, pressure-drop and density-wave oscillations are pure instability types while thermal oscillations are considered as a secondary phenomenon.

The pressure-drop oscillations occur in systems having a compressible volume upstream of, or within, the heated section. In such a system a fundamental static instability triggers a dynamic instability characterized by lower frequency oscillations. The periods of the

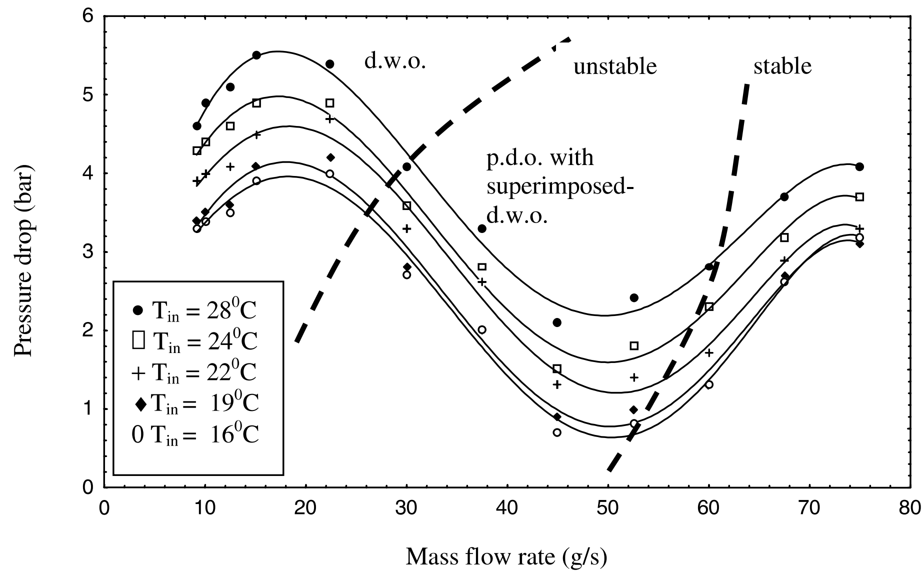

Fig. 7. Steady-state characteristic curves, tube-4.

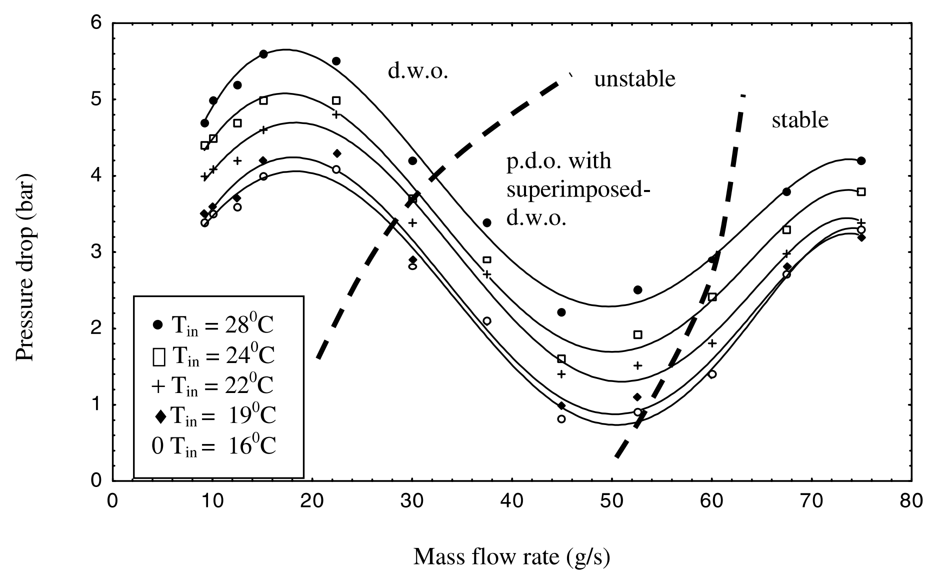

Fig. 8. Steady-state characteristic curves, tube-5.

pressure-drop oscillations vary from 18 to 70 s. The period of oscillations is governed by the volume and the compressibility introduced upstream of the heater by a surge tank, and are determined by the time constant associated with the amount of compressible volume. The periods of these oscillations are generally higher than those experienced in density-wave oscillations (Bergles, 1976; Ding et al., 1995; Çomaklı et al., 2002 etc.).

The density-wave oscillation is the most common dynamic instability in two-phase flows and is caused by the dynamic interaction between flow, density distribution and pressure-drop distribution within the system. A temporary reduction of inlet flow in a heated channel increases the rate of enthalpy rise, thereby reducing the average density. This disturbance affects the pressure drop as well as the heat transfer behavior. For certain combinations of geometrical arrangement, operating conditions, and boundary conditions, the perturbations can acquire appropriate phases and become self-sustained. For boiling systems, the oscillations are due to multiple regenerative feedbacks between the flow rate, vapor generation rate, and pressure drop. The density-wave type oscillations are high frequency oscillations in which the period is approximately one two times the time required for the fluid particle to travel through the heated channel (Bergles, 1976; Ding et al., 1995; Çomaklı et al., 2002 etc.).

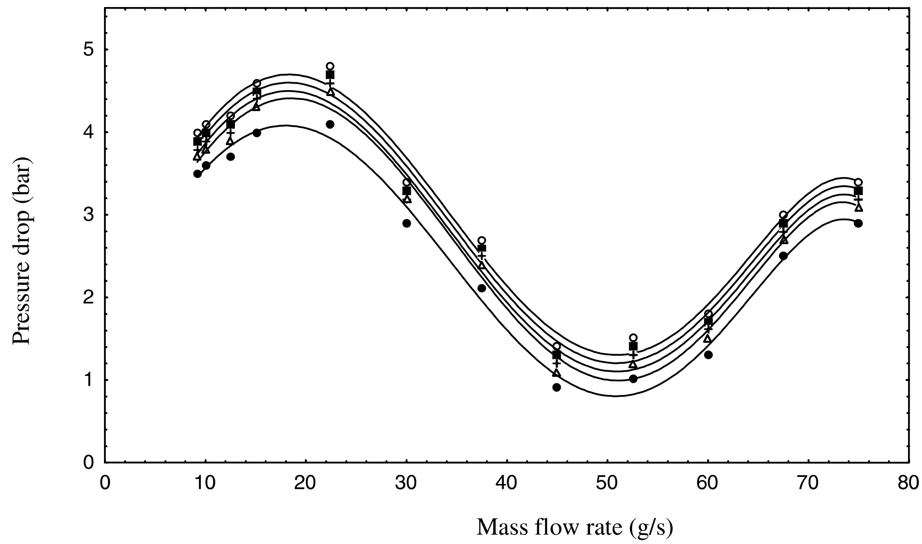

Fig. 9. Steady-state characteristic curves for different tubes with $T_{in} = 22$ °C.

Thermal oscillations are compound dynamic instabilities related to the instability of the liquid film and are accompanied by large temperature fluctuations. The flow oscillates between film boiling and transition boiling at a given point and thus produces large amplitude wall temperature oscillations. The amplitudes and periods of pressure and mass flux oscillations are very small. Density-wave type oscillations are required to trigger the thermal oscillations, which are a relatively rare combination of flow and thermal characteristics. The thermal oscillations can be very dangerous and pressure pulses of 3.4–6.8 bar and temperature variations up to 300 °C are observed (Bergles, 1976; Ding et al., 1995 etc.).

Three basic types of dynamic instabilities, namely pressure-drop type, density-wave type and thermal oscillations were observed during the experiments. Various types of superimposition of pressure-drop type and density-wave type oscillations also, were observed. The boundaries of the oscillations for all heater tubes at different inlet temperatures were collected in Fig. 10. Boundaries at the right of the diagram refer to beginning of the pressure-drop type boundaries, whereas boundaries at the left of the diagram refer to end of the superimposed oscillations and beginning of the pure density-wave type oscillations. These boundaries can also be seen in Figs. 4–8, which are steady-state characteristic curves for different heater tubes. In these fig-

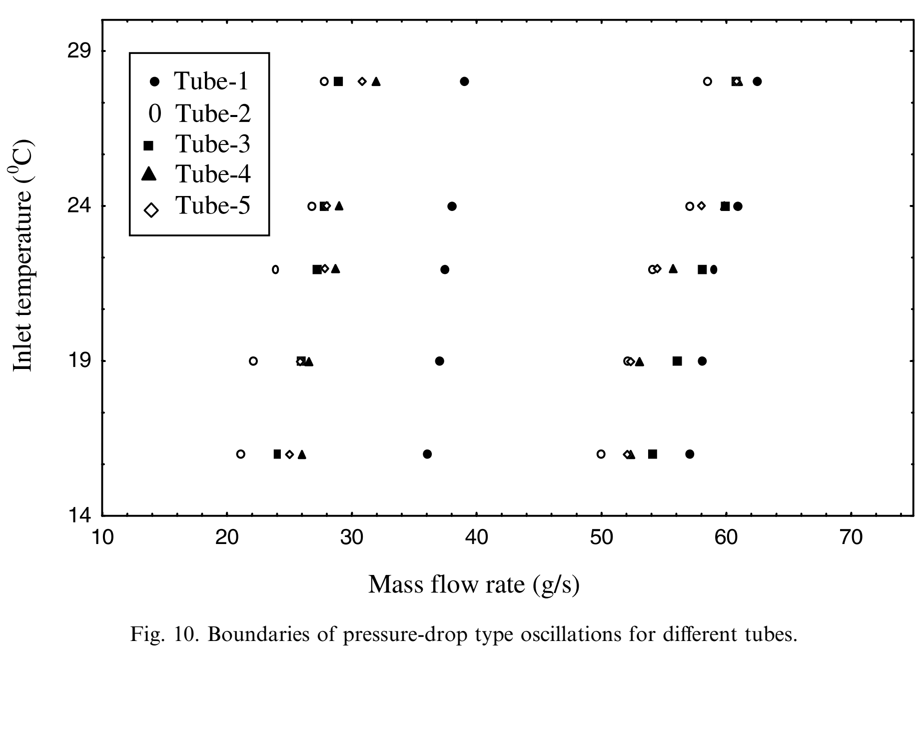

Fig. 10. Boundaries of pressure-drop type oscillations for different tubes.

ures, the boundaries of the oscillations are shown with dashed lines. "p.d.o." seen in the diagram is the abbreviation of pressure-drop type oscillations and d.w.o. is the abbreviation of density-wave type oscillations.

An examination of these figures reveals that the curves of stability boundaries for different heater tubes exhibit approximately a similar pattern on the basis of inlet temperature. The beginning of the pressure-drop type oscillations moves to a lower mass flow rate with lower inlet temperatures. The end of the pressure-drop type oscillations also moves to a lower mass flow rate with lower inlet temperatures. This means that increasing inlet subcooling stabilizes the flow for all heater tube configurations.

For tubes with enhanced surfaces, pressure-drop type oscillations start at a lower mass flow rate than for the bare tube. From the examination of Fig. 10 it may also be seen that the pressure-drop type oscillations in tubes with enhanced surfaces cover a wider range of mass flow rates than the bare tube. Pressure-drop type oscillations ends near the middle point of the characteristic curves for the bare tube, whereas the end of pressure-drop type oscillations goes upwards towards the local maximum point of characteristic curve for four enhanced surfaces. Between these two lines, i.e. the beginning and the end of pressure-drop type oscillations, pressure-drop type oscillations generally occur with superimposed density-wave type oscillations. The bigger this region, the longer are the oscillations. In other words, that the region between two lines is bigger means that flow is less stable. As seen from the Fig. 10 the heater tube covering the shortest mass flow rate region is the bare tube. This means that the bare tube is the most stable configuration. The most unstable configuration is found to be tube-3, because it has the longest distance between the beginning and end of pressure-drop type oscillations. Tubes are ordered on the basis of stability from maximum to minimum as: tube-1; tube-4; tube-5; tube-2; and tube-3 (Table 3). Widmann et al. (1994, 1995) found similar results in a horizontal pipe system namely it was found that the pressure-drop type oscillations in tubes with internal springs covered a wider mass flow rate range than the bare tube. This is a result of unevenness of the tubes with enhanced surfaces creating less stratification of the flow than in a smooth tube. Stratification in the enhanced configurations begins at a lower mass flow rate than the bare tube. The springs or rings inserted inside the tube causes flow turbulence, thus mixing the liquid. As a result of mixing, the colder bulk liquid also touches the tube surface, thus delaying stratification (Widmann et al., 1994, 1995).

The effective diameter of the tube-2, which has internal springs with smaller pitch, is smaller than tube-3, which has internal springs with bigger pitch. On the other hand, it is found that tube-2 is more stable than tube-3. It may be concluded that for tubes with internal spring the stability increases with decreasing effective diameter. Similar results were found for horizontal tube systems with internal springs by Widmann et al. (1994, 1995).

On the other hand among the enhanced configurations tube-4 with equilateral square ring and tube-5 with equilateral triangle ring is more stable than the tube-2 and tube-3 with internal springs. Among the enhanced configurations tube-4 with equilateral square ring is found to be the most stable one. However, on the basis of effective diameter there is no single result such as stability increase/decrease with increasing/decreasing effective diameter. Similar enhanced configuration group such as tubes with internal springs must be evaluated within own group. For this study, if tube-4 with equilateral square ring and tube-5 with equilateral triangle ring are considered one group, because heat transfer enhancement configurations can be accepted similar, same result also becomes valid for this group; namely the stability increases with decreasing effective diameter. Mentes et al. (1983a,b) investigated six different configurations in vertical tubes, and found similar results, expressing that "for tubes with internal springs, the stability increases with decreasing effective diameter, but there is not a consistent pattern for the other tubes".

3.3. The effect of the heat transfer enhancement on the pressure-drop type oscillations

Inlet temperature, inlet pressure, mass flow rate and temperatures were recorded and plotted to find out the amplitudes and the frequency of the oscillations. Oscillation characteristics changed from tube to tube. Figs. 11 and 12 show the periods and amplitudes of the pressure-drop type oscillations for five different tubes for $T_{in} = 16$ °C. It can be seen that when decreasing the mass flow rate, the periods and amplitudes of pressure-drop type oscillations decrease for all heater tubes. The periods and amplitudes of the pressure-drop type oscillations also increase with decreasing inlet temperature for all heater tube configurations.

The range of periods and amplitudes of pressure-drop type oscillations obtained in this study are presented in Table 4. The periods for tubes with enhanced surfaces are higher than those of the bare tube. The minimum inlet pressure period occurred in the bare tube, whereas the maximum inlet pressure period occurred in tube-3 with internal spring having bigger pitch. The order of heater tube configurations on the basis of period of pressure-drop type oscillations is presented in Table 3 and can also be seen from Fig. 11. As seen the following order is found from maximum to minimum: tube-3, tube-5, tube-4, tube-2, and tube-1.

The order of heater tube configurations on the basis of amplitude of pressure-drop type oscillations is presented in Table 3. The following order is found from

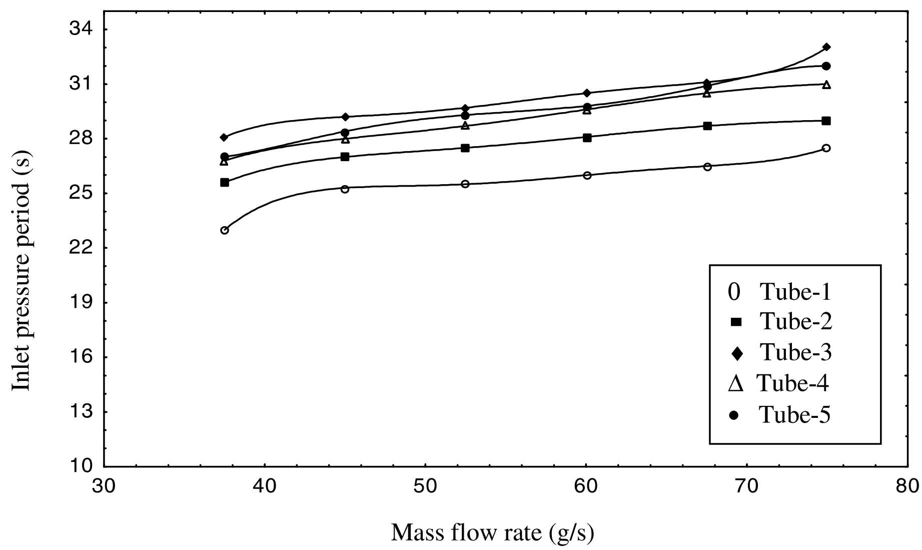

Fig. 11. Periods of pressure-drop type oscillations for different tubes with $T_{in} = 16$ °C.

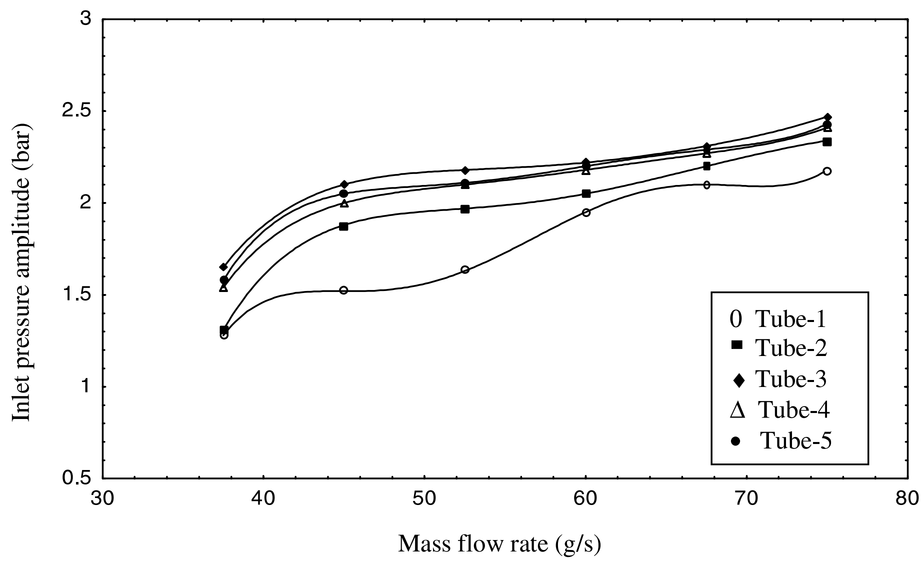

Fig. 12. Amplitudes of pressure-drop type oscillations for different tubes with $T_{in} = 16$ °C.

maximum to minimum: tube-3, tube-5, tube-4, tube-2, and tube-1. As seen the order is the same as the order for the periods of the pressure-drop type oscillations. The amplitudes for tubes with enhanced surfaces are higher than those of the bare tube. The minimum inlet pressure amplitude occurred in the bare tube, the maximum inlet pressure amplitude occurred in tube-3 with internal spring having bigger pitch (Table 4).

3.4. The effect of the heat transfer enhancement on the density-wave type oscillations

Density-wave type oscillation is one of the main instability modes in two-phase flow systems. These oscillations are high frequency oscillations, related to kinematic-wave propagation phenomena, and due to multiple regenerative feedbacks between flow rate, va-

Table 4
The period and the amplitudes of the oscillations

	Period of pressure-drop type oscillations (s)	Amplitude of pressure-drop type oscillations (bar)	Period of density-wave type oscillations (s)	Amplitude of density-wave type oscillations (bar)
Tube-1	23–27.5	1.25–2.2	2.72–2.86	0.34–0.43
Tube-2	25.5–29	1.3–2.3	2.89–2.97	0.36–0.44
Tube-3	28.3–33	1.65–2.45	3.56–3.64	0.39–0.46
Tube-4	26.5–30.5	1.55–2.35	3.20–3.33	0.37–0.45
Tube-5	27–31.6	1.6–2.4	2.97–3.08	0.37–0.44

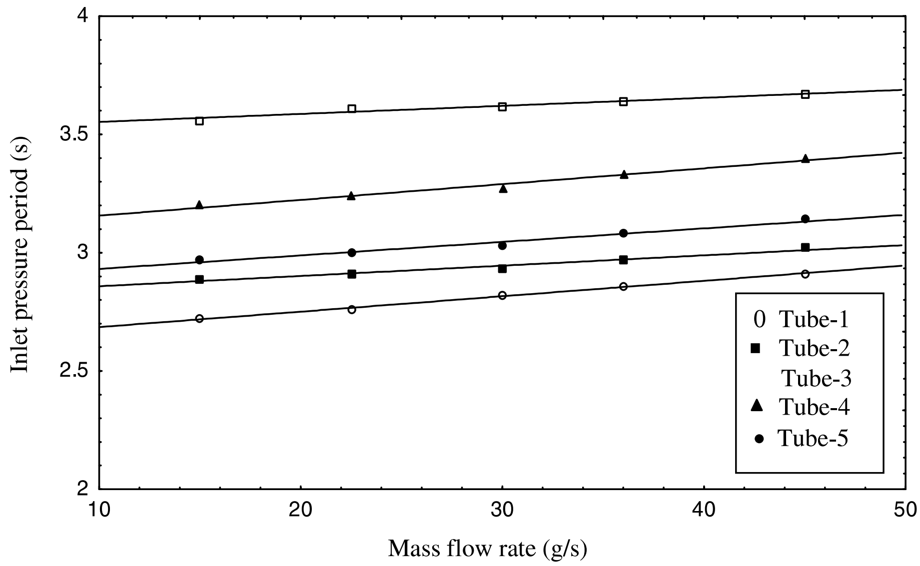

Fig. 13. Periods of density-wave type oscillations for different tubes with $T_{in} = 19$ °C.

por generation rate, and pressure drop. Temperature or enthalpy perturbations cause density or void fraction perturbations, which travel at the kinematic-wave velocity of the mixture. Fluid waves of alternatively higher and lower density mixtures travel across the system during these oscillations. The periods of density-wave type oscillations are proportion to the transit time of a fluid particle through the system (Bergles, 1976; Wang et al., 1994).

It is seen from Fig. 13 that the period of density-wave type oscillations decrease as mass flow rate is decreased for all heater tube configurations. It is also found that the period of density-wave type oscillations increase with decreasing inlet temperature. The range of periods and amplitudes of density-wave type oscillations obtained in this study are presented in Table 4. An examination of Fig. 13 and Table 4 show that the periods of enhanced tubes is higher than that of the bare tube. The order of heater tube configurations on the basis of period of density-wave type oscillations is presented in Table 3. The following order is found from maximum to minimum: tube-3, tube-4, tube-5, tube-2, and tube-1.

Fig. 14 shows that the amplitude of density-wave type oscillations decreases with decreasing mass flow rate for all heater tube configurations. Although there isn't much difference between the amplitudes of the enhanced tubes and bare tube, the amplitudes of the enhanced tubes are higher than that of the bare tube (see also

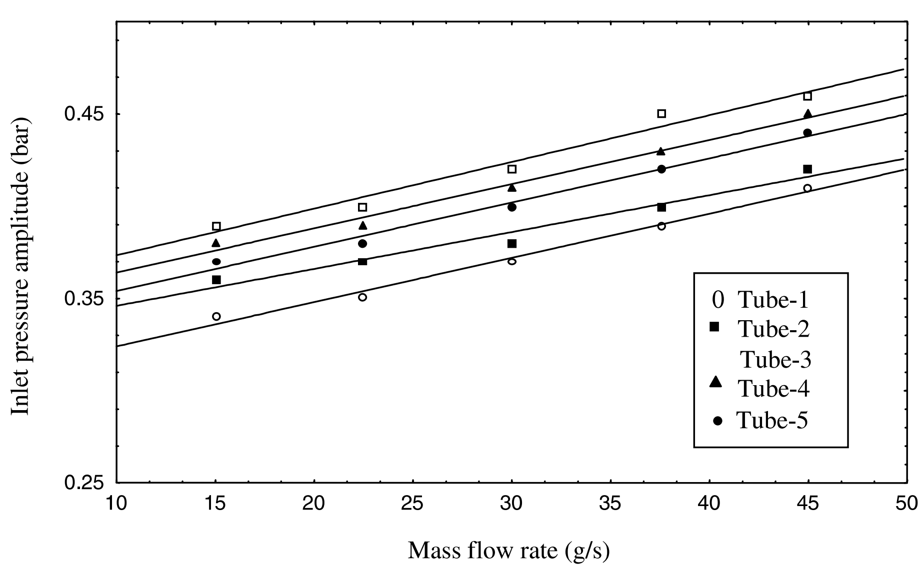

Fig. 14. Amplitudes of density-wave type oscillations for different tubes with $T_{in} = 19$ °C.

Table 4). Maximum amplitudes occurred in tube-3, whereas minimum amplitudes occurred in the bare tube. The order of the heater tubes from maximum to minimum on the basis of amplitude of density-wave type oscillations is found as follows: tube-3, tube-4, tube-5, tube-2, and tube-1.

4. Conclusions

Two-phase flow systems may be unstable under certain operating conditions. This situation may lead unwanted serious problems such as mechanical vibration of components, system control problems, and tube failure, and heat transfer characteristics of the system may also be deteriorated. Knowing the characteristics of these oscillations and boundaries, important conclusions can be drawn for the safe operation of two-phase systems. Five different heater tubes were tested to investigate the effect of heat transfer enhancement on two-phase flow dynamic instabilities. Key findings obtained in this study may be summarized as follows:

1. Three type of oscillations, namely pressure-drop type, density-wave type and thermal oscillations are found for all heater tube configurations and inlet temperatures.
2. Pressure-drop oscillations are observed first as the mass flow is decreased for all heater tube configurations.
3. Stability boundaries move to lower mass flow rates with decreasing inlet temperature, thus system stability increases with decreasing inlet temperature for all heater tube configurations.
4. Unstable region for pressure-drop type oscillations is larger for tubes with enhanced surfaces than for the bare tube. Therefore the bare tube is found to be the most stable configuration, because it has the narrowest unstable region in the steady-state characteristic diagram.
5. Tube with internal springs having bigger pitch is the most unstable one among the tested tubes. Because this heater surface configuration has the longest distance between the beginning and end of pressure-drop type oscillations.
6. Tube-4 with equilateral square ring and tube-5 with equilateral triangle ring are more stable than other enhanced tubes, which are tubes with internal springs, namely tube-2 and tube-3.
7. Periods and amplitudes of pressure-drop type oscillations and density-wave type oscillations decrease for all heater tube configurations as mass flow rate is decreased.
8. Periods and amplitudes of pressure-drop type oscillations and density-wave type oscillations change depending on heater tube configurations. The biggest oscillation period and amplitude occur in tube-3 which has internal springs having bigger pitch.
9. System stability increases with decreasing equivalent diameter for the same type heater tube configurations. However, on the basis of effective diameter there is no single result such as stability increase/decrease with increasing/decreasing effective diameter.

The purpose of heat transfer enhancement is to design a system with higher heat transfer coefficient, lower friction factor, and stable operation characteristics. In this study, however, only the effect of heat transfer enhancement on two-phase flow instabilities was studied to determine conditions under which the stable operation would occur. Therefore, the effect of same heat transfer surface configurations on the heat transfer and friction characteristics will be studied in another paper in detail.

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