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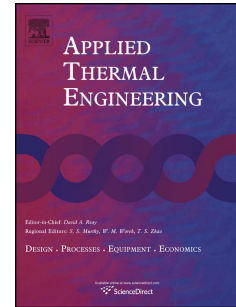
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## Two phase flow instabilities in horizontal straight tube evaporator

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**Abstract:** It is essential to ensure the stability of a refrigeration system if the oscillation in evaporation process is the primary cause for the whole system instability. This paper is concerned with an experimental investigation of two phase flow instabilities in a horizontal straight tube evaporator of a refrigeration system. The relationship between pressure drop and mass flow with constant heat flux and evaporation pressure is measured and determined. It is found that there is a negative slope section in the middle of positive slope pressure drop versus mass flow velocity thus making the velocity a multi-valued function of pressure drop. Three types of dynamic instabilities including the density wave instability, pressure drop instability and thermal instability are found under the conditions of heat flux from 5 to 17 kW/m<sup>2</sup>, mass velocity from 150 to 1500 kg/(m<sup>2</sup>·s), and evaporating pressure from 0.5 to 0.8 MPa. The density wave oscillation occurs at almost all mass velocities; its period is about 1~3 seconds and its amplitude is the lowest. The pressure drop oscillation takes place in a negative slope section; its period is about 10 seconds. The thermal oscillation can be induced at a high mass flow velocity, and its period is of 60 seconds and the amplitude is the highest among the

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three oscillations. The boundary as the onset of pressure drop oscillation and that of thermal oscillation are obtained experimentally. Finally, the empirical equations for the boundaries between three types of oscillations are obtained according to the mathematical model and test data.

**Keywords:** evaporator; instability; refrigeration; two phase flow

### Nomenclature

$C$	constant
$D$	inner diameter, m
$G$	mass velocity, kg/(m <sup>2</sup> ·s)
$g$	acceleration of gravity, m/s <sup>2</sup>
$h$	enthalpy, kJ/kg
$h_{fg}$	latent heat of vaporization, kJ/kg
$f$	friction coefficient
$L$	length, m
$p$	pressure, kPa
$Q$	heat flux, kW/m <sup>2</sup>
$x$	vapour quality
$z$	position on the tube length
$\alpha$	void fraction
$\Delta h_i$	enthalpy rise, kJ/kg

$\Delta p$	pressure drop, kPa
$\rho$	density, kg/ m <sup>3</sup>
$\phi$	two-phase multiplier of pressure drop

### ***Subscripts***

$a$	acceleration
$f$	friction
$g$	gas
$i$	inner
$l$	liquid

## **1. Introduction**

The stability which is an essential condition for a successful and high-efficiency refrigeration system has been discussed by researchers recently [1-5]. The cause of instability may be explained as the influence of the inherent characteristics of two-phase flow in evaporator and also the impact of the system control characteristics [6]. The former emphasizes particularly on the oscillations of two-phase flow of the refrigerant in evaporation process; its result is regarded as the basis of the system instability.

The instability induced by the oscillations of two-phase flow boiling, which conventionally refers mainly water system such as steam generators, boiling water reactors and reboilers, can be classified as either the static instability or the dynamic instability. The static instability, caused by the steady-state performance of a boiling process, mainly includes Ledinegg

instability and flow pattern transition instability; the dynamic instability, initiated by the transient changes in boiling process, mainly covers the density-wave oscillations, thermal oscillations and pressure drop oscillations [7, 8].

Indeed, the results of water system have provided necessary experience and methodology for the researches of refrigeration loop. However, some differences between the two-phase flow instabilities in refrigeration system and that in water system can be identified. For example, water systems are often open circulation and the evaporating pressure is governable as a fixed value with a surge tank or some compressible containers, but the similar containers in refrigeration systems do not have such functions; the heat flux of evaporators tube in water systems is often much higher than those in refrigeration systems; and moreover, the working condition in a refrigeration system is also different from that in water system, such as throttling device and the quality at the inlet and exit. These differences may cause the absence of certain types of oscillations. In addition, they are also obstructions for applying the water two-phase flow instabilities theories and the results into refrigeration systems. Some preceding investigations focused on the thermal character of refrigerant with several types of tubes, but the system setup is the same as the water system. Among these results, only Veziroglu et al. [9] investigated the stability boundaries of sustained density-wave oscillations in an electrically heated single channel, up-flow system with R-11 as the test fluid.

Early experimental observations for two-phase flow instabilities of a refrigeration system were reported by Zahn [1]. Wedekind and Stoecker [2] have found the oscillatory motion of mixture-vapour transition point in evaporator and the motion was considered as an oscillatory nature, even for steady flow condition. Based on experimental and theoretical

analysis, several researchers including Zahn [1], Wedekind & Beck [5], and Barnhart & Peters [10] suggested that the occurrence of slug flow at the inlet of evaporator should have resulted in the oscillation; and actually the slug flow should be a kind of thermal oscillations or the result of density-wave oscillations. Meanwhile, Wedekind and Stoecker [2] believed that it was related to the density-wave oscillation. Moreover, Gruhle and Isermann [11] proposed a probability that it may be influenced by the strong nonlinear character of heat transfer coefficient. However, the variation of heat transfer coefficient may actually be an integrated result of density-wave oscillation and other oscillations like pressure drop oscillation. In general, since only these a few experimental studies have focused on evaporations in refrigeration system, there is not a general explanation and experimental proof for the instabilities in refrigeration system.

This paper presents the results of an experimental study on two phase flow oscillations of R-22 in a refrigeration system with a horizontal straight tube evaporator.

## **2. Experimental measurement**

As shown in Fig. 1, the experimental refrigeration system is consist of four main components, namely the compressor, the water-cooled condenser, the evaporator and the electronic expansion valve (EEV), and other accessorial components. As a main part of the experimental system, the evaporator is a horizontal straight copper tube of 3 m long, 8 mm inner diameter and 9.52 mm outer diameter. The tube is wrapped by a thin zonal heater of 3 kW heat input capacity. The heat flux of the evaporator is controlled by a voltage regulator. All temperatures of the evaporator are measured using copper-constantan thermocouples of

0.5mm diameter. Six copper-constantan thermocouples are fixed evenly on the outer surface of the test tube and other six copper-constantan thermocouples are inserted into the tube to measure the inside refrigerant temperature. The pressure sensors used in the experiment system are MPM480 with the precision of  $\pm 0.25\%$  of full scale.

A compressor with variable frequency and an EEV are used to regulate the mass flow rate of R-22 and evaporation pressure by changing power frequency of the compressor or the EEV opening (or pulse number). The constant temperature water circulator supplies the cooling water for condenser and the water temperature can be controlled upon requirement. The mass flow rate of water or refrigerant is measured with water or refrigerant mass flow meter, respectively. The signals of thermocouples, pressure sensors and mass flow meters are displayed and recorded by the data acquisition system and can be processed in the computer. Detailed information for test devices and their precision is listed in Table 1.

The experiment is performed as follows:

- 1) At first, open all valves and turn on the compressor, R-22 is transferred through the whole system, and the evaporation pressure is adjusted to a fixed value by changing power frequency of the compressor and the EEV opening.
- 2) The heat flux is controlled by setting values with the voltage regulator. The evaporation pressure of R22 will change with the variation of heat flux, and it can be adjusted to a fixed value by changing power frequency of the compressor and the EEV opening again. With constant heat flux and evaporation pressure, change the refrigerant mass flow rate, and observe and measure the parameters of two-phase flow in evaporator.
- 3) Change the setting value of heat flux, and then repeat the step 2.



4) Change the evaporating pressure, and repeat the step 2 and step 3.

### 3. Results and discussion

#### 3.1 Static instability

Fig. 2 show the results of measured pressure drop between inlet and outlet of the evaporator changing with mass flow velocity in the experiment system in which the pressure of evaporation is of 0.7MPa and the heat flux is of 17 kW/m<sup>2</sup>. It can be seen from Fig. 2 that there is a negative slope region (point 2 to 4) in addition to the two positive slope regions (point 1 to 2, point 4 to 5). In general, the pressure drop rises with the increase of mass velocity. This kind of characteristic is called positive slope. The negative slope in two-phase flow of evaporator is formed in the conditions that the pressure increase by the evaporation exceeds the pressure drop of friction. The negative slope region means that there must be several mass velocities (point 1, 3 and 5) for a given pressure drop, which may cause the oscillation due to the mass velocity leap (between point 1 and 3, or between point 3 and 5) at a fixed pressure drop.

The relationship of pressure drop changes with mass velocity of the evaporator as shown in Fig. 2 is similar to the results of water system observed by Ledinegg [12], which is called Ledinegg instability and belongs to the static instability.

Fig. 3 shows the influence of heat flux on pressure drop when the pressure of evaporation remains at 0.7MPa, and heat flux changes from 5 to 17 kW/m<sup>2</sup>. It is obvious that the negative slope appears at larger mass velocity when the heat flux is high. This is because at high mass velocity, larger heat flux is needed to generate enough pressure drop increment by

evaporation to exceed the pressure drop by friction.

### 3.2 Dynamic instability

Under the conditions of heat flux from 5 to 17 kW/m<sup>2</sup>, mass velocity from 150 to 1500 kg/(m<sup>2</sup>·s) and evaporating pressure from 0.5 to 0.8 MPa, three types of dynamic instabilities including the density wave instability, pressure drop instability and thermal instability have been found in our experiments. Typical types of oscillations are chosen for a certain heat flux (17 kW/m<sup>2</sup>) to compare with other different types of oscillations. Experiments data with other heat flux are shown in Table 2.

The density wave oscillation is the results of fluid waves of alternately higher and lower density mixtures due to multiple regenerative feedbacks among the flow rate, vapour generation rate and pressure drop [7]. In the present experiment, the density wave oscillation is observed from very small mass flow rate to large mass flow rate. Figs. 4 and 5 show the measured transient behaviour of evaporation temperature and mass velocity, respectively, with the heat flux of 17 kW/m<sup>2</sup> when the density wave oscillation happens. The pressure of evaporation is referred to the measured value from the pressure sensors in the middle of evaporator, and the temperature of evaporation is the corresponding saturated temperature to the pressure of evaporation. The oscillation period of density wave oscillations is about 4 second, and it is proportional to the transit time of refrigerant particle through the evaporator. And the oscillation amplitude of evaporating temperature is of 0.6 °C, and the according mass velocity is about 60 kg/ (m<sup>2</sup>·s).

It has been pointed out that the pressure drop oscillations occur only when the pressure

drop decreases with the increase of mass flow rate (or mass velocity), and the oscillation period is governed by the volume and the compressibility of vapour in the water system [8]. This means that the necessary condition for the occurrence of the pressure drop type oscillation is the pressure drop curve changing with mass velocity with negative slope (point 2 to 4 in Fig. 2) and the existence of a compressible volume (e.g. surge tank) in the flow circuit. In refrigeration system, most compressible volume usually comes from the accumulator and receiver.

Figs. 6 and 7 show the measured transient behaviour of evaporating temperature and mass velocity, respectively, with the heat flux of  $17 \text{ kW/m}^2$  when pressure drop oscillation happens in the experiments. The period of pressure drop oscillation is about 10 second, and it is governed by the volume and compressibility of the vapour in the refrigeration system. In addition, the amplitude for evaporating temperature is  $1 \text{ }^\circ\text{C}$ ; the amplitude for mass velocity is about  $150 \text{ kg/(m}^2\cdot\text{s)}$ .

The term of ‘thermal oscillation’ is often used to describe the situations of temperature fluctuations occurring in solid interacting with fluid. The thermal oscillation used here is related to the instability of boiling heat transfer [8], and it is characterized by large amplitude fluctuations in the heater wall temperature. The typical character of thermal oscillations is its long period and great amplitude.

In the present experimental study, thermal oscillations have been found at relatively high mass velocity. For example the oscillations are found, respectively, when mass velocity is in the range of  $637\text{-}750 \text{ kg/(m}^2\cdot\text{s)}$  and heat flux is at  $5 \text{ kW/m}^2$ ; and also when mass velocity is in the range of  $959\text{-}1500 \text{ kg/(m}^2\cdot\text{s)}$  and heat flux is  $17 \text{ kW/m}^2$ . Figs. 8 and 9 show the

measured transient behaviour of evaporating temperature and mass velocity respectively during the period of thermal oscillation which occurs when heat flux is of  $17 \text{ kW/m}^2$  and mass velocity is of  $900\text{-}1000 \text{ kg/ (m}^2\cdot\text{s)}$ . The period of thermal oscillations is about 60 second, and the amplitude of evaporating temperature is of  $3 \text{ }^\circ\text{C}$ . There are many short period density wave oscillations in a cycle of the long period thermal oscillation, and the oscillations shown in Figs. 8 and 9 can be seen as an integrated results of thermal oscillations and density wave oscillations.

### 3.3 Instability boundary

It is necessary to determine the boundaries of the regions where the density wave oscillations, pressure drop oscillations and thermal oscillations occur. From a mass of the test data, the boundary as the onset of pressure drop oscillation and that of thermal oscillation can be obtained, and the empirical equations for the boundaries among density wave oscillations, pressure drop oscillations and thermal oscillations are developed according to the mathematical model and test data.

The onset of oscillation is concerned with the inlet subcooling, system pressure, heat flux at inlet and exit, liquid temperature, cross-section of the channel, entrance throttling and the inlet velocity [13, 14]. The stability thresholds and the correlations for density-wave oscillations [15-20], pressure drop oscillations [21] and thermal oscillations [22] in water system were given and the effects of various parameters on stability boundaries and oscillation periods were studied. The equations of instabilities have been reviewed by Kakac and Bon [22]. However, it can be identified that they reviewed equations are aimed only at a certain type of

oscillations; and the aim of the present researches is to simulate the period and amplitude of oscillations. Such theoretical simulations and analyses have the advantages of understanding the mechanism of a certain type of oscillations, but the results are lack of applicability related to practical designing, testing and controlling of refrigeration system. In this paper, a mathematical model is built on the basis of measured instability boundaries to describe the relations between the oscillation boundaries and these parameters which can be measured.

The boundary between density-wave oscillations and pressure drop oscillations is the onset of pressure drop oscillations, and the pressure drop oscillations can occur at the right hand side of the boundary. Similarly, the boundary between pressure drop oscillations and thermal oscillations is the onset of thermal oscillation, and the thermal oscillations can take place at the right hand side of the boundary. The boundary as the onset of pressure drop oscillation (*broken line a*) and that of thermal oscillation (*broken line b*) were obtained according to large numbers of the test data (see Fig.10).

The density wave oscillations (DWO) first appear at very small mass velocity (e.g., 150 kg/(m<sup>2</sup>·s)) and it exists the whole mass velocity range (150 to 1500 kg/(m<sup>2</sup>·s)) in our experiments. From experimental results, it can be concluded that the density wave oscillation takes place when the heat flux is high enough for a given mass velocity.

The pressure drop oscillation (PDO) occurs in the region between the *broken line a* and *broken line b*, where the slope is negative for the pressure drop versus mass velocity. When the heat flux increases, the boundary between the density wave oscillations and pressure drop oscillations moves to higher mass velocity, this means that the region of density wave oscillations enlarges with the increase of the heat flux.

The thermal oscillations (TO) occur in the region of right hand side from *broken line b* and it happens when the heat flux is relatively low at the given mass velocity since the mass velocity is so high that the liquid R-22 cannot be evaporated adequately at a low heat flux.

In refrigeration system, the users always focus on the temperature at evaporator outer wall while the wall temperature is more stable than the actual evaporating temperature. As a result, density wave oscillations can be seen as a kind of oscillations with little harm to system operation because of its low amplitude and short period. However, the pressure drop oscillations and thermal oscillations should be paid more attention for their high amplitude and long period, and the temperature at evaporator outer wall will then appear to be unstable.

#### **4. Mathematical modelling of instability boundary**

The boundaries of the given state were obtained from the experimental results. As the onset conditions of a given oscillations always change along with the working conditions, it is necessary to develop a mathematical model to determine the boundaries at different conditions.

In the present study, a homogeneous phase model is proposed to simplify the governing equations. To develop the mathematical model, the following assumptions have been made :

- 1) the flow inside the tube is one dimensional homogeneous phase flow;
- 2) there is a thermodynamic equilibrium between two phases;
- 3) the fluid is incompressible;
- 4) the dissipated heat by friction, kinetic energy and potential energy is negligible in the energy balance equation.

5) the heat flux is homogeneous.

6) the axial heat conduction of tube wall and the heat stored in tube wall is negligible.

Based on the physical phenomena described, the governing equations of the flow in terms of continuity, momentum and energy equations can be written in Eqs. (1) to (3):

$$\text{Continuity: } \frac{dG}{dz} = 0 \quad (1)$$

$$\text{Momentum: } \frac{dp}{dz} = \left( \frac{dp}{dz} \right)_f + \left( \frac{dp}{dz} \right)_a = - \left( \frac{2fG^2}{D\rho_l} \right) \phi^2 - G^2 \frac{d}{dz} \left( \frac{x^2}{\rho_g \alpha} + \frac{(1-x)^2}{\rho_l(1-\alpha)} \right) \quad (2)$$

$$\text{Energy: } \frac{d(Gh)}{dz} = \pi D Q \quad (3)$$

Considering the above homogeneous phase model, Eqs. (1) to (3) can be transferred into integral equations along the tube length. Then the equations can be converted to dimensionless format as

$$\frac{G^2}{gD\rho_l^2} = C \quad (4)$$

$$\frac{\Delta p}{gD\rho_l} = \frac{G^2}{gD\rho_l^2} \int_{z_1}^{z_0} 2f \frac{\phi^2}{D} dz + \frac{G^2}{gD\rho_l^2} \left( \frac{x^2}{\alpha^2} \left( \frac{\rho_l}{\rho_g} \right) + \frac{(1-x)^2}{(1-\alpha)^2} - 1 \right) \quad (5)$$

$$x + \frac{\Delta h_i}{h_{fg}} = \frac{Q_0}{G_r} \pi G z_0 \quad (6)$$

It is supposed that,

$$a = f(x, \rho_g, \rho_l) \quad (7)$$

$$\phi^2 = f(x, \frac{\rho_g}{\rho_l}) \quad (8)$$

So that the equations can be expressed as:

$$\frac{G^2}{gD\rho_l^2} = C, \quad (9)$$

$$\frac{\Delta p}{gD\rho_l} = f\left(\frac{G^2}{gD\rho_l^2}, x, \frac{\rho_g}{\rho_l}, f, \frac{L}{D}, \frac{\Delta h_i}{h_{fg}}\right), \quad (10)$$

$$\frac{\Delta h}{h_{fg}} = \frac{Q_0\pi Dz_0}{Gh_{fg}} = f\left(x, \frac{\Delta h_i}{h_{fg}}, \frac{L}{D}\right). \quad (11)$$

Then, with these equations, the non-dimensional equations can be further expressed as:

$$\frac{Q}{Gh_{fg}} = f\left(\frac{\Delta h_i}{h_{fg}}, \frac{L}{D}, \frac{G^2}{gD\rho_l^2}, \frac{\Delta p}{gD\rho_l}, \frac{\rho_g}{\rho_l}, f\right). \quad (12)$$

For a given system, the correspondingly change of  $f$  and  $L/D$  can be omitted. So that the final mathematical expression for the oscillation formation condition is:

$$\frac{Q}{Gh_{fg}} = f\left(\frac{\Delta h_i}{h_{fg}}, \frac{G^2}{gD\rho_l^2}, \frac{\Delta p}{gD\rho_l}, \frac{\rho_g}{\rho_l}\right). \quad (13)$$

With the experiment results of evaporating pressure 0.7MPa, empirical correlation for the



onset of pressure drop oscillations and that of thermal oscillation are obtained as Eqs.(14) and (15) respectively.

$$\frac{Q}{Gh_{fg}} = 1.491 \left( \frac{G^2}{gD\rho_l^2} \right)^{0.505} \left( \frac{\Delta h_i}{h_{fg}} \right)^{1.984} \left( \frac{\Delta p}{gD\rho_l} \right)^{0.621} \left( \frac{\rho_g}{\rho_l} \right)^{0.201} \quad (14)$$

$$\frac{Q}{Gh_{fg}} = 1.649 \left( \frac{G^2}{gD\rho_l^2} \right)^{0.852} \left( \frac{\Delta h_i}{h_{fg}} \right)^{2.243} \left( \frac{\Delta p}{gD\rho_l} \right)^{1.174} \left( \frac{\rho_g}{\rho_l} \right)^{0.329} \quad (15)$$

A comparison between the calculated boundaries and measured boundaries is shown in Fig.11. The experimental data is obtained from the test under the conditions of heat flux from 5 ~ 17 kW/m<sup>2</sup>, mass velocity from 150 ~ 1500 kg/ (m<sup>2</sup>·s), and evaporating pressure from 0.5 ~ 0.8 MPa. It can be seen from Fig. 11 that the trend of calculated boundaries is similar to those measured ones. However, there is still an error between the simulation and the measurement, and the corresponding error is about ±30% while the root mean square error is about ±20% . The corresponding comparison between the calculated boundaries and the measured ones is given in Table 3.

Based on Eqs. (5), (6), (14) and (15), the boundary between density-wave oscillations and pressure drop oscillations and the boundary between pressure drop oscillations and thermal oscillations can be calculated for a given refrigeration system when the mass velocity, heat flux, and evaporating pressure are given.

In practical design, a system should be tested before operation. Generally speaking, there are several parameters which can be measured for evaporation, such as the degree of refrigerant subcooling at the expansion valve inlet, the pressure of condensation, the pressure

of evaporation, the heat flux (heat load and heat transfer area), the degree of superheat at evaporator outlet, etc. It is also convenient to record these parameters when the system oscillations occurred. If the refrigerant at expansion valve inlet is subcooled, the period and amplitude of the oscillations will be measured and recorded. Then the oscillations will be classified by the mass velocity, heat flux and evaporating pressure. For the working range (of evaporating pressure), it is possible to draw out such onset boundaries. If possible, it is better to design an evaporator tube with no negative slope in the curve of pressure drop versus mass velocity to avoid pressure drop oscillations. If the evaporator is designed with a negative slope in the curve of pressure drop versus mass velocity, the working range should be set to the region on the left hand side of the pressure drop oscillation onset boundary. If the working range crosses the unstable region (the region on the right hand side of the pressure drop oscillation onset boundary), the system control strategies should be prepared on how to adjust the system parameters to make the system stable when the boundaries are determined.

## 5. Conclusions

In this paper, the static and dynamic instabilities of two phase flow in a horizontal straight tube evaporator of refrigeration system are investigated.

The static instability of evaporator can be expressed by pressure drop versus flow rate. The characteristic curve has a negative slope region in addition to two positive slope regions, thus making the mass velocity a multi-valued function of the pressure drop. The multi-valued characteristic provides the potentiality for static oscillations and dynamic oscillations.

Three types of dynamic oscillations have been found and observed, namely, the density

wave oscillations, pressure drop oscillations and thermal oscillations. The density wave oscillation, which takes place at almost all the mass velocities, has the smallest periods and the lowest amplitude among three types. Pressure drop oscillations are also incidental and the period is about 10 seconds. The thermal oscillation occurs with longest periods and the highest amplitude among three types.

With the experiment results, the experimental onset conditions of pressure drop oscillations and thermal oscillations are obtained. The empirical correlations of the onset of pressure drop oscillations and that of thermal oscillation are also obtained with the homogeneous phase model. The empirical correlations can be used to analyze the onset conditions for pressure drop oscillations and thermal oscillations in a certain R-22 system. The method can be used to guide practical design through some simple tests.

The results in this paper provide a further understanding for the mechanism of two-phase flow instabilities of refrigeration system. These results can also be used to provide guidance for design and control strategy of refrigeration system in order to avoid serious oscillation problems.

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Table 1: Detailed information for the test devices

Components	Model	Remark
Compressor	YZB-21R	Variable frequency from 30 to 120Hz
EEV	DPF 1.6	Variable pulse number from 0 to 550
Evaporator	Self-designed	Horizontal copper tube with 3m long, inner diameter of 8 mm and outer diameter of 9.52 mm
Condenser	SS-075GT	
Refrigerant mass flow meter	Mass 2100	R22 flow rate: 0-1500 kg/h Precision: $\pm 0.1$ kg/h
Water flow meter	LZB-10	Flow rate: 0.36-3.6 m <sup>3</sup> /h Precision: $\pm 0.5\%$

Table 2: Experiment data of different heat flux

Heat flux (kW/m <sup>2</sup> )	Oscillation Type	Period (s)	Average Amplitude (°C or kg/ (m <sup>2</sup> ·s))	
			Temperature	Mass velocity
5	Density wave	0.8	0.9	110
	Pressure drop	9.5	2.1	228
	Thermal	55.4	4.7	1053
9	Density wave	1.0	0.8	100
	Pressure drop	9.9	1.9	195
	Thermal	57.6	4.2	1028
11	Density wave	1.1	0.7	93
	Pressure drop	10.2	1.7	185
	Thermal	58.7	4.0	1008
13	Density wave	1.3	0.7	88
	Pressure drop	10.5	1.6	170
	Thermal	59.3	3.5	968
17	Density wave	1.6	0.6	75
	Pressure drop	10.7	1.4	150
	Thermal	60.4	3.2	950



Table 3: Comparison of calculated boundaries point and experimental boundaries point

Heat flux (kW/m <sup>2</sup> )	Evaporating Pressure (MPa)	Calculated boundaries point (kg/ (m <sup>2</sup> ·s))		Experimental boundaries point (kg/ (m <sup>2</sup> ·s))		Error (%)	
		PDO	TO	PDO	TO	PDO	TO
5.2	0.5			397.4	545.2	7.6	7.5
	0.7	425.4	589.7	470.0	637.5	10.5	8.1
	0.8			524.7	720.4	23.3	22.1
9.0	0.5			484.7	621.5	11.5	13.8
	0.7	547.5	712.8	517.3	686.4	5.6	3.8
	0.8			621.8	812.1	13.6	13.9
11.1	0.5			511.6	655.6	13.4	13.2
	0.7	590.3	755.2	591.4	751.1	0.2	0.5
	0.8			674.5	885.7	14.3	17.3
13.4	0.5			567.5	697.9	11.5	14.7
	0.7	641.6	817.9	624.3	806.1	2.7	1.5
	0.8			726.2	935.9	13.2	14.4
16.9	0.5			657.5	864.6	7.3	3.9
	0.7	709.1	899.6	734.2	958.8	3.5	6.5
	0.8			800.2	1012.4	12.8	12.5

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Fig. 5 Measured transient response of mass velocity in density wave oscillations (heat flux: 17 kW/m<sup>2</sup>)

Fig. 6 Measured transient response of evaporating temperature in pressure drop oscillations (heat flux: 17 kW/m<sup>2</sup>)

Fig. 7 Measured transient response of mass velocity in pressure drop oscillations (heat flux: 17 kW/m<sup>2</sup>)

Fig. 8 Measured transient response of evaporating temperature in thermal oscillations (heat flux: 17 kW/m<sup>2</sup>)

Fig. 9 Measured transient response of mass velocity in thermal oscillations (heat flux: 17 kW/m<sup>2</sup>)

Fig. 10 Measured boundaries of different oscillations (evaporating pressure 0.7 MPa): DWO- density wave oscillations, PDO - pressure drop oscillations, TO- thermal oscillations

Fig. 11 Comparison between the calculated boundaries and measured boundaries

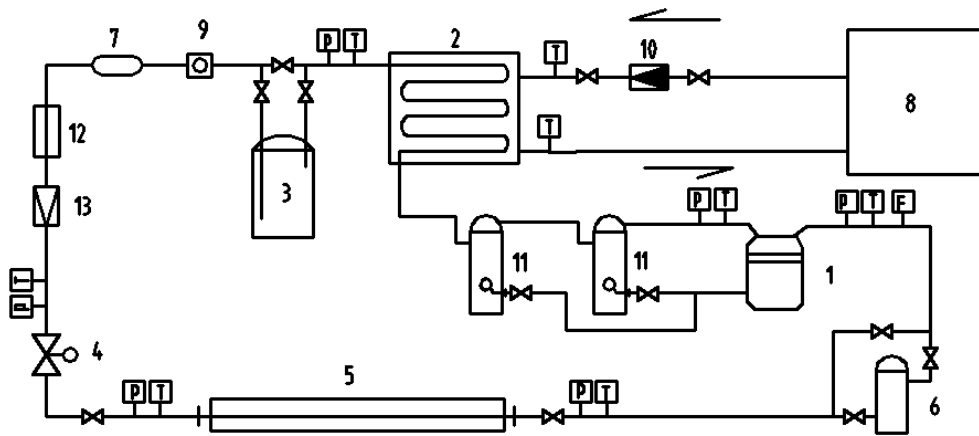


Fig. 1 Experimental system

- 1.compressor 2.condenser 3. receiver 4.EEV 5. horizontal tube electrothermal evaporator 6. accuvulator  
 7. filter drier 8. constant temperature water circulator 9.sight glass 10. water flow meter 11. oil separator  
 12. subcooler 13. mass flow meter

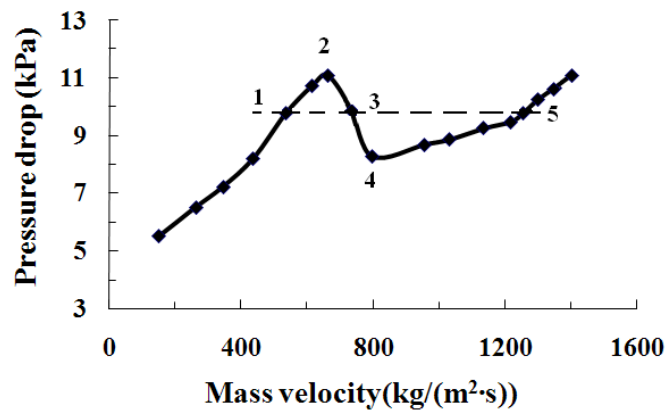


Fig. 2 Measured pressure drop versus mass velocity curve

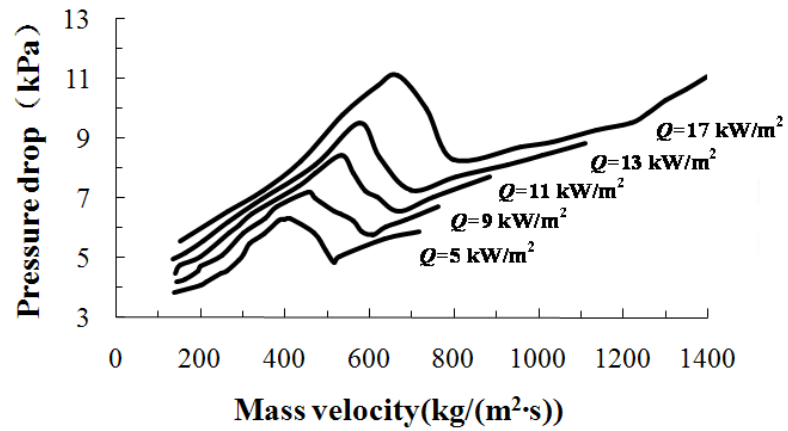


Fig. 3 Influence of heat flux on pressure drop versus mass velocity curve (evaporating pressure: 0.7MPa)

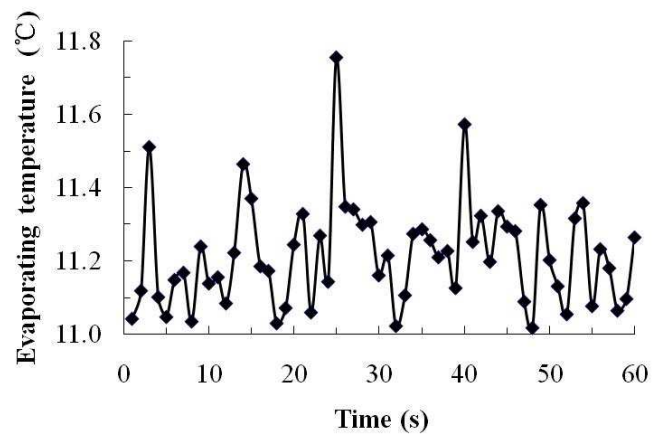


Fig. 4 Measured transient response of evaporating temperature in density wave oscillations

(heat flux:  $17 \text{ kW/m}^2$ )

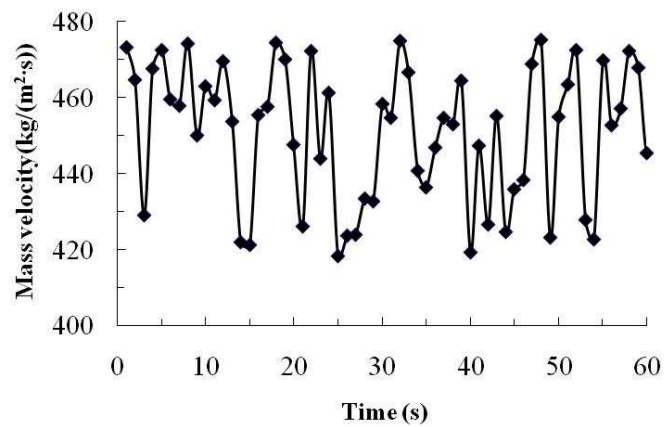


Fig. 5 Measured transient response of mass velocity in density wave oscillations (heat flux: 17 kW/m<sup>2</sup>)

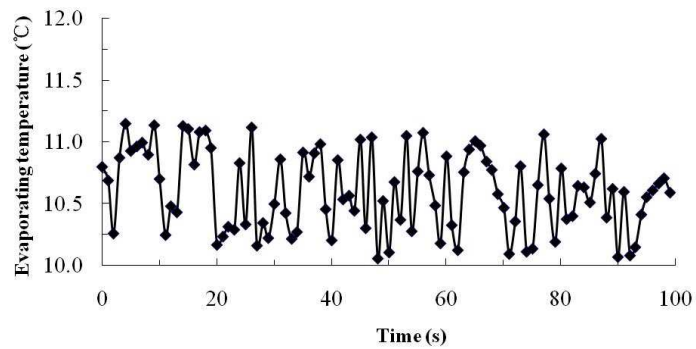


Fig. 6 Measured transient response of evaporating temperature in pressure drop oscillations  
(heat flux:  $17 \text{ kW/m}^2$ )



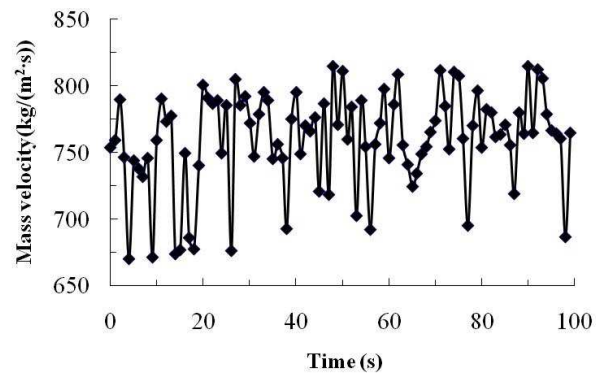


Fig. 7 Measured transient response of mass velocity in pressure drop oscillations (heat flux: 17 kW/m<sup>2</sup>)

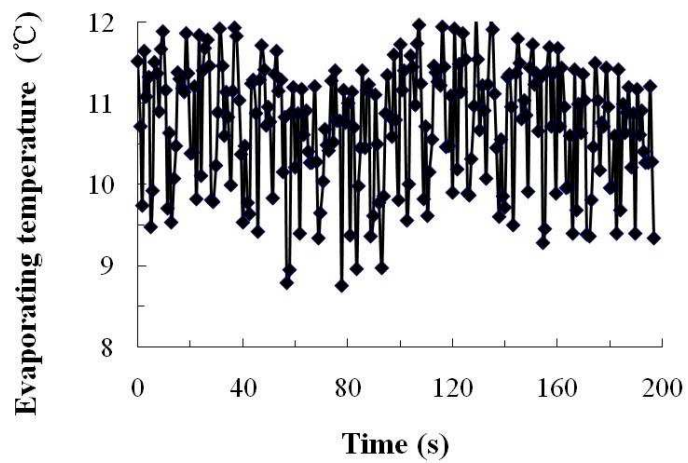


Fig. 8 Measured transient response of evaporating temperature in thermal oscillations (heat flux:  $17 \text{ kW/m}^2$ )

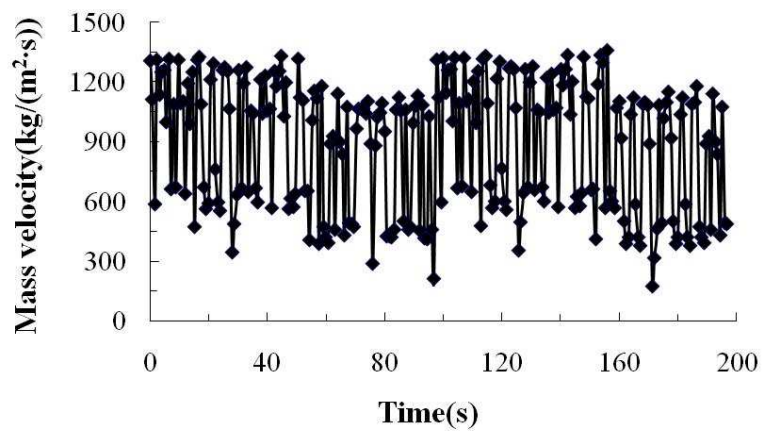


Fig. 9 Measured transient response of mass velocity in thermal oscillations (heat flux: 17 kW/m<sup>2</sup>)

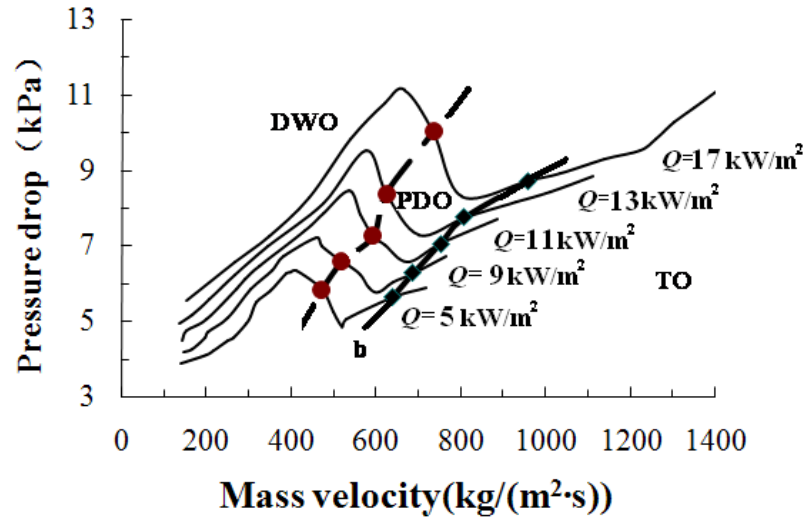


Fig. 10 Measured boundaries of different oscillations (evaporating pressure 0.7 MPa): DWO- density wave oscillations, PDO - pressure drop oscillations, TO- thermal oscillations

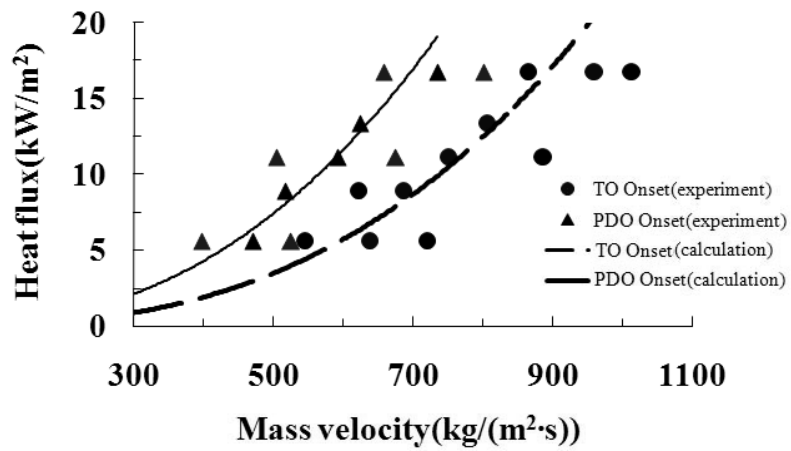


Fig. 11 Comparison between the calculated boundaries and measured boundaries