Characteristic of Pressure Oscillation Caused by Turbulent Vortexes and Affected Region of Pressure Oscillation

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Abstract: Submerged steam jet condensation is widely applied in various fields because of its high heat transfer efficiency. Condensation oscillation is a major character of submerged steam turbulent jet, and it significantly affects the design and safe operation of industrial equipment. This study is designed to reveal the mechanism of the low-frequency pressure oscillation of steam turbulent jet condensation and determine its affected region. First, pressure oscillation signals with low frequency are discovered in the downstream flow field through oscillation frequency spectrogram and power analysis. The oscillation frequency is even lower than the first dominant frequency. Moreover, the critical positions, where the low-frequency pressure oscillation signals appear, move downstream gradually with radial distance and water temperature. However, these signals are little affected by the steam mass flux. Then, the regions with low-frequency pressure oscillation occurring are identified experimentally. The affected width of the low-frequency pressure oscillation is similar to the turbulent jet width. Turbulent jet theory and the experiment results collectively indicate that the low-frequency pressure oscillation is generated by turbulent jet vortexes in the jet wake region. Finally, the angular coefficients of the low-frequency affected width are obtained under different water temperatures. Angular coefficients, ranging from 0.2268 to 0.2887, decrease with water temperature under test conditions.

Keywords: Submerged steam jet; Pressure oscillation; Half affected width; Turbulent jet vortexes

Nomenclature:			
b	Half jet width, m	N	Sample number
$b_{1/2}$	Half jet characteristic width, m	p	Dynamic pressure, kPa
b_p	Half affected region, m	Δp	Maximum absolute error of dynamic pressure
D	Nozzle diameter, m	p_a	Error of dynamic pressure sensor
E	Oscillation power, $\text{Pa}^2 \cdot \text{s}$	R	Radial distance from nozzle center
f	Oscillation frequency, Hz	S	Intercept of boundary line at X-axis
F	Spectrum function obtained with FFT method	T	Pool water temperature, $^{\circ}\text{C}$
F_k	Discrete spectrum function obtained with FFT method	t	Time, s
G	Steam mass flux, $\text{kg} \cdot \text{m}^{-2} \cdot \text{s}^{-1}$	$U_{\max}$	Axial velocity at central line, m/s
H	Submerged depth of nozzle, m	U	Axial velocity, m/s
k_b	Angular coefficient	X	Axial distance from nozzle outlet
Greek symbols and numerical symbol:			
ω	Angular frequency, $\omega = 2\pi f$, Hz	π	Pi
e	Natural logarithm	i	Imaginary unit

1 Introduction

Submerged steam jet exhibits high heat transfer efficiency and energy exchange capacity; hence, it is widely used in various industrial applications, such as steam-water direct contact heat exchangers, steam ejectors, the passive safety systems in advanced pressurized water reactors, gas welding and rocket engine fuel systems. Especially, in nuclear reactor safety system, steam submerged jet condensation is used to release heat and pressure in the circulation loop of nuclear reactor. In case of rapid transients or beyond design bases, such as loss of the coolant accident, the

safety/relief valve is opened and steam is jetted into the pressure suppression pool (PSP) or in-containment refueling water tank (IRWST) to release heat and pressure. The efficiency of heat and pressure releasing depends on the heat transfer and thermal mixture between steam and water in the suppression pool or water tank. However, steam jet and condensation may also induce thermodynamic and hydrodynamic loads on the structures of relevant systems. Such load should be avoided in industrial applications. Thus, research on characteristics of submerged steam jet is very important in the design and application of related equipment.

Turbulent jet characteristics represent an important research topic in the field of submerged steam jet condensation. Such characteristics greatly affect thermal mixture ability, thermal mixture region and jet entrainment ability. Therefore, the steam jet characteristics are highly concerned. And the velocity distribution, temperature distribution and turbulent jet width have been investigated widely [1-3]. According to previous reports [1-4], the turbulent jet flow region can be divided into several parts, as shown in Fig.1. However, when steam is jetted into the subcooled water, it would be condensed. Then the distribution of turbulent velocity and temperature may be different from that with non-condensation gas jet. Van Wissen et al. [5] measured the velocity distribution of submerged steam jet in the main region with particle image velocimetry (PIV) method. The radial distribution of the time-averaged velocity also showed high self-similarity at the main jet region. Moreover, the turbulent steam jet half characteristic width, which is defined as the width where the flow velocity is equal to $U_{max}/2$ (shown in Fig.1), was measured and the half characteristic width was found to be proportional to the axial distance from nozzle outlet. The proportional coefficient (also known as angular coefficient) was approximated to 0.0938. Kim and Youn [6] measured the velocity and temperature distributions in the jet main region. They found that both the velocity distribution and temperature distribution showed self-similarity characteristics. With the variation of pool water temperatures, the jet angular coefficient ranged from 0.10 to 0.12 under the test conditions. Wu et al. [7] investigated the thermal mixture region and jet entrainment characteristics of submerged steam jet with sonic and supersonic nozzles. They found that the thermal mixture region and half characteristic width of sonic and supersonic jet were smaller than the values in previous reports [5, 6]. The angular coefficient of half characteristic width was only 0.068. Table 1 shows the different angular coefficients of jet half characteristic width reported in previous studies [5-8].

Fig.1 Schematic diagram of steam jet [1-4]

Table 1 Previous works on the jet half characteristic width

As mentioned above, submerged steam jet may cause dynamic loads on the structures of relevant systems. When steam is jetted into water and then condensed by water, it will generate some pressure impulses in water pool. The pressure impulses are harmful to the steam jet devices. In particular, the pressure load may cause resonance phenomena which should be avoided in the industrial operations. To properly design and protect apparatuses, many studies have explored condensation oscillation [9-12]. Chan and Lee [10] investigated flow and condensation patterns under low mass flux. They observed a number of typical condensation patterns and proposed a condensation regime map. Simpson and Chan [13] found the periodic process of bubble growth, bubble translation, and bubble separation (necking) under condensation oscillation region. Nariai

and Aya[14] researched the condensation oscillation frequency with three typical bubble shapes, namely cylindrical, spherical and hemispherical. The predicted frequency showed good agreement with the experimental data. Youn et al. [15] proved that pressure oscillation was caused by steam bubble necking and detached process by synchronizing the high speed CCD camera and pressure transducer. Tanskanen et al. [16] simulated steam condensation for chugging regime. They also simulated and identified the characteristics of bubble size and chugging frequency. Hong et al. [17] investigated the steam condensation oscillation under stable condensation region. A one-dimensional oscillation model was proposed to predict the steam condensation oscillation frequency. Qiu et al. [18] investigated the pressure oscillation characteristics of sonic and supersonic nozzles. Both oscillation intensity and dominant frequency were analyzed. Besides the first domain frequency, the second dominant frequency was found under certain test conditions [19]. Zhao et al. [20] investigated the distribution of pressure oscillation intensity and found that its peak at the steam plume tail. Chong et al. [21] released that the second dominant frequency was caused by separated steam bubbles. They also investigated the oscillation power and amplitude propagation characteristics of the second dominant frequency in the water tank.

Although the condensation oscillation characteristics of the first and second dominant frequencies are very important and widely noted, little effort has been exerted to study steam jet condensation oscillation within the frequency band lower than the first dominant frequency. In the industry, the low-frequency oscillation signals are much easier to induce equipment resonance, and they should be detected and strictly restricted. Furthermore, according to previous studies [1-8, 17-21], the characteristics of steam turbulent jet and condensation oscillation of the first and second dominant frequencies have been widely investigated. However, condensation oscillation and turbulent jet flow were dealt as isolated research fields in the past, despite the close relations of condensation oscillation to flow and thermal mixture characteristic of turbulent jet. For example, according to report of Hong et al. [17], the generation of the first dominant frequency is a joint action of steam jet and the water flow in the jet wake region. In addition, previous studies [2, 4, 8, 22-25] indicated the instability of the turbulent jet boundary and the presence of many vortexes in the jet main region. The unstable jet boundary and the vortexes in the jet wake region will also cause pressure oscillation in the water tank.

Therefore, the characteristic and generation mechanism of pressure oscillation within low frequency band are investigated in the present work. Inherent links and the functioning mechanism between turbulent jet flow and steam jet condensation oscillation are explored preliminarily. With the aid of the well-developed turbulent jet flow characteristic and theory, the generation mechanism of pressure oscillation and its affected region are expected to be identified. This study attempts to improve and supplement condensation oscillation theory, as well as eliminate oscillation load and resonance phenomena for industrial apparatuses. Moreover, this study can offer basic data for the size and structural design of equipment and water tanks.

2 Experimental apparatus and method

2.1 Experimental apparatus

The experimental system of submerged steam jet in quiescent water is shown in Fig.2. Steam is generated in a steam generator and delivered to storage collector. Once the control valve opens, the steam flows through the main pipe and jets into water tank through a nozzle. The straight pipe nozzle with an inner diameter of 8mm is tested. The circulating cooling system is set up to

maintain water temperature in water tank. A high frequency dynamic pressure sensor is used to measure oscillation pressure. The maximum sample rate of dynamic pressure sensor is 20 kHz, and its accurate is 0.25% full span. The time constant of the pressure sensor is 3.7 μ s and the rise time is about 5.2 μ s. The dynamic pressure sensor has been calibrated by the institute of metrology & measurement. The noise level is tested and the maximum noise level is turned out to be about 0.01 kPa. Table 2 presents the test conditions.

Fig.2 Schematic diagram of experimental system and measurement point distribution [21]

Table 2 Test conditions in experiment

The dynamic sensor is located at the water level where nozzle axis is, and it is vertical to the nozzle axis. The position of dynamic pressure sensor can be adjusted with a threedimensional mobile scaffold. The horizontal plane, where nozzle centre line locates, is selected as coordinate plane. The centre of nozzle exit is the coordinate origin. Steam jet direction is defined as the X axis, and nozzle radial direction is defined as the R axis. Also, the dimensionless axis distance and dimensionless radial distance are denoted as X/D and R/D , respectively. The details of measurement point distribution are shown in Fig.3.

Fig.3 Measurement points distribution

2.2 Data analysis method

To analyze the oscillation frequency of steam jet condensation, the Fast Fourier Transform (FFT) method is utilized.

$$F_k = \sum_{n=0}^{N-1} p_n e^{-i \frac{2\pi k}{N} n} \quad (1)$$

where p_n is the instantaneous pressure and N is the total number of sampling number.

The power spectrum is calculated based on frequency spectrum with the Parseval's relation [26],

$$E = \int_{t_1}^{t_2} |p(t)|^2 dt = \frac{1}{2\pi} \int_{\omega_1}^{\omega_2} |F(j\omega)|^2 d\omega \quad (2)$$

where $p(t)$ is the pressure oscillation signal in time spectrum, and t_1, t_2 are the lower and upper time bounds, respectively. $F(\omega)$ is the pressure signal in frequency spectrum that is transformed with the FFT method. ω_1 and ω_2 are lower and upper frequency bounds, respectively.

The uncertainty of the experimental results is analyzed. In the experiment, a high frequency dynamic pressure sensor is used to measure pressure oscillation. The accuracy of pressure sensor is 0.4%. The noise level in measurement environment is not more than 0.01 kPa. According to the uncertainty theory, the maximum absolute error of dynamic pressure is

$$\Delta p = \Delta p_a + \Delta p_b \quad (3)$$

The relative error of dynamic pressure is

$$\frac{\Delta p}{p} = \frac{\Delta p_a + \Delta p_b}{p} \times 100\% \quad (4)$$

where Δp_a and Δp_b are measurement and system uncertainties, respectively. Based on the experimental results, the maximum absolute error Δp is about 0.41 kPa. To calculate the relative error, the minimum root mean square dynamic pressure is used. Finally, the maximum uncertainty of dynamic pressure is determined to be less than 15.2%.

3 Experimental results and discussion

With the high frequency dynamic pressure sensor, the condensation oscillation pressure can be measured. Dealing with the FFT program, the steam jet condensation oscillation frequency is analyzed. As reported in the literature [21], there are several oscillation signals with different frequencies in the frequency spectrogram, such as the first and second dominant frequencies, as shown in Fig.4. The second dominant frequency is generated by the separated steam bubble and the oscillation frequency is relatively high. Moreover, the characteristic of the second dominant frequency have been clearly reported by Chong et al. [21], the pressure oscillation signals, whose frequency values are lower than the second dominant frequency band, is investigated to study the turbulent flow oscillation in this study, as shown in Fig.4.

Fig.4 Frequency spectrograms of condensation oscillation [21]

3.1 Pressure oscillation characteristic with low frequency

Figure 5 displays the distribution of frequency spectrograms along the axial direction at the radial position of $R/D=2$ when $G=600\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$, $T=60^\circ\text{C}$. As shown in the frequency spectrograms, when X/D is small, such as $X/D\leq 6$, only one dominant frequency exists in frequency spectrograms. This dominant frequency is called the first dominant frequency and it is generated by the variations of steam jet length [16, 17]. Experimental results show that with rise of water temperature and steam mass flux, the first dominant frequency decreases monotonously. The value of first dominant frequency ranges from 226 Hz to 492 Hz under test conditions, as shown in Fig.6. In addition, the uncertainty of oscillation dominant frequency is analyzed. The maximum uncertainty of oscillation frequency for the first dominant frequency is no more than 8.5%. When $X/D\geq 6$, oscillation amplitude of first dominant frequency decreases gradually and becomes increasingly small as X/D rise.

However, for the frequency spectrograms at the downstream region (such as $X/D\geq 6$ in Fig.5), oscillation pressure signals also exist in the low frequency band, as noted in the light blue rectangular box in Fig.5. The oscillation pressure signals are defined as low-frequency pressure oscillation signals. The amplitudes of those low-frequency signals also increase with further toward the downstream along axial direction. Moreover, the dominant frequency of the low-frequency pressure oscillation signals decreases with rise of water temperature. The maximum frequency value is approximately lower than 200 Hz.

Fig.5 Frequency spectrograms distribution along the axial direction ($R/D = 2.0$)

Fig.6 First dominant frequency versus steam mass fluxes and water temperatures

Similar to the power distribution of the second dominant frequency reported by Chong et al.

[21], the oscillation power within the frequency band, which is lower than the second dominant frequency band is calculated in the present study with Eq.(2). Fig.7 shows the power axial distribution characteristic of the low frequency region under the radial distances of $R/D=2.5$ and $R/D=3$. In the region before the imaginary dash line, the power increases first and then decreases along the axial direction. An oscillation power peak obviously exists because in the region near the nozzle exit, the first dominant frequency is obvious and plays a dominant role. No other frequency oscillation signals appear or these signals may be too small to be observed. The trend of oscillation power reflects the distribution characteristic of the oscillation power of first dominant frequency band. The amplitude of first dominant frequency shows the same trends as that of oscillation power. However, when the position is far away from the nozzle outlet, such as the region of $X/D \geq 6$ in Fig.7, the oscillation power no longer decreases and begins to increase again. With further rise of the X/D , the oscillation power increases rapidly. This phenomenon is quite different from amplitude and power axial distribution of first dominant frequency. The significant contrast indicates that the pressure oscillation is not only caused by the variation of steam jet length, but also caused by other unknown oscillation sources in the downstream flow field away from $X/D \geq 6$. Thus, both the frequency spectrogram and power distribution can prove that new oscillation signals exists in the downstream flow field and that the oscillation frequency of these new signals are even lower than the first dominant frequency.

(a) $R/D=2.5$

(b) $R/D=3.0$

Fig.7 Oscillation power axial distribution for low frequency region

3.2 Distribution characteristic of low frequency oscillation

The distributions of the low frequency oscillation signals are investigated. First, the characteristics of the frequency spectrograms are studied. By increasing R/D , the initial value of X/D , where the low frequency signals become visible, increase as well. Hence, as the radial distance increases, the position of low frequency signals appearing moves downstream gradually. For example, in the frequency spectrograms of $R/D= 3.0$ and $R/D= 4.0$, the axial positions(X/D) where low-frequency oscillation signals appear is even at the position of $X/D=10$ and $X/D=14$, respectively, as shown in Fig.8(a) and Fig.8(b). The position is obviously farther away from nozzle outlet in comparison with the radial position of $R/D=2$ under the same test condition.

(a) $R/D=3.0, X/D=4\sim14$

(b) $R/D=4.0, X/D=10\sim20$

Fig.8 Frequency spectrograms under radial position of $R/D= 3.0$ and $R/D= 4.0$

However, the initial position of the low-frequency oscillation signals is difficult to determine by directly observing the frequency spectrograms. The observational method is too arbitrary and imprecise. To determine the initial position of the low-frequency oscillation, an objective and mathematical method should be proposed. Therefore, a method based on the calculated oscillation power is used. As shown in Fig.7, the condensation oscillation power of the whole frequency band should be very small in the downstream. However, because of the generation of low-frequency oscillation signals, the total oscillation power changes to increase again. Under the test condition, the oscillation power increases gradually with further rise of X/D .

Moreover, this study analyzes the critical oscillation power positions, which can be regarded

as the initial positions of emerging low-frequency signals, at different dimensionless radial distances. Before the critical positions, the effect of low-frequency pressure oscillation can be neglected. However, after the critical position, the effect of low-frequency pressure oscillation becomes significant. To determine the critical oscillation power positions, the curve of the axial oscillation power distribution is fitted with the least squares method. The local extreme point in the downstream region is obtained with the partial derivative method. Fig.9 shows the turning points of dimensionless axial critical power under different test conditions. Under the same radial position, the critical positions of dimensionless oscillation power move downstream as water temperature increases.

(a) $R/D=2.0$

(b) $R/D=3.5$

Fig.9 Dimensionless axial critical positions of power changing

Figure 10 compares the critical positions of dimensionless axial oscillation power under different radial distances. Clearly, the critical positions of dimensionless axial oscillation power (X/D) increases with rising of dimensionless radial distance. The tendency is consistent with the results observed in frequency spectrograms.

(a)

(b)

Fig.10 Comparison of dimensionless axial critical positions with different radial distances

In addition, the uncertainty of critical position of dimensionless axial power is analyzed. The error of the three-dimensional mobile mounting bracket is about 0.01mm. As a result of the deflection of installed stick, the maximum pressure sensor displacement is no more than 0.77 mm. Then, the maximum uncertainty of the pressure sensor position is about 4.9%.

3.3 Turbulent jet pressure oscillation region

As mentioned previously, the change of the total power trend is caused by the addition of oscillation power of the low-frequency oscillation pressure signals. The axial critical positions in Fig.9 and Fig.10 can be used as a criterion to determine the affected region of low-frequency pressure oscillation. According to the spatial distribution of the critical positions (X/D) of low-frequency oscillation signals shown in Fig.10, the affected region of low-frequency pressure oscillation can be determined by connecting the axial critical turning points at different radial distances. In the affected region, the low frequency pressure oscillation is obvious and should be considered in the industrial applications.

Moreover, according to Fig.9, the low-frequency pressure oscillation affected region is greatly influenced by the water temperature, but is weakly affected by the steam mass flux. Therefore, the low-frequency pressure oscillation regions are investigated separately for different water temperatures. Similar to the half jet characteristic width [5-8], the boundary of pressure oscillation region refers to the dimensionless half affected width of the pressure oscillation, b_p . With rise of axial distance, the pressure oscillation half affected width expands linearly, approximately, as shown in Fig.11. Thus, the half affected width of pressure oscillation can be expressed as Eq.(5). The angular coefficient, k_b , and intercept term, S , can be calculated on the basis of the experimental data under different water temperatures. The assessment of the fit accuracy is presented in Fig.11.

$$b_p = k_b \frac{X}{D} + S \quad (5)$$

Fig. 11 Half affected width of pressure oscillation

According to Fig.11, under different water temperatures, all the intercept terms, S , are located in the front of nozzle exit. The result is similar to that reported by Van Wissen et al.[5]. However, the half affected width of the low-frequency pressure oscillation and the corresponding angular coefficients are quite different when water temperature changes.

Fig.12 compares the angular coefficient of pressure oscillation affected width under different water temperatures. The angular coefficient ranges from 0.2268 to 0.2887. Compared with the turbulent jet velocity distribution, the value of the low-frequency pressure oscillation affected width approximates to 2.5 times those of the half characteristic width of the steam turbulent jet reported by previous studies [5-6, 8], as shown in table 1. In practice, the value of $2.5b_{1/2}$ is usually used to estimate the turbulent jet half width. That is to say, the affected width of the low-frequency pressure oscillation is close to the turbulent jet velocity boundary.

Fig.12 Angular coefficient versus water temperature

The generation mechanism of the low-frequency pressure oscillation signals is analyzed. The pressure oscillation region is similar to the main region of the turbulent jet. Thus, the pressure oscillation must be generated by the main region of turbulent jet. As reported by previous researchers [2, 3, 8, 22-23], several turbulent vortexes and found in the turbulent jet wake region and jet boundary. According to the turbulent jet and turbulent vortex theories, the jet vortexes can cause significant pressure fluctuation. Moreover, the pressure oscillation frequency caused by the turbulent jet vortexes is relatively low. Thus, it can be concluded that the low frequency oscillation pressure of the submerged steam jet is generated by turbulent jet vortexes.

Fig.12 shows that with the rise of water temperature, the angular coefficient of the half affected width of pressure oscillation region, k_b , decreases gradually at first, and then decreases sharply when water temperature ranges from 40°C to 45°C, approximately. Then, a semi-empirical correlation is proposed to predict the angular coefficient of pressure oscillation half width for different water temperatures. Fit accuracy is shown in Fig.12.

$$k_b = -0.0006T + 0.3055 \quad T \leq 40^\circ\text{C} \quad (6)$$

$$k_b = -0.0028T + 0.3921 \quad T > 40^\circ\text{C} \quad (7)$$

The sharp decreasing phenomena can mainly be attributed to the change in flow pattern. As observed with the high speed camera, when water temperature reaches to 40~45°C, the shape of the steam jet condensation would change from an expansion contraction shape to a double expansion contraction shape, which was also observed by Wu et al.[27](Fig.13). The flow pattern map is divided under the condition of water temperature approximated to 40~45°C, which is noted with a red dash line. With the transformation of the steam jet shape, the condensation region reported by Van Wissen et al. [5] changes. The main region and the position of turbulent vortexes change as well. Thus, the turbulent pressure oscillation affected width should adjust

synchronously.

As a result of low-frequency oscillation generated by the turbulent jet vortexes, the flow characteristic and thermal mixture capacity influences the oscillation characteristic. The change in the steam jet pattern reflects the variation in the state of turbulent jet flow and thermal mixture at the initial region, as shown in Fig.1. Then, the state of turbulent jet flow and thermal mixture at the jet wake region change.

Fig.13 Flow pattern map [27]

4 Conclusions

The low-frequency condensation oscillation of the submerged steam turbulent jet is an important component of steam jet condensation oscillation. Its characteristics and its affected region provide information that is necessary in industrial applications. This study experimentally investigates the characteristic of low-frequency condensation oscillation pressure. The affected region is determined and fitted on the basis of the analysis of oscillation power and turbulent jet theory. Moreover, the mechanism of low-frequency condensation oscillation pressure generated by the turbulent vortexes is explored. The main results are summarized as follows:

- (1) The oscillation frequency spectrogram and oscillation power of the low frequency region are acquired. Near the nozzle exit, the power of the first dominant frequency oscillation plays a dominant role and it increases at first and then decreases along the axial direction.
- (2) The newly discovered low-frequency pressure oscillation signals are generated in the jet wake region. The oscillation power increases gradually as axial distance increases. With the rising of radial distance and water temperature, the critical position, where the new oscillation pressure signals appear, moves downstream gradually. However, this movement is barely affected by the steam mass flux.
- (3) The half affected widths of the low-frequency oscillation region are determined under different temperatures. The affected width is approximate to the turbulent jet width, and the angular coefficients of the half affected width decrease with the rising of water temperature. The value ranges from 0.2268 to 0.2887 under test conditions.
- (4) Low-frequency pressure oscillation signals are generated by the turbulent jet vortexes in the jet wake region. The oscillation characteristic and affected region are greatly affected by the turbulent jet speed and thermal mixture capacity. For example, the half affected width of the low-frequency oscillation changes with variations in jet flow pattern.

Based on the analysis of the low-frequency pressure oscillation of the submerged steam jet, the mechanism and characteristics of low-frequency pressure oscillation are revealed, and the affected region is determined. The results enrich the research on the steam jet condensation oscillation. Moreover, this work can aid in the design and safe operation of industrial equipment and thereby help avoid the resonance caused by low-frequency oscillation. The determined relationship between condensation oscillation and turbulent jet characteristics will benefit further research on flow characteristics, the thermal mixture of turbulent jet, and steam condensation oscillation.

Acknowledgements:

This work was supported by the national science foundation of china (No. 51506167, and No. 51476128), the national key basic research development plan (No.2015CB251504).

Figure Captions

Fig.1 Schematic diagram of steam jet[1-4]

Fig.2 Schematic diagram of experimental system[21]

Fig.3 Measurement point distribution

Fig.4 Frequency spectrograms of condensation oscillation [21]

Fig.5 Frequency spectrograms distribution along the axial direction ($R/D=2$)

Fig.6 First dominant frequency versus steam mass fluxes and water temperatures

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Fig.8 Frequency spectrograms under radial position of $R/D= 3.0$ and $R/D= 4.0$

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Fig.10 Comparison of dimensionless axial critical positions with different radial distances

Fig.11 Half affected width of pressure oscillation

Fig.12 Angular coefficient versus water temperature

Fig.13 Flow pattern map [27]

Table Captions

Table 1 Previous works on the jet half characteristic width

Table 2 Test conditions in experiment

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Table 1 Previous works on the jet half characteristic width

Authors	Width	$k_{b_{1/2}}$	Flow field
Van Wissen[5]	$b_{1/2}$	0.0937	Velocity distribution (PIV)
YS Kim[6]	$b_{1/2}$	0.10-0.12	Velocity and Temperature distribution
Wu[7]	$b_{1/2}$	0.068	Velocity and Temperature distribution
YJ Choo[8]	$b_{1/2}$	0.0898–0.979	Velocity (PIV)

Table 2 Test conditions in experiment

Parameters	value	units
Steam mass flux, G	400-800	$\text{kg}\cdot\text{m}^{-2}\cdot\text{s}^{-1}$
Water temperature T_w	20-60	$^{\circ}\text{C}$
Nozzle diameter, D	8	mm
Submerged depth of nozzle, H	500	mm