Pressure oscillation of submerged steam condensation in condensation oscillation regime

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Abstract: The condensation oscillation of submerged steam was investigated theoretically and experimentally at the condensation oscillation regime. It was found that pressure oscillation frequency was consistent with the bubble oscillating frequency and there was a quasi-steady stage when bubble diameters remained constant. A thermal-hydraulic model for the condensation oscillation regime was proposed based on potential flow theory, taking into account the effects of interface condensation and translatory flow. Theoretical derivations indicated that oscillation frequencies were mainly determined by bubble diameters and translatory velocity. A force balance model was applied to the calculation of bubble diameters at quasi-steady stage, and the oscillation frequencies were predicted with the calculated diameters. Theoretical analysis and experimental results turned out that oscillation frequencies at the condensation oscillation regime decreased with the increasing steam mass flux and pool temperature. The predicted frequencies corresponded to the experimental data well with the discrepancies of $\pm 21.7\%$.

Keywords: bubble dynamics; direct contact condensation; frequency; bubble diameter; condensation oscillation

Nomenclature

1. Introduction

Much attention has been focused on the phenomenon of direct contact condensation (DCC) for the wide application in military industry, nuclear power and some other fields. The mass exchange and momentum transfer enhance the energy transfer across the interface between vapor and water resulting in high efficiency of heat transfer. Particularly, the complicated phenomenon was seen in safety system of advanced nuclear power plant, when the reactor pressure was high, the overpressure steam discharged into the subcooled water pool. Pressure oscillation occurs during the injecting process and does harm to apparatuses in the safety system. Therefore, the pressure oscillation of DCC has drawn much attention in the last few decades.

Pressure oscillation of DCC has been discussed by different investigators. In view of various flow patterns at different steam mass fluxes, the investigations can be classified into high steam mass flux regime and low steam mass flux regime. The condensation regime map is shown in Fig. 1.

Fig. 1. Condensation regime map by Cho et al. [1]

(C—chugging, TC—transitional region from chugging to CO, CO—condensation oscillation, SC—stable

condensation, BCO—bubble condensation oscillation, IOC—interfacial oscillation condensation)

The investigations on high steam mass flux are focused on the stable condensation regime. Hong et al. [2] investigated the pressure oscillation in the stable condensation regime, and pointed out that the pressure oscillation was originated by the periodic motion of the steam plume and the dominant frequency was inversely proportional to pool temperature. In addition, he adopted the turbulent jet theory to develop a one-dimensional model which could predict the dominant frequencies well for the steam mass flux between 400 and 900 kg m⁻² s⁻¹. Qiu et al. proposed a theoretical model on the pressure oscillation amplitude [3] and conducted research on pressure oscillation of sonic steam jet in comparison with supersonic jet [4].

The investigations on low steam mass flux include flow patterns of condensation oscillation, transient chugging and chugging. The oscillation at low steam mass flux was found to be caused by bubble motions; however, the phenomenon in condensation oscillation regime was distinct from chugging, in which the pool water entered the nozzle periodically. The two regimes were studied separately in the previous research due to different thermal hydraulics. Simpson and Chan [5] investigated the pressure oscillations of subsonic steam jets and indicated that the dynamics of subsonic jets was constituted by bubble growth, bubble translation, and bubble detachment. An empirical dimensionless correlation was presented to predict the oscillation frequency based on experimental data. Furthermore, Chan and Lee [6] mainly studied the subsonic steam injection at lower mass fluxes which consisted of chugging and oscillatory bubble. They proposed a condensation regime map with approximate boundaries for various flow patterns and observed that pressure oscillations were characterized by large amplitude concurring with bubble detachment and collapse at frequencies lower than 100Hz. Damasio et al. [7] compared pressure oscillation frequencies with vapor–liquid interface fluctuation frequencies measured by electrical resistive probes, and proposed a dimensionless correlation for frequencies in chugging and condensation oscillation regime. Nariai and Aya [8] classifying the oscillating patterns into detailed regimes including internal chugging, small chugging, bubbling at the steam mass flux less than 200 kg m⁻² s⁻¹, and developed a theoretical bubble model for oscillation frequency at low steam mass flux on the basis of the original Rayleigh equation with a modified continuity equation. Youn et al. [9] discussed the dynamic oscillation of high pressure pulses with low frequency in the chugging region experimentally and observed that a negative pressure was originated by sudden condensation after bubble collapse. Li et al. [10] simulated the flow phenomenon at low mass flux by using the VOF model and LES turbulence model, four typical stages—the initial stage, the maximum stability stage, the oscillatory stage and the detachment stage—observed in the simulation were consistent with the experimental results of Chan and Lee [6], however, the calculated frequency didn't match well with the experimental frequency.

In view of the above research, the condensation oscillation was influenced by the periodic bubble motions.

Regarding the bubble dynamics, much work has been devoted to it theoretically. A great number of modifications have been performed on the basic equation to apply to different conditions. Several theoretical models were proposed in the research on the bubble motions at a submerged nozzle or orifice, among which the force balance theory, the Rayleigh equation and the potential flow theory were the most popular. Some analytical studies done by various authors are summarized in Table 1.

Many investigations based on the force balance theory were analyzed with empirical detachment distance of neck [11-14]. Due to the lack of empirical research, the force balance theory was not applicable for the dynamic analysis of vapor bubble motions. Moreover, the generalized Rayleigh equation in [15] was derived from continuity equation which neglected the condensation effect. Therefore, the Rayleigh equation should be modified to apply to the flow conditions in condensation oscillation.

Table 1

Research on bubble motions at a submerged nozzle or orifice

Above all, the pressure oscillation at the condensation oscillation regime was mainly experimentally studied, but there were no abundant photographic results and the mechanism related to the bubble dynamics was still to be revealed. In this paper, we conducted an experimental and theoretical investigation on the pressure oscillation at the condensation oscillation regime. Periodic motions of vapor bubbles and pressure oscillation were recorded in the experiment. A thermal-hydraulic model based on potential flow theory was proposed to describe the bubble oscillation in the regime, taking into account the effects of interface condensation and translatory flow. A force balance model including the condensation effect was introduced to predict the diameter at the quasi-steady stage. Consequently, the oscillation frequencies at different thermal hydraulic conditions were predicted with the calculated diameters and comparisons were made between theoretical and experimental results.

2. Experimental apparatus and method

2.1. Experimental apparatus

The experimental system is similar to the system in [4], ~~shown in Fig. 2~~. Three dynamic pressure sensors were installed on different locations of the tank. All the sensors were at the same submerged depth of nozzle exit. The details of the three sensors' position are shown in Fig. 2. The No.1 sensor was installed on the tank wall with a distance to the axis l_1 , $l_1/D_h=10$. The No.2 sensor was mounted in the axial direction with a distance to the nozzle exit l_2 , $l_2=3$ m. The No.3 sensor was attached to the observation window, the distance to the axis $l_3=0.6$ m.

111 The high speed camera recorded with the frame rate of 5000 fps which was consistent with pressure sampling
112 frequency. The experimental nozzles were straight-pipe nozzles with through hole and the structure of the nozzles
113 are shown in Fig. 3.

114 ~~Fig. 2. Schematic of experimental apparatus [4]~~

115 ~~Fig. 2. Location of dynamics pressure sensors on the wall (top view)~~

116
117 ~~Fig. 3. Structure of the nozzle [24]~~

118 119 2.2. Experimental method and conditions

120 The saturated steam continuously produced from the steam generator flowed through the steam line and was
121 injected into the subcooled water through a nozzle. The steam valve was adjusted to control the steam mass flux.
122 Record the data for 4 seconds at different test points after clearing out the water and gas in the steam line by
123 discharging the pipe for certain time. Then video the steam plumes with sampling frequency of 5000 Hz to obtain
124 the oscillating patterns in an oscillation cycle. Fast Fourier Transform (FFT) was employed to transform the time
125 domain signal to frequency domain in which the dominant frequency could be found more easily. Test conditions
126 are listed in Table 2.

127 **Table 2**

128 Test conditions

129 130 131 2.3. Uncertainties

132 In the experiment, the accuracy of steam flow meter is 1.0%. The accuracy of dynamic pressure sensors is
133 0.25% with the pressure range of -100–100 kPa. Based on the uncertainty analysis of Moffat [25], the maximum
134 uncertainty of oscillation frequency and bubble diameter were 14% and 15% respectively.

135 136 3. Experimental results and discussion

137 3.1. Bubble motions

138 The flow region can be divided into three parts—vapor, water and two-phase region—according to the
139 descriptions in previous research [2, 7, 26]. The schematic diagram of the steam plume at the condensation
140 oscillation regime is shown in Fig. 4. Vapor and water are separated by a boundary named vapor–liquid interface.
141 And the two-phase region is a mixture of condensed vapor and subcooled water flowing downstream.

Fig. 5 exhibits the process of bubble oscillation in a period of 2.6 ms at steam mass flux $300 \text{ kg m}^{-2} \text{ s}^{-1}$ and temperature $35 \text{ }^{\circ}\text{C}$. The periodic oscillation cycle consisted of growth (No. 1–5 in Fig. 5), quasi-steady stage (No. 6–9 in Fig. 5), detachment (No. 10 and No. 11 in Fig. 5), and motions after detachment (No. 11–14 in Fig. 5). During the quasi-steady stage, the bubble diameter remained almost invariant, i.e., the radial oscillating rate of bubble was much less than the translatory velocity [27]. At the quasi-steady stage, bubble deformations were relatively small that the bubble was assumed to be in quasi-steady equilibrium with comprehensive effects of steam momentum and condensation. However, as the neck was elongated, the interface area between vapor and water expanded, enhancing the condensation capability and causing the imbalance. Then, the water intruded on the neck from the radial direction and detached the bubble. In the detachment stage, the bubble underwent fierce turbulences and shrank rapidly [28]. After some deformations, the detached bubble imploded (No. 14 in Fig. 6). At the same time, the remaining attached vapor was cut off from the nozzle exit due to the shock front generated by bubble implosions [29–31], and a new cycle began as another bubble grew from the nozzle.

Fig. 4. Schematic diagram of the steam plume at the condensation oscillation regime

Fig. 5. Sequent Photos at temperature $35 \text{ }^{\circ}\text{C}$ and steam mass flux $300 \text{ kg m}^{-2} \text{ s}^{-1}$ with time interval of 0.2 ms

3.2. Oscillation frequency

The dominant frequencies at the steam mass flux of $150\text{--}400 \text{ kg m}^{-2} \text{ s}^{-1}$ were investigated. Fig. 6 shows the FFT results of pressure oscillation gauged by the dynamic pressure sensors. Although the oscillating pressure varies in amplitudes due to different installation positions, the dominant frequencies tested by the different sensors are almost identical. Additionally, the oscillation frequency 384 Hz corresponds with the bubble oscillation cycle of 2.6 ms in Fig. 5. The diverse frequencies at various test conditions are illustrated in Fig. 7. For the condensation oscillation region, the frequency decreased with the increasing steam mass flux and pool temperature. However, there was a sudden rise of the frequency at the regime transition between condensation oscillation and stable condensation due to the translation of flow patterns which was observed in the experiments, as shown in Fig. 8. When the pool temperature was less than $25 \text{ }^{\circ}\text{C}$, the critical mass flux of the transition was $250\text{--}300 \text{ kg m}^{-2} \text{ s}^{-1}$ (A in Fig. 8); when the pool temperature was $30\text{--}50 \text{ }^{\circ}\text{C}$, the critical mass flux increased to $300\text{--}350 \text{ kg m}^{-2} \text{ s}^{-1}$ (B and C in Fig. 8); when the pool temperature got higher, the critical mass flux rose up to $350\text{--}400 \text{ kg m}^{-2} \text{ s}^{-1}$ (D in Fig. 8). Thus, the critical mass flux for the transients is proportional to the liquid temperature, and the condensation oscillation regime is similar to the one in Fig. 1.

Fig. 6. FFT results at pool temperature 35 °C and steam mass flux 300 kg m⁻² s⁻¹

Fig. 7. Dominant frequencies of 10mm nozzle

Fig. 8. Regime transitions at different temperatures

Note that the increasing steam mass flux decreases the dominant frequency in condensation oscillation. The higher steam mass flux implies the higher jet momentum which intensifies the momentum effect, so it needs longer time for quiescent water to decelerate the jet. As a result, the bubble grows larger for prolonged growth time and the oscillation frequency is lower. It is also illustrated in the previous investigations that the longer oscillation period corresponds with the larger bubble diameters [13, 32]. The same tendency in stable condensation regime is explicated in an analogous theory that the longer condensation length caused by the higher mass flux requires more time for the jet oscillation cycling [2].

4. A thermal–hydraulic model for condensation oscillation

Since the phenomenon of submerged steam condensation is complicated due to the mass exchange and heat transfer across the vapor–liquid interface, it is difficult to describe every detail of the phenomenon. Potential flow theory was applied to develop formula which could predict the flow of gas discharged from an orifice or a hole into a bubble accurately [16, 21]. Fujikawa and Akamatsu [30] conducted numerical research on the collapse of a vapor bubble based on velocity potential equation, taking into account the effects of condensation by introducing a condensation rate term to the continuity equation of velocity potential. However, an attempt to include compressibility of the liquid, heat conduction and temperature discontinuity made the computing procedure quite complicated. Besides, the model in [30] focused on the cavitation bubble dynamics in the absence of translatory flow, and it ignored the translatory velocity of bubble gained from momentum of steam injection. In this paper, some modifications regarding the translatory velocity and condensation were made on the previous model to develop a thermal–hydraulic model for bubble oscillation in condensation oscillation regime, and a simplified prediction for oscillation frequency was obtained analytically.

4.1. Basic assumptions

Based on the photographic results, the bubble oscillation in condensation oscillation regime can be simplified as a periodic oscillator fluctuating in the vicinity of an equilibrium point at quasi–steady stage. The geometric

configuration of steam bubble at the condensation oscillation regime is shown in Fig. 9.

Fig. 9. Schematics of bubble element at quasi-steady stage

Accordingly, the main assumptions are made as follows:

- (1) The bubble remains spherical and thickness of the vapor-liquid interface is negligible.
- (2) Pressure inside the bubble is uniform and follows a polytropic law.
- (3) Condensation occurs only on the interface.
- (4) The external flow near the bubble surface is irrotational and gravitational effects are negligible.
- (5) Translatory velocity of bubble center is considered to be constant during the oscillation cycle (Verified by photographic observations in Fig. 5).

4.2. Derivation of the governing equation

According to the potential flow analysis [16, 18, 21, 33], the potential function for the flow field external to a spherical bubble is

$$\Phi(r, \theta) = \frac{R^2}{r} \frac{dR}{dt} + \frac{UR^3}{2r^2} \cos \theta \quad (1)$$

where R is the bubble radius, U the translatory velocity of bubble center and r, θ the coordinates taking the bubble center as the origin. Here in this paper, the acoustic effects are disregarded due to the relatively low frequencies and small diameters [34].

However, the above velocity potential equation doesn't take into account the effect of interfacial condensation rate. Based on the boundary conditions and continuity at the interface [30]

$$\left(-\frac{\partial \Phi}{\partial r}\right)_{r=R} = \frac{dR}{dt} + \frac{C_r}{\rho} + U \cos \theta \quad (2)$$

where C_r is the rate of condensation, and it can be derived in line with energy conservation [35].

$$C_r = \frac{h_i(T_s - T_\infty)}{h_{fg}} \quad (3)$$

Here in this paper, the heat transfer coefficient h_i adopted the experimental results from [5]. As the interface condensation mainly affected bubble motions in the radial direction, Eq. (1) can be modified as below.

$$\Phi(r, \theta) = \frac{R^2}{r} \left(\frac{dR}{dt} + \frac{C_r}{\rho} \right) + \frac{UR^3}{2r^2} \cos \theta \quad (4)$$

The pressure external to the interface can be derived from the general Bernoulli theorem

$$\frac{p - p_\infty}{\rho} = \frac{\partial \Phi}{\partial t} - \frac{1}{2} v^2 \quad (5)$$

where p is the liquid pressure at coordinate (r, θ) , p_∞ the ambient pressure and v the absolute velocity of the

external liquid flow, $v = \sqrt{(\partial\Phi/\partial r)^2 + (\partial\Phi/r\partial\theta)^2}$. Thus, the governing equation is obtained

$$\frac{p - p_\infty}{\rho} = \frac{R^2}{r} \frac{d^2 R}{dt^2} + \left(\frac{2R}{r} - \frac{R^4}{2r^4}\right) \left(\frac{dR}{dt} + \frac{C_r}{\rho}\right)^2 + \left(\frac{5R^2}{2r^2} - \frac{R^5}{r^5}\right) \left(\frac{dR}{dt} + \frac{C_r}{\rho}\right) U \cos \theta$$

$$+ \left[\frac{(3\cos^2 \theta - 1)R^3}{2r^3} - \frac{(3\cos^2 \theta + 1)R^6}{8r^6} \right] U^2 \quad (6)$$

On the basis of the uniform assumption, the pressure inside the bubble p_b is equal to the sum of the average pressure $\bar{p}_{sur}$ at the interface and the pressure differences caused by surface tension and viscosity.

$$\bar{p}_{sur} = \frac{\int_0^\pi p_{sur} 2\pi R^2 \sin \theta d\theta}{4\pi R^2} = \frac{\int_0^\pi p(R, \theta) \sin \theta d\theta}{2} \quad (7)$$

$$p_b = \bar{p}_{sur} + \frac{2\sigma}{R} + \frac{4\mu}{R} \frac{dR}{dt} \quad (8)$$

Substituting Eq. (5) and Eq. (6) into Eq. (7) gives

$$R \frac{d^2 R}{dt^2} + \frac{3}{2} \left(\frac{dR}{dt} + \frac{C_r}{\rho}\right)^2 - \frac{U^2}{4} + \frac{2\sigma}{\rho R} + \frac{4\mu}{\rho R} \frac{dR}{dt} = \frac{p_b - p_\infty}{\rho} \quad (9)$$

The governing equation is similar to the generalized Rayleigh equation from which no analytical closed-form solutions can be derived [15]. However, the equation can be transformed into the typical equation of a damped harmonic oscillator by letting

$$R = R_b(1 + x), \quad |x| < 1 \quad (10)$$

where R_b is the bubble radius at the quasi-steady stage, x is a time-dependent dimensionless number for the ratio of radius variation, including the bubble growth at the exit and the subsequent bubble motions. The multiplication result of x 's derivatives of time and oscillating periods is in the same magnitude with x . ~~oscillations, and its derivatives are in the same order of magnitude.~~ Substitution of Eq. (9) into Eq. (8) derives

$$R_b^2(1+x) \frac{d^2 x}{dt^2} + \frac{3}{2} R_b^2(1+x)^2 \left(\frac{dx}{dt}\right)^2 + 3R_b(1+x) \left(\frac{C_r}{\rho}\right) \frac{dx}{dt} + \frac{3}{2} \left(\frac{C_r}{\rho}\right)^2 - \frac{U^2}{4} + \frac{2\sigma}{\rho R_b} + \frac{4\mu}{\rho} \frac{dx}{dt} = \frac{p_b - p_\infty}{\rho} \quad (11)$$

According to the pressure equilibrium, bubble pressure at the quasi-steady stage can be acquired by letting $dx/dt = 0, d^2x/dt^2 = 0$

$$p_b^{static} = p_\infty + \frac{2\sigma}{R_b} + \frac{3\rho}{2} \left(\frac{C_r}{\rho}\right)^2 - \frac{\rho U^2}{4} \quad (12)$$

Following the polytropic law, pressure inside the bubble is

$$\begin{aligned}
p_b &= p_b^{static} \left(\frac{R_b}{R} \right)^{3\gamma} \\
&= \left(\frac{1}{1+x} \right)^{3\gamma} \left[p_\infty + \frac{2\sigma}{R_b} + \frac{3\rho}{2} \left(\frac{C_r}{\rho} \right)^2 - \frac{\rho U^2}{4} \right] \\
&= [1 - 3\gamma x + O(x^2)] \left[p_\infty + \frac{2\sigma}{R_b} + \frac{3\rho}{2} \left(\frac{C_r}{\rho} \right)^2 - \frac{\rho U^2}{4} \right]
\end{aligned} \tag{13}$$

Neglecting all powers of x and of its derivatives higher than the first [15, 34], from Eq. (11) and Eq. (13) we obtain

$$\frac{d^2x}{dt^2} + \left(\frac{3}{R_b} \frac{C_r}{\rho} + \frac{4\mu}{\rho R_b^2} \right) \frac{dx}{dt} + \frac{3\gamma}{\rho R_b^2} \left[p_\infty - \frac{\rho U^2}{4} + \frac{2\sigma}{R_b} + \frac{3\rho}{2} \left(\frac{C_r}{\rho} \right)^2 \right] x = 0 \tag{14}$$

In comparison with the equation of a damped harmonic oscillator

$$\frac{d^2x}{dt^2} + 2\beta \frac{dx}{dt} + \omega_0^2 x = 0 \tag{15}$$

where β is the damping constant and ω_0 the natural angular velocity. Condensation and viscosity are the two main factors for the damping $\beta = \beta_{con} + \beta_{vis}$, in which $\beta_{con} = \frac{3C_r}{2\rho R_b}$ and $\beta_{vis} = 2\mu/\rho R_b^2$ are the effects of interfacial condensation rate and viscosity respectively.

The natural frequency f is defined as below

$$f = \frac{\omega_0}{2\pi} = \frac{1}{2\pi R_b} \sqrt{\frac{3\gamma}{\rho} \left[p_\infty - \frac{\rho U^2}{4} + \frac{2\sigma}{R_b} + \frac{3\rho}{2} \left(\frac{C_r}{\rho} \right)^2 \right]} \tag{16}$$

Four terms within the radical indicate four different factors dominating the oscillation process: p_∞ implies the effect of ambient pressure, $\rho U^2/4$ the momentum of external liquid gained by momentum transfer, $2\sigma/R_b$ the effect of surface tension and $3C_r^2/2\rho$ represents the influence of condensation rate at the interface. ~~Among the four influence factors, surface tension and condensation rate at the interface are relatively small so that the two terms can be neglected for simplification.~~

According to the calculation results at thermal hydraulic conditions of condensation oscillation regime, the oscillation frequency is primarily dependent on the bubble diameter and translatory velocity of bubble center. Furthermore, these two parameters are closely relevant that the frequency calculation can be developed into a set of simultaneous equations in which there exists only one variable determined by thermal hydraulic conditions.

4.3. Force balance model for diameters

The bubble diameter at quasi-steady stage can be calculated from translatory velocity based on the force balance. According to the models proposed in [12] and [14], the bubble is subject to six forces in the horizontal direction, shown in Fig. 10. The buoyancy force is so small to the other forces that force analysis in the vertical is

279 ignored.

280

281 **Fig. 10. Forces acting on the bubble**

282 The overall force balance equation is

283
$$F_p + F_m = F_s + F_d + F_i + F_{con} \quad (17)$$

284 The theoretical descriptions of each force are illustrated as follows:

285 Pressure force: $F_p = \frac{\pi}{4} D_h^2 (p_s - p_\infty)$, where D_h is the nozzle diameter and p_s is the steam pressure at nozzle
286 exit.

287 Vapor momentum force: $F_m = \frac{\pi}{4} D_h^2 \rho_s v_s^2$

288 Surface tension force: $F_s = \pi D_h \sigma$

289 Drag force: $F_d = \frac{\pi}{4} D^2 c_d \frac{\rho v^2}{2}$, where D is the bubble diameter, v the external flow velocity, and c_d the drag
290 coefficient. In accordance with the bubble velocity measured in the experiments, the relevant Reynolds numbers is
291 about 2×10^5 . By adopting the imperial correlation $c_d(\text{Re}, A_c) = (1 + 0.25 A_c) c_d(\text{Re}, 0) + 0.55 A_c$ from [36], where
292 $c_d(\text{Re}, 0) = 0.5$ and A_c is acceleration number counted as 0.2 in the experimental conditions, the drag coefficient
293 is calculated as 0.635.

294 Inertia force: $F_i = (\rho_s V + \rho V_l) a$, where a is the bubble acceleration, V is the vapor bubble volume and V_l is
295 the liquid volume attached to the interface between vapor and water which is $0.5V$ in theory [36, 37].

296 Force due to condensation: $F_{con} = \frac{\pi D^2}{4 \rho_s} \left[\frac{h_i (T_s - T_\infty)}{h_{fg}} \right]^2$ [23]. It can be substituted as $F_{con} = \frac{\pi D^2 C_r^2}{4 \rho_s}$ with
297 Eq. (4).

298 In accordance with the discussion in Section 3.1, the bubble volume remains invariant in the quasi-steady
299 equilibrium period so that the radial oscillating rate equals zero and $v=U$. Based on the assumption (4) in Section
300 4.1, $a=0$. Substitution gives

301
$$\left(\frac{2C_r^2}{\rho_s} + c_d \rho U^2 \right) D^2 = 2D_h^2 (p_0 - p_\infty + \frac{1}{2} \rho_s v_s^2) - 8D_h \sigma \quad (18)$$

302 where $p_0 = p_s + \frac{1}{2} \rho_s v_s^2$ is the stagnation pressure.

303 Therefore, the theoretical bubble diameters at quasi-steady stage will be derived from Eq. (18).

304

4.4. Verification of model

The theoretical model proposed in Section 4.2 predicts the frequency of the duration from bubble growth to the formation of new bubble from the exit, which is consistent with dominant frequency of pressure oscillation.

Based on the fact that the damping coefficient β is smaller than the natural angular velocity ω_0 at the thermal hydraulic conditions in this experiment, the generalized solution to Eq. (15) is

$$x = Ae^{-\beta t} \sin(\omega t + \varphi) \quad (19)$$

where $\omega = \sqrt{\omega_0^2 - \beta^2}$ and there are two coefficients needed to be determined by the experiment.

Here, the oscillation process in Fig. 5 is discussed. In light of the bubble motions described in Section 3.1, the processes before and after the bubble detachment are quite different. The thermal hydraulic parameters also witness great variations between the two stages, for example, the time-average heat transfer coefficient after the detachment is about 4 times of that at the growth and transformation stages [5]. Therefore, the analysis is divided into two stages. The period before detachment is defined as stage 1 and the one after detachment is stage 2.

The dominant frequency measured by FFT results is 384Hz. Critical time point is obtained in Fig. 5: detachment time $t_d = 1.8\text{ms}$. The natural angular velocity ω_0 for the whole oscillation process is identical to 2562.26 rad/s, therefore, the calculated period is 2.452ms and the deviation of frequency prediction is 6.2%.

Stage 1: Based on the heat transfer coefficient in [5], the damping coefficient is $\beta_1 = 4.6768$. The theoretical solution can be obtained by adopting the experimental radius to calculate the undetermined coefficients, $A_1 = -0.3020$, $\varphi_1 = 1.9447$.

Stage 2: The heat transfer coefficient increases from 0.496 to 1.826MW m⁻² K⁻¹. Accordingly, the damping coefficient $\beta_2 = 17.1283$ and other coefficients are calculated as $A_2 = -0.4244$, $\varphi_2 = 1.8689$.

The comparisons between radius variations of theoretical and experimental at different time are illustrated in Fig. 11.

Fig. 11. Comparison between radius variations of theoretical and experimental at steam mass flux 300 kg m⁻² s⁻¹ and temperature 35 °C

It can be seen from the figure that the oscillating trends of theoretical and experimental results are consistent despite some deviations in certain points. Below are corresponding explanations.

At the beginning of Stage 1 (0ms), bubble growth is in its hemispherical shape. However, the bubble radius is calculated with the method of discrete integral in 2-D binary image [28]. Therefore, the counted radius is less than the real one.

In the middle of Stage 1 (1.0–1.4ms), the discrepancy is caused by the linear treatment [15, 34] for the governing equations—neglecting all powers of x and of its derivatives higher than the first. As the amplitude of x rises, the deviation increases. However, this amplitude has little influence on the oscillation period. The calculated trend of variations shows good agreement with the motions of bubble radius at Stage 1.

At Stage 2 (1.8–2.452ms), the deviations after the detachment may be caused by the dramatic increase of capillary waves formed on the bubble surface. These waves intensify the roughness of bubble surface, which greatly enhances the shear stress between bubbles moving with translatory velocity and motionless cool bulk [28, 29]. Thus, the viscosity near the interface is the dominant factor for the dissipation caused by shear stress. However, this factor is inevitably absent due to the lack of relevant research. Therefore, the damping coefficient should be much greater than the one calculated with normal hydraulic parameters. Moreover, the viscosity increases greatly with the time, which accelerates the oscillation process [28, 29]. The above two factors will induce parametric discrepancies between experiments and predictions at Stage 2.

Above all, due to the fact that $\beta \ll \omega_0$, the damping effects only influence the variation amplitude and have little impact on the oscillation frequency. Thus, the angular velocity $\omega = \sqrt{\omega_0^2 - \beta^2}$ can be calculated as ω_0 approximately. In conclusion, the analytical results agree reasonably with the experimental, the discrepancies between the experimental and calculated frequency are within $\pm 6.2\%$.

5. Comparison of experimental and theoretical results

5.1. Diameters at quasi-steady stage

The diameters at quasi-steady stage were computed based on Eq. (18), in which the translatory velocity U is measured in a sequence of processed photos with time interval of 0.2ms. The image processing procedure was similar to the one in Tang's research [28]. The identification errors of the bubble centroid are within ± 1 pixel. Thus, the location error is less than ± 0.182 mm (110 pixels for 20 mm in the captured photo) and the velocity error is less than ± 0.9091 m s⁻¹. Fig. 12 exhibits the measured bubble velocity with error bars at steam mass flux of 250 kg m⁻² s⁻¹.

To verify the force balance model, diameters at quasi-steady stage were also measured in processed images. Fig. 13 shows the measured and calculated diameters at the steam mass flux of 250 kg m⁻² s⁻¹, and it indicates that the bubble diameters are larger when the pool temperature increases. On thermal dynamics, since larger heat transfer area is expected to balance the greater latent heat with the decreasing temperature difference, the bubble diameter is larger with the increasing pool temperature. Moreover, it will take longer distance for the external

365 liquid to attenuate the greater steam momentum with the increasing steam mass flux at the nozzle exit, leading to
366 larger growth diameters. Above all, the diameters show a tendency of increment with the increasing pool
367 temperature and steam mass flux in light of Eq. (18).

368

369 **Fig. 12.** Measured bubble velocity at the steam mass flux of $250 \text{ kg m}^{-2} \text{ s}^{-1}$

370

371 **Fig. 13.** Bubble diameters at the steam mass flux of $250 \text{ kg m}^{-2} \text{ s}^{-1}$

372 **Fig. 14** shows the comparison between theoretical and experimental diameters. The theoretical diameters
373 agree with the experimental well, and the discrepancies are within $\pm 18.3\%$.

374

375 **Fig. 14.** Discrepancies between experimental diameters and predictions using Eq. (18)

376

377 5.2. Oscillation frequency

378 The oscillation frequencies are computed by combining ~~the theoretical equations of frequencies~~ Eq. (16) and
379 ~~the predicted~~ bubble diameters at quasi-steady stage in Section 5.1. The predicted frequencies and experimental
380 results are shown in Fig. 15. The frequencies vary from 200 to 550 Hz for the 8 mm nozzle, and the values are
381 greater than that of 10mm nozzle in the same experimental conditions because bubbles released from 10mm
382 nozzle are larger. The oscillation frequency decreases with higher pool temperature because lower condensing
383 capability of subcooled water prolongs the condensation process, which can be also inferred from the theoretical
384 analysis that larger diameter at higher pool temperature results in lower frequencies. Additionally, the frequency
385 shows an increasing tendency as steam mass flux intensifies. Apart from the fact that higher steam mass flux
386 enlarges the bubble diameter, greater steam momentum will enhance the momentum of external flow. Therefore,
387 the oscillation frequency will decrease based on Eq. (16).

388

389 **Fig. 15.** Frequencies at different test conditions

390 **Fig. 16** exhibits the comparisons between results of simultaneous equations and experiments, and the
391 discrepancies are within $\pm 21.7\%$.

392

393 **Fig. 16.** Prediction accuracy of simultaneous equations for oscillation frequency

394

395 6. Conclusions

396 In this study, the pressure oscillation at the condensation oscillation regime was investigated experimentally
397 and theoretically. Periodic bubble motions were recorded by the high speed camera and oscillation frequencies
398 were obtained by Fast Fourier Transform of the original pressure signal in the experiment. Theoretically, a
399 thermal–hydraulic model based on potential flow theory was presented to characterize the bubble motions at the
400 condensation oscillation regime. The major results are summarized as follows:

- 401 (1) The experimental investigations indicated that pressure oscillation frequency was consistent with the
402 bubble oscillating frequency and the bubble underwent a quasi–steady stage during the oscillation.
- 403 (2) A thermal–hydraulic model based on potential flow theory was developed for the condensation
404 oscillation regime, integrating the effects of condensation and translatory flow. The predicted
405 frequencies using the calculated diameters fit the experimental data well with the deviations less than
406 $\pm 21.7\%$.
- 407 (3) The bubble diameter and translatory velocity were the dominant variables for frequency calculation in
408 light of derived formula. The frequency at condensation oscillation regime was found to decrease with
409 the increasing steam mass flux and pool temperature theoretically and experimentally.
- 410 (4) A force balance model including the condensation effect was established to predict the bubble diameters
411 at the quasi–steady stage. The predictions agree well with the experimental diameters, and the
412 discrepancies are within $\pm 18.3\%$.

413

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417

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492

493 **Figure Captions**

494 **Fig. 1.** Condensation regime map by Cho et al. [1] (C—chugging, TC—transitional region from chugging to CO,
495 CO—condensation oscillation, SC—stable condensation, BCO—bubble condensation oscillation,
496 IOC—interfacial oscillation condensation)

497 ~~**Fig. 2.** Schematic of experimental apparatus [4]~~

498 **Fig. 2.** Location of dynamics pressure sensors on the wall (top view)

499 **Fig. 3.** Structure of the nozzle [24]

500 **Fig. 4.** Schematic diagram of the steam plume at the condensation oscillation regime

501 **Fig. 5.** Sequent Photos at temperature 35 °C and steam mass flux $300 \text{ kg m}^{-2} \text{ s}^{-1}$ with time interval of 0.2 ms

502 **Fig. 6.** FFT results at pool temperature 35 °C and steam mass flux $300 \text{ kg m}^{-2} \text{ s}^{-1}$

503 **Fig. 7.** Dominant frequencies of 10mm nozzle

504 **Fig. 8.** Regime transitions at different temperatures

505 **Fig. 9.** Schematics of bubble element at quasi—steady stage

506 **Fig. 10.** Forces acting on the bubble

507 ~~**Fig. 11.** Comparison between radius variations of theoretical and experimental at steam mass flux $300 \text{ kg m}^{-2} \text{ s}^{-1}$
508 and temperature 35 °C~~

509 ~~**Fig. 12.** Measured bubble velocity at the steam mass flux of $250 \text{ kg m}^{-2} \text{ s}^{-1}$~~

510 **Fig. 13.** Bubble diameters at the steam mass flux of $250 \text{ kg m}^{-2} \text{ s}^{-1}$

511 **Fig. 14.** Discrepancies between experimental diameters and predictions using Eq. (18)

512 ~~**Fig. 15.** Frequencies at different test conditions~~

513 **Fig. 16.** Prediction accuracy of simultaneous equations for oscillation frequency

514

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527
528

529 **Fig. 5.** Sequent Photos at temperature 35 °C and steam mass flux 300 kg m⁻² s⁻¹ with time interval of 0.2 ms

530

531

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548 and temperature $35 \text{ }^{\circ}\text{C}$

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555

556

557 **Fig. 14.** Discrepancies between experimental diameters and predictions using Eq. (18)

558

559

560 **Fig. 15.** Frequencies at different test conditions— $250\text{ kg m}^{-2}\text{ s}^{-1}$

561

562

563 **Fig. 15.** Frequencies at different test conditions— $300\text{ kg m}^{-2}\text{ s}^{-1}$

564

565

566 **Fig. 16.** Prediction accuracy of simultaneous equations for oscillation frequency

567

568

569 **Table Captions**

570 **Nomenclature**

571 Table 1 Research on bubble motions at a submerged nozzle

572 **Table 2 Test conditions**

573

Nomenclature			
c_d	drag coefficient	R_b	bubble radius at quasi-steady stage (m)
C_r	rate of condensation ($\text{kg m}^{-2} \text{s}^{-1}$)	r	distance from bubble center to external liquid (m)
D	bubble diameter (m)	T_s	steam saturation temperature (K)
D_h	nozzle diameter (m)	T_∞	pool water temperature (K)
F_{con}	condensation force (N)	U	translatory velocity of bubble center (m s^{-1})
F_d	drag force (N)	v	velocity of external liquid flow (m s^{-1})
F_i	inertia force (N)	V	vapor velocity at the nozzle exit (m s^{-1})
F_m	steam momentum force (N)	x	dimensionless parameter for oscillation
F_p	pressure force (N)		
F_s	surface tension force (N)		
f	oscillation frequency (s^{-1})	<i>Greek symbols</i>	
G	steam mass flux ($\text{kg m}^{-2} \text{s}^{-1}$)	β	damping constant (s^{-1})
g	gravity (m s^{-2})	β_{con}	condensation damping constant (s^{-1})
h_{fg}	latent heat (J kg^{-1})	β_{vis}	viscosity damping constant (s^{-1})
h_i	heat transfer coefficient ($\text{J m}^{-2} \text{K}^{-1} \text{s}^{-1}$)	σ	surface tension of water (N m^{-1})
l	distance from the location of the pressure sensor to the axis (m)	θ	angle between the position vector and the horizontal axis of symmetry
p	liquid pressure of external flow (Pa)	μ	viscosity (N s m^{-2})
p_b	bubble pressure (Pa)	γ	adiabatic constant ($\gamma=1.32$)
p_s	vapor pressure at nozzle exit (Pa)	ρ	water density (kg m^{-3})
p_0	vapor stagnation pressure (Pa)	ρ_s	steam density (kg m^{-3})
p_∞	ambient pressure (Pa)	ω	angular velocity (s^{-1})
R	bubble radius (m)	ω_0	natural angular velocity (s^{-1})

Investigators	Gas	Theoretical Model	Characteristic
McCann and Prince [16]	Air	Potential flow theory Orifice equation	Bubble size, frequency and weeping rates Criterion for paring and doublet formation
Ramakris.S et al. [17]	Air	Force balance	Two step mechanism of expansion stage and detachment stage
Wraith and Kakutani [18]	Air	Potential flow theory	Governing equation of two spheres moving in parallel Maximum orifice pressure occurring at cut-off point
Marmur and Rubin [11]	Air	Navier–Stokes equation	Specified differential interface element method with “added mass” concept
Pinczewski [19]	Air	Rayleigh equation	Assumption of a circulating toroidal vortex inside the connected bubble
Gaddis and Vogelpohl [12]	Air	Force balance	Bubble detachment diameter prediction with extensive validity
Longuethiggins et al. [13]	Air	Force balance Curvature function	Theoretical calculation of oscillation frequency Formation of a jet following detachment
Oguz and Prosperetti [20]	Air	Potential flow theory	Boundary integral numerical solution with validity to very small bubbles
Zhang and Shoji [14]	Air	Force Balance	Influence of previous bubble wake on aperiodic bubble formation
Zhang and Tan [21]	Air	Potential flow theory Weeping theory	Modification to potential function Prediction of wake pressure
Guy and Ledwidge [22]	Vapor	Rayleigh equation Mass transfer equation	Numerical simulations for nonspherical bubble Mass transfer equation for evaporation
Clerx and Van der Geld [23]	Vapor	Mass conservation Force Balance	Added mass coefficients dependent on distance to wall Added term for momentum loss due to condensation

579 **Table 2**
580 **Test conditions**

Parameters	Value
Steam mass flux G , $\text{kg m}^{-2} \text{s}^{-1}$	150, 200, 250, 300, 350, 400
Steam initial pressure p_0 , MPa	0.12–0.32
Pool temperature T_∞ , °C	15–60
Exit diameter of nozzle D_h , mm	8.0, 10.0
Submerged depth of nozzle exit H , m	0.5
Ambient pressure p_a , MPa	0.097
Pool size(length×width×height), m×m×m	3×1.2×1

581