Flow boiling heat transfer enhancement under ultrasound field in minichannel heat sinks

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A B S T R A C T

The enhancement of the heat transfer assisted by ultrasound is considered to be an interesting and highly efficient cooling technology, but the investigation and application of ultrasound in minichannel heat sinks to strengthen the flow boiling heat transfer are very limited. Herein, a novel installation of ultrasound transducers in the flow direction of a minichannel heat sink is designed to experimentally study the characteristics of heat transfer in flow boiling and the influence of operation parameters (e.g., heat flux, mass flux rate) and ultrasound parameters (e.g., frequency, power) on the flow boiling heat transfer in a minichannel heat sink with and without ultrasound field. Bubble motion and flow pattern in the minichannel are analyzed by high-speed flow visualization, revealing that the ultrasound field induces more bubbles at the same observation position and a forward shift of the onset of nucleation boiling along the flow direction, as ultrasonic cavitation produces a large number of bubbles. Moreover, bubbles hitting the channel wall on the left and right sides are found, and the motion speed of the bubbles is increased by 31.9% under the ultrasound field. Our results demonstrate that the heat transfer coefficient obtained under the ultrasound field is 53.9% higher than in the absence of the ultrasound field under the same conditions, and the enhancement ratio is decreased in the high heat flux region due to the change of the flow regime with increasing heat flux. This study provides a theoretical basis for the application of an ultrasound field in minichannel heat sinks for the enhancement of flow boiling heat transfer.

1. Introduction

Enhancement of the dissipation of large amounts of heat from small surfaces is required for many rapidly developing modern applications, such as hybrid vehicle power electronics, avionics, as well as laser and microwave military systems. Minichannel heat sink combining flow boiling heat transfer technology is an ideal solution for the heat dissipation of the high heat flux facility. However, due to the increasing power density, the thermal performance of the radiator still needs to be further improved, which makes thermal management a great challenge for the future [1]. In recent decades, many novel techniques have been used in micro/minichannel heat sinks to further improve the heat transfer performance, such as passive techniques, e.g., mechanical sanding technique [2], coating technique [3], and nanofluids [4], and active techniques, e.g., electric field [5], magnetic field [6], and acoustic field [7]. Since 1965, the research of Bergles et al. focused on the enhancement of the heat transfer performance of water by the ultrasound field, and ultrasound has been increasingly applied in heat transfer fields [8]. Nevertheless, the ultrasound field has not attracted much more attention until 2000 as a possible thermal enhancement technology. Legay et al. [9] summarized that the research on the enhancement of heat transfer by ultrasound in the years from 2000 to 2010 mainly focused on the field of pool boiling and convection heat transfer, while relatively few studies have been reported on the enhancement of flow boiling heat transfer.

Convection heat transfer under the ultrasound field has been the focus of an increasing number of studies in the past 20 years. Tam et al.
[10] placed three ultrasonic oscillators in a horizontal stainless-steel tube to study the influence of ultrasound on convective heat transfer, achieving an improvement of the heat transfer performance by about 10%. Rahimi et al. [11] studied the influence of oscillator arrangement methods (three oscillators placed on the bottom of the container, two oscillators placed on the container wall) on convection heat transfer in a pool and found that the heat transfer coefficient of the oscillator arrangement in the container wall was higher than that at the bottom of the container. Tajik et al. [12] reported that the ultrasound field can increase the heat transfer coefficient to 390%, and the enhancement of the heat transfer increased with increasing ultrasonic power. Legry et al. [9] compared the heat transfer performance of the heat sink with ultrasonic field with that without ultrasound field, showing that the heat transfer performance improved by 2.3 times. Gondrexon et al. [13,14] and Dhanalakshmi et al. [15] also showed that the convective heat transfer could be enhanced by the ultrasound field.

In the presence of an ultrasound field, pool boiling presents one of the most studied modes of heat transfer enhancement [16,17]. Most studies mainly focus on the influence of acoustic radiation distance [18], vibration frequency [19], ultrasonic vibration power, and boiling surface structures on the boiling heat transfer [21]. According to the above studies and based on experimental observation, acoustic streaming and acoustic cavitation are the main reasons for the enhancement of the heat transfer coefficient [22,23]. Wong et al. [19] conducted a pool boiling experiment using water and ethanol as the working medium at ultrasonic frequencies from 20.6 to 306 kHz, and the results showed that the enhancement of the heat transfer coefficient in the nucleate boiling region can be ignored, and the natural convective heat transfer coefficient increased by 8 times. The heat transfer coefficient enhancement was caused by acoustic streaming turbulence and acoustic cavitation bubbles, as observed by a high-speed camera. Baffigi et al. [20] experimentally explored the influence of an ultrasound field (the ultrasound frequency was 40 kHz, and the ultrasonic oscillator power was in the range of 300 to 500 W) on the pool boiling heat transfer. The increment of the heat transfer coefficient under saturated pool boiling was smaller than that under subcooled boiling. Moreover, in the high heat flux region, the critical heat flux could be reduced in the presence of the ultrasound field but not in the absence of the ultrasound field. Zhang et al. [24] concluded that the mechanism of heat transfer improvement in the acoustic field is different in regions of low and high heat flux. In the region of low heat flux, the heat transfer improvement was attributed to acoustic field cavitation bubbles, while in the region of high heat flux, it was attributed to acoustic streaming. Li et al. [25] investigated the influence of the acoustic field on the heat transfer characteristics of three copper-tube surface structures (smooth surface, fish-scale structure, and thread structure), revealing that acoustic streaming could increase the bubble escape frequency.

Although some important studies have been reported on the heat transfer enhancement by ultrasound, their focus was on the field of pool boiling and convection heat transfer. Relatively few studies focus on the introduction of an ultrasound field to enhance the flow boiling heat transfer and applied the ultrasound field to micro-mini-channel heat sinks to strengthen the flow boiling heat transfer. Therefore, the heat transfer mechanism of the ultrasound field in minichannels has not been elucidated yet. In the present paper, a novel installation of ultrasound transducers in the flow direction of the minichannel heat sink is designed (this novel installation has been patented, No. ZL201721270954.X). Bubble movement and flow pattern in the minichannel with and without ultrasound were analyzed by high-speed visualization. In addition, we studied the effect of operation parameters (e.g., heat flux, mass flux rate) and ultrasound parameters (e.g., frequency, power) on the enhancement of the heat transfer. The main purpose of this paper is to study the feasibility of the ultrasound field in enhancing the flow boiling heat transfer in minichannel heat sinks and to provide a theoretical basis for the application of an ultrasound field in flow boiling heat transfer.

2. Experimental apparatus and method

2.1. Experimental system

The experimental system with ultrasound transducers for flow boiling heat transfer is shown in Fig. 1. The experimental system was mainly composed of a heating device control system, data acquisition system, ultrasound generation system, and liquid circulation control system. The heating device was composed of a power regulator, power display, six single-headed heating tubes, and a heating body. The rated power of each heating tube was 150 W. The data acquisition system was composed of a computer, high-speed visualization camera, Agilent 34970A data acquisition instrument, thermocouples, and pressure sensors. The signal output of two parallel rows of thermocouples with an accuracy of 0.1 °C on the test section was recorded by a data acquisition instrument (Agilent-34970A). At the inlet and outlet of the test section, pressure sensors (HC3160-HVG4) were installed to measure the pressure values in the flow boiling process. The measuring range of the HC3160-HVG4 pressure sensor was 0–100 kPa with an accuracy of 0.1 kPa. The ultrasound field generation system was mainly composed of an ultrasonic generator (KMD-D1) and ultrasonic transducers. The rated power of the generator was 1000 W, and the frequency range was 20–60 kHz. The liquid circulation control system included a liquid injection device, fluid reservoir, variable-frequency pump, preheating tank, and cooling device. The liquid circulation control system was a closed vapor–liquid conversion loop system. The working medium was injected into the fluid reservoir, and then the liquid was driven by the variable-frequency pump. The liquid was preheated to the required inlet temperature by the preheating tank, and then the liquid was transported to the

### Nomenclature

| Symbol | Description |
|--------|-------------|
| $A_i$  | Heating area, m$^2$ |
| $c_{p,l}$ | Liquid specific heat, J/(kg·K) |
| $M$ | Mass flow rate, kg/s |
| $f$ | Ultrasonic frequency, kHz |
| $G$ | Mass flux, kg/(m$^2$·s) |
| $h_{tu}$ | Heat transfer coefficient without sound field, kW/(m$^2$·K) |
| $h_{tu}$ | Heat transfer coefficient with sound field kW/(m$^2$·K) |
| $h_{lg}$ | Latent heat of vaporization, J/kg |
| $H_{ch}$ | Channel depth, m |
| $k_{al}$ | Aluminum thermal conductivity, W/(m·K) |
| $k_{st}$ | Heat sink thermal conductivity, W/(m·K) |
| $k_{sr}$ | Silica thermal conductivity, W/(m·K) |
| $L_{sub}$ | Subcooled section length, m |
| $q_e$ | Heat flux, kW/m$^2$ |
| $Q$ | Input power produced, W |
| $Q_l$ | Heat absorbed by working fluid, W |
| $P_u$ | Ultrasonic power, W |
| $T_w$ | Wall temperature, °C |
| $T_{in}$ | Inlet temperature, °C |
| $T_{out}$ | Outlet temperature, °C |
| $T_{sat}$ | Fluid saturation temperature, °C |
| $T_f$ | Upper wall temperature, °C |
| $W_d$ | Minichannel interval spacing, m |
| $W_{ch}$ | Channel spacing, m |
| $x_{out}$ | Outlet vapor quality |
| $Z_r$ | Distance between the $r$ temperature test points and inlet, m |

### Symbols

- $h_{tu}$: Heat transfer coefficient without sound field
- $h_{tu}$: Heat transfer coefficient with sound field
- $h_{lg}$: Latent heat of vaporization
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- $Z_r$: Distance between the $r$ temperature test points and inlet
experimental section. In the experimental section, the working medium was heated by a heating device control system to form a vapor–liquid two-phase mixture. This vapor–liquid two-phase mixture was condensed into a liquid by a condenser and finally transferred back to the fluid reservoir under the driving force of the pump.

2.2. Test section

Fig. 2 (a) shows the schematic exploded view diagram of the test section, which was composed of five transducers, a stainless-steel upper cover plate with four visualization windows, four Pyrex glasses, a 3D printing minichannel heat sink, a sealing ring, Teflon housing, an aluminum alloy heating element, 12 pairs of bolts and nuts, inlet and outlet pressure meter, inlet and outlet thermocouple temperature sensors, two rows of twelve thermocouple temperature sensors, a lower cover plate, and six cartridge heaters. Fig. 2 (b) shows the cross-sectional view of the assembled test section. The heating element was packaged in Teflon housing to prevent heat loss. The minichannel heat sink was placed at the top center of the heating element. Five ultrasonic transducers were mounted on the upper cover plate (the distance between the ultrasonic transducers is shown in Fig. 2(b)), and four visualization windows were located between the five ultrasonic transducers. The thickness of the upper cover plate was 1.5 mm in the installation area of the transducer, and the ultrasonic transducers were connected to the ultrasonic generator. In the inlet and outlet plenum, temperature sensors and pressure meters were installed to measure the inlet and outlet temperature and pressure, respectively. Local temperature test points of a test section along the flow direction were measured using K-type thermocouples. The distances between the measuring points are shown in Fig. 2(b). The distance between the measuring points at the bottom and upper wall temperature $T_{j}$ of the minichannel was 8.61 mm. The position relationship among transducers, visualization windows, and temperature measurement points is shown in Fig. 2(b).

2.3. Estimation of heat loss

To verify the accuracy of the data in the experimental process, the heat loss needed to be estimated before the flow boiling heat transfer experiment. Pure refrigerant R141b was used for the flow heat transfer experiment to obtain the heat loss ratio ($\varepsilon$). It was necessary to ensure that the working medium experienced no phase change during the whole experimental process. The heat loss ratio ($\varepsilon$) was calculated by Eq. (1), where $Q$ is the total input power, $Q_e$ is the heat absorbed by the pure refrigerant R141b, $M$ is the mass flux, $c_{p,l}$ is the specific heat capacity, $T_{in}$ is the inlet plenum liquid temperature, and $T_{out}$ is the outlet plenum temperature.

$$\varepsilon = \frac{|Q - Q_e|}{Q} \times 100\%$$  \hspace{1cm} (1)

$$Q_e = M c_{p,l} (T_{out} - T_{in})$$  \hspace{1cm} (2)

Fig. 3 shows the experimental results of the heat loss estimation at the mass flow rates of 237.8 and 267.6 kg/(m² s). The heat loss ratio ($\varepsilon$) eventually stabilized between 8% and 11%, which is similar to 10%–25% reported by Tang et al. [26] and 10%–30% reported by Deng et al.
Therefore, the experimental data were in agreement with the accuracy requirements.

2.4. Data acquisition and uncertainties

In the present experimental system, the input power of the single cartridge heater was adjusted by a power regulator to control the heat flux of the test section. The test section was wrapped in a layer of insulating cotton to reduce heat loss. The effective heat flux $q_e$ is expressed by Eq. (3), where $A_s$ is the longitudinal section area of the minichannel heat sink.

$$q_e = (1 - \epsilon)Q/A_s$$

Fig. 2. Schematic diagram of the test section (a) Exploded view, (b) Assembly view.
The local heat transfer coefficient \( h \) can be expressed by Eq. (5), where \( \eta \) is the heat transfer efficiency of the channel fins, \( T_w \) is the bottom wall temperature, and \( T_f \) is the fluid temperature at the measurement point.

\[
h = \frac{q_i(W_w + W_a)}{(T_w - T_f)(W_a + 2\eta H_{in})} \quad (5)
\]

The bottom wall temperature \( T_w \) is expressed by Eq. (8). Herein, the distance between the upper temperature measuring point \( T_f \) and the thermal conductive silica was 7.0 mm, the thickness of the thermal conductive silica was 0.11 mm, and the channel thickness was 1.5 mm. The variables \( k_{al}, k_{ch}, \) and \( k_{in} \) are the thermal conductivity of the heating element, heat sink, and silica, respectively.

\[
T_w = T_f - q_i R_i 
\]

(8)

\[
R_i = \frac{7.0}{k_{al} + 1.5 + \frac{0.11}{k_{in}}} \times 10^{-3} 
\]

(9)

The liquid temperature \( T_l \) along the flow direction of the channel is expressed by Eq. (10), where \( \Delta \rho \) is the horizontal distance between the temperature measuring points, \( c_p \) is the specific heat capacity of the liquid, and \( T_{sat} \) is the saturated temperature of the liquid.

\[
T_l = T_{in} + \frac{q_i(W_w + 2\eta H_{in})\Delta \rho}{M c_p} x_s < 0
\]

(10)

\[
T_f = T_{sat} \; \text{if} \; 0 \leq x_s < 1
\]

(11)

The thermodynamic equilibrium dryness \( x_e \) is expressed by Eqs. (12 and 13), where \( H_{fg} \) is the latent heat of vaporization, \( Z_r \) is the distance between the \( r \) temperature test points and the inlet, \( L_{sub} \) is the length of the subcooled section, and \( G \) is the mass flow rate in a single minichannel.

\[
x_e = -\frac{c_p(T_{sat} - T_f)}{H_{fg}}, \quad x_e < 0
\]

(12)

\[
x_e = \frac{q(W_w + 2\eta H_{in})(Z_r - L_{sub})}{H_{fg} M}, \quad 0 \leq x_e \leq 1, r = 1, 2, 3, 4, 5, 6
\]

(13)

\[
G = \frac{M/A_{in}}{\rho \varepsilon_0 N_t} \times \frac{V}{W \times L_{sub}}
\]

(14)

\[
L_{sub} = \frac{M c_p(T_{sat} - T_{in})}{q_i(W_{in} + 2\eta H_{in})}
\]

(15)

In addition, the enhancement ratio \( Eh \) is expressed by Eq. (16), where \( h_{in} \) is the heat transfer coefficient of the ultrasound field, and \( h_{out} \) is the heat transfer coefficient without ultrasound field.

\[
Eh = (h_{in} - h_{out}) \times 100% / h_{out}
\]

(16)

Herein, the uncertainties of the physical quantities were estimated based on the analysis of error propagation by Moffat et al. [28]. The uncertainties of temperature, pressure, mass flow rate, mass flux, heat flux, and heat transfer coefficient were ± 0.1 °C, ±0.5 Pa, ±1.5%, ±1.6%, ±4.82%, and ± 7.53%, respectively.

### 3. Results and discussion

**3.1. Flow visualization analysis**

Fig. 4 illustrates the schematic of the reconstructed flow regime in no ultrasound field (NU) and ultrasound field (WU) heat sinks at \( f = 40 \) kHz, \( P_n = 40 \) W, \( T_{in} = 26.4-26.8 \) °C, \( G = 268.8 \) kg/(m²·s), and \( q_e = 18.2 \) kW/m², respectively. The required images of the reconstructed flow regime were captured by a high-speed camera. The visual windows were located between the measurement points \( T_2 \) and \( T_3 \) (second visual window), as well as \( T_4 \) and \( T_5 \) (third visual window). Under the same experimental conditions, the reconstructed flow regime showed that the vapor quality of the NU heat sink was larger than that of the NU heat sink due to the generation of more bubbles. Taking the second visual window (between the measurement points \( T_2 \) and \( T_3 \)) as an example, we interestingly found more bubbles at the same observation position in the channel with ultrasound field than without ultrasound field. From two visual windows in the reconstructed diagram of Fig. 4, about 17 and 63 bubbles (containing 38 tiny bubbles) were counted for NU and WU, respectively. The reason for this phenomenon is that the forces of ultrasound can promote the detachment of bubbles from the heating wall surface, which increases the frequency of bubble detachment [29]. In addition, we found that the onset of nucleation was slightly downstream shifted in the WU heat sink compared with the NU heat sink, as ultrasound cavitation can produce a large number of tiny bubbles, and even the channel wall covered by elongated bubbles generated many tiny bubbles, as shown in Fig. 4. This production of many tiny cavitation bubbles has not been observed in the NU heat sink. Therefore, the boiling incipience in minichannels with ultrasound field can be initiated at very low superheat temperatures due to the ultrasound field.

A simplified model for bubble motion in the ultrasound field is shown in Fig. 5. It assumes that a bubble is subjected to the intra-vesicular force and the extra-bubble force. In the ultrasound field, the bubble radius is \( R_0 \), and the coordinates are \( Q(x, y, z) \). The force of the bubble at \( Q(x, y, z) \) is the superposition of the acoustic wave \( L_0 \). The distance between the ultrasound emission source and \( L_i \) is \( R_i \). Here, the inside of the bubble was assumed to be an ideal gas, and the gravity of the bubble could be ignored.

The force of the micronucleus bubble at a certain time \( P \) is expressed by Eq. (17), where \( P_{in} \) and \( P_{out} \) are the intra-vesicular force and the extra-bubble force of the micronucleus bubble, respectively. \( P_n \) is expressed by Eq. (18), where \( P_r \) and \( P_f \) are the vapor pressure in a micronucleus and the gas pressure, respectively.

\[
P = P_{out} - P_{in}
\]

(17)
Fig. 4. Schematic of reconstructed flow regimes and representative flow image.

NU (no ultrasound field)  WU (with ultrasound field)

Visual experimental conditions: \( G = 268.8 \text{ kg/(m}^2\text{s) }, q = 18.2 \text{ kW/m}^2 \)
\[ P_{\text{in}} = P_h + P_v \]  

where \( P_h \) and \( \sigma \) are the hydrostatic pressure and surface tension, respectively.

\[ P_{\text{out}} = P_h + \frac{2\sigma}{R_0} \]  

When \( P_{\text{in}} = P_{\text{out}} \), the micronucleus bubble is in equilibrium (Fig. 5(b)), which is expressed by Eq. (20)

\[ P_h + P_v = P_h + \frac{2\sigma}{R_0} \]  

When \( P_{\text{in}} > P_{\text{out}} \), the micronucleus bubble will expand, and the radius \( R \) will increase (see details in Fig. 5(c)). When \( P_{\text{in}} < P_{\text{out}} \), the micronucleus bubble will be compressed under the oppression of an extra bubble force, and the radius \( R \) will decrease (Fig. 5(d)). This force is expressed in Eq. (21). \( P_h \) is equal to \( P_0 \) without ultrasound disturbance. Herein, sinusoidal pressure waves were adopted as the input of ultrasonic vibration.

\[ G(R) = P_h + A\sin(\omega t) - P_h + P_v - \frac{2\sigma}{R} \]  

It was assumed that the work done by the internal and external forces of the bubble is converted into kinetic energy of the bubble, which is expressed by Eq. (23).

\[ W = \int_{R_0}^{R} 4\pi R^2 (P_{\text{out}} - P_{\text{in}}) dR \]  

\[ E_k = \int_{R_0}^{R} \frac{1}{2} \times 4\pi^2 p\rho v^2 (\frac{\Delta r}{\Delta t})^2 dR = \int_{R_0}^{R} 4\pi R^2 (P_v + A\sin(\omega t)) + P_v - \frac{2\sigma}{R} dR \]  

According to the above description, cavitation occurs when the external pressure of the bubbles surpasses the pressure threshold, which is demonstrated in Fig. 6 (see Figs. 4 and 6 for details). Many tiny cavitation bubbles were detected with the high-speed camera, and these bubbles absorbed heat from the surrounding liquid and grew in the channel. Fig. 6 shows high-speed photographs of the first visual window (between measurement point \( T_1 \) and channel entrance) in WU heat sinks at the ultrasound frequency \( f = 40 \) kHz, ultrasound transducer power \( P_u = 40 \) W, fluid inlet temperature \( T_{\text{in}} = 26.4-26.8 \) °C, mass flow rate \( 268.8 \) kg/(m²·s), and heat flux of \( 18.2 \) kW/m². Under these conditions, many tiny bubbles were found at the channel entrance. However, this position was not the thermodynamic onset of boiling (ONB), as it was mainly attributed to the action of ultrasonic cavitation. The first ultrasonic transducer was installed in the first visual window (see installation...
Fig. 6. Cavitation bubble phenomenon under ultrasound field.
location of the ultrasonic transducer in Fig. 2 for details). At the time t = 0.5 ms, the ultrasonic pressure of the working medium at the channel entrance was assumed to be negative. Here, the negative pressure only reflected the direction of acoustic streaming. The dramatic decrease of the acoustic pressure gradient resulted in supersaturation of the original vaporized microparticles, which spilled out from the working medium to form cavitation bubble, as shown in Fig. 6. Then, the cavitation bubble absorbed heat and grew in the channel (see the results for the times t = 0.5 ms to t = 10.5 ms in Fig. 6). This phenomenon has not been found in the NU heat sink, indicating that the presence of the ultrasound field explains the slight downstream shift of the onset of nucleation in the WU heat sink compared with the NU heat sink. Most importantly, this is also one of the main reasons for the enhanced flow boiling heat transfer in the presence of the ultrasound field.

Furthermore, the motion trajectories of the bubble was changed by ultrasonic streaming. Fig. 7 shows the high-speed photographs at a mass flow rate of 268.8 kg/(m²·s), heat flux of 12.8 kW/m², ultrasound field frequency of 40 kHz, and transducer power of 50 W. Taking the bubble indicated by the arrow in Fig. 7 as an example, the bubble was close to the right channel wall at time t, in the middle of the channel at time t + 1.0 ms, close to the left channel wall at time t + 1.0 ms, and bounced back to the left wall at time t + 2.5 ms. The bubbles constantly hit the channel wall on the left and right sides under the ultrasound field, and the speed of the bubble motion was increased. It was assumed that the bubble radius changed from 1.07 to 0.964 mm between t and t + 1 ms, close to the left channel wall at time t + 1.0 ms, and bounded back to the left wall at time t + 2.5 ms. The bubbles hit the channel wall on the left and right sides under the ultrasound field, and the speed of the bubble motion was increased. It was assumed that the bubble radius changed from R₀ to R under the action of the ultrasound field (here, R₀ is greater than R), and the bubble contraction transformed the kinetic energy of the bubble. The work done by the external force can be calculated by Eq. (22), where Pₐ is the ultrasound pressure, and the kinetic energy of the bubble Eₖ is expressed by Eq. (23). Herein, the software COMSOL Multiphysics was used to numerically analyze the acoustic field, a left/right vibration of the bubbles oriented perpendicular to the structure boundary between the transducer and the working medium is expressed in Eq. (24) [30], where n is the normal vector, aₙ is the normal acceleration, and ρ₁ is the transducer floor density.

\[
n \cdot \left( \frac{1}{ρ} \nabla Pₐ + q' \right) = aₙ \tag{24}
\]

\[
\n \cdot \left( \frac{1}{ρ} \nabla Pₐ + q' \right) + \alpha \frac{aₙ}{ρc²} = 0 \tag{25}
\]

The acoustic wave transmission equation is shown in Eq. (25), where q’ is the dipole, c is the acoustic velocity, and ρ is the heat transfer density of the working medium.

Fig. 8 shows the distribution of the sound pressure with the channel depth obtained by numerical calculation. In Fig. 7, the bubble diameter changed from 1.07 to 0.964 mm between t and t + 1.5 ms (the bubble diameter was obtained by referring to the channel width of 1.5 mm). Here, based on the parameters of the transducer, the sound pressure value obtained by Fig. 6 at a depth of 1 mm, and Eq. (23), it could be concluded that the bubble motion speed was increased by 31.9%. The force exerted on the bubble came from multiple directions, as the actual ultrasound sound pressure was spatially distributed. Therefore, bubbles hitting the channel wall on the left and right sides were observed in the high-speed video, as shown in Fig. 7. The increase of the motion speed of the bubbles strengthened the flow field disturbance, which was conducive to the mixing of cold and hot working medium. At the same time, the flow field disturbance enhanced the heat absorption of the heating surface, which improved the heat transfer performance of the whole channel. Boziuk et al. [31] reported similar results for sound fields.

3.2. Effects of operation parameters on flow boiling heat transfer

Fig. 9 displays variations of the local heat transfer coefficient (hₐ, n = 1, 2, 3, 4, 5, 6) along the flow direction of the minichannel with and without ultrasound field at f = 40 kHz, P_u = 40 W, and T_in = 26.4–6.8 °C for mass flow rates of 268.4 and 298.2 kg/(m²·s). The distances between the temperature measuring points and the minichannel entrance were 22, 52, 82, 112, 142, and 172 mm, respectively (see Fig. 2 for details). All temperature measuring points were in the phase-change region, as the length of the single-phase section was 8 to 15 mm according to Eq. (15). Fig. 9 shows that the heat transfer coefficients of NU and WU heat sinks increased slightly with the increase of the axial distance (Z) until Z = 82 mm, which could be explained by the convective heat transfer in this region. The heat transfer coefficient increased gradually until Z = 142 mm. In this area, the formation of some bubbles started, and only bubbly flow patterns were found in this axial region. With the increase of Z to values larger than 142 mm, the heat transfer coefficient significantly decreased, as bubbly flow to slug and annular flow patterns were observed.

Interestingly, the local heat transfer coefficients of WU heat sinks at each temperature measuring point were much larger than the corresponding coefficients of NU heat sinks at f = 40 kHz, P_u = 40 W, T_in = 26.4–6.8 °C, and G = 268.4 kg/(m²·s). These results indicate that the ultrasound could effectively improve the flow boiling heat transfer performance of the minichannel heat sink (see Fig. 9 for details). The enhancement ratio Eh showed a non-linear relation with Z and increased with the increase of Z for the heat fluxes of 21.3 and 12.5 kW/m² up to Z = 112 mm. At values higher than Z = 112 mm, Eh decreased with Z. The maximum enhancement ratio Eh was obtained at the fourth temperature measuring point (Eh = (hₐ(max)/hₐ(min)), and the maximum enhancement ratios Ehₐ(max) were 21.3% and 14.6%. The average enhancement ratios Ehₐ were 10.5% and 10.4% for heat fluxes of 21.3 and 12.5 kW/m², respectively. This result was explained by the ultrasound field, which was mainly effective for bubbly flow, while the enhancement ratio was weakened for elongated bubbly flow, and ultrasound field had no effect on slug and annular flow patterns (see Fig. 13 for details). Taking f = 40 kHz, P_u = 40 W, and a high heat flux of 29.1 kW/m² as an example, Eh slightly increased with the increase of Z until Z = 92 mm (here, Ehₐ(max) was obtained at the third temperature measuring point), the maximum enhancement ratio Ehₐ(max) was 20.7%, and the average enhancement ratio Ehₐ was 10.7%. Eh decreased earlier for a heat flux of 29.1 kW/m² than for heat fluxes of 21.3 and 12.5 kW/m², as the flow boiling process has a longer elongated bubbly and annular flow region for a high heat flux of 29.1 kW/m².

Fig. 11 shows the variations of the local heat transfer coefficient (hₐ) with the heat flux of WU and NU heat sinks at the experimental conditions of G = 238.6, 268.4, and 298.2 kg/(m²·s), T_in = 26.4–26.8 °C, P_out = 121 kPa, and f = 40 kHz. The results reveal that the local heat transfer coefficient (hₐ) increased with the increase of the heat flux for NU and WU heat sinks at different mass flow rates. The flow boiling heat transfer coefficients were higher in the presence of the ultrasound field under the same experimental conditions (Fig. 11). The heat transfer coefficients of WU heat sinks for the mass flow rates of 238.6, 268.4, and 298.2 kg/(m²·s) increased by 5.9%–46.3%, 12.4%–33.4%, and 11.3%–29.7% compared with those of NU heat sinks under the same experimental conditions, respectively, and the average enhancement ratios Eₐ were 25.5%, 21.4%, and 19.4% for the mass flow rates of 238.6, 268.4, and 298.2 kg/(m²·s), respectively. Moreover, Fig. 12 shows that the enhancement ratio Eh in the low heat flux region was larger than in the high heat flux region. Taking the mass flow rate of 298.2 kg/(m²·s) as an example, the heat transfer coefficient increased by 5.93%–23.4% for the heat flux of 8.03–17.2 kW/m², and the heat transfer coefficient increased by 5.93%–23.4% for the heat flux of 20.7–28.4 kW/m². The reason for this finding is that the flow boiling in the minichannel mainly has a bubbly flow pattern in the region with a low heat flux of 8.03–17.2 kW/m² (see Fig. 13(b) for details). Under the action of the ultrasound field, a left/right vibration of the bubbles oriented perpendicular to the fins along the flow direction was observed (Fig. 7), which intensified the disturbance of the working medium. Therefore, the enhancement ratio
Fig. 7. Bubble left-right hitting wall phenomenon under ultrasound field.
Eh was significantly improved. However, with the increase of the heat flux, mainly elongated bubbles (slug and annular flow patterns) were observed in the channel (see Fig. 13(a) for details). The force of the ultrasound field only changed the morphology of the bubble liquid film but did not trigger bubbles hitting the channel wall on the left and right sides due to the size effect of the minichannel, and the effect of the ultrasound field on bubble oscillation was obviously weakened. Therefore, the enhancement ratio Eh was decreased in the region with a high heat flux of 20.7–28.4 kW/m².

Fig. 14 shows the variations of the local heat transfer coefficient (h₄) with the mass flow rate under the ultrasound field. The results reveal that h₄ decreased with the increase of the mass flow rate for NU and WU heat sinks, and the flow boiling heat transfer coefficient of the WU minichannel heat sink was significantly increased compared with that of the NU heat sink under the same conditions. Ehₘₐₓ was 19.8% for the mass flow rate of 298.2 kg/(m²·s), ultrasound frequency of 30 kHz, ultrasonic power of 50 W, and heat flux of 20.5 kW/m², and the average enhancement ratio Ehav was 14.5%. Fig. 15 shows that the enhancement ratio Eh increased slightly in the range of 208–298.2 kg/(m²·s), and the enhancement ratio Eh decreased significantly when the mass flow rate exceeded 298 kg/(m²·s). These results can be explained by the decrease of the effect of ultrasonic streaming on the flow field disturbance when the mass flow rate is too large.

3.3. Effects of ultrasound parameters on flow boiling heat transfer

Fig. 16 shows the variations of the local heat transfer coefficient (h₄) with the heat flux for five ultrasound powers (10, 20, 30, 40, and 50 W) of 20.5 kW/m², Tᵢn = 26.4–26.8 °C, P_out = 121 kPa, and f = 30 kHz. Fig. 15 demonstrates the variations of the enhancement ratio Eh of the local heat transfer coefficient (h₄) with the mass flow rate under the ultrasound field. The results reveal that h₄ decreased with the increase of the mass flow rate for NU and WU heat sinks, and the flow boiling heat transfer coefficient of the WU minichannel heat sink was significantly increased compared with that of the NU heat sink under the same conditions. Ehₘₐₓ was 19.8% for the mass flow rate of 298.2 kg/(m²·s), ultrasound frequency of 30 kHz, ultrasonic power of 50 W, and heat flux of 20.5 kW/m², and the average enhancement ratio Ehav was 14.5%. Fig. 15 shows that the enhancement ratio Eh increased slightly in the range of 208–298.2 kg/(m²·s), and the enhancement ratio Eh decreased significantly when the mass flow rate exceeded 298 kg/(m²·s). These results can be explained by the decrease of the effect of ultrasonic streaming on the flow field disturbance when the mass flow rate is too large.
at a mass flow rate of 298.2 kg/(m²·s), \( T_{\text{in}} = 26.4^\circ\text{C} \), \( P_{\text{out}} = 121 \text{kPa} \), and \( f = 30 \text{ kHz}, 40 \text{ kHz} \). Fig. 10 demonstrates that the flow boiling heat transfer coefficients of WU heat sinks obtained for five power levels of the ultrasonic vibrator were improved compared with those obtained for NU heat sinks under the same experimental conditions. The heat transfer coefficients of WU heat sinks obtained for five power levels of the ultrasonic vibrator increased by 19.3%, 33.8%, 43.4%, 50.2%, and 52.9% compared with those obtained for NU heat sinks under the same experimental conditions, and the corresponding average enhancement ratios \( E_{h_{\text{av}}} \) were 6.7%, 14.4%, 22.6%, 27.7%, and 29.9%, respectively. At the same time, the local heat transfer coefficient \( (h_4) \) increased with the increase of the ultrasonic power under the experimental conditions.

Fig. 13. Flow patterns for different heat flux region under ultrasound field.
as the higher power of the ultrasonic vibrator could prevent the appearance of more elongated bubbles (slug and annular flow patterns) due to the stronger action of the ultrasound field forces. Fig. 17 shows the variations of the enhancement ratio $Eh$ with the heat flux for five power levels of each ultrasonic vibrator. The results show that the enhancement ratio $Eh$ increased proportionally with the increase of the ultrasonic power: the higher the ultrasonic power, the higher will be $Eh$ at the same heat flux. For the ultrasonic vibrator power of 20 W, the enhancement ratio $Eh$ was in the range of 0.037–0.338. However, for the ultrasonic vibrator power of 50 W, the enhancement ratio $Eh$ was in the range of 0.158–0.529. In addition, the enhancement ratio $Eh$ for five power levels of the ultrasonic vibrator gradually declined with the increase of the heat flux, as the flow regime changed from a bubbly flow to elongated bubbles (slug and annular flow patterns) in the high heat flux region.

Fig. 18 shows the variations of the local heat transfer coefficient ($h_4$) with the heat flux at four ultrasonic frequencies (10, 20, 30, and 40 kHz) at $G = 298.2$ kg m$^{-2}$s$^{-1}$, $T_{in} = 26.4–26.8$ °C, $P_{out} = 121$ kPa, and $P_u = 40$ W. The results show that the local heat transfer coefficient of the WU heat sink was improved for four ultrasonic frequencies (10, 20, 30, and 40 kHz) compared with that of the NU heat sink under the same experimental conditions, and the heat transfer coefficient $h$ increased proportionally with the increase of the heat flux. In agreement with the results shown in Fig. 19, the maximum enhancement ratios $Eh_{max}$ for four ultrasonic frequencies were 53.9%, 43.7%, 34.6%, and 28.8%, respectively, and the average enhancement ratios $Eh_{av}$ were 27.7%, 22.2%, 16.0%, and 11.2%, respectively. These results show that the enhancement ratios $Eh$ for four ultrasonic frequencies gradually declined with the increase of the heat flux, and the influence of the ultrasound field on the flow boiling heat transfer was weakened in the region of high heat flux. It should be noted that $h$ decreased with the increase of the ultrasound frequency at the same experimental conditions, and similar results have been reported in ref. [32]. This decrease of $h$ may be explained by the more drastic coalescence of bubbles due to a rapid change from positive to negative pressure induced by the higher frequency of the ultrasound [33]. This indicates that, with the increasing ultrasound frequency, the growth of the bubbles occurred rather by coalescence than by heat absorption. Therefore, less heat was absorbed at a high ultrasound frequency than at lower frequencies in the process of bubble growth.
4. Conclusions

In the present paper, a novel installation of ultrasound transducers in the flow direction of a minichannel heat sink is designed to experimentally study the characteristics of flow boiling heat transfer and the influence of operation parameters (e.g., heat flux, mass flux rate) and ultrasound parameters (e.g., frequency, power) on the flow boiling heat transfer in the minichannel heat sink. In addition, bubble motion and flow pattern in the minichannel with and without ultrasound are analyzed by high-speed visualization. Some important investigation results are summarized in the following.

(1) Flow visualization analysis revealed that the onset of nucleation boiling is slightly downstream shifted, as ultrasonic cavitation produces a large number of tiny bubbles. In addition, bubbles hitting the channel wall on the left and right sides are observed, and the motion speed of the bubble is increased by 31.9% under the action of ultrasonic streaming, which strengthens the disturbance of the flow field.

(2) Ultrasound has a significant effect on the flow boiling heat transfer coefficient of the minichannel heat sink. The heat transfer coefficient of the minichannel heat sink with ultrasound is 52.9% higher than that of the minichannel heat sink without ultrasound under the same conditions, and the enhancement ratio $Eh$ is decreased in the high heat flux region due to the change from a bubbly flow regime to slug and annular flow patterns.

(3) Different trends of the influence of ultrasound parameters (e.g., frequency, power) on the flow boiling heat transfer in the minichannel heat sink can be observed under the experimental conditions. The local heat transfer coefficient $h$ decreases with the increase of the ultrasound frequency, and $h$ increases with the increase of the ultrasonic power. The heat transfer coefficient of the minichannel heat sink at an ultrasonic transducer power of 50 W is increased by 52.9% compared with that without ultrasound field under the same conditions, and the maximum enhancement ratio $Eh_{max}$ of 53.9% is obtained at the ultrasound frequency of 10 kHz.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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