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Acoustics Vibrations to Enhance Flow Boiling in Micro Channels

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Abstract

Heat transfer coefficients were measured for forced convection refrigerant mixture (R-407C) flow in micro channels enhanced by acoustics vibration. Flow in horizontal micro channels (ID 100.0 μm), and a length of 50 mm for subcooled and saturated boiling conditions are reported in this study. Experiments were conducted at a heat flux of 6 kW/m^2 for subcooled boiling; saturated boiling was carried at heat fluxes of 15 and 29 kW/m^2. The mass flow rate was varied from 0.45 to 1.85 kg/min. The frequency was varied, freq= 5, 10, and 15 kHz. An experimental setup composed of heating elements provided heat flux variations in the micro channels and excitation elements were attached to the side of the channels to produce the acoustics vibrations. The heat transfer coefficient was found to be dependent on both the heat flux as well as mass flux levels. Results show that acoustics vibration enhanced the heat transfer performance by 21% as compared to regular flow convective boiling process.

Keywords: *Micro Channels, Flow Boiling, Enhancement, Acoustics, Refrigerant Mixture.*

1. Introduction

The use of acoustics vibration in enhancing heat transfer characteristics for refrigerant mixtures at the micro-scale has an important impact on various daily used thermal systems. The use of acoustics vibration was investigated, but there is a need for predicting the heat transfer coefficients enhancement for subcooled and saturation boiling processes in such micro-scale. Therefore, combining micro-scale channels along with acoustics vibration for subcooled and saturated boiling needs to be investigated in conjunction with refrigerant mixtures.

The relationship between the flow behavior induced by ultrasonic vibration and the consequent heat transfer enhancement in natural convection and pool boiling regimes were investigated by Kim et al. Experimental results showed that the effects of ultrasonic vibration on flow behavior are vastly different depending on the heat transfer regime and the amount of dissolved gas. In the natural convection and subcooled boiling regimes, behavior of cavitation bubbles strongly affected the degree of heat transfer enhancement. In saturated boiling, no cavitation occurred thus the reduced thermal bubble size at departure and acoustic streaming were major factors in enhancing the heat transfer rate. The highest enhancement ratio was obtained in natural convection regime where the effect of ultrasonic vibration is manifested through violent motion of cavitation bubbles [1].

The effects of ultrasonic vibration on critical heat flux (CHF) have been experimentally investigated by Jeong and Kwon under natural convection condition. Flat bakelite plates were coated with thin copper layer and distilled water were used in their experiment as heated specimens and working fluid, respectively. Measurements of CHF on flat heated surface were made with and without ultrasonic vibration applied to working fluid similarly to the study done but with the addition of size and working fluid impact on the heat transfer coefficient. An inclination angle of the heated surface and water subcooling were varied as well by Jeon and Kwon as they examined water subcoolings at 5°C, 20°C, 40°C and the angles are 0°, 10°, 20°, 45°, 90°, 180°. The measurements showed that ultrasonic wave applied to water enhances CHF and its extent is dependent upon inclination angle as well as water subcooling this particular comparison was not done as in this experimental investigation a flat horizontal setting is investigated. The rate of increase in CHF increases with an increase in water subcooling while it decreases with an increase in inclination angle. Visual observation showed that the cause of CHF augmentation is closely related with the dynamic behaviour of bubble generation and departure in acoustic field [2].

Other investigations were done by Tillery et al on two-phase cooling cell based on channel boiling and a vibration-induced cavitation jet whose collective purpose is to delay the onset of critical heat flux by forcibly dislodging the small vapor bubbles attached to a solid surface during nucleate boiling and propelling them into the cooler bulk liquid within the cell. The submerged turbulent cavitation jet is generated by a vibrating piezoelectric

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diaphragm operating at resonance. The piezoelectric driver induced pressure oscillations in the liquid near the surface of the driver, resulting in the time-periodic formation and collapse of cavitation bubbles that entrain surrounding liquid and generate a strong liquid jet. The resultant jet, which was directed at a heated surface, enhances boiling heat transfer by removing attached vapor bubbles that insulate the surface while providing additional forced fluid convection on the surface. By introducing a crossflow within the cell, the heat transfer was increased even further due to the fact that the flow swept the bubbles downstream while keeping the temperature of the water within the cell regulated [3].

The effects of ultrasonic vibration on heat transfer were investigated by Shinfuku et al in a natural convection region and a nucleate boiling region. The Nusselt number by ultrasonic vibration, cavitation intensity using aluminum sheet, and electric input power to oscillator were measured systematically in their study. As the height of heating elements was varied, both the profile of the electric input power and the cavitation intensity were periodically changed due to standing waves formed in the water vessel. These results agreed with the profile of the augmentation ratio estimated from the Nusselt number by ultrasonic vibration. The augmentation ratio for a downward facing surface was found to be greater than that for vertical surface in a natural convection region, however, for nucleate boiling, it was less than the augmentation ratio for the vertical surface. The ultrasonic enhancement effect became negligible in a well developed nucleate boiling region. The mechanism of heat transfer enhancement by ultrasonic vibration was the agitation generated by cavitation in a natural convection region, and the removal effect of the vapor bubbles by acoustic streaming in a nucleate boiling region [4].

Experimental investigations were done by Kim et al on the effects of tube vibration on critical heat flux (CHF) in order to gain an understanding of the relationship between CHF and flow-induced vibration (FIV). The experiment was carried out in the following range of parameters: diameter (D)=0.008 m; heated length (L)=0.2, 0.4 m; pressure (P)=101 kPa; mass flux (G)=403–2,551kg/m2·s; quality (x)=−0.045–0.289; amplitude (a)=0.0001–0.001 m; frequency (f)=0–70 Hz which is similar to the presented results of this paper but differ with the working fluid. The CHF generally increases with vibration intensity, which was represented by vibrational Reynolds number (Rev); the CHF enhancement was more dependent on amplitude than on frequency. CHF enhancement seems to come from the reinforced flow turbulent mixing effect by vibration in the vicinity of heat transfer surface. Based on the experimental results, an empirical correlation was proposed by Kim et al for the prediction of CHF enhancement by tube vibration. The correlation predicted the CHF enhancement ratio (En) with reasonable accuracy, with an average error rate of −2.18% and 27.75% for RMS [5].

In order to gain an understanding of the relationship between critical heat flux (CHF) and flow-induced vibration (FIV), an experimental investigation was carried out by Lee et al with vertical round tube at the atmosphere. In the both condition of departure from nucleate boiling (DNB) and the liquid film dryout (LFD), CHF increases up to 12.6% with vibration intensity, represented by vibrational Reynolds number (Rev). CHF enhancement by tube vibration seems to come from the reinforced flow turbulent mixing and the increment of deposition of droplet into the liquid film. Based on the experimental results presented by Lee et al, an empirical correlation was proposed for the prediction of CHF enhancement ratio. The correlation predicted the CHF enhancement ratio (En) with reasonable accuracy, with an average error rate of 4.5 and 26.5% for RMS. Vibration is an effective method for heat transfer enhancement as well as CHF. Nonetheless, the risk of system failure by FIV has made it very difficult to take advantage of vibration in heat transfer facilities. Therefore, it is necessary to find out optimal design enhancing the CHF but preventing damage in an acceptable vibration range [6].

2. Content

The defined range for acoustic vibration according to Wikipedia.org as shown in Fig 1, is between 20 Hz to 20 kHz, as in this range of lower frequency vibration mode which is more advantageous to use versus higher frequency, because the penetration of these waves at the prescribed frequency will give a more mixing of the bubbly flow while the refrigerant is increased in temperature.

2.1. Experimental Set-up

An experimental setup has been built for measuring heat transfer and pressure drop under various flow conditions in micro- and meso-channels. In Fig 2, the schematic diagram of the experimental setup is shown. It consists of a storage tank, pump, by-pass loop, flowmeter, preheater, test section, VARIAC power controller, a condenser, chiller, and a data acquisition system. The setup has been designed and built such that it is flexible enough to use with different refrigerants. The pump circulates the refrigerant in the loop at a rated capacity of 3 gpm. The by-pass loop diverts most of the flow exiting the pump from entering the test section and redirects it to the condenser. The amount of refrigerant entering the test section is measured using a digital flowmeter with a capacity of 0-2000 ml/min. Heating tape is wrapped around a pipe segment approximately 1 m long and controls the temperature of the refrigerant entering the test section. The heating tape has a capacity of 750 W at 110 V, and is supplied with a controller that changes the power from 0-100%. The temperature of the refrigerant entering the preheater section is monitored by a T-type thermocouple.

The test section consists of the temperature/pressure measurement stations at the inlet and outlet of the mini-channels, where for the temperature measurements, T-type thermocouples are used at strategic locations on the test section's surface. The pressure at the inlet and outlet are measured using both pressure gauges and pressure transducers, where the pressure gauges have a range of 0-600 psi. The pressure transducers are made of thin film vapor deposited strain gauges having a range of 0-6.8 bar. The refrigerant enters the inlet header of the test section, which distributes the flow to the channels where the refrigerant was heated with the heating elements (five cartridge heaters were used), and exits through the outlet header of the test section, then flows towards the condenser, and next to the chiller. Four thermocouples located in an array on the test section provide information on the surface temperature of the channels. The heat input to the test section is controlled using a VARIAC with an input of 110 V / 50 Hz, and it gives a variable output of 0-110 V / 50 Hz. The output is controlled in 5V increments, and the voltage output is measured directly using a voltmeter. For cooling the refrigerant as it exits the test section, two stations are used in the process. The first station is the condenser, which is composed of a steel shell and integrally finned copper tubes, and its function is to reduce the temperature of the refrigerant exiting the test section and the by-pass loop before it enters the chiller. Condenser water was circulated through the tubes while heated refrigerant flows through the shell of the condenser.

Copper-constantan thermocouples (type-T) were used for this experiment. Thermocouple output was interfaced to a personal computer through a National Instruments Data Acquisition System. LabView was used to control the system with a software graphical interface that allows the user to select the input sensor type, data acquisition rate, filtering and gain, and analysis and storage of the data. The micro channels test section (Figure 3) were micro-machined by precision cutters, and five heaters were used to heat the test section. The length of the channels in all test sections is kept constant at 50 mm. There are 8 parallel channels for each test section. The advantage of using an n number of arrays in a parallel formation gives a broad spectrum of temperature averages and clearly shows which side of the test section is affected. The length of the channels in all test sections is kept constant at 50 mm. Copper was chosen because of its superior heat conduction characteristics, and R-407C is stable in the presence of copper over the normal operating temperature range.

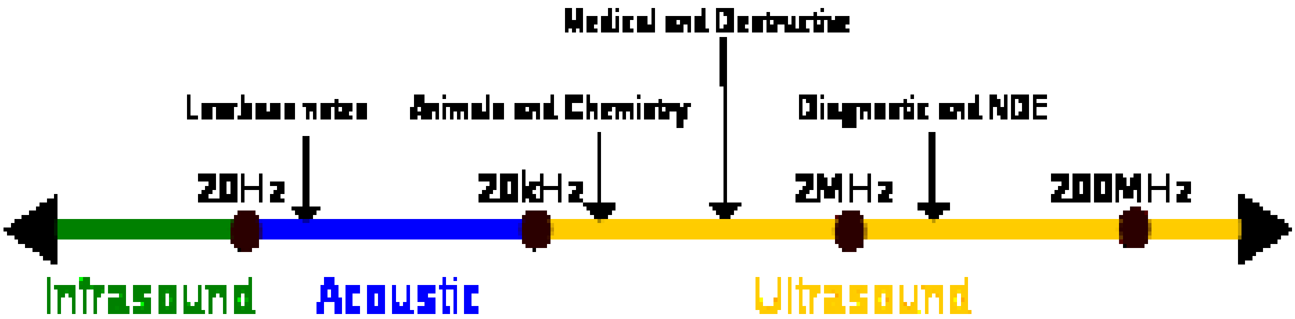

Fig. 1. Approximate frequency ranges corresponding to ultrasound, with rough guide of some applications [7]

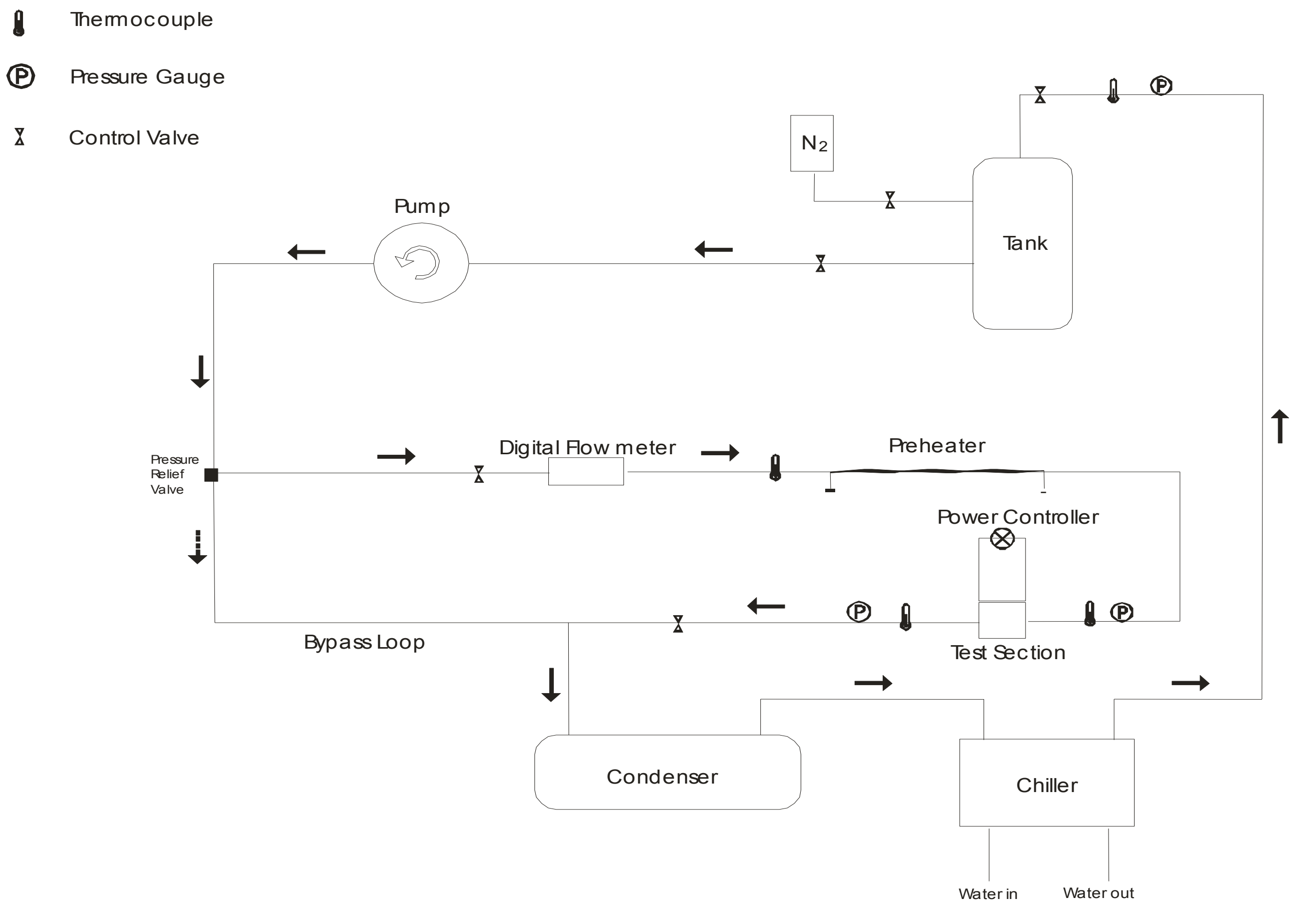

Fig. 2. Schematic diagram of the experimental set-up

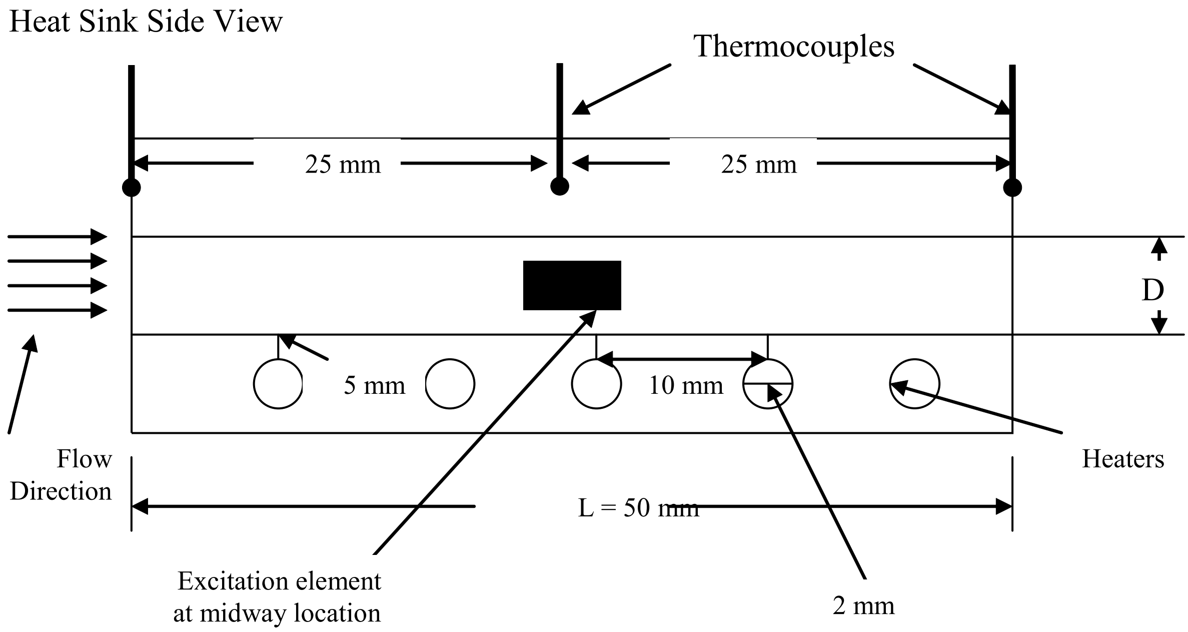

Fig 3. Detailed schematic of the micro channels

The channels are spaced equally across the heaters, to ensure uniform energy distribution to all channels and minimize temperature gradients. The channel surface temperature is measured by averaging the temperature of the four thermocouples inserted into strategic locations in the test section. Different ultrasonic vibrations were produced by the excitation element installed on the middle side of the micro channel and was varied between 5 to 15 kHz.

2.2. Experimental Procedure

The data acquisition software, LabView, loads on the pc with a direct connection to the T-type thermocouples on different strategic locations with in the experimental loop. The circulation pump was switched on followed by the preheater and the test section heaters controlled by the VARIAC controller, this is done till it reaches 25^{o}C. The power settings are adjusted according to the levels of heat fluxes desired. The flowrate was adjusted to the desired flowrate using the needle valves next to the digital flowmeter.

The preheater is adjusted so that the inlet temperature of the refrigerant to the test section was about 10^{o}C below saturation temperature of the refrigerant (~32^{o}C @ atm P). The process is carefully monitored to reach a steady state at which time data can be logged into the file. The data recorded into the spreadsheet file generates a list for the current surface, inlet, outlet temperatures which calculates all relevant parameters of the refrigerant such as heat input, heat removed, Reynolds number, heat transfer coefficient, and Nusselt number. The flow instability somewhat increased at higher flow rates when data are taken by keeping the flow rate constant and varying the power supply to the test section. This is due to pressure fluctuations and as were different from regular flow in the same test section. The power input is increased until the critical heat flux condition is encountered, as indicated by a sudden and large increase in surface temperature.

2.3. Error and Uncertainty Analysis

A major emphasis of experimental error is based on critical measurements of flow-rate and temperature. The flow-meter is calibrated based on actual flow rates, where the calibration curve has a confidence level of 95% with tolerance limits of +/-0.0027kg/s. It should also be noted that there is a component of human error, which can be considered insignificant and that it is completely unknown. The thermocouples used in the experiment have been calibrated at the ice point (0^{o}C). However, channel surface temperatures at each station as measured by four different thermocouples were found to vary within a range of +/-1^{o}C. The best estimate of the true value is taken as the arithmetic mean of these four values. A typical set of data is analyzed for precision of measurement. Precision of the temperature measurement is found to be of the order of 0.42^{o}C. In addition, the errors considered are for convective heat transfer calculations, therefore, conduction and radiation effects are not considered, where conduction was not involved in the measured part of the test section as the variations in measured temperatures between the mixed refrigerant flow and the surface of the channels, similarly, radiation was not involved in the measurements and the experiment was done indoors in a laboratory environment and ignoring such effects does not have a large impact on the final results.

2.4. Reduction of Data

Test section surface temperatures were measured at four locations. Three of the thermocouples were located on the top and between of the channels and the fourth was located above the middle heater on the side of the test section. The average surface temperature was obtained by taking the arithmetic average of the thermocouple readings as follows

$$T_s = \frac{T_1 + T_2 + T_3 + T_4}{4} \qquad (1)$$

The power input (electrical power) to the test section was calculated based on the voltage and current measurements. For a purely resistive load the power factor for AC voltage is 1

$$P = VI \qquad (2)$$

The mass flow rate of the refrigerant was calculated using the flow rates as measured by the flow meters as follows

$$m = \rho\ Q_1 + Q_2) \qquad (3)$$

The energy removed by the refrigerant was calculated using the simple expression

$$q = m\ c_p\ (T_{out}\text{-}T_{in}) \qquad (4)$$

And the heat transfer coefficient, based on the channel surface area A, and the average refrigerant temperature, can be calculated as

$$h = \frac{q}{A\ (T_s - \frac{(T_{out} + T_{in})}{2})} \qquad (5)$$

3. Results and Discussion

Experiments were carried out as described earlier and the results were reduced to provide heat transfer coefficient and heat flux characteristics for subcooled and saturated boiling processes. These results are discussed in the following two sections.

3.1. Subcooled Boiling

Experiments were performed over a range of mass flow rates inside horizontal channels. Subcooled tests were performed on the test section for three types acoustics vibration levels (5, 10, and 15 kHz). The heat fluxes for the subcooled boiling experiments were maintained at 6 kW/m^2. Figures 4 show an increase in the value of the heat transfer coefficient with the mass flow increase is observed at both values of heat flux (6 kW/m^2), with the heat transfer coefficient increasing dramatically at a mass flux of 0.95 kg/min. □he data is plotted for a constant value of heat flux and a mass flow ranging from 0.48 to 1.85 kg/min. It is seen that the heat transfer coefficient increases at higher mass fluxes.

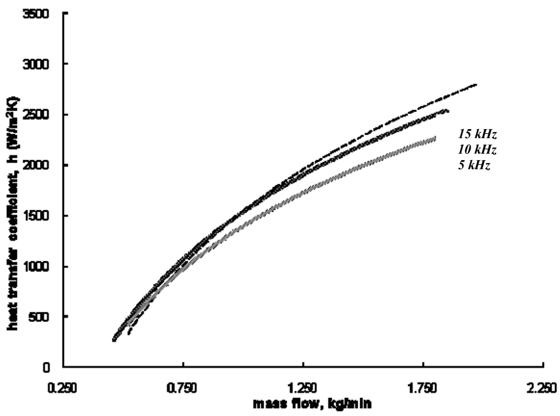

Fig 4. Effect of vibration levels on heat transfer in micro channels with a constant heat flux of q"=6 kW/m^2

The figure also illustrates the relative influence of the various parameters on the thermal hydraulic characteristics, including the impact due to the channel size D. We can notice that increasing the frequency from 5 to 15 kHz affects the heat transfer coefficient. The heat transfer coefficient for large frequency increases up to 14% the value for smaller frequency values.

3.2. Saturated Boiling

Convective saturated flow boiling experiments were conducted on the horizontal test section for three vibration levels (5, 10, and 15 kHz) over the same mass flow range (0.45 to 1.85 kg/min) as in subcooled boiling tests. The heat fluxes for the saturated boiling experiments were maintained at 15 and 29 kW/m^2. Figures 5 and 6 show the data trends at different vibration levels for each channel tested and two different values of heat flux. An increase in the value of the heat transfer coefficient with a mass flow increase is observed at both values of heat flux (q"=15, and 29 kW/m^2), with the heat transfer coefficient increasing dramatically at mass flux of 0.6 kg/min due to the increase in flow causing a mixing effect with increase freq. □he data is plotted for a constant value of heat flux and mass flow ranging from 0.48 to 1.85 kg/min. It is seen that the heat transfer coefficient increases at higher mass fluxes.

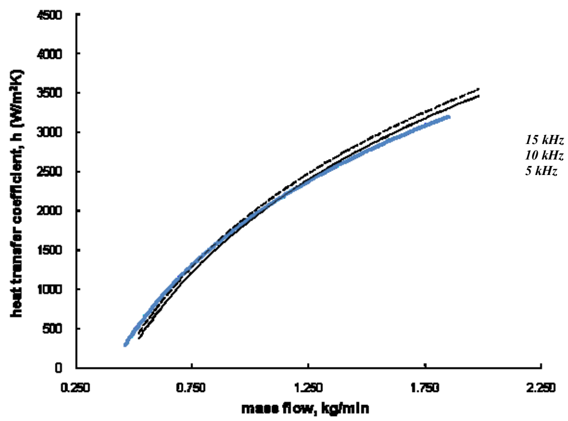

Fig 5. Effect of vibration levels on heat transfer in micro channels with a constant heat flux of q"=15kW/m^2

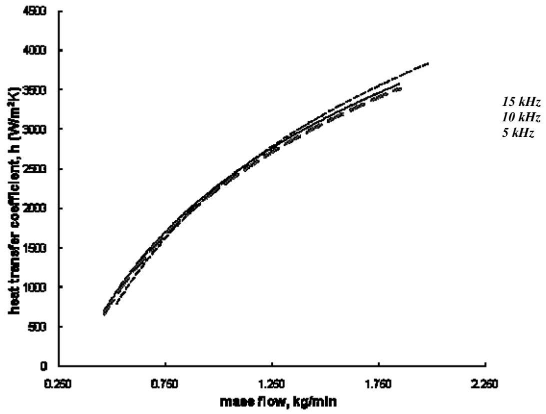

Fig 6. Effect of vibration levels on heat transfer in micro channels with a constant heat flux of q"=29 kW/m^2

In Figure 5, the experimental heat transfer coefficient is plotted at different values of freq. at a heat flux of 15 kW/m^2. The heat transfer coefficient for a constant heat flux of 15 kW/m^2 ranged from 465 to 520 W/m^{2}K at lower mass fluxes ranging from 0.45 to 0.85 kg/min and 520 to 1619 W/m^{2}K for higher mass fluxes ranging from 0.85 to 1.85 kg/min. For a heat flux at 29 kW/m^2 the heat transfer coefficient ranged between 495 and 795 W/m^{2}K at lower mass fluxes ranging from 0.45 to 0.85 kg/min and 798 to 1587 W/m^{2}K for higher mass fluxes ranging from 0.85 to 1.85 kg/min.

The figures also illustrate the relative influence of the various parameters on the thermal hydraulic characteristics, including the impact due to the channel size. Comparing Figures 5 and 6, we can notice the influence of increasing the heat flux from 15 to 29 kW/m^2 on the heat transfer coefficient. The heat transfer coefficient increases in value for the lower heat flux ranging from 0.45 to 0.85 kg/min. Comparing both figures for freq. values and the influence of the heat transfer coefficient increase to channel size indicates increasing heat flux from 15 to 29 kW/m^2 increased the heat transfer coefficient by an average of 28%.

4. Conclusion

This paper outlines heat transfer characteristics of subcooled boiling and saturated boiling of the refrigerant mixture R-407C in micro channels. The mass flow rate has a positive influence on the 'h' values, and more so near the onset of nucleate boiling. The heat fluxes for the channels are also noticeably higher with the enhancement vibration effects on the flow. Furthermore, it also can be concluded that increasing heat flux by 45% increases the heat transfer coefficient by approximately 13% for the subcooled boiling process, but increasing heat flux by 53% increases the heat transfer coefficient by 21% for the saturated boiling process.

Enhancement of heat transfer was also studied by implementing vibration levels (5, 10, and 15kHz) on the micro channels. Increasing freq. from 5 to 10kHz increased the heat transfer coefficient up to 13% for the subcooled process, where increasing freq. from 10 to 15kHz increased the heat transfer coefficient up to 21% for the saturated boiling process. Using larger gradients in the freq. values with smaller diameters will increase the heat transfer coefficient.

Nomenclature

A	internal surface area (m^2)
c_p	specific heat (J/kgK)
D	micro channel diameter (mm)
f	friction factor (-)
Freq	frequency (Hz)
h	heat transfer coefficient (W/m^{2}K)
I	current (amps)
k	thermal conductivity (W/mK)
m	mass flowrate (kg/min)
P	power (W)
Q	volumetric flowrate (m^3/s)
Q_1	volumetric flowrate at entrance of channel (m^3/s)
Q_2	volumetric flowrate at exit of channel (m^3/s)
q	heat transfer (W)
q"	heat flux (W/m^2)
T_s	average surface temperature (K)
$T_\#$	temperature of # thermocouple (K)
T_{in}	inlet temperature (K)
T_{out}	outlet temperature (K)
v	flow velocity (m/min)
V	voltage (volts)

Greek Symbols

ρ	density (kg/m^3)

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