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Impulse Acousitic Enhancement of Flow Boiling in Micro Channels

Yasir M. Shariff*

Mechanical Engineering Dept., College of Engineering, P.O. Box 1020, Taibah University, Madina, Saudi Arabia

Abstract

Measurements were done for the heat transfer coefficients h in forced convection refrigerant mixture (R-407C) flow within micro channels enhanced by impulse acoustic vibration. Flow in horizontal micro channels (ID 100.0 m), and a length of 50 mm for subcooled and saturated boiling conditions are reported in this study. The heat flux was ranged from 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 varied from 0.45 to 1.85 kg/min. The frequency was maintained at a constant value of 20 kHz. The impulse was varied in three ranges of 20, 40, and 60 dBs. An experimental setup composed of heating elements provided heat flux variations in the micro channels and excitation elements attached to the side of the channels to produce the impulse acoustic vibrations. The heat transfer coefficient found to be dependent on both the heat flux as well as mass flux levels. Results show that impulse acoustic vibration enhanced the heat transfer performance by 12% as compared to regular flow convective boiling process.

Keywords: *Heat transfer, flow boiling, micro channels*

1. Introduction

Using new techniques in enhancing heat transfer characteristics for refrigerant mixtures at the micro-scale has an important impact on various daily used thermal systems specially using impulse acoustic vibration. Using impulse acoustic vibration to enhance the heat transfer coefficient is a new phenomenon and needs investigation towards improving micro-mechanical devices. Hence, 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 impulse acoustic vibration for subcooled and saturated boiling needs investigation in conjunction with refrigerant mixtures. Recent advances in the microelectronics industry have led to increasingly smaller and faster electronic integrated circuits. The micro scale and increase in speed have resulted in an increase in heat generation, and with it, the need for improved heat dissipation. In older or less sophisticated systems, cooling is generally accomplished using natural or forced convection. Typically these systems are only capable of heat dissipation on the order of 10 W/cm^2[1]. Since heat fluxes from modern integrated circuits are projected to exceed 1000 W/cm2 in the next five to ten years [2], new methods for electronic cooling have been developed, such as liquid convection and two-phase cooling as is compared to experimental results done on recent work [3][4]. Liquid cooling methods are used because liquids tend to have a much greater thermal conductivity than air, so natural and forced convection are enhanced by different methods and techniques. Two-phase heat exchange in the form of boiling also utilizes the constant temperature heat sink by a fluid or an absorber, which undergoes a liquid-to-air phase change to remove excess heat from the cooled device. The process of bubbles forming on the channel surface and then moving into the refrigerant results in fluid motion which further increases heat transfer coefficient. However, boiling heat transfer is limited by the critical heat flux. If this maximum heat flux limit is increased, a vapor film will form over the channel's surface and will cause an insulation layer over the entire heat transfer surface resulting in a dramatic increase in the surface temperature and could cause deformations to the cooled device. Enhanced boiling seeks to improve the heat transfer from a surface at a given surface temperature or extend the critical heat flux limit. One of the used methods for doing this is to facilitate better dissipation of the bubbles from the heated surface using acoustic vibrations and yet impulse rates. This could be achieved by introducing an exciter that produces the desired frequency and timed impulse. The impulse will increase in releasing the bubbles from the heated surface and hence will make reaching the boiling point more raped [5]. Research at some leading institutions has focused on the use of miniaturized enhanced with acoustic excitation elements to help bubbles be elevated from the surface of the micro device by the creation of either a synthetic jet [6][7][8] or pressure waves [9].

* Corresponding author. Tel.: +966
Fax: +; E-mail: yshariff@taibahu.edu.sa

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2. Experimental Set-Up

An experimental setup has set up for measuring heat transfer and pressure drop under various flow conditions in micro-channels. In Fig 1, 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.

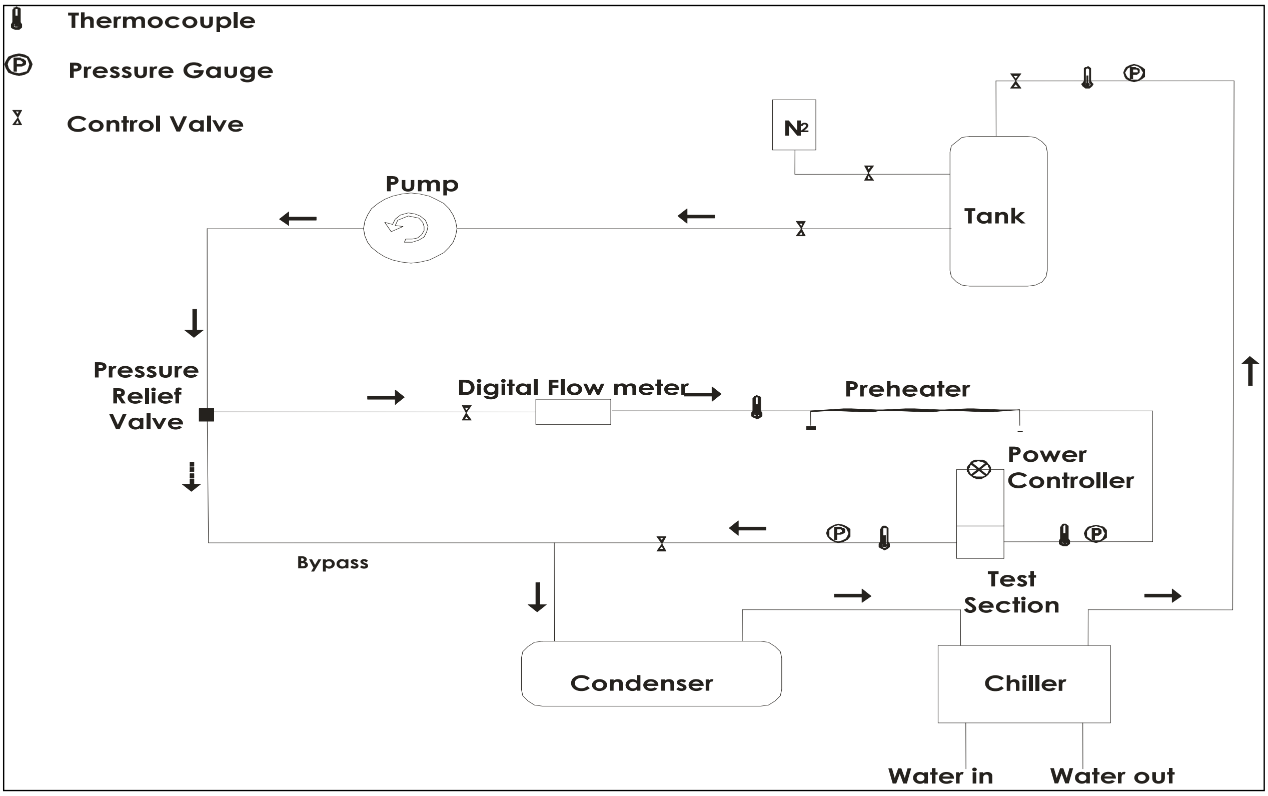

Fig. 1 Schematic diagram of the experimental set-up

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, in the test section of Figure 2, 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.

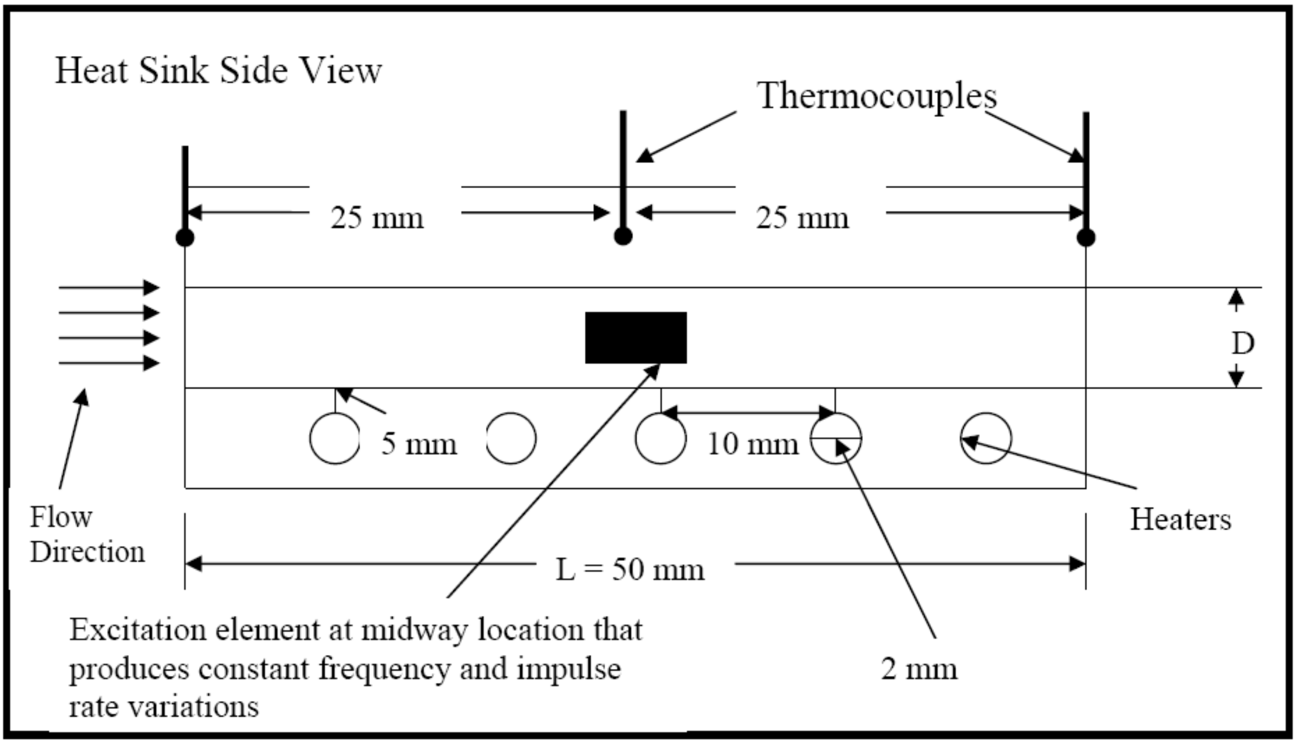

Fig. 2 Detailed schematic of the micro channels

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 impulse acoustic vibrations were produced by the excitation element installed on the middle side of the micro channel and was varied between 20 to 60 dBs at a constant frequency of 60 kHz.

2.1. 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 25oC. 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 10oC below saturation temperature of the refrigerant (~32oC @ 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.

2.2. 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.

2.3. 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

$$T_s = \frac{T_1 + T_2 + T_3 + T_4}{4} \tag{1}$$

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

$$P = VI \tag{2}$$

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

$$m = \rho\,(Q_1 + Q_2) \tag{3}$$

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

$$q = m\, c_p\, (T_{out} - T_{in}) \tag{4}$$

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

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

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 three test sections for three types impulse vibration levels (20, 40, and 60 dB) for the constant acoustic frequency of 20 kHz. The heat fluxes for the subcooled boiling experiments were maintained at 6 kW/m^2. Figures 3 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 0.85 kg/min. The data is plotted for a constant value of heat flux and a mass flow ranging from 0.5 to 1.9 kg/min. It is seen that the heat transfer coefficient increases at higher mass fluxes.

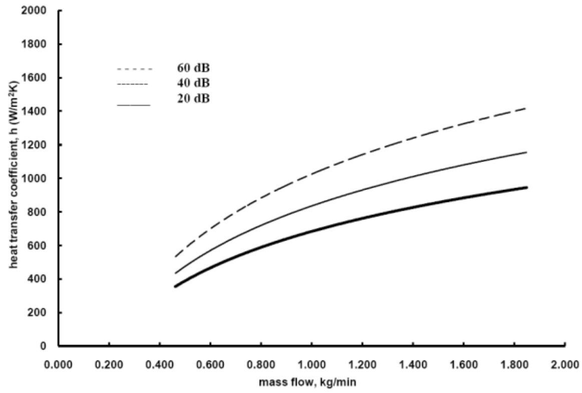

Fig. 3 Effect of vibration levels on heat transfer in micro channels with a constant acoustic frequency of 60 kHz and a heat flux of q"=6 kW/m^2

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 impulse from 20 to 60 dBs affects the heat transfer coefficient. The heat transfer coefficient for large frequency increases up to 8% against the value for smaller impulse rate.

3.2. Saturated boiling

Convective saturated flow boiling experiments were conducted on the horizontal test section for three impulse levels (20, 40, and 60 dB) 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 4 and 5 show the data trends at different impulse 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 0.9 kg/min due to the increase in flow causing a mixing effect with increase impulse rate. he data is plotted for a constant value of heat flux and mass flow ranging from 0.5 to 1.9 kg/min. It is seen that the heat transfer coefficient increases at higher mass fluxes.

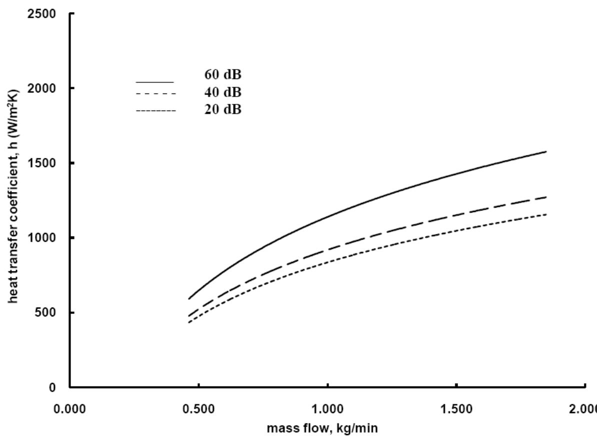

Fig 4 Effect of vibration levels on heat transfer in micro channels with a constant acoustic frequency of 60 kHz heat flux of q"=15 kW/m^2

In Figure 4, the experimental heat transfer coefficient is plotted at different values of impulse rates 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 510 to 545 W/m^{2}K at lower mass fluxes ranging from 0.45 to 0.85 kg/min and 580 to 1650 W/m^{2}K for higher mass fluxes ranging from 0.9 to 1.9 kg/min. For a heat flux at 29 kW/m^2 the heat transfer coefficient ranged between 545 and 840 W/m^{2}K at lower mass fluxes ranging from 0.5 to 0.9 kg/min and 880 to 1645 W/m^{2}K for higher mass fluxes ranging from 0.9 to 1.9 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, which is at the micro scale versus larger diameters. Comparing Figures 4 and 5, we can notice the influence of increasing the heat flux from 15 to 29 kW/m^2 on the heat transfer coefficient and is believed to be due to the impact of the impulse rate increase. The heat transfer coefficient increases in value for the lower heat flux ranging from 0.5 to 0.9 kg/min. Comparing both figures for impulse rates 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 12%.

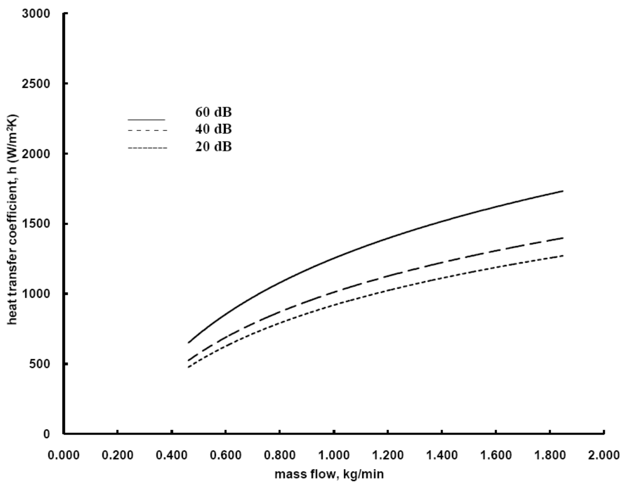

Fig. 5 Effect of vibration levels on heat transfer in micro channels with a constant acoustic frequency of 60 kHz heat flux of q"=29 kW/m^2

4. Conclusion

The study in heat transfer characteristics improvement of subcooled boiling and saturated boiling of the refrigerant mixture R-407C in micro channels is presented. The mass flow rate has a positive influence on the 'h' values, and more so near the onset of nucleate boiling especially with the excitation elements as the value of impulse was increased. The heat fluxes for the channels are also noticeably higher with the enhancement with impulse acoustic vibration effects on the flow. Furthermore, it also can be concluded that increasing heat flux by 35% increases the heat transfer coefficient by approximately 8% for the subcooled boiling process, but increasing heat flux by 42% increases the heat transfer coefficient by 12% for the saturated boiling process.

Enhancement of heat transfer was also studied by implementing impulse acoustic effect rates at (20, 40, and 60 dB) on the micro channels. Increasing the impulse rates from 20 to 40 dBs at a constant acoustic frequency of 60 kHz increased the heat transfer coefficient up to 8% for the subcooled process, where increasing the impulse rate from 40 to 60 dBs increased the heat transfer coefficient up to 12% for the saturated boiling process. Using larger gradients in the impulse rate 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)
i	impulse (dB)
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	
ρ	density (kg/m^3)

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