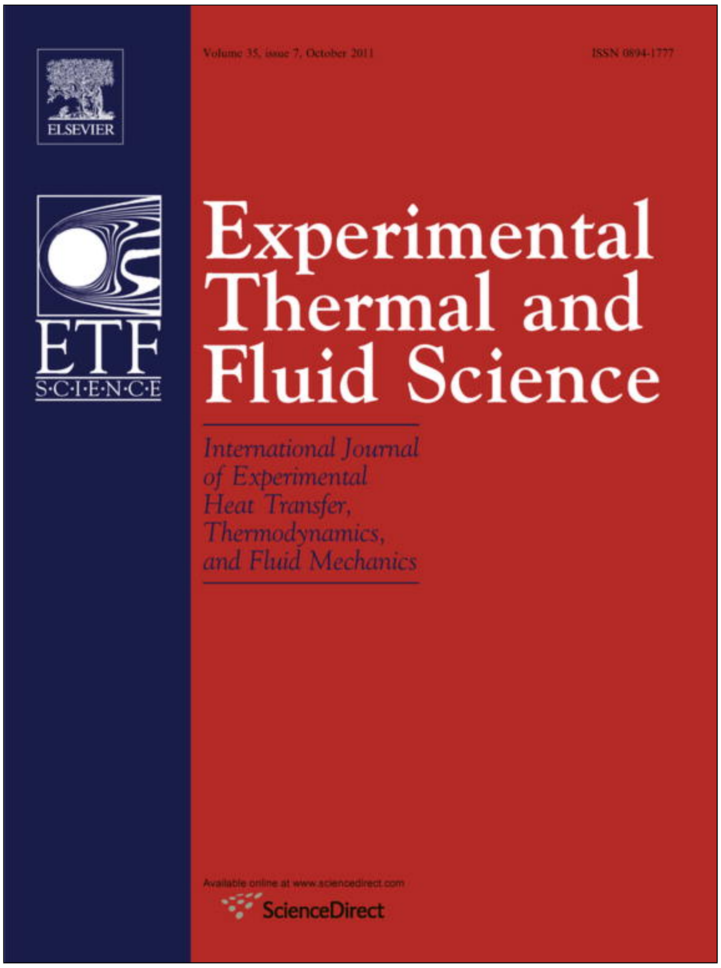
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Characteristics of R-123 two-phase flow through micro-scale short tube orifice for design of a small cooling system

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ABSTRACT

We present a new discharge coefficient correction method for the orifice equation for R-123 two-phase flows. In this method, an evaporator is mounted after the orifice as a vapor refrigeration cycle, and the evaporator is used to measure the quality of downstream flow through the orifice. Quality is estimated from the measured temperature and pressure of the evaporator inlet and outlet, respectively, instead of by direct measurement of quality. The condition of upstream flow of the orifice is the liquid state at 3 bar and 60 °C. The liquid flow is changed to two-phase flow after passing through the orifice. Orifice diameters of 300, 350, 400, and 450 μm are used for the experiment, and the results are analyzed. Experiments are conducted for various conditions of flow rate between 20 and 70 ml/min and for cooling loads of 60, 80, and 100 W. The results show that the quality of flow downstream from the orifice can be calculated using the enthalpy difference between the inlet and outlet of the evaporator. An equation to determine the discharge coefficient is formulated as a function of quality. We expect that these results can be used to help design a small cooling system.

1. Introduction

Recently, heat dissipation in portable electronic devices has become a crucial issue [1–3]. As integrated electronics become faster, more heat is generated. Passive cooling systems such as fan-fins and heat pipes are commonly used; however, they have limitations due to the available heat transfer efficiency. Therefore, small coolers based on a vapor compression refrigeration cycle have been suggested as a possible solution. We proposed a stack-type active cooler with a size of $50 \times 50\ \text{mm}^2$ [4]. Recently, a prototype of the cooler was designed and tested as shown in Fig. 1.

In the vapor-compression refrigeration cycle, an expansion device is necessary for the isenthalpic process between the condenser and the evaporator. The expansion device receives liquid refrigerant from the condenser and converts the liquid into two-phase flow composed of gas and liquid. Since wet steam causes poor compressor performance, it is important that the flow downstream of the orifice has sufficient quality to maintain constant superheat at the outlet of the evaporator. An orifice is commonly used as a simple expansion device in a small cooling system. As a passive expansion device, the design of the shape and dimensions of the orifice is very important because these parameters significantly affect performance. Liu et al. [5] experimentally investigated and analyzed effects of orifice design parameters of R744 two-phase flow. Nilpueng and Wongwises [6] investigated two-phase flow characteristics of HFC-134a through short tube orifices.

The discharge coefficient (C_d) in the orifice equation is a specific characteristic of the orifice and is typically derived from experimental data. Generally, C_d for two-phase flow is not defined as a constant, so C_d needs a correction method based on experimental data. Kim and O'Neal [7,8] applied a nonlinear regression technique to correct C_d according to the experimental data. Diener and Schmidt [9] applied a homogeneous nonequilibrium method to estimate C_d. Tu et al. [10] revised C_d based on empirically determined constants in single-phase flow and two-phase flow. Han et al. [11] analyzed two-phase flow characteristics through a short tube orifice by numerical modeling approach. The common variable in these methods is the quality of the orifice downstream flow. However, quality is difficult to measure directly.

We suggest a simple C_d correction method using an evaporator. A micro-evaporator [12] was mounted on the orifice outlet, and the temperature and pressure of the evaporator were measured instead of measuring quality directly. Temperature and pressure can easily and accurately be measured using commercial sensors. The enthalpy difference can be obtained from the superheat at the evaporator outlet, and the quality of the flow at the evaporator inlet can be calculated by the enthalpy difference, which is the same as the quality of the orifice outlet. A correlation equation between the quality and the C_d of the orifice can be derived from experimental data. The corrected C_d, which is a function of quality,

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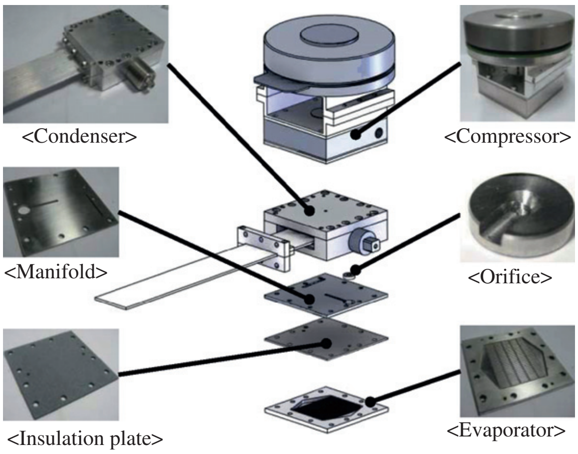

Fig. 1. Design of a small cooling system.

is used to predict the status of the two-phase flow through the orifice.

2. Problem definition

The quality of flow downstream from an orifice is an essential factor in correcting the value of C_d in the orifice equation in two-phase flow. Measuring the superheat of an orifice-mounted evaporator outlet is a substitute for measuring the quality of downstream flow of the orifice. Proof is given by the following theoretical background. A model of an orifice and an evaporator system is shown in Fig. 2.

2.1. Theoretical background

2.1.1. Orifice equation

An orifice is characterized by the aspect ratio of length and diameter L/D, where L is the orifice length and D is the orifice diameter. In single-phase flow through a thin-walled orifice ($L/D < 0.125$), C_d is calculated using Eq. (1) with the mass flow rate, the pressure difference between the inlet and outlet of the orifice, the orifice cross-sectional area, and the density of the refrigerant. In a thick-walled orifice ($0.125 < L/D < 2$) and a short tube orifice or orifice tube ($L/D > 2$), the conventional orifice equation described by Eq. (1) is still applicable. Typically, C_d is a function of diameter ratio (β) and Reynolds number (Re) for a single phase. The diameter ratio β is given by the orifice diameter D divided by the pipe diameter D'. Ramamurthi and Nandakumar [13] showed that the C_d of two-phase flow through a short tube orifice is not a function of the Reynolds number in a separated flow. Moreover, C_d needs additional correction when the flow has a phase change through a short tube orifice. The main characteristic of two-phase flow is vapor quality; thus, many papers suggest that C_d can be adjusted by the quality of the flow [7–10]. We have

$$\dot{m} = C_d A\sqrt{2\rho\Delta P(1-\beta^2)}, \quad \beta = \frac{D}{D'}, \tag{1}$$

where A is the area of the orifice hole, ρ is the density of the flow, and ΔP is the differential pressure between the inlet and outlet of the orifice.

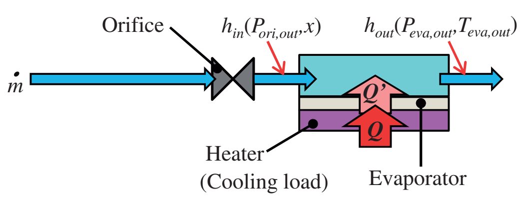

Fig. 2. Diagram of system modeling: h_{in} and h_{out} are the enthalpy of the refrigerant at the inlet and outlet of the evaporator, respectively; $P_{ori,out}$ and x are the pressure and vapor quality of the downstream flow through the orifice; $P_{eva,out}$ and $T_{eva,out}$ are the pressure and temperature at the evaporator outlet; Q is the cooling load; Q' is the heat transfer inside evaporator; and $\dot{m}$ is the mass flow rate.

2.1.2. Evaporator modeling at steady state

The evaporator mounted after the orifice transfers heat from the cooling load to the refrigerant. In the steady state, the enthalpy at the inlet is a function of quality and pressure. The enthalpy at the outlet is a function of pressure and temperature. Heat exchange (Q') occurs due to the difference in enthalpy. Here, the cooling load (Q) was assumed to be equal to the heat exchange (Q') based on three assumptions: (i) the thermal conductivity is high enough; (ii) the bottom plate is very thin; and (iii) the housing of the evaporator is well insulated. Then, Eq. (2) represents the relation between the enthalpy difference and the cooling load as follows:

$$Q = Q' = \dot{m}(h_{out} - h_{in}). \tag{2}$$

2.1.3. Correction of C_d based on superheat at the evaporator outlet

The quality of downstream flow through the orifice can be derived using the enthalpy of the refrigerant vapor at the evaporator inlet. The enthalpy of the evaporator outlet can be calculated using the temperature and pressure of the evaporator outlet. The cooling load of the heater is a variable that can be controlled by an operator. Since the cooling load is the same as the heat exchange, the enthalpy of the evaporator inlet can be calculated using Eq. (2). If the pressure of the evaporator inlet (which is same pressure as the downstream flow through the orifice) can be measured, the vapor quality of the evaporator inlet can be estimated using the enthalpy. Then, C_d can calculated using Eq. (1) from the mass flow rate and the pressure difference between the inlet and outlet of the orifice. We can measure every variable in Eq. (2) except C_d. Therefore, it is straightforward to calculate C_d using Eq. (1). This procedure is summarized in Fig. 3. We derived the correlation between the vapor quality and C_d from the measured values. The correction method for C_d was derived as a second-order function of the quality, and the resulting correction function for C_d varies linearly according to the cooling load. The function is described in more detail in Section 4.2.

3. Experimental

A diagram of the experimental apparatus is shown in Fig. 4. The apparatus consists of a transfer section, an orifice section, and an

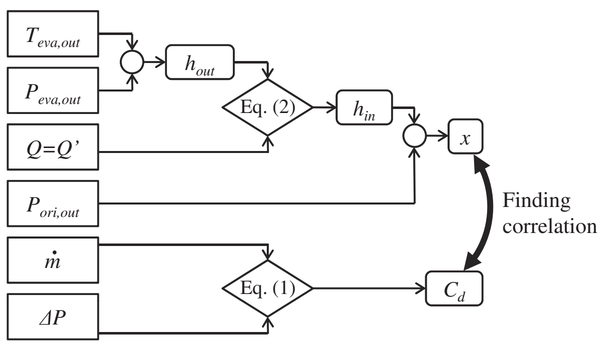

Fig. 3. Analysis procedure.

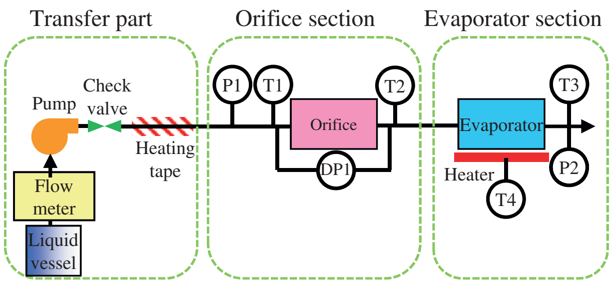

Fig. 4. Schematic diagram of the experimental apparatus.

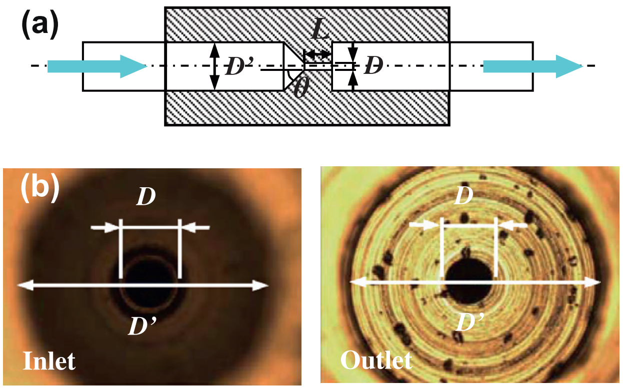

Fig. 5. Orifice design: (a) section view; (b) microscope image of the orifice (350 μm diameter).

evaporator section. The transfer section is composed of the liquid vessel, a flow meter, a liquid pump, and heating tape. The R-123 is liquid at room temperature and pressure; therefore, we used a liquid volumetric pump (KPV-22-SF-S, Cheon-Sei Co., Korea) that controlled the flow rate in a range of 0–100 ml/min. The float flow meter (custom made, KITS, Korea) for liquid R-123 was located between the liquid vessel and the pump. The heating tape worked as a pre-heater to adjust the experimental conditions of the orifice inlet.

The orifice section was comprised of the orifice sample and sensors. A pressure sensor (P1, Model 206, Setra Systems, Boxborough, USA), differential pressure transducers (DP1, Model 230, Setra Systems) and two type-K thermocouples (T1, T2) were used to measure the pressures and the temperatures of the refrigerant at the inlet and the outlet of the orifice. The estimated accuracy of the temperature measurements was ±0.5% full scale. The pressure transducer was calibrated to guarantee 0.044% accuracy. The structure of the orifice is shown in Fig. 5a. The entrance angle (θ) is 45∘, and the aspect ratio (L/D) is about 2. The diameters of the orifice samples were 300, 350, 400, and 450 μm. As shown in Fig. 5b, the diameters of the orifices were inspected using a microscope. The inside diameter of the pipe (D') at the inlet, and the outlet of the orifice, were both 2 mm. The geometric parameters of the orifice sample are summarized in Table 1.

A micro-evaporator with a lateral gap, which was used in a previous study by the authors, was installed in the evaporator section [12]. A cartridge heater with a thermocouple (T4) was attached under the evaporator to apply a cooling load. A direct current (DC) power supply (0–30 V, 0–10 A) supplied constant power to the cartridge heater. The pressure transducer (P2) and the thermocouple (T3) measured the pressure and temperature of the refrigerant at the outlet of the evaporator. All of the experimental equipment was well insulated with expanded polystyrene tubes and polyether ether ketone (PEEK) housings.

Table 1
Geometric parameters of the orifice samples.

	Parameter	Values			
D	Orifice diameter	300 μm	350 μm	400 μm	450 μm
L	Orifice length	Determined from the aspect ratio (L/D) of 2.			
θ	Entrance angle	45°			
D'	Pipe diameter	2 mm			

Table 2
Experimental conditions and control variables.

	Property	Value
Experimental conditions	Refrigerant	R-123
	Inlet temperature	60 °C
	Inlet pressure	3 bar
Control variables	Orifice diameter	300, 350, 400, 450 μm
	Flow rate	20–70 ml/min
	Cooling load	60, 80, 100 W

3.1. Pre-experiment

3.1.1. Single-phase flow through the orifice

A single-phase flow experiment without the heating tape and the evaporator was conducted as a preliminary test. The volumetric pump supplied liquid refrigerant at various flow rates. The pressure difference between the inlet and outlet was measured for each orifice sample. Using the orifice equation (Eq. (1)), C_d was calculated for each orifice sample. Our experimental results show the change of C_d according to the diameter in single-phase flow through the orifice. These experimental results are going to be used as comparison data set to explain the reason that the correction of C_d is required in two-phase flow. Also, the feasibility of our orifice samples and experimental equipments can be verified by comparing the measured C_d with the results from Ramamurthi and Nadakumar [13].

3.1.2. Two-phase flow through the orifice (liquid upstream/two-phase downstream)

We examined the performance of the orifices in two-phase flow without the evaporator. The heating tape upstream from the orifice gave inlet conditions of liquid 60 ± 1 °C and 3 ± 0.1 bar. The downstream flow became two-phase vapor since the pressure of the orifice outlet was atmospheric pressure. These conditions are determined from the refrigeration cycle of a small cooling system which we aims to develop [4]. C_d was calculated at various flow rates and orifice diameters (300, 350, 400, and 450 μm) with fixed inlet conditions. Contrary to the experimental results for a constant C_d in single-phase flow, the experimental results for two-phase flow shows why C_d needs correction in multiphase flow.

3.2. Two-phase flow through the orifice and evaporator

The evaporator section was mounted after the orifice section as shown in Fig. 3. The inlet conditions were liquid flow at 60 ± 1 °C and 3 ± 0.1 bar. The cartridge heater under the evaporator supplied a cooling load of 60, 80, and 100 W. The volumetric pump controlled the liquid flow rate in the range from 20 to 70 ml/min for orifice samples with diameters of 300, 350, 400, and 450 μm. The orifice diameter, flow rate, and cooling load were controllable variables. The inlet conditions and the range of control variables are summarized in Table 2.

All parameters were measured at steady state. Data when the temperature of the cartridge heaters were increased infinitely and the pressure of the upstream flow of the orifices were in-

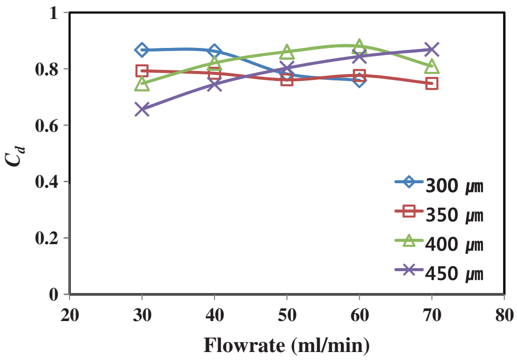

Fig. 6. C_d of each orifice diameter in single-phase flow.

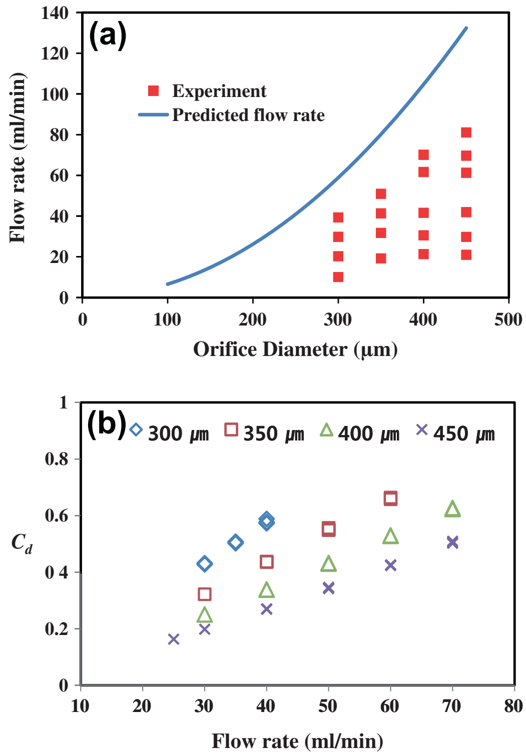

Fig. 7. (a) Comparison of predicted and measured flow rate, and (b) C_d of each orifice diameter in two-phase flow.

creased infinitely due to low flow rate were not used in the analysis. At a cooling load of 60 W, the performance of flow through the orifice with a diameter of 400 μm did not yield consistent data compared to the other experiments. Thus, the data for this experiment were not included in the analysis.

4. Result and analysis

4.1. Result of pre-experiment

4.1.1. C_d in single-phase flow

In the single-phase flow experiment, both the upstream and downstream conditions of the orifice are in the liquid state. The C_d for single-phase flow was calculated using Eq. (1) based on the measured pressure difference, flow rate, and orifice diameter. The resulting C_d is shown in Fig. 6.

The C_d is known as a function of Reynolds number and aspect ratio in single-phase flow. In a specific condition, the C_d can be treated as a constant. Ramamurthi and Nadakumar reported that the C_d of the 300 μm orifice of water flow is a constant when the Reynolds number is over 6×10^3; it means the liquid flow through the orifices separated [13]. Tu et al. [10] also reported constant C_d in the liquid flow in a specific range. In the liquid R-123 flow experiment, the ranges of Reynolds number of a constant C_d are 6.8×10^3–1.4×10^4, 5.8×10^3–1.4×10^4, 5.1×10^3–1.2×10^4, and 4.5×10^3–1.1×10^4 for 300, 350, 400, and 450 micro-orifices, respectively. In the cases of 300, 350 micro-orifices, the plot of Fig. 6 is very similar to the result reported by Ramamurthi and Nadakumar [13]. We note the relatively small C_d of 450 μm diameter orifice in 30 ml/min flow rate is due to the small Reynolds number of the flow. The C_d of single-phase flow has much less deviation than that of two-phase flow. The average of the C_d is 0.8 and the standard deviation is 0.06. We conclude that the micro- scale short tube orifice can use a constant C_d within limited range of flow rate of the liquid R-123.

4.1.2. Comparison between the orifice equation and experimental results in two-phase flow

The temperature and pressure of the liquid upstream flow were kept constant at 60 °C and 3 bar by the heating tape. The downstream flow was two-phase vapor at atmospheric pressure. By controlling the pump, the flow rate was varied from 20 to 70 ml/min. Fig. 7 shows the experimental results. In Fig. 7a, the blue[1] line shows the predicted flow rate from Eq. (1) using the C_d of 0.8 from the single-phase flow experiment. In the single-phase flow, the flow rate is determined to be a constant when the pressure drop and the orifice diameter are fixed. On the contrary, the results in the two-phase flow, denoted as a red dots in Fig. 7a, has distributed range of flow rate for each orifice though the pressure drop is fixed. The experimental data were measured as lower values than the predicted flow rate for single-phase flow. The lower flow rate also suggests that there was a pressure drop due to phase change while the flow passed through the orifice. The relation between C_d and flow rate is shown in Fig. 7b. Note that C_d was not a constant in two-phase R-123 flow; thus, C_d requires correction according to the flow condition.

4.2. Results of two-phase flow through the orifice and evaporator

4.2.1. Derivation of quality

Vapor quality is very important property of two-phase flow. Derivation of quality explains the variation of flow rate in constant pressure drop. In order to calculate qualities of various flow rate, we installed an evaporator after the orifice and measured the difference of enthalpy.

Enthalpy at the evaporator outlet can be obtained from the measured temperature and pressure. Using the evaporator model in Eq. (2), the enthalpy difference was calculated from the flow rate and the cooling load. Since enthalpy at the evaporator inlet is a function of pressure and quality, the quality of the evaporator inlet was derived from measured pressure and temperature. Fig. 8 shows the effect of flow rate, orifice diameter, and cooling load on quality. The fact that quality increased as the flow rate increased is as we expected. Note that the quality of two-phase flow through the short tube orifice was not affected by the orifice diameter, as shown in Fig. 8. We conclude that the quality was strongly related to the flow rate.

As shown in Fig. 8, we measured different vapor quality values for each cooling load. We believe that one of the main reasons for the change is the pressure variation between the orifice and the evaporator. As the cooling load was increased, the inside pressure

[1] For interpretation of color in Fig. 7, the reader is referred to the web version of this article.

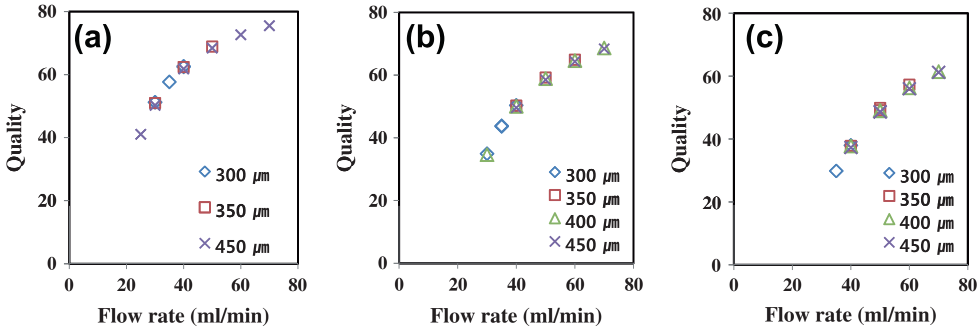

Fig. 8. Relationship between quality and flow rate for each orifice diameter: (a) 60 W, (b) 80 W, (c) 100 W.

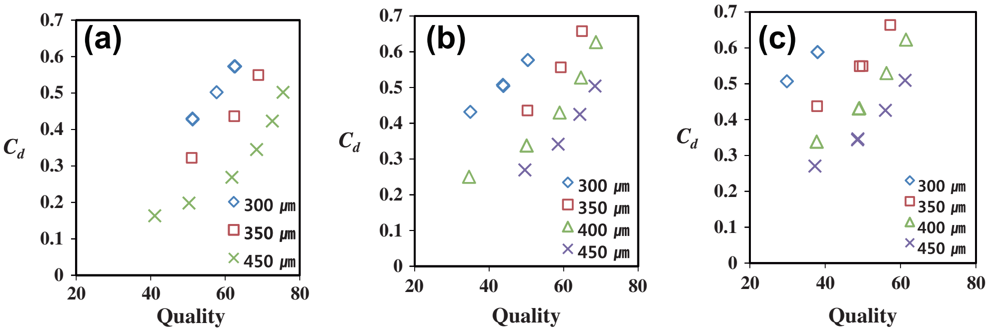

Fig. 9. C_d for orifices with different cooling loads: (a) 60 W, (b) 80 W, (c) 100 W.

of the evaporator increased. High pressure in the evaporator caused back-pressure on the orifice. Since the orifice was connected directly to the evaporator, pressure between the orifice and the evaporator cannot be controlled. The experimental data show that orifice modeling should account for the coupling effect with the evaporator in order to design the cooling system.

4.2.2. Correction of C_d of the orifice equation

In two-phase flow, the correction of C_d is required to compensate the variation of flow rate in constant pressure drop. In the previous research as Tu et al. [10], the C_d was commonly corrected as a function of vapor quality. The correction should be performed based on extensive empirical data.

By substituting the measured values into Eq. (1), C_d for each case can be calculated. The resulting C_d shows a proportional relation to the quality of downstream flow through the orifice, as shown in Fig. 9. C_d increased as the orifice diameter decreased because C_d is a function of diameter ratio, β, in Eq. (1). We suggest the C_d as a function of the quality at the orifice outlet and the diameter ratio of the orifice as previously explained in Fig. 3.

The correction equation for C_d can be deduced from the experimental data by interpolation, as shown in Eq. (3). A second-order polynomial function of quality x was used to estimated C_d to minimize the estimation error. Diameter ratio ($\beta = D/D'$) is one of the variables to calculate C_d. The final C_d is shown in Eq. (3). Three constants (a, b, and c) were determined to minimize the error. The values of a and b were determined from the experimental results: $a = 2.8$ and $b = 2$. Note that value of constant c varied by 0.048, 0.064, and 0.080 according to cooling loads of 60, 80, and 100 W, respectively. This correction method is capable of predicting C_d with an average accuracy of 91% with standard deviation error of 8% based on our experimental results. Fig. 10 shows the estimation results of C_d for various orifice diameters. A semi-empirical flow model for two-phase flow through a micro-scale short tube orifice could be developed based on the correction of C_d for the orifice equation:

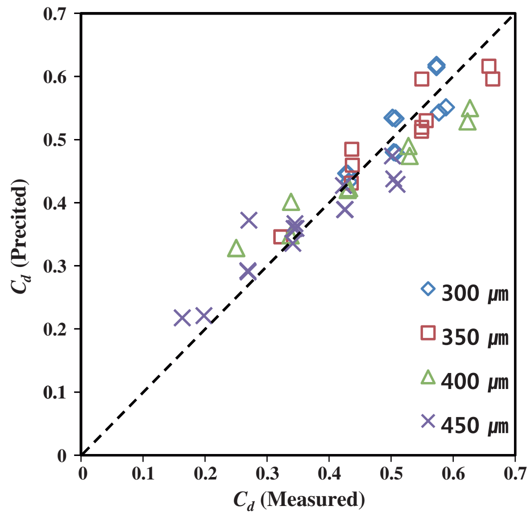

Fig. 10. Comparison between Eq. (3) and experimental data.

$$C_d = a(x - b \cdot \beta)^2 + \frac{c}{\beta}. \qquad (3)$$

We would like to note this correction equation is limited to the experimental condition of the setting. The results cannot be directly applied to other cases since this is semi-empirical model for the pre-defined refrigeration cycle. Refrigerant is also limited to R-123. Upstream of the orifice is liquid flow of 60° and 3 bar, and downstream of orifice is two-phase flow in atmospheric pressure. We expect the simple correction method of the C_d of a orifice by using an evaporator can be applied to other cases. Moreover, the empirical correction equation is going to be used to design the small size cooling system. The correction method can increase the robustness of the small size cooler since the equation not only considers the characteristics of the orifice but also gives us correlated data with the evaporator.

5. Conclusions

We suggest a correction method for C_d based on flow quality. Instead of measuring the quality of downstream flow through the orifice directly, the quality was calculated using experimentally measured temperature and pressure data from the evaporator mounted after the orifice. The correction equation for C_d was derived from a correlation of C_d, quality, and the orifice diameter ratio. The correction equation for C_d needs modification as the cooling load of the evaporator varies. The equation can predict C_d with an average accuracy of 91% with standard deviation error of 8% for a given set of experimental conditions. As part of the initial design of a small cooling system, the proposed correction method could be used to determine the specifications of other components such as compressors and condensers.

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