MICROCHANNEL HEAT EXCHANGERS WITH CARBON DIOXIDE

Final Report

Date Published – September 2001

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EXECUTIVE SUMMARY

The objective of the present study was to determine the performance of CO₂ microchannel evaporators and gas coolers in operational conditions representing those of residential heat pumps. A set of breadboard prototype microchannel evaporators and gas coolers was developed and tested. The refrigerant in the heat exchangers followed a counter cross-flow path with respect to the airflow direction. The test conditions corresponded to the typical operating conditions of residential heat pumps. In addition, a second set of commercial microchannel evaporators and gas coolers was tested for a less comprehensive range of operating conditions. The test results were reduced and a comprehensive data analysis, including comparison with the previous studies in this field, was performed. Capacity and pressure drop of the evaporator and gas cooler for the range of parameters studied were analyzed and are documented in this report. A gas cooler performance prediction model based on non-dimensional parameters was also developed and results are discussed as well.

In addition, in the present study, experiments were conducted to evaluate capacities and pressure drops for sub-critical CO₂ flow boiling and transcritical CO₂ gas cooling in microchannel heat exchangers. An extensive review of the literature failed to indicate any previous systematic study in this area, suggesting a lack of fundamental understanding of the phenomena and a lack of comprehensive data that would quantify the performance potential of CO₂ microchannel heat exchangers for the application at hand.

All experimental tests were successfully conducted with an energy balance within $\pm 3\%$. The only exceptions to this were experiments at very low saturation temperatures

(-23 °C), where energy balances were as high as 10%. In the case of evaporators, it was found that a lower saturation temperature (especially when moisture condensation occurs) improves the overall heat transfer coefficient significantly. However, under such conditions, air side pressure drop also increases when moisture condensation occurs. An increase in airflow rate also increases the overall heat transfer coefficient. Air side pressure drop mainly depends on airflow rate. For the gas cooler, a significant portion of the heat transfer occurred in the first heat exchanger module on the refrigerant inlet side. The temperature and pressure of CO₂ significantly affect the heat transfer and fluid flow characteristics due to some important properties (such as specific heat, density, and viscosity). In the transcritical region, performance of CO₂ strongly depends on the operating temperature and pressure.

Semi-empirical models were developed for predictions of CO₂ evaporator and gas cooler system capacities. The evaporator model introduced two new factors to account for the effects of air-side moisture condensate and refrigerant outlet superheat. The model agreed with the experimental results within $\pm 13\%$. The gas cooler model, based on non-dimensional parameters, successfully predicted the experimental results within $\pm 20\%$.

Recommendations for future work on this project include redesigning headers and/or introducing flow mixers to avoid flow mal-distribution problems, devising new defrosting techniques, and improving numerical models. These recommendations are described in more detail at the end of this report.

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NOMENCLATURE

A	surface area, m^2
AFR	air flow rate, m^3/h
Bta	superheat related factor, $Bta = \frac{T_{ref-out} - T_{ref-sat}}{T_{air-in-dry} - T_{ref-sat}}$
C	specific heat, kJ/kgK
Cta	moisture condensate factor, $Cta = \frac{T_{air-in-dew} - T_{ref-sat}}{T_{air-in-dry} - T_{ref-sat}}$
D	diameter, m
D_h	hydraulic diameter, m
DP	pressure drop, kPa
DT	temperature difference, $^{\circ}C$
eff	heat exchanger effectiveness
G	mass flux, kg/m^2s
h	specific enthalpy, kJ/kg
h_{fg}	latent heat, kW/kg
j	fin related coefficient
k	thermal conductivity, W/mK
K	flow coefficient
L	length, m
LMTD	log mean temperature difference
$\dot{m}$	mass flow rate, kg/s
MFR	mass flow rate of refrigerant, gr/s
NTU	number of exchanger heat transfer units
Nu	Nusselt Number, (hD/k)
P	pressure, kPa
Pr	Prant Number
Q	rate of heat transfer, kW
Re	Reynolds Number, (GD/μ)

S	specific entropy, kJ/kgK
St	Stanton Number, $Nu/(Re \cdot Pr)$
T	temperature, °C
U	overall heat transfer coefficient, W/m ² K
x	vapor quality
Z	capacity rate ratio

Greek Symbols

μ	viscosity, Pa·s
β	Diameter ratio
δ	fin spacing, m
ρ	density, kg/m ³

Subscripts

air	air stream
ave	average
cri	critical point
dew	dew point
dry	dry bulb
face	face area of heat exchangers
ideal	ideal conditions
in	inlet
local	local values
out	outlet
p	constant pressure
ref	refrigerant
RTD	resistance temperature detector
sat	saturation
sup	superheat
total	total heat transfer rate
tp	two phase
wet	wet bulb

CHAPTER 1 INTRODUCTION

The objective of the present study was to determine the performance of CO₂ microchannel evaporators and gas coolers in operational conditions representing those of residential heat pumps.

A set of breadboard prototype microchannel evaporators and gas coolers was developed and tested. The refrigerant in the heat exchangers followed a counter cross-flow path with respect to the airflow direction. The test conditions corresponded to the typical operating conditions of residential heat pumps. Capacity and pressure drop of the evaporator and gas cooler for the range of parameters studied were analyzed and are documented in this report. Semi-empirical models were also developed for prediction of CO₂ evaporator and gas cooler capacities. The experimental results are discussed in this report. This chapter presents an overview of natural refrigerants.

1.1 Phase-out of Refrigerants

Refrigerants are the working fluids in refrigeration, air-conditioning, and heat pump systems. An "ideal" refrigerant is chemically stable and inert, has excellent thermal and fluid flow characteristics, is compatible with common materials, is soluble in lubricating oils, is nontoxic, nonflammable, has low cost, and is environmentally acceptable. Since no single fluid meets all these attributes, a variety of refrigerants have been developed and applied in HVAC&R systems.

The Montreal Protocol is an international treaty that controls the production of ozone-depleting substances, including refrigerants containing chlorine and/or bromine

(U.N. 1994, 1996). The first version of the Protocol was signed September 16, 1987, by the European Economic Community (currently the European Union) and 24 nations, including the United States. As described in Chapter 18 of the ASHRAE Handbook of Fundamentals (2001), the Montreal Protocol was enacted on January 1, 1989, and limits the 1998 production of specified carbofluorocarbons (CFCs) to 50% of their 1986 levels. Starting in 1992, the production of specified halons (including R-13B1) was frozen at 1986 levels. Developing countries were granted additional time to meet these deadlines.

On June 14, 1994, the Copenhagen Amendment to the Montreal Protocol, ratified by 58 parties, was enacted. It called for a complete cessation of the production of CFCs by January 1, 1996, and of halons by January 1, 1994. Continued use from the existing (reclaimed or recycled) stock is permitted. Allowance is also provided for continued production for very limited "essential uses." In addition, hydrocarbofluorocarbons (HCFCs, including R-22) are to be phased out--according to a 1989 reference level for developed countries. Production was frozen at the reference level on January 1, 1996. Production will be limited to 65% of the reference level by January 1, 2004; to 35% by January 1, 2010; to 10% by January 1, 2015; and to 0.5% of the reference level by January 1, 2020. Complete cessation of the production of HCFCs is called for by January 1, 2030. In addition to the international agreement, individual countries may have domestic regulations for ozone-depleting compounds.

The production and use of hydrofluorocarbon (HFC) refrigerants (such as R-32, R-125, R-134a, and R-143a and their mixtures) are not regulated by the Montreal Protocol because they are not considered ozone depleting compounds. However, HFCs do have global warming potential because of their carbon content. Some individual

countries are beginning to regulate HFCs. Denmark, for example, is moving away from the use of HFC refrigerants. These facts indicate that the extensive use of synthetic refrigerants may be limited in the future and therefore should be used with caution. They also suggest that an alternative to HFC refrigerants could be useful if, in the future, these are phased out.

1.2 The Natural Refrigerants

"Natural refrigerants" refer to those naturally occurring substances, such as air, ammonia, carbon dioxide, isobutane, propane, and water. An overview of selected natural refrigerants is provided in Table 1.1. Since these substances are naturally occurring in our atmosphere, the use of these substances is expected to have minimal adverse effects on the environment.

The original application of natural refrigerants dates back to the middle of the nineteenth century when Linde, Perkins, Harrison, and others introduced pioneering refrigeration systems. In 1834, Perkins introduced the first refrigerant, sulfuric ether. From the 1840s through 1920s, the main refrigerants in practical use were ammonia (NH_3) for large and medium size stationary systems, sulfur dioxide (SO_2) for household refrigerators and small commercial plants, and carbon dioxide (CO_2) for ship installations, with brine as a secondary refrigerant. CO_2 was also often used in stationary systems (Elefsen et al., 1995).

Midgley and Henne (1930) published papers on fluorochemical refrigerants as a result of searching for stable, nontoxic, nonflammable, efficient refrigerants. In 1931, dichlorodifluoromethane, CFC-12, was commercially produced (Downing, 1966). After

the introduction of fluorochemical refrigerants, the early refrigerants, including CO₂, were replaced by many other CFCs, and later HCFCs. This course of action led to a drastic decline in the use of refrigerants other than CFCs and HCFCs after World War II. Only ammonia remained in use, though it was predominantly used only in large industrial systems.

Ohadi and Mo (1998) conducted a detailed review of natural refrigerants. The role of natural refrigerants in preventing or mitigating the problems associated with global warming and ozone depletion was addressed. Thermophysical properties and cycle performance of selected natural refrigerants were discussed and compared with their counterpart HCFC and HFC refrigerants. It was concluded that while HFC blends have been able to address acceptable Ozone Depletion Potential (ODP), their Global Warming Potential (GWP) is high enough to warrant a continued search for environmentally friendly refrigerants. The use of natural refrigerants appears to be one solution to this problem for immediate, as well as future applications. Research work on natural refrigerants is receiving renewed attention.

Table 1.1 Overview of selected natural refrigerants

Refrigerant	General Characteristics	Major Advantages	Major Disadvantages
Ammonia (NH ₃)	Ammonia is a well-known refrigerant in large scale industrial refrigeration plants. It has been used as a refrigerant for more than 120 years, but until now it has not been widely used in small plants.	<ul style="list-style-type: none"> -More than 120 years of practical use -Excellent thermodynamic and thermophysical properties -Higher energy efficiency in most temperature ranges -Well known oil tolerance -Great tolerance to water contamination -Simple and immediate leak detection -No ODP or GWP -Lower cost -Smaller pipe dimensions leading to lower plant investments 	<ul style="list-style-type: none"> -Toxic at low concentrations in air (above 500ppm) -No tolerance to some materials, e.g. copper -No miscibility with most known oils -High discharge temperatures -Flammable at 15-30% Vol.
Water (H ₂ O)	Environmentally attractive, water has potential as a long-term acceptable refrigerant. Water offers high plant energy efficiency.	<ul style="list-style-type: none"> -Higher Carnot COP due to the use of direct heat exchanger -High mechanical efficiency of compressor -Production of vacuum ice -Low energy consumption -No ODP or GWP 	<ul style="list-style-type: none"> -Process under vacuum -Good only for cooling/refrigeration above 0°C
Carbon Dioxide (CO ₂)	In recent years, after the Montreal Protocol, much development activity has been devoted to CO ₂ as a refrigerant. This development is based on new material technology which allows high pressures in the thermodynamic cycle. CO ₂ is quite harmless; it is environmentally attractive, and is neither toxic, flammable nor explosive.	<ul style="list-style-type: none"> -Low weight and small dimensions of plant -Large refrigeration capacity -Tolerance with well known oils -Low compression ratio -Low environmental impact -Low price, ample supply 	<ul style="list-style-type: none"> -High pressure -Low critical temperature (31°C)
Hydro-carbons	The measures taken to find suitable “natural” refrigerants as substitutes for CFC and HCFC have called attention to two hydrocarbons (propane and isobutane) that have properties similar to the most widely used CFCs and HCFCs.	<ul style="list-style-type: none"> -Compatible with materials normally used in refrigeration plants, such as copper and mineral oil -Similar physical properties to CFC-12 (isobutane) and HCFC-22 (propane) -Small amount of refrigerant needed -Lower prices than HFCs -Low environmental impact 	<ul style="list-style-type: none"> -Flammable at concentration of 1-10% v/v (requires additional safety measures) -Smaller volumetric cooling capacity

1.3 Carbon Dioxide as Working Fluid

The development of refrigeration systems using CO₂ as a refrigerant dates back to 1866 when Thaddeus created an ice production machine using CO₂ (Thevenot, 1979). In 1880, Windhausen designed the first CO₂ compressor (Gosman, 1927). After the late 1800s, the use of CO₂ refrigeration systems increased. As a result of continuous efforts to improve efficiency, two-stage CO₂ machines were developed in 1889 in Great Britain (Thevenot, 1979). Later, the multiple-effect CO₂ cycle was developed by Voorhess in 1905 (Thevenot, 1979).

Although the use of CO₂ as a refrigerant declined drastically in the 1930s due to the appearance of CFCs and HCFCs, CO₂ as a natural and environmentally favorable refrigerant has gained more attention recently. One of the solutions to minimize ozone depletion and global warming concerns is the use of natural refrigerants. Hydrofluorocarbons (HFCs) are replacement candidates, but have the disadvantages of relatively high GWP, high cost, and unresolved issues regarding environmental impact. Natural refrigerants, such as NH₃ and CO₂, have a low GWP, no ODP, and no adverse environmental effects (Lorentzen, 1995). Its low toxicity, nonflammability, and low cost make CO₂ the preferred refrigerant when compared with NH₃.

In addition to its environmental advantages, CO₂ offers certain attractive thermal characteristics. General physical and chemical properties of CO₂, as well as comparisons with those of other refrigerants, are listed in Chapter 18 of the ASHRAE Handbook of Fundamentals (1997). The thermophysical properties listed include the standard designation of characteristic properties, such as molecular mass, critical points, etc. Electrical properties, performance comparisons with other refrigerants, a safety

classification, and CO₂'s effect on several other materials are also provided. A brief chart and a saturated table of thermodynamic properties are given in Chapter 19 of the ASHRAE Handbook of Fundamentals (1997).

As a refrigerant, CO₂ offers more variation in thermodynamic and transport properties than other conventional refrigerants, as mentioned by Devotta et al. (2000). Since the critical temperature of CO₂ is 31°C for air-conditioning applications, the heat rejection above 31°C is not by condensation, as in conventional systems, which will necessarily lead to some changes in system design. Due to its higher volumetric capacity, which is five times higher than that of HCFC-22, the size of the system can be reduced considerably.

Some important thermophysical properties of CO₂ and comparisons with R-134a are listed in Table 1.2. Carbon dioxide has a much smaller surface tension and liquid viscosity than R-134a. Lower surface tension facilitates bubble formation, thus resulting in higher heat transfer coefficients. Lower liquid viscosity causes a smaller pressure drop when CO₂ flows in a tube or channel.

Table 1.2 Thermophysical properties of CO₂ and R-134a at 5 / 10 / 15 °C

Refrigerant	CO ₂	R-134a
P _{sat} (MPa)	3.969/4.502/5.086	0.350/0.414/0.488
Latent Heat (kJ/kg)	214.6/196.8/176.2	194.8/190.9/186.7
Surface Tension (mN/m)	3.53/2.67/1.88	11.0/10.3/9.6
Liq. Density (kg/m ³)	899.6/861.5/821.3	1277.1/1260.2/1242.8
Vap. Density (kg/m ³)	114.8/135.3/161.0	17.1/20.2/23.7
Liq. Viscosity (μPa·s)	95.9/86.7/77.2	270.3/254.3/239.7
Vap. Viscosity (μPa·s)	15.4/16.1/17.0	11.2/11.4/11.7
Liq. C _p (kJ/kg·K)	2.73/3.01/3.44	1.35/1.37/1.38
Vap. C _p (kJ/kg·K)	2.21/2.62/3.30	0.91/0.93/0.96

1.4 Benefits

The research work performed here will provide a design methodology platform for enhanced/compact heat exchangers for high pressure working fluids such as CO₂. The research results will have significant effects on the heat exchanger industries, from design and manufacturing/operation to integration in practical CO₂ systems. The findings in this study will contribute to the development and production of a new generation of high performance heat exchangers that are suitable for high pressure refrigerants such as CO₂, while offering significantly reduced size, weight, and consumed materials for the heat exchanger. A more in-depth understanding of the corresponding heat transfer and pressure drop phenomena and their empirical modeling is another inherent benefit of this research.

CHAPTER 2 RESEARCH BACKGROUND

This chapter presents a brief review of the fundamentals of heat transfer in boiling and an extensive overview of the prior research efforts that have been conducted to investigate the heat transfer characteristics of pure refrigerant and refrigerant/oil mixtures in smooth tubes, enhanced tubes, and microchannels.

2.1 Previous Studies on CO₂ Heat Transfer

The open literature on CO₂ heat transfer is limited. Bredesen et al. (1997) investigated flow boiling of CO₂ in a smooth tube. The test section was a 7 mm-diameter aluminum tube with direct heating. Temperatures were measured at 10 different positions and the local heat transfer coefficient was calculated. Bredesen et al. found that CO₂ has a much higher heat transfer coefficient and much lower pressure drop than that experienced with halocarbons. In addition, their experimental results showed that as the heat flux was increased, the heat transfer coefficient increased considerably without pressure loss penalty. Moreover, high heat transfer coefficients could be obtained even with smaller mass flux and pressure drop. However, the explanation of this unexpected phenomenon was unconvincing in the paper.

Based on the experimental results of Bredesen et al. (1997), Hwang et al. (1997) investigated the applicability of six commonly used empirical correlations reported by Chen (1966), Bennett-Chen (1980), Gungor-Winston (1987), Shah (1976), Schrock-Grossman (1959), and Liu-Winterton (1991). It was found that the correlations had a large deviation (from 20% to 80%) when predicting the boiling heat transfer coefficient of CO₂. Hwang et al. proposed a new empirical model, the Modified Bennett-Chen

correlation, for CO₂ flow boiling in horizontal smooth tubes. They claimed that the new correlation could predict the heat transfer coefficients, consistent with Bredesen's results, to within a mean deviation of 14%.

Zhao et al. (1997) studied the boiling heat transfer characteristics of ammonia and CO₂ in horizontal smooth tubes. The test section was a tube with an inner diameter of 5.44 mm and a length of 1.78 m. A water-heating method was applied, and the average heat transfer coefficient was determined. Zhao et al. reported a slightly smaller heat transfer coefficient compared to that of Bredesen et al. (1997). The deviation between the two results could be due to different thermal-boundary conditions, i.e. constant heat flux vs. the convection boundary condition, and different boiling temperatures. Their results, however, showed the same trend as Bredesen's data. They also compared the typical values of the heat transfer coefficient of CO₂ with those for R-134a, R-12, and R-22, and found that the transfer coefficient of CO₂ is substantially higher.

Olson and Allen (1998) investigated the heat transfer characteristics of supercritical CO₂ in turbulent flow in a heated horizontal tube. The tested tube was 10.9 mm ID and was heated over a length of 2.47 m. Operating pressure was varied from 7.8 MPa to 13.1 MPa. It was found that the measured Nusselt number agreed with the constant-property Petukhov-Gnielinski correlation for turbulent tube flow to within 6.6% at high operation pressure. As the pressure was reduced toward the critical pressure (7.38 MPa), the measured Nusselt number diverged from the constant-property correlation. At low pressures, the heat transfer coefficient increased with increasing mass flux and/or heat flux.

A review of the existing literature on CO₂ studies indicates that most of the CO₂ research studies conducted so far have been focused on heat exchanger design for heat pumps or refrigerators. This fact indicates that experimental investigation of the heat transfer coefficient and the modeling of CO₂ has received less attention.

2.2 Previous Studies on Microchannel Heat Transfer

Heat transfer and fluid flow in microchannels have wide practical applications in highly specialized fields, such as bioengineering, microfabricated fluidic systems, and microelectronics. Lately, microchannels have been intensively used by the automotive air conditioning industry. The advantage of the microchannel lies in its high heat transfer coefficient and significant potential in decreasing the size of heat exchangers. Microchannels have almost completely replaced circular tubes in automotive condensers and have recently become the subject of study for automotive evaporators.

Compared with channels of normal size, microchannels have many heat transfer advantages. Since microchannels have an increased heat transfer surface area and a large surface-to-volume ratio, they provide a much higher heat transfer. This feature allows heat exchangers to become compact and light-weight. In addition, microchannels can support high heat flux with small temperature gradients. However, microchannels also have weaknesses, such as large pressure drop, high cost of manufacture, dirt clogging, and flow mal-distribution, especially for two-phase flows.

The hydraulic diameters of microchannels are quite small, typically 1 μm to 2000 μm , and the fluid flow and heat transfer in microchannels are expected to be, in some cases, substantially different from those encountered in the normal-sized tubes and

channels. A review of microchannel heat transfer indicates that the previous studies can be divided into single-phase and two-phase (condensation and evaporation) forced convection. In two-phase flows, the studies are concentrated on heat transfer coefficient, pressure drop, and critical heat flux.

Wang and Peng (1994), Peng and Wang (1994), and Peng et al. (1996, 1998) performed a series of tests on rectangular microchannels (with hydraulic diameters of 0.133 – 0.747 mm) machined into stainless steel plates. They found that the flow transition for single-phase flow occurred at a Reynolds number ($Re = \rho u D_h / \mu$) of 200 – 700. This critical Re for the flow transition was strongly affected by the hydraulic diameter of the microchannel, and it decreased for smaller hydraulic diameters. In addition, the range of flow transition was diminished and the fully developed turbulent flow occurred at a much lower Re . For flow boiling in microchannels, the small size of the microchannel resulted in a dramatically higher heat flux for liquid nucleation when the microchannel was sufficiently small. However, microchannel size, flow velocity, and inlet sub-cooling temperature had no significant effect on the heat transfer coefficient in the fully nucleate boiling regime.

Ravigururajan et al. (1996, 1998) studied the impact of size and geometry of microchannels on their heat transfer characteristics. For single-phase flow, they found that microchannels provided significantly higher heat transfer coefficients. They indicated that the higher heat-transfer coefficients might be attributed to the thinning of the boundary layer in the microchannels, although the use of microchannels increased the surface area significantly. They also found that parallel geometry microchannels could give a better heat transfer coefficient than diamond geometry microchannels. For flow

boiling, they inferred that the large number of channels per unit width (e.g. typically 25 or more channels per inch) results in a significantly higher heat transfer area. The two-phase flow heat transfer coefficients strongly depend on the channel's geometry, surface, and shape. The heat transfer coefficient decreases with increasing saturation temperature, and the pressure drop increases with increasing heat flux.

Tran et al. (1996) investigated laminar and turbulent boiling heat transfer in small circular and rectangular channels. They found that for wall superheats larger than 2.75°C , the nucleate boiling mechanism dominates the forced convection effect. Yang and Webb (1996) compared commonly used correlations with their experimental results. They found that Shah's (1979) correlation overpredicts the heat transfer coefficient, and that the correlation by Akers et al. (1959) is suitable for small mass fluxes. They also indicated that pressure drop increases with increasing mass flux and heat flux. Surface tension was found to play an important role in heat transfer.

Bau (1998) numerically investigated the optimization of channel shape in micro heat exchangers. An approximate theory was derived to compute the thermal resistance of flat-plate micro heat exchangers whose surfaces are heated with uniform flux. It was demonstrated that the thermal resistance could be minimized by proper selection of uniform channel geometry. The maximum hot surface temperature and its gradient could be further reduced by changing the channel cross-sectional dimensions as a function of the axial coordinate.

Tong et al. (1997) studied pressure drop with highly subcooled flow boiling in small-diameter tubes. In designing heat-removal systems utilizing high-heat-flux subcooled boiling, pressure drop is an important consideration. Tong et al. performed an

experimental investigation to identify the important parameters affecting pressure drop across small-diameter tubes in highly subcooled flow boiling. The effects of mass flux, inlet temperature, exit pressure, tube internal diameter, and length-to-diameter ratio on both single and two-phase pressure drop were studied and evaluated. The experimental results indicated that mass flux, tube diameter, and length-to-diameter ratio were the major parameters that altered the pressure-drop curves. Both single- and two-phase pressure drops increased with increasing mass flux and length-to-diameter ratio, but decreased with increasing internal diameter. Inlet temperature and exit pressure were shown to have a significant effect on two-phase pressure drop but a negligible effect on single-phase pressure drop.

Tables 2.1 and 2.2 summarize the studies of fluid flow and heat transfer in microchannels. A review of the single-phase and two-phase heat transfer characteristics in microchannels indicates that the two-phase heat transfer in microchannels is superior to single-phase. As indicated by Bowers and Mudawar (1994), single-phase microchannel heat exchangers react to high surface heat fluxes by a large stream-wise increase in coolant temperature, and a corresponding stream-wise increase in the heat sink temperature. This temperature increase is often detrimental to temperature-sensitive devices, such as electronic components. Two-phase heat sinks, on the other hand, rely mainly on latent heat, and maintain a stream-wise uniform coolant and heat sink temperature at a level set by the coolant saturation temperature. To diminish the detrimental effect of a stream-wise temperature increase, microchannel heat sinks that operate in single-phase often need a large flow rate. Two-phase heat sinks, on the other hand, utilize the latent heat of liquid evaporation, and require minimal coolant flow rates.

However, since flow boiling has a critical heat flux, if the applied heat flux exceeds the critical heat flux, the dry-out phenomenon may occur. Under dry-out conditions, the heat transfer coefficient will be dramatically reduced, resulting in a rapid increase in wall temperature. Therefore, in two-phase microchannel heat sinks, a safety factor should be considered.

Table 2.1 Summary of studies on single-phase flow in microchannels

Investigator	Channel Geometry and Size (μm)	Reynolds Number, Fluid	Remarks
Lancet, 1959	Gap: 580 to 640	8,000 - 40,000, air	<ul style="list-style-type: none"> Experimental f is much larger than the correlation prediction value up to 100%
Gambill and Bundy, 1961	Rectangular channel D_h : 1910 – 2670	9,000 - 270,000, water	<ul style="list-style-type: none"> Experimental f matches correlation prediction Experimental Nu is only slightly smaller than correlation prediction
Wu and Little, 1983	Trapezoidal channel D_h : 56 – 83	100 - 15,000, N_2 , H_2 , Ar	<ul style="list-style-type: none"> Friction factor depends on roughness Critical Re decreases with increasing surface roughness ($350 < Re < 900$) Experimental f is larger than correlation's prediction
Wu and Little, 1984	Trapezoidal channel D_h : 56 – 83	400 - 15,000, N_2	<ul style="list-style-type: none"> Critical Re from 1000 to 3000 Turbulent Nu higher than standard correlation prediction Reynolds analogy not valid
Acosta, 1985	Rectangular channel D_h : 960 – 380	500 - 15,000, water	<ul style="list-style-type: none"> Experimental f and Nu match correlation's prediction
Pfahler et al., 1991	Trapezoidal channel D_h : 0.96 – 39.7	0.0005 – 70	<ul style="list-style-type: none"> Experimental f is slightly smaller than the prediction value (by less than 25%)
Choi et al., 1991	Circular channel D_h : 3 to 81.2	20 - 2500	<ul style="list-style-type: none"> The critical Re for flow transition is 2300 For both laminar and turbulent flows, real f is 25% smaller than correlation prediction Experimental Nu is larger than that predicted by Dittus-Boelter correlation
Peng and Peterson, 1996	Rectangular channel D_h : 311 to 367	50 – 4000, water	<ul style="list-style-type: none"> Critical Re in the range of 200 to 1500 Flow transition occurs at smaller Re as the size of channel is decreased Friction factor depends on the height-to-width ratio of the channel.
Wang and Peng, 1994	Rectangular channel D_h : 311 to 747	50 – 4000, water, methanol	<ul style="list-style-type: none"> Experimental Nu is lower than that predicted by the Dittus-Boelter correlation The critical Re is in the range of 1000 to 1500
Yang and Webb, 1996	Rectangular channel D_h : 1564 to 2637	2500 - 25,000, R-12, R-134a	<ul style="list-style-type: none"> Experimental data agree with the predictions of the Petukhov correlation (within 10%)
Webb and Zhang, 1997	Rectangular channel D_h : 960 to 1310	5000 - 25,000, R-134a	<ul style="list-style-type: none"> Experimental results match those predicted by the Petukhov and Dittus-Boelter correlations
Ravigururajan et al., 1996	Rectangular channel D_h = 425	4650 – 3990, R-124	<ul style="list-style-type: none"> The thinning of the boundary layer is the major contributor to high heat transfer coefficient. Channel arrangement affects the heat transfer coefficient
Adams et al., 1998	Rectangular channel D_h : 760 to 1090	2600 - 23,000, water	<ul style="list-style-type: none"> Experimental Nu is larger than that predicted by the Gnielinski correlation, and the deviation increases with increasing D_h and Re

Table 2.2 Summary of studies on two-phase flow in microchannels

Investigator	Approach	Fluid	Channel Geometry (mm)	Studied
Bowers and Mudawar (1994)	Experimental	R-113	Circular $D = 0.51$	Boiling, h , ΔP , Critical Heat Flux
Yang and Webb (1996)	Experimental	R-12, R-134a	Rectangular $1.564 < D_h < 2.637$	Condensation, h , ΔP , Correlations
Tran et al. (1996)	Experimental	R-12, R-113	Circular and rectangular $D_h = 2.46$	Boiling, flow regime, Correlations
Peng et al. (1996)	Experimental	Water / methanol mixture	Rectangular $0.133 < D_h < 0.367$	Boiling, h , concentration of mixtures, and number of channels
Kureta (1997)	Experimental	water	Circular $2.0 < D < 6.0$	h , ΔP , and correlations
Tong et al. (1997)	Experimental	R-134a	Rectangular $1.05 < D_h < 2.44$	Flow boiling, ΔP
Petterson et al. (1998)	Experimental	CO ₂	Microchannel heat exchanger	Overall heat transfer coefficient
Peng et al. (1998)	Numerical	Water, methanol, R-12	Rectangular $0.15 < D_h < 0.646$	Boiling, presence of bubbles, new definitions: evaporating space, fictitious boiling
Ravigururajan (1998)	Experimental	R-124	Rectangular and diamond $D_h = 0.425$	Boiling heat transfer and pressure drop

2.3 Recent Microchannel Experimental Work in Our Laboratory

Extensive tests in the Advanced Heat Exchangers Laboratory at the Center for Environmental Energy Engineering at the University of Maryland have revealed attractive features of flow boiling of CO₂ in commercial microchannels. Figures 2.1 and 2.2 compare the heat transfer coefficient and pressure drop of CO₂ and R-134a for flow boiling in the same microchannels and for the same test conditions. The saturation temperature was 283 K. The inlet and outlet vapor qualities were 0.05 and 0.30,

respectively. Figure 2.1 indicates that the heat transfer coefficient of CO₂ is much higher (up to 200%,) than that of R-134a. Figure 2 indicates that the pressure drop of CO₂ is much lower (60%) than that of R-134a. Thus, CO₂ exhibits outstanding heat transfer characteristics compared to R-134a. The excellent characteristics of CO₂ are attributed to its unique thermal properties. At 283 K, the surface tension of CO₂ is 2.67 mN/m, which is only 1/4 that of R-134a. Moreover, the viscosity of CO₂ at 283 K is 86.7 $\mu\text{Pa}\cdot\text{s}$, which is much smaller than that of R-134a (254.3 $\mu\text{Pa}\cdot\text{s}$). Therefore, CO₂ has a much higher heat transfer coefficient and lower pressure drop than R-134a.

Figure 2.1 Heat transfer coefficient of CO₂ and R-134a (from experiments performed in our Advanced Heat Exchangers Laboratory)

Figure 2.2 Pressure drop of CO₂ and R-134a (from experiments performed in our Advanced Heat Exchangers Laboratory)

2.4 Microchannel Heat Exchangers for CO₂

Microchannel heat exchangers for CO₂ are different from those for an R-134a system in design and characteristics. This is because the operating pressure of a CO₂ system is much higher than that of an R-134a system. And also, since the typical operating conditions of CO₂ are near its critical region, the performance and heat transfer characteristics of the two kinds of heat exchangers are expected to be different.

The advantage of CO₂ microchannel heat exchangers lies not only in the high performance of microchannel heat transfer and the environmentally friendly nature of CO₂, but also on the fact that microchannels and CO₂ can offset the weaknesses of each other. One of the main weaknesses of microchannels is the tremendous flow resistance. Fortunately, CO₂ has very low viscosity, as shown in Table 1.2. Lower viscosity

corresponds with a lower pressure drop as refrigerant flows through the exchangers. As shown in Figure 2.2, the pressure drop of CO₂ is much lower (60%) than that of R-134a. This suggests that mass flow rate of CO₂ in microchannel heat exchangers can be designed to be much larger. In addition, CO₂ systems have high operating pressures. Higher system operating pressure for a microchannel evaporator means the system can tolerate larger refrigerant pressure drop without affecting saturation temperature significantly. On the other hand, microchannels are also suitable for high operating pressure, which is one of the main disadvantages of CO₂. As discussed above, smaller diameter tubes can withstand higher system pressures.

Research on microchannel heat exchangers for CO₂ is relatively new and the available information is limited. Pettersen et al. (1998) developed a microchannel heat exchanger for CO₂ and experimentally evaluated the overall heat transfer coefficient. They indicated that refrigerant-side heat transfer coefficients are higher than those of fluorocarbons, and therefore, the internal surface areas of heat exchangers can be reduced. Smaller tube and manifold dimensions reduce the heat exchanger size compared to those using R-134a. The temperature difference between the inlet air and the outlet refrigerant is much lower in CO₂ gas coolers than in baseline HFC and/or HCFC system condensers of equal size and capacity. The reduced refrigerant exit temperature has a noticeable influence on the coefficient of performance. It appears that the microchannel heat exchanger has the best overall heat transfer coefficient.

Cutler et al. (2000) developed a transcritical carbon dioxide environmental control unit by applying microchannel heat exchangers. They reported that the capacity of a microchannel evaporator increases with increasing refrigerant mass flow rates.

Pitla et al. (2000) numerically analyzed heat exchangers for transcritical CO₂ systems. They suggested that experimental results were hard to predict when the operating conditions were close to the critical point.

Ortiz and Groll (2000) developed a finite-element model to study a microchannel CO₂ evaporator. The model was based on the assumption that a refrigerant-side heat transfer coefficient has a negligible effect on volumetric capacity. They concluded that the volumetric cooling capacity of the microchannel evaporator increases with increasing air-side heat transfer coefficients and is nearly constant with respect to refrigerant-side heat transfer coefficients.

A search of the literature indicates that a clear understanding of the performance and potential of CO₂ microchannel heat exchangers is lacking. Therefore, the intention of this project is to characterize the performance of a current generation of CO₂ heat exchangers based on tests at controlled operating conditions.

CHAPTER 3 TEST FACILITIES AND SYSTEM COMPONENTS

The test facility used in this study measures the capacities of microchannel heat exchangers, including the evaporator and gas cooler. During system operation, the gas cooler and evaporator will be separated from each other through the use of different air ducts in separate rooms, thus allowing for independent fine control of the inlet air stream conditions (including temperature and relative humidity) for each heat exchanger. Since the sizes of the available microchannel heat exchangers are large, available facilities (indoor loop and outdoor duct) at the CEEE Heat Pump Laboratory could not satisfy the test requirements. In order to fulfill the tasks set forth, a new indoor loop and an outdoor duct were built and tested.

3.1 Indoor Loop

As shown in Figure 3.1, the indoor loop contains an air handler unit, a fan, an upstream screen, an upstream thermocouple grid (3x3 thermocouples), an upstream air sampling tree, a microchannel evaporator, a downstream thermocouple grid (3x3 thermocouples), a downstream air sampling tree, a screen, and a flow nozzle. The air handler unit is used to adjust the inlet air conditions (temperature and humidity), while the fan circulates the airflow inside the loop. The fan speed is adjustable, and thus airflow rate can be controlled. The screens help by allowing a more uniform airflow. The thermocouple grids are made from nine thermocouples, arranged 3 x 3 uniformly on the cross sectional area of the air duct, so that the mean temperature of the nine thermocouple readings can represent the bulk temperature of the air stream. Two dew point meters

(chilled mirror type with an accuracy of ± 0.2 °C) are used to determine the humidity of the upstream and downstream airflow. Single point measurement of the humidity might not be sufficient to determine the bulk humidity of the air stream since the humidity may slightly vary across the cross sectional area of the air duct. To determine the bulk humidity more accurately, air-sampling trees, shown in Figure 3.2, were designed to suck air uniformly from nine positions in the cross section of the air duct. The flow nozzle was used to measure the airflow rate inside the indoor loop.

The air duct was constructed from polypropylene and insulated with 25.4 mm thickness of thermal insulation material ($k = 0.04$ W/mK).

Figure 3.1 Schematic of indoor loop

Figure 3.2 Air sampling tree

3.2 Outdoor Duct

The outdoor duct, shown in Figure 3.3, was built inside another environmental chamber. The outdoor duct houses the microchannel gas cooler. It is also constructed from polypropylene. The duct contains screens, upstream and downstream thermocouple grids, a gas cooler, an air mixer, an obstruction meter, and a large fan. Similar to that of the indoor loop, screens are used to make the air stream uniform while thermocouple grids measure the bulk temperatures of the air stream. The fan, controlled by a variable speed motor, was placed at the outlet of the duct where it draws air through the duct. The air duct is also insulated with 25 mm thickness of thermal insulation material ($k = 0.04$ W/mK).

Figure 3.3 Schematic of outdoor duct

Due to the large cross sectional area of the outdoor duct (1.219 m x 0.914 m), CEEE's current flow rate measurement devices (typically flow nozzles) were not suitable for this task. For this reason, an obstruction flow meter (shown in Figure 3.4) was designed, fabricated, and calibrated. The flow meter was made of 117 circular holes with a diameter of 25 mm. Since these small holes were uniformly deployed across almost the entire cross sectional area of the test duct, the flow and thermal fields were relatively uniform for the present situation.

Fin strip heaters were used to calibrate the obstruction flow meter. Figure 3.5 shows the energy balance for the calibrated obstruction flow meter. For the airflow rate range of interest to this project, the obstruction flow meter can measure airflow rate within $\pm 2\%$. For more details, see Appendix I.

Figure 3.4 Energy balance for the calibrated obstruction flow meter

Figure 3.5 The obstruction flow meter

3.3 Microchannel Heat Exchangers

Microchannel heat exchangers were provided by Hydro Aluminum. The microchannel used is shown in Figure 3.6.

Figure 3.6 Microchannels from Hydro Aluminum

Both the evaporator and the gas cooler are made from several microchannel unit slabs. A schematic diagram and picture of one of these unit slabs are shown in Figures 3.7 and 3.8. It is important to note that the two halves of the heat exchange slab are non-communicative, so that flexibility in choosing refrigerant paths may be ensured. Moreover, the stubs providing refrigerant access to the headers were placed in the middle of each section in order to reduce the possibility of flow mal-distribution, especially for the evaporator. The specifications of each unit slab are as follows:

- One unit has two passes of 17 parallel microchannels, overall surface area of 3 m² (see Figures 3.7 and 3.8)
- Al 3003-O stub, 9.5 mm OD, 5.4 mm ID (see Figure 3.9)
- Header is single tube, 348 mm long, 21.3 mm OC, 17.7 mm ID (see Figure 3.9)
- Louvered fin density: 16 fins/25.4 mm, fin height: 8.0 mm (see Figure 3.10)
- Total flow cross sectional area: $17 \times 10 \times (3.14 \times 1 \times 1) / 4 = 133.5 \text{ mm}^2$
- Refrigerant side heat transfer area: $34 \times 10 \times (3.14 \times 1) \times 430 = 0.46 \text{ m}^2$

Figure 3.7 Schematic of a microchannel unit slab

Figure 3.8 Picture of a unit slab

Figure 3.9 Picture of header and stub

Figure 3.10 Picture of fins and microchannels

3.3.1 Evaporator

The evaporator is comprised of two microchannel unit slabs. Figure 3.11 shows a schematic of the two unit slabs placed side by side with the microchannels in a vertical orientation to allow for gravity driven condensation on the air side. Refrigerant flows in a serpentine fashion through each of the two halves of the two slabs.

Figure 3.11 Schematic of the evaporator stack layout

3.3.2 Gas Cooler

The gas cooler, shown in Figure 3.12, is comprised of 10 microchannel unit slabs. Two slabs were set into a frame side by side, and then stacked parallel to the air flow five units deep. The refrigerant from the discharge of the compressor was brought to the rear of this unit, divided by two, and then routed through the two parallel stacks of slabs until it exited at the front of the stacks where the two streams were recombined. This exit point was the point where the incoming air stream entered. This design is a counter-crossflow pattern, which improves the heat transfer performance. Eleven thermocouples were mounted to measure the refrigerant temperatures along the flow path, as shown in Figure 3.12 by the letter T. These thermocouples help to determine the heat transfer rate of each individual slab.

Figure 3.12 Schematic of gas cooler layout

3.4 Refrigerant Loop

The refrigerant loop, shown in Figure 3.13, is made up of two parts that are located in separate chambers: an indoor chamber and an outdoor chamber. The two parts are piped together by stainless steel tubes with an inner diameter of 12.3 mm. In the indoor chamber, the refrigerant loop contains an expansion valve, a microchannel evaporator, an accumulator, thermocouples, and pressure transducers. The refrigerant loop inside of the outdoor chamber is comprised of a CO₂ compressor, an oil separator, a microchannel gas cooler, a Coriolis mass flow meter, a suction line heat exchanger, thermocouples, and pressure transducers.

Figure 3.13 Schematic of refrigerant loop

3.5 Compressor

The tests were carried out with the use of a prototype Dorin semi-hermetic CO₂ compressor. The compressor is a constant speed/fixed displacement unit that runs at 1750 RPM and can provide a mass flow rate of CO₂ up to 60 g/s. The compressor, which has logged more than 1500 hours of run time, and has shown no excessive wear, is shown in Figure 3.14. Its specifications are as follows:

Type:	Reciprocating, Semi-hermetic
Piston Number:	2
Displace Volume:	30.9 cm ³
Clearance:	1.2% of cylinder volume
Motor:	3 kW, 1750 RPM, 3 Phase, 208-230 V, 60 Hz

Figure 3.14 A photograph of the Dorin semi-hermetic CO₂ compressor

3.6 Data Acquisition System

Signals from the air- and refrigerant-side measurement devices were fed to

LabView data acquisition software through the use of National Instruments' Field Point DAQ modules. These modules allow for flexibility in instrumentation, as additional channels may be added or removed easily if needed. These modules (rather than the computer) may also be placed close to the individual parts of the experiment, eliminating both excessive cable lengths (through the use of an RS-485 networking scheme) and problems arising from incorrect wiring.

Currently, 96 channels of data are collected (64 thermocouple and 32 analog inputs) and sent to the computer for collation and instantaneous on-screen visualization of system parameters (e.g. pressures, temperatures, air flow rates, etc.). The smallest tested sampling rate of this system is 1 second.

Figure 3.15 shows a screen shot of the data acquisition program written for the experiment. Numeric outputs monitored include air side temperatures, air flow rates, dew points, performance (including COP, compressor work, and both latent and sensible cooling loads), refrigerant pressures, mass flow rate, and in-stream temperatures. The graphical portion of the program monitors the history of many of these same measurements in addition to a few additional data sets.

Figure 3.15 Screen shot of the data acquisition program written for the system

3.7 Instrumentation and Measurements

3.7.1 Pressure measurement

Pressure measurements are made using two types of instruments: absolute pressure transducers and differential pressure transducers. The absolute pressure transducers were used at the compressor suction and discharge and the expansion valve inlet and outlet. These measurements were made in conjunction with differential pressure transducers used to more accurately measure pressure drop across the heat exchanger stacks. The differential pressure transducers used for the evaporator and gas cooler stacks have an accuracy of 0.2% and 0.1% of full-scale output, respectively.

3.7.2 Air Stream Temperature Measurement

Air stream temperatures are measured by thermocouple grids. Each grid has 9 (3x3) T-type thermocouples deployed uniformly across the cross sectional area. The thermocouples were calibrated before installation. Since the air side capacity is determined by the difference between average upstream temperature and average downstream temperature, specific tests were conducted to determine the deviations of the average upstream temperature and average downstream temperature with respect to RTD readings. The results are shown in Figure 3.16, where it is suggested that the deviations between the average upstream temperature and average downstream temperature are within 0.05 °C.

Figure 3.16 Deviations between upstream average temperature and downstream average temperature

3.7.3 Coriolis Mass Flow Meter

Refrigerant mass flow was measured with a Coriolis type mass flow meter (Micro Motion R-Series) placed downstream of the single-phase gas cooler outlet. The output signal of 4-20 mA was adjusted to correspond to a range of 0-100 g/s, the operating range expected in the experimental system.

The Micro Motion R-Series flowmeter measures the mass and volume of liquids, gases and slurries within 0.5% of flow rate. It has no moving parts to wear out or break down, and mechanical installation is made easier with no straight run or flow conditioning requirements.

3.7.4 Dewpoint Temperature Measurement

The dewpoint temperatures of the air stream were measured by dewpoint meters. The dew point meters are Model M2 Plus from General Eastern. The Model M2 Plus is a general purpose chilled mirror hygrometer with advanced capabilities, suitable for use in a large number of applications. It utilizes advanced microprocessor control and includes General Eastern's patented PACER™ technology. PACER™ is designed to reduce contamination induced errors and to provide long, unattended operation in industrial applications. 4-20mA and 0-5 VDC analog outputs and RS-232C digital output are provided. The RS-232C port allows the analog outputs to be scaled and provides direct communication with a host computer. It can support an accuracy of +/- 0.2°C on dewpoint measurement.

CHAPTER 4 EXPERIMENTAL RESULTS FOR MICRO-CHANNEL EVAPORATOR

4.1 Experimental Test Conditions

Experimental test conditions for evaporation, as specified in the proposal of Microchannel Heat Exchangers with Carbon Dioxide (Ohadi, et al., 1999), were tabulated in Table 4.1.

Vapor qualities at the inlet of the evaporator are specified in the proposed test conditions as 20% and 50% respectively. Since direct measurement of vapor quality at the inlet of the evaporator is almost impossible, inlet vapor qualities were determined by assuming an adiabatic process through the expansion valve. The state of the refrigerant at the outlet of the gas cooler can be easily determined by measuring the refrigerant pressure and temperature as long as a supercritical state of the refrigerant is secured in the gas cooler. The piping line and expansion valve were insulated by thermal insulation materials with a thermal conductivity of 0.04 W/mK.

To obtain the specified vapor quality at the inlet of the evaporator, the required states at the outlet of the gas cooler were calculated according to the adiabatic process through the expansion valve. The required states at the outlet of the gas cooler as well as all other proposed test conditions were converted into SI units and tabulated in Table 4.2. In the proposal, the refrigerant saturation temperatures for test conditions series No. 19 – 22, were proposed as $-23\text{ }^{\circ}\text{C}$, and air side conditions were $-8\text{ }^{\circ}\text{C}/-9\text{ }^{\circ}\text{C}$ (dry bulb/wet bulb). However, the present system could not reach a steady state because of frozen moisture on the outer surface of the evaporator. Thus, air conditions were adjusted to $-8\text{ }^{\circ}\text{C}$ dry air to avoid the formation of frost on the coil.

Table 4.1 Experimental test conditions for evaporation (English Units)

Tests	Serial No.	Test conditions							
		CO ₂ (Saturated)			Air				
		X _{in}	T _{in} (°F)	T _{out} (°F)	Flow rate (CFM)	T (°F)	P (Psi)		
Evaporator	1	20%	35	55	330	Dry bulb :80 Wet bulb :67	Around 14.7		
	2				380				
	3				430				
	4		45	65	330				
	5				380				
	6				430				
	7	55	75	330					
	8			380					
	9			430					
	10	50%	35	55	330				
	11				380				
	12				430				
	13		45	65	330				
	14				380				
	15				430				
	16	55	75	330					
	17			380					
	18			430					
	19	20%	-10	10	250 (ft/min)	Dry bulb: 17 Wet bulb: 15			
	20	50%			500 (ft/min)				
	21				250 (ft/min)				
	22				500 (ft/min)				

Table 4.2 Experimental test conditions for evaporation (SI Units)

Tests	Serial No.	Test conditions							
		CO ₂ (Saturated)			Air				
		X _{in}	P/T (MPa/°C)	T _{out} (°C)	Flow rate (m ³ /h)	T (°C)	P (kPa)		
Evaporator	1	20%	3.6/1.7	12.8	561	Dry bulb: 27 Wet bulb: 20	Around 100		
	2				646				
	3				731				
	4		4.2/7.2	18.3	561				
	5				646				
	6				731				
	7		4.8/12.8	23.9	561				
	8				646				
	9				731				
	10	50%	3.6/1.7	12.8	561				
	11				646				
	12				731				
	13		4.2/7.2	18.3	561				
	14				646				
	15				731				
	16		4.8/12.8	23.9	561				
	17				646				
	18				731				
	19	20%	1.4/-32	-22.2	1200	Dry bulb: -8 Dry air			
	20	50%			2400				
	21				1200				
	22				2400				

4.2 Experimental Test Results

The inlet and outlet air-side and refrigerant-side pressures and temperatures were measured during the evaporator tests. The measured data were used to determine the capacity and pressure drop. The air side capacity consists of two parts: the sensible heat part and latent heat part. The sensible heat part was calculated from the dry air enthalpy change and mass flow rate. The dry air enthalpy change was determined by the temperature difference multiplied by the air-specific heat. The latent heat part refers to the heat due to the moisture condensate. It was calculated from latent heat h_{fg} and the condensate flow rate $\dot{m}_{water-cond.}$ determined by collecting condensed water over a certain period (typical collecting period is 30 - 60 minutes). The determined condensate flow rates were double-checked by calculating the condensate flow rate based on the humidity difference of air stream across the evaporator measured by the two dewpoint meters (chilled mirror type with an accuracy of ± 0.2 °C)

The refrigerant-side capacity was determined by the mass flow rate and enthalpy difference between the inlet and outlet of the evaporator. Since the proposed inlet test conditions lay in the two-phase region, the enthalpy at the inlet of the evaporator could not be determined directly. However, the enthalpy at the inlet of the evaporator can be considered the same as the enthalpy at the outlet of the gas cooler since the piping line and expansion valve were perfectly insulated, and the process of expansion can be assumed as a constant enthalpy process. The data reduction uses the following equations:

$$Q_{air-sensi.} = \dot{m}_{air} \cdot C_p \cdot (T_{air-out} - T_{air-in}) \quad (4.1)$$

$$Q_{air-latent} = \dot{m}_{water-cond.} \cdot h_{fg} \quad (4.2)$$

$$Q_{air} = Q_{air-sensi.} + Q_{air-latent} \quad (4.3)$$

$$Q_{ref} = \dot{m}_{ref} \cdot (h_{ref-gc-out} - h_{ref-evp-out}) \quad (4.4)$$

$$EB = \frac{Q_{air} - Q_{ref}}{Q_{air}} \quad (4.5)$$

Table 4.3 Experimental data for evaporation (SI Units)

Table 4.3 Experimental results for evaporation (SI Units) (Cont.)

Serial No.	Air					Refrigerant					Heat				
	Upstream		Downstream		AFR	DP	X _{in}	T _{sat}	Superheat	MFR	Air				
	T _{dry} (°C)	T _{dew} (°C)	T _{dry} (°C)	T _{dew} (°C)	(m ³ /h)	Pa		(°C)	(°C)	G/s	Sens. (kW)	Latent (kW)	Total (kW)	Ref.	EB %
12a	26.5	15.8	14.5	12.7	727	6.6	0.40	2.2	8.9	29.1	3.01	1.27	4.28	4.37	-2.1
12b	26.9	15.9	14.6	12.2	731	6.6	0.39	2.2	8.8	28.9	3.09	1.47	4.57	4.43	3.1
13a	26.5	15.9	16.1	13.4	563	4.1	0.45	7.3	5.6	21.5	2.00	0.81	2.82	2.73	2.9
13b	26.9	16.2	15.4	13.0	561	4.7	0.46	7.3	5.1	25.2	2.20	1.04	3.24	3.16	2.6
13c	26.6	16.2	15.2	12.9	562	4.7	0.45	7.2	4.8	25.3	2.20	1.06	3.26	3.17	3.0
14	27.0	15.9	16.0	13.3	654	5.7	0.45	7.4	5.5	26.2	2.45	0.96	3.41	3.35	1.8
15a	26.8	15.5	15.7	12.7	729	6.5	0.43	7.1	5.4	28.7	2.75	1.14	3.89	3.78	2.6
15b	26.9	15.5	15.7	12.8	732	6.5	0.43	7.1	5.3	28.9	2.81	1.10	3.91	3.80	2.8
16	26.8	16.5	20.7	16.1	572	4.5	0.60	12.2	6.2	14.3	1.16	0.16	1.33	1.33	0.0
17a	26.8	16.2	19.9	15.8	658	5.3	0.59	12.8	5.3	18.5	1.53	0.19	1.73	1.70	1.4
17b	26.8	16.3	20.5	15.8	668	5.5	0.64	12.3	6.0	18.4	1.41	0.21	1.62	1.57	2.8
18a	26.9	16.1	20.0	15.6	745	6.3	0.56	12.8	5.0	19.4	1.72	0.21	1.93	1.88	2.4
18b	26.9	16.1	20.0	15.7	743	6.4	0.55	12.8	5.1	19.4	1.72	0.22	1.93	1.89	2.2
19	-8.3	-	-13.6	-	1109	20.2	0.30	-24.1	4.1	11.7	2.27	-	2.27	2.44	-7.4
20	-7.9	-	-11.7	-	2456	71.4	0.25	-25.1	2.5	17.8	3.65	-	3.65	3.94	-8.1
21	-7.9	-	-13.3	-	1116	21.4	0.69	-25.4	4.1	26.3	2.36	-	2.36	2.61	-10.6
22	-7.8	-	-11.8	-	2436	72.2	0.55	-25.2	8.1	29.0	3.82	-	3.82	4.10	-7.3

4.3 Data Reduction and Discussion

Energy balances were checked for all reported experimental data and found to be within $\pm 3\%$ except for test series 19 – 22, where the refrigerant saturation temperature was around $-23\text{ }^{\circ}\text{C}$. Since a saturation temperature of $-23\text{ }^{\circ}\text{C}$ is beyond the normal operating conditions of the present experimental setup, it was difficult for the system to become stabilized. Also, the air stream might still contain a very small amount of moisture and thus influence the energy balance results. Figure 4.1 shows that the energy balance for test series 1 – 18 is within $\pm 3\%$.

Figure 4.1 Energy balance for evaporation test results

Figure 4.2 depicts the cooling capacity of the evaporator vs. refrigerant mass flow rate for refrigerant inlet vapor qualities of 20% and 50% and certain specified outlet superheats. It is clearly shown that the cooling capacity of the evaporator is proportional to the refrigerant mass flow rate when the refrigerant outlet superheats are maintained. Increasing the inlet vapor quality shifts the curve to the right.

Ordinarily, air side conditions dominate the overall heat transfer coefficient of an evaporator since the air side thermal resistance is much larger than the refrigerant side when the refrigerant experiences phase changes. However, the capacity of an evaporator depends not only on the overall heat transfer coefficient, but also on the temperature difference between the air and refrigerant. This means that the two-phase region in an evaporator influences the capacity of the evaporator significantly by affecting the temperature difference. The capacity of an evaporator increases if the two-phase region occupies a larger area of the evaporator. However, it should be noted that this is only true when the refrigerant is superheated at the outlet of the evaporator.

Figure 4.2 Capacity of the evaporator vs. CO₂ mass flow rate (MFR)

To better understand the heat transfer process through the evaporator, overall heat transfer coefficients were calculated. The overall heat transfer coefficient is defined as U

$= q''/DT$, where q'' refers to the mean heat flux and DT refers to the temperature difference between the average air temperature (the mean value of upstream and downstream temperatures) and the refrigerant saturation temperature. Since pressure drops of refrigerant side are very small (less than 20 kPa, with the corresponding temperature change less than 0.2 °C) in the present tests, inlet saturation temperature was selected to represent the refrigerant saturation temperature.

Figure 4.3 depicts the influence of the airflow rate and saturation temperature on the overall heat transfer coefficient for an inlet quality of 20%. Saturation temperature strongly affects the overall heat transfer coefficient. A lower saturation temperature results in a higher overall heat transfer coefficient. This is because the moisture is condensed on the outer surface of the heat exchanger. It is well known that the overall heat transfer coefficient of an evaporator strongly depends on the air-side heat transfer coefficient. The air-side heat transfer is always very weak if the air is dry. However, if the moisture condenses or evaporation takes place, the heat transfer of the air-side will be enhanced significantly since latent heat is much more efficient than sensible heat in a heat transfer process. This will improve the air-side heat transfer coefficient significantly. The lower the saturation temperature, the higher the moisture condensate rate, and thus, higher enhancement on the air-side heat transfer. To verify the explanation experimentally, the ratio of the latent heat to total air-side heat transfer rate is depicted in Figure 4.4. These results clearly show that a lower saturation temperature is associated with a larger percentage of latent heat.

Figure 4.3 Overall heat transfer coefficient vs. air flow rate (AFR)

Figure 4.4 Ratio of latent heat to total heat of air side vs. air flow rate (AFR)

Figures 4.5 and 4.6 present the overall heat transfer coefficient and ratio of the latent heat to the total heat exchange of the air-side for a refrigerant inlet vapor quality of 50%, respectively. In Figure 4.5, the overall heat transfer coefficient for $T_{\text{sat}} = 1.7\text{ }^{\circ}\text{C}$ is comparable to that of $T_{\text{sat}} = 7.2\text{ }^{\circ}\text{C}$. This is due to the fact that, during the experiments, it was very difficult to precisely maintain the superheat at certain values. The experimental results for $T_{\text{sat}} = 1.7\text{ }^{\circ}\text{C}$ have larger outlet superheat ($> 8\text{ }^{\circ}\text{C}$) than the other two test serials (around $5\text{ }^{\circ}\text{C}$). Since larger superheats means larger heat transfer surface area occupied by superheated vapor, the overall heat transfer coefficient reduces.

Figures 4.7 and 4.8 present the air-side pressure drop as a function of air flow rate at different saturation temperatures for an inlet vapor quality of 20% and 50%, respectively. The air-side pressure drop is expected to increase with an increasing air flow rate. The pressure drop is slightly higher for lower saturation temperatures. This can be explained by the fact that more water condenses on the outer surface of the evaporator, retarding the air flow to a greater extent.

Figure 4.5 Overall heat transfer coefficient vs. air flow rate (AFR)

Figure 4.6 Ratio of latent heat to total heat of air side vs. air flow rate (AFR)

Figure 4.7 Pressure drop of air vs. air flow rate

Figure 4.8 Pressure drop of air vs. air flow rate

4.4 Capacity Predictive Model for Microchannel Evaporators

The present experiments provide extensive data under different test conditions. The data can be used as a platform to develop prediction models for the evaporator capacity.

The capacity of an evaporator can be expressed as:

$$Q = U \cdot A \cdot DT \quad (4.6)$$

From Figs 4.3 - 4.6, it is clear that overall heat transfer coefficient U depends on many factors such as moisture condensation, refrigerant outlet superheat, air flow rate, fin geometry, etc.

The experimental results show that a larger amount of moisture condensation will improve air side heat transfer coefficient and thus resulting in higher overall heat transfer

coefficient. To reflect the moisture condensate effect, a new factor, C_{ta} , is defined as follows:

$$\text{For } T_{\text{air-in-dew}} \geq T_{\text{ref-sat}}, \quad C_{ta} = \frac{T_{\text{air-in-dew}} - T_{\text{ref-sat}}}{T_{\text{air-in-dry}} - T_{\text{ref-sat}}} \quad (4.7)$$

$$\text{For } T_{\text{air-in-dew}} < T_{\text{ref-sat}}, \quad C_{ta} = 0$$

Figure 4.9 shows the overall heat transfer coefficient versus C_{ta} . Generally, the overall heat transfer coefficient increases with increasing C_{ta} . However, there is some data scatter as shown in Figure 4.9. This is due to the effects of some other factors. To develop a successful model, those effects also need to be addressed.

Figure 4.9 Overall heat transfer coefficient vs. C_{ta}

To reflect the effects of refrigerant superheat at the outlet of evaporator, another factor B_{ta} is defined as:

$$Bta = \frac{T_{ref-out} - T_{ref-sat}}{T_{air-in-dry} - T_{ref-sat}} \quad (4.8)$$

As shown in Figure 4.10, the overall heat transfer coefficient generally decreases with increasing Bta. It reflects the fact that, in a typical evaporator, the heat transfer performance decreases with increasing the refrigerant outlet superheat.

Figures 4.9 and 4.10 reflect the effects of moisture condensate and refrigerant superheat on the heat transfer performance of the evaporator.

Figure 4.10 Overall heat transfer coefficient vs. Bta

Therefore, a new model is proposed for predicting the capacity of the microchannel evaporator that accounts for the above-mentioned factors. It is as follows:

$$Q = A \cdot DT \cdot \frac{G_{air} \cdot C_p}{P_{1,air}^{2/3}} (0.0185 - 3.467 \times 10^{-5} Re_{air}) (1 + 6.343 Cta)^{0.8} (1 - 0.75 Bta)^{0.65} \quad (4.9)$$

$$\text{Re}_{air} = \frac{G_{air} \delta}{\mu_{air}} \quad (4.10)$$

$$G_{air} = \frac{\dot{m}_{air}}{A_{air-face}} \quad (4.11)$$

$A_{air-face}$ is face area of the evaporator regardless of the microchannel tubes and fins. δ refers to the fin spacing. DT is the difference between the average temperature of air upstream and downstream and refrigerant saturation temperature.

Figure 4.11 presents comparison of the predicted capacity with the experimental results. The proposed empirical model can predict experimental results within $\pm 13\%$ (for $Cta < 0.6$ and $Bta < 0.45$).

Figure 4.11 Comparison between the model predictions and experimental results

CHAPTER 5 EXPERIMENTAL TESTS FOR MICROCHANNEL GAS COOLER

In this chapter, the experimental data for the gas cooler test conditions will be presented. Then, data analysis and prediction models will be discussed.

5.1 Experimental Test Conditions

Experimental tests for the microchannel gas cooler were conducted in an outdoor chamber. The system was allowed about two hours to reach a steady-state condition for any specific test conditions. Data was taken once the steady-state conditions were achieved and remained constant for 30 minutes. Data was recorded for 40 minutes for each test condition and averaged over time.

Experimental test conditions for the gas cooler, as specified in the proposal of Microchannel Heat Exchangers with Carbon Dioxide (Ohadi, et al., 1999) are tabulated in Table 5.1. In the proposal, all parameters are given in SI Units. Table 5.2 lists the proposed test conditions in SI units.

The test conditions of $P_{in} = 13.79$ MPa (test series 15 – 21) were beyond the capacity of the current test facilities. In the present test setup, the Dorin compressor could only permit a high-side pressure of 12.5 MPa. Therefore, during the experiments, the refrigerant pressure at the inlet of the gas cooler for test serial No. 15 – 21 was set at around 12.5 MPa.

In Chapter 5, all properties related to air were based on dry air properties.

Table 5.1 Experimental test conditions for gas cooler (English Units)

Table 5.2 Experimental test conditions for gas cooler (SI Units)

Tests	Serial No.	Test conditions				
		CO ₂ (Superheated)		Air		
		P _{in} (MPa)	T _{in} (°C)	Flow rate (m ³ /h)	T (°C)	P (kPa)
Gas Cooler	1	7	78.9	1682	21.1	Around 100
	2			1937		
	3			2192		
	4			1298	27.8	
	5			2596		
	6			1298		
	7			2596	35	
	8	1682	21.1			
	9	1937				
	10	2192				
	11	1298	27.8			
	12	2596				
	13	1298		35		
	14	2596				
	15	1682	21.1			
	16	1937				
	17	2192				
	18	1298	27.8			
	19	2596				
	20	1298		35		
	21	2596				

5.2 Experimental Test Results

Table 5.3 lists all raw experimental data for the gas cooler test conditions. The first column in the table indicates the test serial number. Because refrigerant mass flow rates are not specified in the proposal, some test series were repeated at the same proposed conditions, but at a different refrigerant mass flow rate. This can help us to better understand the effect of refrigerant mass flow rate.

Since the inlet conditions (pressure and temperature) of the refrigerant at the gas cooler strongly depend on the discharge conditions of the compressor, pressure and temperature of the refrigerant at the gas cooler inlet are related to each other to some extent. Therefore, the refrigerant inlet pressure and temperature are very difficult to set perfectly simultaneously. This means that each experimental data point might have a very accurate match (normally within 0.5 °C for temperature, 5% for pressure) on either the proposed pressure or the temperature, but have a less accurate match on the other. Experimental tests showed that the pressure of the refrigerant has a stronger effect on the heat transfer process of the gas cooler than temperature.

Table 5.3 Experimental data for gas cooling (SI Units)

Serial No.	Air				Refrigerant													Heat		
	T _{in}	T _{out}	AFR	DP	P _{in}	P _{out}	MFR	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	Q _{ref}	Q _{air}
	°C	°C	M ³ /h	Pa	kPa	kPa	g/s	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	kW	kW
1a	21.2	35.2	1656	53	8134	8126	32.8	78.0	43.1	38.1	35.6	33.7	31.8	28.4	24.9	23.6	22.4	21.6	7.73	7.58
1b	21.3	29.8	1670	49	6948	6943	20.5	79.3	35.2	30.5	29.0	28.5	28.1	28.0	28.0	27.6	27.1	23.6	4.87	4.77
1c	21.4	29.8	1661	49	6947	6942	20.4	79.5	35.1	30.6	29.0	28.5	28.1	28.0	28.0	27.5	27.0	23.6	4.86	4.72
2a	20.9	32.4	1932	70	7703	7695	30.5	78.4	39.2	34.6	32.8	31.1	29.5	26.4	23.4	22.5	21.6	21.2	7.33	7.40
2b	20.8	28.8	1912	64	6881	6875	21.1	78.3	33.6	29.4	27.9	27.8	27.7	27.7	27.6	25.3	23.0	21.5	5.15	5.12
3a	21.5	32.4	2099	80	7896	7888	31.2	80.3	39.6	35.3	33.6	30.9	28.2	25.7	23.3	22.7	22.1	21.8	7.51	7.59
3b	20.3	32.4	2103	84	8937	8927	35.4	79.9	42.4	39.0	32.7	28.2	23.8	22.4	21.1	20.9	20.7	20.6	8.33	8.40
3c	20.3	32.4	2095	83	8989	8978	35.5	79.8	42.3	38.8	32.0	27.8	23.5	22.3	21.1	20.9	20.7	20.6	8.34	8.40
3d	20.8	28.2	2077	76	6981	6975	20.9	79.5	33.0	29.5	28.3	28.3	28.2	25.3	22.5	21.8	21.2	21.0	5.16	5.14
3e	20.9	28.5	2074	75	7000	6993	21.8	78.2	33.4	29.6	28.5	28.4	28.4	25.8	23.2	22.3	21.4	21.1	5.34	5.33
3f	20.9	28.6	2071	75	6999	6993	21.8	78.2	33.3	29.6	28.5	28.4	28.4	25.7	23.1	22.2	21.4	21.1	5.33	5.35
4	27.7	47.5	1273	33	9473	9461	39.4	79.8	56.2	50.2	46.4	44.4	42.4	40.2	37.9	35.1	32.3	29.7	8.02	7.83
5a	28.4	38.1	2554	113	8836	8825	35.9	83.1	45.4	41.0	38.4	36.0	33.6	31.8	30.0	29.5	29.0	28.8	7.75	7.95
5b	28.6	38.1	2576	113	8850	8839	35.9	83.2	45.4	41.1	38.4	36.0	33.6	31.9	30.1	29.6	29.1	28.9	7.73	7.96
6	35.1	44.5	1272	40	8773	8759	22.9	82.6	51.9	46.3	43.4	42.3	41.1	40.4	39.7	39.0	38.4	37.3	3.80	3.74
7	35.1	40.5	2545	122	8514	8501	26.0	81.1	47.6	42.8	40.5	39.7	38.8	38.3	37.9	37.5	37.1	36.4	4.24	4.36

Table 5.3 Experimental data for gas cooling (Cont.)

Serial No.	Air				Refrigerant														Heat	
	T _{in}	T _{out}	AFR	DP	P _{in}	P _{out}	MFR	T1	T2	T3	T4	T5	T6	T7	T8	T9	T10	T11	Q _{ref}	Q _{air}
	°C	°C	M ³ /h	Pa	kPa	kPa	g/s	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	°C	kW	kW
8	19.8	35.5	1675	58	10341	10324	33.2	98.3	48.7	42.1	31.2	26.8	22.4	21.3	20.3	20.1	19.9	19.8	8.53	8.61
9	21.2	34.6	1937	73	10661	10645	33.6	99.3	48.3	40.4	30.3	26.7	23.2	22.5	21.7	21.6	21.5	21.5	8.45	8.50
10	21.5	33.5	2186	88	10598	10582	33.7	98.0	46.8	38.8	29.1	26.1	23.1	22.6	22.0	21.9	21.8	21.8	8.42	8.62
11a	28.8	43.3	1303	27	10343	10326	26.4	98.5	54.8	48.6	42.7	38.6	34.5	32.5	30.5	29.8	29.2	29.2	6.10	5.98
11b	28.9	43.5	1302	27	10350	10332	26.4	98.6	54.8	48.6	42.7	38.6	34.5	32.5	30.5	29.9	29.2	29.3	6.10	5.97
12	29.8	39.3	2594	116	11203	11185	34.6	98.9	51.0	43.5	36.4	33.8	31.2	30.7	30.2	30.1	30.1	30.1	7.76	7.92
13	35.8	55.4	1318	38	11845	11826	39.6	96.4	66.9	59.2	53.2	49.6	46.0	43.3	40.6	39.1	37.6	36.9	7.75	7.67
14a	35.2	42.4	2577	111	10857	10837	26.8	102.3	52.1	46.3	41.2	39.0	36.9	36.3	35.8	35.7	35.5	35.6	5.74	5.83
14b	35.5	43.0	2567	111	10669	10648	28.8	98.5	52.2	46.7	42.0	39.8	37.5	36.8	36.1	35.9	35.8	35.8	5.98	6.14
15a	20.8	32.5	1671	60	11138	11118	22.6	117.8	43.7	33.2	24.1	22.8	21.5	21.3	21.1	21.1	21.1	21.0	6.31	6.49
15b	21.6	31.7	1673	53	13476	13450	21.5	118.7	39.0	28.6	23.4	22.8	22.1	22.1	22.0	22.0	21.9	22.0	5.69	5.63
15c	20.1	29.1	1674	55	13268	13242	17.9	130.1	31.2	24.3	21.0	20.7	20.4	20.4	20.3	20.3	20.3	20.5	5.13	5.06
15d	21.6	31.7	1673	53	13476	13450	21.5	118.7	39.0	28.6	23.4	22.8	22.1	22.1	22.0	22.0	21.9	22.0	5.69	5.63
16	21.0	29.9	1925	68	13543	13516	21.2	118.2	35.8	26.2	22.3	21.8	21.4	21.4	21.3	21.3	21.3	21.4	5.63	5.70
17a	21.4	29.5	2082	78	13592	13566	21.1	118.5	34.9	26.0	22.6	22.2	21.8	21.8	21.7	21.7	21.7	21.8	5.58	5.60
17b	21.5	29.4	2092	78	13668	13641	20.7	119.8	34.3	25.9	22.5	22.2	21.8	21.8	21.8	21.8	21.8	21.9	5.50	5.55
18	27.9	43.1	1291	39	12117	12093	24.3	122.0	55.8	46.6	35.5	32.6	29.7	29.0	28.4	28.3	28.1	28.3	6.35	6.20
19	28.1	36.0	2576	121	11940	11917	24.5	124.7	47.1	36.6	30.5	29.5	28.5	28.4	28.2	28.2	28.2	28.4	6.50	6.56
20	35.7	47.8	1286	43	12143	12118	21.1	120.1	57.9	50.3	41.8	39.5	37.2	36.7	36.1	36.0	35.9	36.1	4.99	4.85
21a	35.3	41.9	2582	124	12031	12005	22.4	119.7	51.0	42.7	37.5	36.6	35.7	35.6	35.4	35.4	35.4	35.5	5.31	5.47
21b	35.1	41.2	2579	124	12847	12819	19.3	133.7	48.5	40.0	36.3	35.8	35.3	35.3	35.3	35.3	35.3	35.4	4.94	5.01

5.3 Data Reduction and Discussion

The inlet and outlet air-side and refrigerant-side pressures and temperatures were measured during the gas cooler tests. The measured data were used to determine the capacity and pressure drop. The air-side capacity was calculated from air enthalpy change and mass flow rate. The air enthalpy change was determined by the temperature difference multiplied by air specific heat. Air specific heat was based on dry air properties. The values are listed in Table 5.3.

$$Q_{air} = \dot{m}_{air} \cdot C_p \cdot (T_{air-out} - T_{air-in}) \quad (5.1)$$

The refrigerant-side capacity was determined by the mass flow rate and enthalpy difference between the inlet and outlet of the gas cooler, as shown in Equation 5.2. The refrigerant side capacity is also listed in Table 5.3.

$$Q_{ref} = \dot{m}_{ref} \cdot (h_{ref-in} - h_{ref-out}) \quad (5.2)$$

Energy balances were checked for all reported experimental data using Equation 5.3:

$$EB = \frac{Q_{ref} - Q_{air}}{Q_{ref}} \quad (5.3)$$

Figure 5.1 depicts the energy balance results for all reported experimental results, clearly indicating that energy balances for all reported data were within $\pm 3\%$.

Figure 5.1 Energy balance for gas cooler test results

Figure 5.2 shows the gas cooler capacity vs. mass flow rate of the refrigerant for an air inlet temperature of 21 °C. The refrigerant-side inlet pressure was varied (P_{in}) from 6.9 MPa to 12.5 MPa. As seen there, the mass flow rate of the refrigerant has a significant effect on the capacity of the gas cooler. This is expected because the refrigerant capacity rate ($\dot{m}_{ref} \cdot C_p$) is much smaller than that of the air-side ($\dot{m}_{air} \cdot C_p$). Increasing refrigerant mass flow rate results in the augmentation of the capacity of the heat exchanger. These results also show that refrigerant mass flow rate plays a more important role than those of the inlet pressure and temperature of the refrigerant. Thus, the mass flow rate is the one of the primary factors that affects the capacity of a gas cooler.

Figure 5.2 Capacity vs. refrigerant mass flow rate

Figure 5.3 shows typical temperature profiles for the air and refrigerant streams in the gas cooler. The two arrows indicate the flow directions of refrigerant and air. A schematic of the gas cooler layout is shown in Figure 3.10. The flow pattern is a cross counter flow. Eleven thermocouples were installed in the refrigerant path to precisely measure and monitor the temperature of the refrigerant. The abscises in Figure 5.3 indicate the thermocouple locations. The air stream temperatures were determined by applying an energy balance to specific heat exchanger units. The temperature difference between the refrigerant and air reaches a maximum at the refrigerant inlet while the difference approaches zero at the air inlet. This is because the air capacity rate ($\dot{m}_{air} \cdot C_p$) is much larger than the refrigerant capacity rate ($\dot{m}_{ref} \cdot C_p$). The temperature profiles also

suggest that most of the heat transfer occurs at the heat exchanger units that are near the refrigerant inlet side. The last three slabs do not transfer any heat since there is no temperature difference.

Figure 5.3 Typical temperature profiles of air stream and refrigerant

Based on the temperature profile of the refrigerant, heat transfer rates for each HX unit can be determined. Figure 5.4 shows typical heat transfer rate profiles at various refrigerant pressures and temperatures. The results show that a large portion of the total heat transfer takes place in the heat exchanger no. 1 (HX1) that is located at the refrigerant inlet side with the higher refrigerant inlet pressure and temperature. In other words, the slope of Q_i/Q_{total} is steeper at the refrigerant inlet side for higher refrigerant inlet temperature and pressure.

Figure 5.4 Typical heat transfer rate profile for each HX unit

Figure 5.5 depicts the percentage of heat (Q_i/Q_{total}) transferred by each HX as a function of refrigerant inlet pressure for all gas cooler test results. It should be noted that higher refrigerant inlet pressures are associated with higher refrigerant inlet temperatures. The percentage of heat transferred by HX1 increases with increasing refrigerant inlet pressures. The HX5 does not transfer any heat, thus it did not appear in the figure. This indicates that over-sizing a gas cooler does not improve its capacity. It was found that no significant improvement in the capacity was obtained when more than four HXs were used in the current experiments (i.e., the number of HXs was oversized). Four HXs have a total surface area of 24 m^2 . For high refrigerant inlet pressures ($> 10 \text{ MPa}$), the first

three HXs transfer up to 95% of the heat. One benefit associated with this characteristic is that the size of the gas cooler can be reduced if it operates at high gas cooling pressures.

Another observation from Figure 5.5 is that for low refrigerant inlet pressures, the HXs that are near the outlet of refrigerant also play a significant role in the heat transfer process. This may be explained by the fact that the specific heat of CO₂ increases rapidly in the temperature range of 30 – 50 °C for low refrigerant inlet pressures, as shown in Figure 5.6. Since refrigerant capacity rate ($\dot{m}_{ref} \cdot C_p$) is proportional to specific heat, the heat transfer amount for low refrigerant inlet pressures is still significant for the temperature range of 30 – 50 °C.

Figure 5.5 Percentage of heat transferred by each HX vs. refrigerant inlet pressure

Figure 5.6 Specific heat of CO₂ vs. temperature

Because the specific heat of CO₂ changes significantly with variations in temperature, logarithmic mean temperature difference (LMTD defined in Equation 5.4) cannot be applied.

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \quad (5.4)$$

where ΔT_1 and ΔT_2 refer to the temperature differences between refrigerant and air at the inlet and outlet of the gas cooler, respectively.

LMTD is based on the assumptions that the overall heat transfer coefficient is constant, that the specific heat of refrigerant is constant through the heat exchanger, and that no heat losses occur. Also, as shown in Figure 5.2, the temperature profiles of air and refrigerant are not linear. Therefore, the average temperature difference is not helpful in defining the overall heat transfer coefficient. In fact, it is nearly impossible to calculate

the overall heat transfer coefficient such that it can reflect the performance of the heat exchanger. In the present experiments, our trial-and-error attempts to determine the overall heat transfer coefficient were unsuccessful since the results do not have any significance in studying the performance of the heat exchanger.

Another important parameter that reflects the performance of a heat exchanger is the heat transfer effectiveness *eff*, which is defined as:

$$eff = \frac{h_{r,in} - h_{r,out}}{h_{r,in} - h_{r,ideal}} \quad (5.5)$$

where $h_{r,ideal}$ refers to the enthalpy of the refrigerant at the air inlet temperature. The term $h_{r,in} - h_{r,ideal}$ indicates the capacity potential of the refrigerant side.

Figure 5.7 shows the heat transfer effectiveness as a function of the heat exchanger size for all test conditions except those of $P_{ref} = 7$ MPa. Since there are 11 thermocouples installed along the refrigerant path as shown in Figure 3.10, it is convenient to calculate the heat transfer effectiveness for the gas coolers made from a different number of HXs (such as HX1, HX1+HX2, HX1+HX2+HX3, HX1+HX2+HX3+HX4, and the entire five HXs, respectively). The x-axis represents the size of the heat exchanger. For each curve depicted in the figure, the heat transfer effectiveness increases in a very similar pattern with an increase in the size of the heat exchanger. This figure is important and very helpful in developing empirical models to predict the capacity of a heat exchanger.

Figure 5.7 Heat transfer effectiveness vs. number of HXs

Figure 5.8 shows the heat transfer effectiveness vs. number of HXs for test conditions of $P_{ref} = 7 \text{ MPa}$. The data shown in Figure 5.8 shows a similar trend to those of Figure 5.7. However, when compared to Figure 5.7, two curves in Figure 5.8 do not show a similar trend. This can be explained by the fact that the specific heat of CO_2 changes rapidly in the temperature range of $30 - 40 \text{ }^\circ\text{C}$, as shown in Figure 5.6. At $P_{ref} = 7 \text{ MPa}$, which is close to the critical pressure of CO_2 ($P_{cri} = 7.3 \text{ MPa}$), the temperature of CO_2 is typically in the range of $30 - 40 \text{ }^\circ\text{C}$ ($T_{cri} = 31 \text{ }^\circ\text{C}$) at later HX units such as HX3, HX4, or HX5. Since the capacity of CO_2 in this temperature range increase rapidly, even a small

temperature variation corresponds to a significant amount of heat. In fact, at this section, the local refrigerant capacity rate ($\dot{m}_{air} \cdot C_p$) becomes very large, thus enhancing the heat transfer. In addition, the heat transfer process becomes more complicated when the test conditions of CO₂ approach its critical point. Therefore, to some extent, inconsistent trends are expected since the refrigerant properties change drastically.

Figure 5.8 Heat transfer effectiveness vs. number of HXs for a CO₂ pressure of 7 MPa

Figure 5.9 shows the effect of the refrigerant mass flow rate on the heat transfer effectiveness. It shows that refrigerant mass flow rate plays an important role in determining the performance of microchannel heat exchangers with CO₂.

Figure 5.9 Heat transfer effectiveness vs. number of HXs for Serial No. 1 condition

Figure 5.10 shows the pressure drop of the air side as a function of air velocity. The results indicate that the pressure drop of air is approximately proportional to the velocity squared. The effects of other parameters such as air temperature, heat flux, and refrigerant conditions are insignificant on the air side pressure drop. Since air properties remain rather constant for the range of parameters tested in the present test conditions, the pressure drop appears to be a function of velocity alone.

Figure 5.10 Air side pressure drop as a function of air velocity

Refrigerant pressure drop is much more complicated due to the rapid changes in the properties of CO_2 with temperature. Basically, for a certain heat exchanger, refrigerant pressure drop is related to the refrigerant viscosity and velocity. Figure 5.11 depicts the viscosity of CO_2 as a function of temperature at different pressures. During gas cooler tests, the temperature of the refrigerant changes as it flows through each heat exchanger unit. The range of operating temperatures of the refrigerant for the present study corresponds with the temperature range in which the viscosity is changing rapidly, as shown in Figure 5.11. Moreover, refrigerant velocity depends on the refrigerant mass flux and refrigerant density. Figure 5.12 shows the density of CO_2 as a function of temperature for different refrigerant pressures. As shown, the density of CO_2 strongly

depends on the refrigerant temperature. Therefore, refrigerant temperature is expected to be one of the dominant factors in refrigerant pressure drop.

Figure 5.11 Viscosity of CO₂ vs. temperature

Figure 5.12 Density of CO₂ vs. temperature

5.4 A Model for Predicting the Capacity of a Gas Cooler

As discussed in the previous sections, it is difficult to determine the overall heat transfer coefficient for a CO₂ gas cooler since the properties of CO₂ change significantly with temperature. Particularly, the specific heat of CO₂ changes tremendously in the present experimental temperature range. To overcome this problem, the concept of heat exchanger effectiveness was introduced to develop an empirical correlation for predicting a CO₂ gas cooler capacity.

The capacity of a gas cooler can be expressed as:

$$Q = Q_{ref} = \dot{m}_{ref} \cdot (h_{ref-in} - h_{ref-out}) \quad (5.8)$$

The heat transfer effectiveness (*eff*) is defined as:

$$eff = \frac{h_{ref-in} - h_{ref-out}}{h_{ref-in} - h_{ref-ideal}} \quad (5.9)$$

where $h_{ref-ideal}$ refers to the enthalpy of refrigerant at the air inlet temperature. *eff* represents the performance of a heat exchanger. Then, we have

$$Q = \dot{m}_{ref} \cdot (h_{ref-in} - h_{ref-ideal}) \cdot eff \quad (5.10)$$

In the above equation, $\dot{m}_{ref}$, h_{ref-in} , and $h_{ref-ideal}$ are normally available when designing a gas cooler. If we can find *eff*, the capacity of the gas cooler can be predicted.

Theoretically, for a heat exchanger,

$$eff = f(NTU, Z, flowarrangement) \quad (5.11)$$

where *Z* refers to the ratio of capacity rate between the two fluids, and *NTU* is a function of $\int U dA$. *U* refers to the local overall heat transfer coefficient.

The effectiveness for counterflow heat exchangers can be expressed as:

$$eff = \frac{1 - \exp(-NTU \cdot (1 - Z))}{1 - Z \cdot \exp(-NTU \cdot (1 - Z))} = 1 - \frac{(1 - Z) \cdot \exp(-NTU \cdot (1 - Z))}{1 - Z \cdot \exp(-NTU \cdot (1 - Z))} \quad (5.12)$$

The present gas cooler is a counter-cross flow type. Recall from the previous section that the effectiveness of the present heat exchanger can be orderly plotted as a function of the number of HXs.

For the present type of gas cooler, we assume the effectiveness has the form:

$$eff = 1 - \frac{1}{f(U_{local}) \cdot \exp(a_1 \cdot (L/D)^{a_2})} \quad (5.13)$$

where $f(U_{local})$ is expected to depend on the air-side and refrigerant heat transfer coefficients. Or,

$$f(U_{local}) = f(St_{air}, Nu_{ref}) \quad (5.14)$$

The present experiment provides extensive data for various test conditions. Since the refrigerant temperature was measured for each exchanger slab, the amount of heat transfer from each HX unit can be determined. Thus, the present experiment also enables us to study a CO₂ gas cooler of different sizes (HX1 alone, HX1+HX2, HX1+HX2+HX3, HX1+HX2+HX3+HX4, and HX1+HX2+HX3+HX4+HX5).

An empirical model is developed as follows:

$$Q = \dot{m}_{ref} \cdot (h_{ref-in} - h_{ref-ideal}) \cdot eff \quad (5.15)$$

where eff depends only on non-dimensional parameters.

$$eff = 1 - \frac{1}{(1 + 1.254 \times 10^{-8} St_{air}^{-4.493} Nu_{ref}^{-2.958}) \cdot \exp(0.5829 \cdot (2.325581 \times 10^{-4} \frac{L}{D})^{1.269})} \quad (5.16)$$

$$St_{air} = \frac{j}{Pr^{0.677}} \quad (5.17)$$

$$Nu = \frac{(Re-1000) Pr \zeta / 8}{1 + 12.7 \sqrt{\zeta / 8} (Pr^{2/3} - 1)} \quad (5.18)$$

$$\zeta = (0.79 \ln(Re) - 1.64)^{-2} \quad (5.19)$$

where L refers to the length of the refrigerant flow path while D is the hydraulic diameter of the microchannels.

Figures 5.13 and 5.14 present the predicted effectiveness as a function of number of HXs. Compared with Figures 5.8 and 5.9, it shows that the model can predict the test results of higher refrigerant pressure (10 ~ 13 MPa) with higher accuracy. Figure 5.15 compared the predicted effectiveness with experimental results. The empirical model can predict experimental results within $\pm 20\%$ except for the tests that are very close to the critical point of CO_2 .

Figure 5.13 Predicted effectiveness vs. number of HXs

Figure 5.14 Predicted effectiveness vs. number of HXs

Figure 5.15 Comparison between the model predictions and experimental results

CHAPTER 6. CONCLUSIONS AND FUTURE WORK

6.1 CONCLUSIONS

An experimental and computer simulation study was conducted to characterize the performance of CO₂ microchannel evaporators and gas coolers over a range of operating conditions found in residential heat pumps. An extensive review of the literature failed to indicate any previous systematic study in this area, suggesting a lack of fundamental understanding of the phenomena and a lack of comprehensive data that would quantify the performance potential of CO₂ microchannel heat exchangers for the application at hand.

In the present study, experiments were conducted to evaluate capacities and pressure drops for sub-critical CO₂ flow boiling and transcritical CO₂ gas cooling in microchannel heat exchangers. All experimental tests were successfully conducted with an energy balance within $\pm 3\%$. The only exceptions to this were experiments at very low saturation temperatures ($-23\text{ }^{\circ}\text{C}$), where heat balances were as high as 10%.

In the case of evaporators, it was found that a lower saturation temperature (especially when moisture condensation occurs) improves the overall heat transfer coefficient significantly. However, under such conditions, air-side pressure drop also increases when moisture condensation occurs. An increase in airflow rate also increases the overall heat transfer coefficient. However, this in turn is associated with higher pressure drops at higher air flow rates. A semi-empirical model was developed for prediction of CO₂ evaporator system capacity by introducing two new factors (C_{ta} and B_{ta}). The model agrees within $\pm 13\%$ with the experimental test results.

For the gas cooler, a significant portion of the heat transfer occurred in the first heat exchanger module on the refrigerant inlet side. For example, over 80% of the heat transfer occurred in the first heat exchanger module for the refrigerant with T_{in} of 120 °C and P_{in} of 12.5 MPa, and nearly 50% occurred in the first module for T_{in} of 79 °C and P_{in} of 7 MPa. The temperature and pressure of CO₂ significantly affect the heat transfer and fluid flow characteristics due to some important properties (such as specific heat, density, viscosity). In the transcritical region, heat transfer of CO₂ strongly depends on temperature and pressure. A semi-empirical model was developed based on non-dimensional parameters for the prediction of CO₂ gas cooler system capacity. The model successfully predicted the experimental results within $\pm 20\%$. Recommendations for future work on this project are listed at the end of this report.

6.2 Future Work

During the course of this project, many questions, ideas, and opportunities have become apparent and are included here. The work includes recalling and discussing important phenomena of CO₂ and microchannel heat exchangers, proposing problems for future study, and including some guidelines for designing microchannel heat exchangers.

6.2.1 Two-Phase Flow Mal-distribution for a Microchannel Evaporator

Flow mal-distribution is one of the major concerns in designing heat exchangers. Moreover, when heat exchangers experience phase change, such as condensation and evaporation, flow mal-distribution becomes even more important, especially for microchannel heat exchangers, which normally have to distribute the fluid of a two-phase

mixture into many parallel microchannels. Since the entrance of each microchannel is very small, and two-phase fluid typically exists as a non-homogenous pattern, the entrances of some microchannels may come into contact with the refrigerant only in vapor phase, while others may meet pure liquid all the time. The channels that meet only pure vapor will have a very poor heat transfer coefficient, which will reduce the total heat transfer performance of the heat exchanger. Therefore, header design is a key issue in designing microchannel heat exchangers that experience phase change. Two-phase flow mal-distribution is a research topic in our advanced heat exchangers laboratory. Zhao et al. (2000) has conducted some preliminary tests focused on two-phase flow mal-distribution in microchannels. The tests indicated that two-phase mal-distribution was caused by the phase separation in the inlet header. To suppress two-phase flow mal-distribution, some flow mixers could be introduced into the inlet header.

6.2.2 Defrosting Techniques

The present experimental conditions cover some tests at a saturation temperature of $-23\text{ }^{\circ}\text{C}$, which is significantly below the freezing temperature of water. In such cases, icing may occur if the air stream has a certain level of moisture. Moisture frozen on the outer surface of a heat exchanger may block airflow, thus reducing the gas cooler performance significantly. For a typical heat pump in winter, the system requires periodic system operation of the defrost cycle. Defrost techniques are currently being researched in our laboratory.

6.2.3 Model Developments and Improvements

The present study developed two models for predicting the capacity of an evaporator and a gas cooler with CO₂. The evaporator model results agree with experimental results within $\pm 13\%$ when coefficients are introduced to account for the effects of moisture condensate and superheat. The gas cooler model implements non-dimensional parameters and can predict the experimental data within $\pm 20\%$. More accurate model can be developed after a better understanding of the heat transfer process of CO₂ gas coolers is gained and a detailed investigation of the gas cooler is initiated. Furthermore, since the properties of supercritical CO₂ change significantly with temperature, a numerical model that accounts for that will better represent the practical situation.

As discussed above, flow mal-distribution is a key factor in the performance of a microchannel evaporator. In order to develop a more accurate model for predicting the performance of an evaporator, the flow mal-distribution problem needs to be investigated thoroughly. In addition, experimental tests need to be done on the air-side heat transfer coefficient, specifically when moisture condensation or freezing occurs.

6.2.4 Recommendations for Heat Exchanger Design

Based on the present experimental study, some recommendations for the microchannel heat exchanger design are listed as follows:

1. Since the flow pattern for two-phase flow is always heterogeneous, the flow mal-distribution problem needs to be considered. Using an elbow to replace the 90 degree sharp turn may improve this problem.

2. Considering the velocity profile of the air stream, the central area of the heat exchanger normally has higher air velocity, and thus has higher cooling potential. Therefore, the heat exchanger should be designed so that the refrigerant can be fed at the middle, as shown in Figure 6.1.

Figure 6.1 Schematic diagram of the proposed header design

3. To decrease the mal-distribution effect, limit the length of each pass.
4. Other important issues on heat exchanger design:
 - Cross section optimization (oval vs. square or round)
 - Header Design (minimization of mal-distribution and pressure drops)
 - Heat exchanger circuitry
 - Air-side enhancement

APPENDIX A. DESIGN OF AN OBSTRUCTION FLOW METER

A.1 Introduction

To test a microchannel heat exchanger as a gas cooler, an outdoor air duct was built. The duct was 1.22 m wide, 0.91 m high, and 2.74 m long. It consisted of an obstruction meter, an air mixer, a speed-variable fan, two thermocouple grids (one upstream, the other downstream), two screens, and a heat exchanger as shown in Fig. A.1. The duct was sealed carefully with silicone and insulated with a 25 mm thick thermal insulation material ($k = 0.04 \text{ W/mK}$). Since the cross sectional area of the duct is very large (1.22 m by 0.91 m), a large air flow rate was needed to provide the specified air velocity as required by the project.

1. Primary or quantity methods such as weight or volume tanks, burettes, positive-displacement meters;
2. Flowmeters including obstruction meters, variable-area meters, vortex-shedding meters, turbine and propeller meters, magnetic flowmeters;
3. Velocity probes such as Pitot-static tubes, hot-wire and hot-film anemometers, Laser-Doppler anemometer, Ultrasonic anemometer;
4. Flow-visualization techniques including smoke trails, dye injection, laser-induced fluorescence, and refractive-index change.

The above outline does not exhaust the list of flow-measuring methods, but it includes those of primary interest to the mechanical engineer. Among those methods, primary methods capture and record the volume or mass that flows during a fixed time interval. The primary methods, especially weight tanks, are typically used for steady-state calibration of liquid flow-meters. They are not suitable for on-line measurement applications. For on-line measurements, flowmeters and velocity probes are the most commonly used devices. Flowmeters determine volume or mass flow rates through tubes and ducts, whereas velocity-sensing probes measure fluid speed at a point in the flow. Although velocity-sensing probes can be used as building blocks for flowmeters, the converse is rarely true. Flow-visualization techniques sometimes can be employed to obtain an image of the overall flow field; however, these methods usually have special requirements on the experimental test section.

As flow meters, obstruction meters are the most often used flow measurement devices in industrial practice. Compared with other measurement methods such as

velocity probes, obstruction meters are the most convenient methods for on-line volume/mass measurement, and they can also provide very high accuracy.

A.2 Obstruction Meters

There are three basic types of obstruction meters, such as the venturi, the flow nozzle, and the orifice. An obstruction meter acts as an obstacle placed in the path of the flowing fluid, causing localized changes in velocity. Concurrently with velocity change, there will be pressure change. At the points of maximum restriction, and hence maximum velocity, minimum pressures are found. A certain portion of this pressure drop becomes irrecoverable owing to dissipation of kinetic energy; therefore, the output pressure will always be less than the input pressure.

For an incompressible flow, a relation between pressure drop across an obstruction meter and flow rate can be obtained by applying the mass conservation equation and Bernoulli's equation (Momentum equation).

For incompressible fluids,

$$\rho_1 = \rho_2 = \rho \quad (1)$$

$$G = A_1 V_1 = A_2 V_2 \quad (2)$$

$$\frac{P_1}{\rho g} + Z_1 + \frac{V_1^2}{2} = \frac{P_2}{\rho g} + Z_2 + \frac{V_2^2}{2} \quad (3)$$

If we let $Z_1 = Z_2$ and substitute $V_1 = (A_2/A_1) V_2$ in Eq. (3), we get

$$P_1 - P_2 = \frac{V_2^2 \rho}{2} \left[1 - \left(\frac{A_2}{A_1} \right)^2 \right] \quad (4)$$

and

$$G_{ideal} = A_2 V_2 = \left[\frac{A_2}{\sqrt{1 - \left(\frac{A_2}{A_1} \right)^2}} \right] \sqrt{\frac{2(P_1 - P_2)}{\rho}} \quad (5)$$

For a given meter, A_1 and A_2 are established values, so Eq. (5) can be rewritten as

$$G_{ideal} = A_2 V_2 = EA_2 \sqrt{\frac{2(P_1 - P_2)}{\rho}} \quad (6)$$

where

$$E = \frac{1}{\sqrt{1 - (A_2 / A_1)^2}}$$

In reality, G_{actual} is not the same as G_{ideal} because of some irreversible factors. Two additional factors are introduced for obstruction meter calculations. The first one is the discharge coefficient, C , and the other is flow coefficient, K . They are defined as following:

$$C = \frac{G_{actual}}{G_{ideal}} \quad (7)$$

$$K = CE = \frac{C}{\sqrt{1 - (A_2 / A_1)^2}} \quad (8)$$

Finally, the equation for an obstruction meter is

$$G_{actual} = KA_2 \sqrt{\frac{2(P_1 - P_2)}{\rho}} \quad (9)$$

A.3 Design of a Special Obstruction Meter

In the present experiments, the cross sectional area of the outdoor air duct is very large (1219 mm wide and 914 mm high). A venturi or a flow nozzle is not suitable for the present experimental setup since they both need long installation space that is very valuable for the present situation. In addition, large air flow rates are required for the present experiments. However, a typical orifice may affect the flow field significantly and thus affect the downstream temperature measurement. Unfortunately, a large diameter orifice is very costly to fabricate.

In the present study, a new type of obstruction meter was designed and fabricated as seen in Figure A.2. The meter was made from of 117 holes with an inner diameter of 25 mm. Since these small holes are uniformly arranged over the entire cross sectional area of the outdoor air duct, the flow and thermal fields are rather uniform for the present situation.

Figure A.2 Obstruction meter

A.4 Calibration of the Obstruction Meter

Equation 9 is a general equation for an obstruction meter. The present obstruction meter also satisfies Equation 9, although it has a special geometry. Therefore, in the process of calibrating the obstruction meter, the flow factor (K) was also determined.

Six finned strip heaters were installed in the outdoor air duct to calibrate the obstruction meter. The heat input to the heaters could be adjusted between 0.75 kW and 4.2 kW. The outdoor air duct was carefully sealed and insulated with 25 mm thick thermal insulation material.

To calibrate the obstruction meter, the heat loss through the outdoor air duct was first estimated. A 1-D heat conduction model was used to estimate the heat loss analytically. Since the outdoor air duct and the air duct of the indoor loop were designed and fabricated according to the same procedure and using the same material and same thickness, the overall heat transfer coefficient for the two air ducts should be approximately the same. The overall heat transfer coefficient of the air duct of the indoor loop can be determined from the energy balance. The overall heat transfer coefficients determined analytically strongly agree with those of obtained from the indoor loop within $\pm 9\%$.

Therefore, the heat loss through the outdoor air duct can be determined by the overall heat transfer coefficient and temperature difference between the air inside the duct and the room. The results are shown in Figure A.3.

Figure A.3 Heat losses from the outdoor air duct

As the starting point, the present obstruction meter can be approximated as 117 parallel orifice meters. The total airflow rate is the sum of the airflow rates through these 117 orifices. Each orifice can be considered as a standard orifice, and flow coefficient K can be determined from Figure A.4.

Figure A.4 Flow coefficient as function of Reynolds number

where $\beta = \frac{D_{orifice}}{D_{duct}} \approx 0.0243$. From the above figure, flow coefficient K was determined to be approximately 0.59. This value remains almost independent of Reynolds number, for Re ranged from 10^4 to 10^7 .

The airflow rate can be calculated from

$$G_{actual} = 117KA_{hole}\sqrt{\frac{2(P_1 - P_2)}{\rho}} \quad (10)$$

where

$$A_{hole} = \frac{\pi D_{hole}^2}{4} = 0.000507 \text{ m}^2$$

Based on the airflow rate determined from the above equation and the temperature difference between the upstream and downstream thermocouples, the heat absorbed by the airflow through the duct can be calculated from

$$Q_{air} = G\rho C_p (T_{downstream} - T_{upstream}) \quad (11)$$

Two series of experimental tests were performed, and the results are shown in Figures A.5 and A.6. These results show a relatively large difference between Q_{heater} and Q_{air} . This reflects that the flow factor K for a standard orifice was not directly applicable to the present obstruction meter. However, more careful observation of the test results suggested that the difference between Q_{air} and Q_{heater} is nearly a constant value and is independent of the temperature difference between the air inside the duct and ambient air. A rather constant ratio between the Q_{air} and Q_{heater} was observed and found to be 1.29 for both of the two test series. Since the present obstruction meter is not a standard orifice meter, the flow factor determined for standard orifices might not be applicable directly. However, since the obstruction meter was made from many standard orifices, it is likely to have some correlation between the present obstruction meter and standard orifice meters. Thus, a correction factor based on the flow factor for standard orifice meter was developed for the present obstruction meter. The constant was found to be 1.29. The flow factor for the obstruction meter is equal to 1.29 times 0.59, which is the value for a single standard orifice with ID of 25 mm.

Figure A.7 depicts the comparison between the Q_{air} and Q_{heater} after the flow coefficient was corrected.

Figure A.7 Comparison between the heating determined by air and by heaters after correction

New experimental tests were conducted based on the modified flow factor. The energy balance results are shown in Figure A.8. It was found that the energy balance was within $\pm 3\%$. However, energy balance is within $\pm 3\%$ when temperature difference between the upstream and downstream thermocouples is large enough. This reflects that a relatively larger error in the energy balance might be due to the insufficient temperature difference (around 1 C). It also shows that the flow factor determined above is accurate for the present obstruction meter.

Figure A.8 The results of energy balance

Figure A.9 depicts the air flow rate as a function of the fan frequency. Three different test series showed the same results. The repeatability is within $\pm 2\%$.

Figure A.9 Volumetric flow rate vs. fan frequency

In summary, a new type of obstruction flow meter was designed, constructed and calibrated experimentally. The flow meter was constructed by uniformly drilling 117 holes with an inner diameter of one inch on a plate. Extensive experimental tests were conducted at different test conditions, heat input, and air flow rate.

It was shown that typical flow coefficients for flat-plate orifices could not be applied directly to the present obstruction meter. This is due to the special geometry of the present obstruction meter. However, since the present meter is similar to a typical orifice meter, experimental results showed that a typical procedure used to obtain the flow coefficient for a single orifice could be applied with some modifications.

The final energy balance results showed that the results were within $\pm 3\%$ for the different test conditions ranging from 1.5 kW to 4.2 kW, and air flow rate from 1000 to 4000 m³/h.

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