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Transcritical Carbon Dioxide Microchannel Heat Pump Water Heaters: Part I - Validated Component Simulation Modules

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ABSTRACT

An experimental and analytical study on the performance of carbon dioxide heat pumps for water heating was conducted. The performance of compact, microchannel, water-coupled gas coolers, evaporator, and suction line heat exchanger (SLHX) were evaluated in an experimental facility. Analytical heat exchanger models accounting for the flow orientation and changing CO₂ thermophysical properties were developed and validated with data. Heat transfer coefficients were predicted with correlations available in the literature and local heat duty calculated using the effectiveness-NTU approach. The gas cooler, evaporator, and SLHX models predicted measured heat duties with an absolute average error of 5.5%, 1.3%, and 3.9%, respectively. Compressor isentropic and volumetric efficiency values were found to range from 56% to 67% and 62% to 82%, respectively. Empirical models for compressor efficiency and power were developed from the data. The resulting component models are implemented in a system model in a companion paper (Part II)

1. INTRODUCTION

A desire to utilize low global warming potential (GWP) fluids in HVAC&R equipment has generated much interest in carbon dioxide (CO₂) due to its low GWP, favorable thermophysical properties, non-toxicity and low cost. CO₂ was once a widely used refrigerant, but fell out of favor with the advent of halocarbons, before being reintroduced as a viable refrigerant by Lorentz and Pettersen (1993). Perhaps one of the most attractive applications of CO₂ transcritical systems is for water heating applications. Compared to synthetic refrigerants (*e.g.* R134a), CO₂ exhibits a distinct advantage in water heating systems due to the lack of a temperature “pinch” on the high-side of the system. For an R134a system, the condensation temperature must be higher than the desired water delivery temperature (> 60°C), resulting in a high pressure ratio, reduced system efficiency, and a large condenser. In CO₂ systems, the non-isothermal temperature glide through the gas cooler matches well with the high water temperature lift required, resulting in smaller heat exchangers. Additionally, as the high-side temperature and pressure are uncoupled, CO₂ systems can operate at high temperature lifts with a much lower pressure ratio than conventional subcritical systems. Commercial systems are on the market in Japan with COPs in the range of 3 to 4 (Kim *et al.*, 2004). Many researchers (Neksa *et al.*, 1998; Cecchinato *et al.*, 2005; Kim *et al.*, 2005; Stene, 2005; Sarkar *et al.*, 2006; Laipradit *et al.*, 2008; Sarkar *et al.*, 2009) have developed experimental and analytical models of systems for providing hot water, space heating, air conditioning, or a combination of all three. Nearly all prior investigations of CO₂ water heating systems utilize concentric tube-in-tube counter flow heat exchangers for both the evaporator and gas cooler (Neksa *et al.*, 1998; Cecchinato *et al.*, 2005; Kim *et al.*, 2005; Rigola *et al.*, 2005; Stene, 2005). The ease of construction and modeling of this heat exchanger geometry has made the design a logical choice for experiments.

One of the stated benefits of CO₂ is the ability to use compact, microchannel heat exchanger geometries to reduce system size. Microchannel heat exchangers will be necessary for widespread commercial acceptance of CO₂ heat pumps due to their ability to provide high heat transfer coefficients, reduce material inventory of the heat exchangers and safely contain the high pressures associated with CO₂. Little research has been published on CO₂ water heater modeling and performance with the use of microchannel heat exchangers. The focus of this paper is to develop and validate analytical models for compact, microchannel heat exchangers for use in a CO₂ water heating heat pump

system model. The models must provide accurate results, while maintaining computational speed to have value in an iterative system model. Additionally, empirical compressor relations are developed from the data and utilized in the system performance model. All developed models are validated using data obtained from an experimental CO₂ heat pump facility. In Part II of this work, an overall system model incorporating the component-level models is developed. In this model, the component sizes are fixed and performance simulated by varying the gas cooler and evaporator inlet conditions. The results are analyzed to compare different heat pump designs, operating conditions and water heating strategies.

Figure 1: Experimental System Schematic

2. EXPERIMENTAL APPROACH

2.1 Experimental Facility

The experimental system consisted of a refrigerant loop coupled to chilled and heated water loops at the gas cooler and evaporator. The system was operated in two modes; with and without a suction line heat exchanger (SLHX). A schematic of the system is shown in Fig. 1. The evaporation and gas cooling temperatures could be controlled by controlling the flow rate and temperature of the closed water loops. Two hermetic, reciprocating compressors, each with a swept volume of 2.46 cm³ and fixed speed of 3450 RPM at 120 VAC, were operated in parallel. Each compressor was installed with a nominal charge of 40 mL of polyolester (POE) lubricant. A high pressure “u-tube” type accumulator prevented liquid refrigerant from entering the compressor. The heat pump was comprised of three heat exchangers, a cross flow water-coupled aluminum microchannel brazed plate gas cooler, a cross-counter flow water-coupled aluminum microchannel brazed plate evaporator and a counterflow brazed microchannel SLHX. A photograph of the heat exchangers under investigation is shown in Fig. 2.

In the present study, gas coolers of “5-plate” and “7-plate” design were used. Each plate contained a set of offset-strip fins. Water entered one side of the heat exchanger and flowed through each subsequent plate in a serpentine manner. The plates were wrapped with an array of 16 aluminum microchannel tubes, resulting in a total of 64 circular ports with $D = 0.89$ mm. The resulting flow is a cross-counterflow orientation. The evaporator is of a similar aluminum plate construction. However, the water enters the evaporator, splits between the plates and makes one pass through the heat exchanger, resulting in a cross flow orientation. Overall, tube-side and fin-side dimensions of the heat exchangers under investigation are given in Table 1. The suction line heat exchanger consisted of three aluminum microchannel tubes ($370 \times 25.4 \times 3.2$ mm) stacked and brazed together, with high temperature refrigerant from the gas cooler outlet flowing through the center tube, and lower temperature refrigerant from the evaporator outlet flowing through the two outside tubes in counterflow. The inner tube had 11 ports, while the outer tubes each had 17 ports, all with $D = 0.89$ mm.

To determine heat duties and thermodynamic state points, temperature, pressure and flow rate measurements were obtained at various points in the refrigerant and water loops as indicated in Fig. 1. Electrical power consumption of the compressors was also measured. All refrigerant and water temperatures were measured with T-type thermocouples ($\pm 0.5^\circ\text{C}$) immersed in the flow. Refrigerant pressure transducers had an uncertainty of ± 28 kPa on the high-side and ± 9 kPa on the low-side. The refrigerant mass flow rate was measured with a Coriolis type meter ($\pm 0.035\%$ of reading), while the volumetric flow rates of the gas cooler and evaporator water loops were measured with magnetic type ($\pm 0.5\%$ of reading) and turbine type ($\pm 0.5\%$ of reading) flow meters, respectively. The combined electrical

Figure 2: Gas Cooler and Evaporator

Table 1: Gas Cooler and Evaporator Exchanger Dimensions

Overall			
	5-plate Gas Cooler	7-plate Gas Cooler	Evaporator
Number Of Refrigerant Tubes	16	16	24
Number Of Refrigerant Passes	4	6	6
Length × Width × Height (mm)	191 × 84 × 38	191 × 84 × 54	245 × 84 × 45
Refrigerant Side			
	5-plate Gas Cooler	7-plate Gas Cooler	Evaporator
Total Refrigerant Tube Length (mm)	343	516	516
Ports Per Tube	4	4	4
Port Diameter (mm)	0.89	0.89	0.89
Tube Width (mm)	6.35	6.35	6.35
Tube Thickness (mm)	0.38	0.38	0.38
Heat Transfer Area (mm ²)	6150	9230	13040
Water Side			
	5-plate Gas Cooler	7-plate Gas Cooler	Evaporator
Fin Pitch (mm)	2.26	2.26	2.25
Fin Height (mm)	6.4	6.4	4.9
Fin Thickness (mm)	0.031	0.031	0.031
Fin Depth (mm)	3.2	3.2	3.2
Heat Transfer Area (mm ²)	27510	38510	43630

power input to the dual compressor setup was determined with a 0 to 4 kW (± 20 W) watt meter.

2.2 Experimental Conditions

Data were obtained with the heat pump operating in the SLHX and non-SLHX configuration over a variety of operating conditions to characterize compressor performance and validate individual component and system modeling. Data with no SLHX installed that were originally obtained to characterize gas cooler performance and validate a detailed analytical model by Fronk and Garimella (2009) are used to validate models developed in the current study. Data points were obtained at (nominal) three gas cooler refrigerant inlet temperatures (85, 100, and 115°C), two gas cooler water inlet temperatures (5 and 20°C), three gas cooler water flow rates (0.95, 2.38, and 5.68 L min⁻¹), four refrigerant flow rates (8, 12, 16 and 21 g s⁻¹) and two gas cooler sizes (5- and 7-plate). Evaporator data were obtained simultaneously, with conditions adjusted to meet the required gas cooler inlet conditions. Testing of the system with SLHX installed was performed by controlling both the gas cooler and the evaporator water inlet temperatures. At high gas cooler water flow rates, the approach temperature difference at the gas cooler exit was low (1 to 3°C), thus, the inlet temperature to the SLHX could be controlled with good resolution. Likewise, the evaporator water inlet temperature provided a good approximation of the superheated low pressure refrigerant inlet temperature to the SLHX. Combinations of gas cooler water inlet temperatures of 10, 23, 30, and 40°C and evaporator water inlet temperatures of 10, 15, 20, 25, and 30°C were used to test the SLHX. Evaporator superheat was maintained at 5°C by varying the system refrigerant charge.

A series of tests at different pressure ratios and suction-side superheat values were conducted to characterize compressor performance and enable development of an empirical performance model to be incorporated into the system model. Testing was conducted at suction pressures of 2,200, 3,000, and 4,000 kPa., and discharge pressures of 8,000, 9,500, and 11,000 kPa. The suction-side superheat was nominally maintained at 5, 12.5, and 20°C. At a suction pressure of 2200 kPa, only the 5°C superheat condition was tested. Testing at 12.5 and 20°C superheat at this suction pressure would have resulted in compressor discharge temperatures above the rated maximum. Thus, 21 data points were collected to characterize compressor performance.

3. EXPERIMENTAL RESULTS

From the experiments described above, compressor operating data including oil circulation rate, volumetric and isentropic efficiencies, and the heat duties of each heat exchanger (gas cooler, evaporator and suction line heat

exchanger) were obtained. These data were used to validate individual component models and assist in the development and validation of a system level model for predicting overall performance described in Part II.

3.1 Compressor Performance Results

Twenty-one data points were obtained to characterize compressor volumetric efficiency, isentropic efficiency, and electrical consumption as a function of suction and discharge parameters. The experimental volumetric efficiency was calculated from Equation (1), using the refrigerant density calculated at the measured suction inlet temperature and pressure, the measured refrigerant/lubricant mass flow rate, compressor speed and the total cylinder displacement of the two compressors.

$$\eta_{vol} = \frac{\dot{m}_{ref}}{V_d \cdot N \cdot \rho_{suc}} \quad (1)$$

The trend in compressor volumetric efficiency is shown in Fig. 3. It should be noted that by using the combined cylinder displacement in Equation (1), it is assumed that the mass flow was evenly distributed between each compressor and that each compressor had equivalent volumetric efficiency. Fig. 3 shows that pressure ratio is the dominant governing factor in volumetric efficiency for the compressors under investigation, with the efficiency ranging from 63% at a pressure ratio of 5 to 82% at a pressure ratio of 2. Isentropic efficiency of the compressor setup was calculated as shown in Equation (2) at varying pressure ratio, suction pressure and suction superheat.

$$\eta_s = \frac{i_{ref,out}^* - i_{ref,in}}{i_{ref,out,s} - i_{ref,in}} \quad (2)$$

The refrigerant inlet enthalpy was calculated as a function of the measured suction pressure and temperature, while the refrigerant isentropic outlet enthalpy was calculated at that outlet pressure and with specific entropy equal to that at the suction inlet. The outlet enthalpy in the numerator ($i_{ref,out}^*$) is corrected for the effect of circulating lubricant on the discharge temperature. At the same outlet pressure, the presence of lubricant lowers the discharge temperature compared to pure refrigerant, artificially inflating isentropic efficiency. By estimating oil circulation rate (OCR), it is possible to calculate a corrected compressor outlet enthalpy that yields a more representative isentropic efficiency for use in a predictive system model. The nominal OCR for the compressors under investigation was expected to be 5% (Hoehne, 2007). Using property data from the lubricant manufacturer (Fuchs, 2007), and accounting for heat loss from the compressors to the environment, an overall energy balance was used to obtain an estimate of OCR. Estimates of OCR ranged from 7.9% to 16%, with 90% of the data being between 7.9% and 11.5%. The compressors were designed for operation in a much larger volume air-coupled system and for operation as standalone units, indicating that the compressors were probably initially overcharged with lubricant for an entirely liquid-coupled design, which may explain these higher calculated OCR values. Using the OCR, the corrected isentropic efficiency versus pressure ratio is shown in Fig. 4.

Figure 3: Volumetric Efficiency Results

Figure 4: Isentropic Efficiency Results

3.2 Heat Exchanger Results

Temperature and flow rate measurements were taken on the evaporator and gas cooler on both the water and refrigerant sides of each heat exchanger. This allows for two independent capacity calculations for each heat exchanger, ensuring an energy balance. The refrigerant-side capacity of the gas cooler was calculated using Equation (3).

$$\dot{Q} = \dot{m}(i_{out} - i_{in}) \quad (3)$$

The gas cooler inlet and outlet enthalpies are a function of measured temperature and pressure only, determined from the equation of state of Span and Wagner (1996). Evaporator testing was conducted with no SLHX installed, thus, with the assumption of isenthalpic expansion; the inlet enthalpy was set equal to the outlet enthalpy of the gas cooler. The enthalpy of the superheated refrigerant at the evaporator outlet was calculated from the local temperature and pressure. The water-side heat duty was also found from an energy balance, where the mass flow rate was determined from the measured volumetric flow rate and the density of the water at the flow meter inlet.

The average heat duties for the gas coolers varied from 1.5 to 6.5 kW and are shown for the 5- and 7-plate heat exchangers in Fig 5. The absolute average deviation between calculated refrigerant and water-side heat duty was 5.4%. A complete discussion of the gas cooler experimental results, trends and uncertainties is available in Fronk and Garimella (2009). For the evaporator, high water flow rates resulted in small temperature differences and high uncertainties in water-side heat duty. Thus, there was not good agreement between the refrigerant- and water- side heat duties. Therefore, only the refrigerant side heat duty was used in validation of the evaporator model. It should be noted that the original focus of the study, as reported in Fronk and Garimella (2009), was for gas cooler performance characterization; therefore, the independently controlled parameters were set to satisfy the gas cooler test matrix. Accuracy in the energy balance of the evaporator was of secondary importance.

Seventeen data points of varying conditions were used to characterize the performance of the SLHX. Assuming an average OCR rate of 7.9% as calculated previously, the heat duty for the high pressure-side as calculated using Equation (4) and the low pressure-side from Equation (5).

$$\dot{Q}_{slhx,high} = (1-OCR)\dot{m}_{sys}(i_{slhx,high,in} - i_{slhx,high,out}) + (OCR)\dot{m}_{sys}(i_{oil,high,in} - i_{oil,high,out}) \quad (4)$$

$$\dot{Q}_{slhx,low} = (1-OCR)\dot{m}_{sys}(i_{slhx,low,out} - i_{slhx,low,in}) + (OCR)\dot{m}_{sys}(i_{oil,low,out} - i_{oil,low,in}) \quad (5)$$

The refrigerant inlet and outlet enthalpies were calculated at the local temperature and pressures. The inlet and outlet oil enthalpies were calculated at the local temperature from the property data provided by the manufacturer (Fuchs, 2007). The measured heat duty values range from 0.11 to 0.78 kW with an average absolute energy balance error of 4.4%. SLHX UA was calculated from the average heat duty and the log-mean temperature difference.

Figure 5: Gas Cooler Experimental Results

Increased refrigerant flow rate leads to larger UA values due to the increase in the heat transfer coefficient at higher mass flow rates. The UA varies from 0.038 kW K^{-1} at 0.0122 kg s^{-1} to 0.084 kW K^{-1} at 0.0231 kg s^{-1} .

4. COMPONENT MODEL DEVELOPMENT AND RESULTS

Performance prediction models were developed for the four key components of the heat pump system: gas cooler, evaporator, suction line heat exchanger (SLHX), and compressor. When incorporated into an overall system model, each component model is solved multiple times in an iterative manner to produce overall system heat duties, water delivery temperature, and coefficient of performance (COP). The three heat exchanger models were developed using a segmented analysis to account for the varying refrigerant properties and complex geometries and were validated with the data presented above. The compressor models were developed empirically from the data. Due to the iterative nature of the system model, a secondary goal of the performance models was computational efficiency.

Figure 6: Gas Cooler Segmented Geometry

4.1 Gas Cooler Performance Model

Fronk and Garimella (2009) developed a detailed analytical model for the gas cooler geometry used in this experimental setup that demonstrated good agreement with data. This model used a segmented analysis method using local property values and heat transfer coefficients to obtain local heat duties throughout the heat exchanger. Due to the complicated flow paths of the refrigerant and water, the segments used in the model interconnect in a complex manner resulting in extended solution times not practical for application in a system model. To reduce solution time, a simplified approximation of the heat exchanger geometry was used, as shown in Fig. 6 where the water flow makes a single pass and the refrigerant is in cross flow over each segment. Refrigerant and water-side heat transfer areas and flow rates were calculated according to the geometry of the actual gas cooler. A system of coupled equations in which the local refrigerant and water thermodynamic and transport properties are evaluated in each segment and the individual segment duties are determined using the effectiveness-NTU method was solved using the non-linear equation solver capabilities of *Engineering Equation Solver* (EES) (Klein, 2007) software.

Refrigerant and water thermophysical properties in each segment were calculated from the segment average temperature and pressure and inlet temperature, respectively. The water-side Nusselt number was calculated using a correlation proposed by Manglik and Bergles (1995) for the Colburn factor (j) for offset-strip fin geometry as given by Equation (6), where α , δ , and γ are dimensionless fin geometry ratios.

$$j = 0.6522 \text{Re}_{\text{water}}^{-0.5403} \alpha^{-0.1541} \delta^{0.1499} \gamma^{-0.0678} \times \left[1 + 5.269 \times 10^{-5} \text{Re}_{\text{water}}^{1.340} \alpha^{0.504} \delta^{0.456} \gamma^{-1.055} \right]^{0.1} \quad (6)$$

The refrigerant-side heat transfer coefficient (h_{ref}) was calculated using a single-phase turbulent correlation proposed by Gnielinski (1976). The UA value of the segment (UA_{seg}) was calculated from Equation (7); the aluminum wall resistance was neglected.

The number of transfer units (NTU) for each segment was determined from the local UA and minimum thermal capacitance rate. Segment effectiveness (ϵ_{seg}) was calculated assuming fluid flow through the heat exchanger segment was in a cross flow orientation (Incropera and Dewitt, 2002). Finally, segment heat duty was calculated using Equation (8).

$$\frac{1}{UA_{\text{seg}}} = \frac{1}{h_{\text{ref}} A_{\text{ref,seg}}} + \frac{1}{h_{\text{water}} (A_{\text{base,seg}} + \eta_{\text{fin}} A_{\text{fin,seg}})} \quad (7)$$

$$\dot{Q}_{\text{seg}} = \epsilon_{\text{seg}} C_{\text{min}} (T_{\text{ref,in}} - T_{\text{water,in}}) \quad (8)$$

The total gas cooler heat duty was found by summing all of the segment heat duties. Fig. 7 shows a comparison between the measured refrigerant-side gas cooler heat duty for the 5 and 7-plate gas cooler design and the calculated heat duty using the simplified gas cooler model. The average absolute error between the predicted and measured

Figure 7: Gas Cooler Model Results

Figure 8: Evaporator Segmented Schematic

data for the 7-plate case was 5.7% with 88.2% of the predictions within 10% of the measured values. For the 5-plate case, the average absolute error was 5.2% with 82.3% of the predictions within 10% of the measured values.

4.2 Evaporator Model

The methodology for modeling the evaporator was similar to that used for the gas cooler: the water and refrigerant properties and heat transfer coefficients in each segment are determined, and the local heat duty calculated using the effectiveness-NTU method. The evaporator segmented schematic is shown in Fig 8. The once-through geometry of the evaporator allows for the heat exchanger computations to be conducted without iteration. A sensitivity analysis was performed to determine the number of segments needed to account for property changes of the two-phase refrigerant. It was also ensured that the transition from subcooled liquid to two-phase flow and two-phase flow to superheated vapor was handled appropriately within segments where this transition occurred.

The water-side heat transfer coefficient was calculated in an identical fashion to the gas cooler using the correlation provided by Manglik and Bergles (1995) and the appropriate geometric parameters and flow rates. Refrigerant could enter into a segment in one of three conditions: subcooled liquid, two-phase mixture, or superheated vapor. If the fluid entered as a subcooled liquid or superheated vapor, the refrigerant-side heat transfer coefficient is calculated using the Gnielinski (1976) correlation. The required property values are again calculated at the refrigerant inlet enthalpy ($i_{ref,in}$) and the evaporator pressure (P_{evap}). The segment thermodynamic quality was calculated from the evaporator pressure (P_{evap}) and the segment inlet refrigerant enthalpy ($i_{ref,in}$). If the refrigerant entered the segment as a two-phase mixture, the two-phase boiling heat transfer coefficient correlation suggested by Wattelet *et al.* (1994) was used (Eqn. (9)). The refrigerant-side heat transfer coefficient is composed of a nucleate boiling and a convective component (h_{nb} and h_{cb} , respectively).

$$h_{ref} = \left(h_{nb}^{2.5} + h_{cb}^{2.5} \right)^{1/2.5}$$

$$h_{nb} = 55M^{-0.5} q^{0.67} P_r^{0.12} \left(-\log_{10}(P_r) \right)^{-0.55}$$

$$h_{cb} = F h_{liq} R$$
(9)

An iterative procedure is implemented to calculate the local heat flux (q'') since it is dependent on the overall segment heat duty, which is dependent on both the refrigerant and water-side heat transfer coefficients. After calculation of the segment heat duty, the local heat flux is updated and the refrigerant-side heat transfer coefficient is recalculated. This procedure was repeated until the change in local heat flux between iterations was less than 1%. The convective boiling term is a function of the liquid-phase heat transfer coefficient (h_{liq}), the two-phase multiplier (F), and the convective boiling correction factor (R) as detailed by Wattelet *et al.* (1994).

The procedure used for calculating gas cooler heat transfer coefficient and heat duty was used for evaporator segments where the fluid was in a single phase. If the refrigerant inlet conditions represented a two-phase mixture, NTU and UA were calculated in the same manner, but the effectiveness was determined by Equation (10).

$$\varepsilon_{seg} = 1 - \exp(-NTU)$$
(10)

Figure 9: Evaporator Model Results

segment heat duties. The average absolute error between measured and predicted values is 3.9% with a maximum error of 5.9% and 82.4% of the data points within a 5% error range.

$$\frac{1}{UA_{seg}} = \frac{1}{h_{ref,high} A_{surf,seg,high}} + \frac{t_{wall}}{k_{wall} A_{wall,seg}} + \frac{1}{h_{ref,low} A_{surf,seg,low}} \quad (11)$$

4.4 Compressor Models

Empirical models of the dual compressor setup were developed to predict isentropic and volumetric efficiency, and electrical power consumptions. When used in the system level model, the empirical models can predict compressor outlet conditions, refrigerant mass flow rate and power consumption as a function of the inlet conditions and outlet pressure. A biquadratic equation was fit to the volumetric (12) and electrical power consumption (13) data with suction and discharge pressures as variables. The average absolute error in η_{vol} was 0.6%. For W_{comp} , the average absolute error was 0.7%. The resulting R^2 values were 0.986 for the η_{vol} model and 0.995 for the compressor power model. Thus, both models provided good results without further correction for different superheat values. Values for the constants are given in Table 2.

$$\eta_{vol} = a_1 + a_2 P_{suc} + a_3 P_{suc}^2 + a_4 P_{dis} + a_5 P_{dis}^2 + a_6 P_{suc} P_{dis} \quad (12)$$

$$\dot{W}_{comp} = b_1 + b_2 P_{suc} + b_3 P_{suc}^2 + b_4 P_{dis} + b_5 P_{dis}^2 + b_6 P_{suc} P_{dis} \quad (13)$$

The model for isentropic efficiency was found to depend on suction and discharge pressure and suction superheat. Two biquadratic equations accounting for the effects of inlet/outlet pressure and suction superheat were developed and the isentropic efficiency was calculated using both these equations. The average absolute error between the model and the measured values was 1.0% with a maximum deviation of 2.4%, with 85.7% of the predicted values being within 2% of the measured values.

$$\begin{aligned} \eta_{s,SH} &= c_1 + c_2 P_{suc} + c_3 P_{suc}^2 + c_4 P_{dis} + c_5 P_{dis}^2 + c_6 P_{suc} P_{dis} \\ CF_{SH} &= d_1 + d_2 P_{suc} + d_3 P_{suc}^2 + d_4 SH + d_5 SH^2 + d_6 P_{suc} SH \\ \eta_s &= CF_{SH} \eta_{s,SH} \end{aligned} \quad (14)$$

Table 2: Constants for Compressor Empirical Models

	n = 1	2	3	4	5	6
a_n	0.6135	$1.6356 \cdot 10^{-4}$	$-2.0528 \cdot 10^{-8}$	$-3.0485 \cdot 10^{-5}$	$-2.8690 \cdot 10^{-11}$	$3.6903 \cdot 10^{-9}$
b_n	0.5264	$3.0196 \cdot 10^{-5}$	$-4.7542 \cdot 10^{-8}$	$6.2063 \cdot 10^{-5}$	$-3.7849 \cdot 10^{-9}$	$4.0339 \cdot 10^{-8}$
c_n	0.3156	$-1.6465 \cdot 10^{-4}$	$2.2676 \cdot 10^{-8}$	$8.9522 \cdot 10^{-5}$	$-5.4486 \cdot 10^{-9}$	$7.4165 \cdot 10^{-9}$
d_n	0.7642	$1.6048 \cdot 10^{-4}$	$-1.9661 \cdot 10^{-8}$	$9.5087 \cdot 10^{-3}$	$6.7124 \cdot 10^{-4}$	$-7.9741 \cdot 10^{-6}$

5. CONCLUSIONS

Microchannel heat exchanger models that can be for the design and simulation of CO₂ heat pump water heaters and other heat pumps were developed using a segmented approach. Due to the complex nature of the flow paths in the gas cooler and evaporator, simplifications to these paths were made to reduce computational time without any adverse impact on accuracy. An experimental CO₂ heat pump facility was developed and used to test and validate the analytical models for each component over a range of expected operating conditions. The gas cooler models were compared with data collected on heat exchangers of two sizes (5-plate and 7-plate), and accurately predicted the heat duties within an average error of 5.5% over a heat duty range of 2 to 6 kW. The evaporator model predicted the data with an average error of 1.3% over a heat duty range of 1.5 to 5 kW. The SLHX model predicted the data with an average error of 3.9% over a heat duty range of 0.1 to 0.8 kW.

The oil circulation rate was calculated using the data collected from the compressor performance testing. No direct measurement of the OCR was possible due to the high operating pressure of the CO₂ cycle and the small refrigerant charge in the system. Through the measurement of the electrical power consumption of the compressor and estimation of the heat loss from the compressor shell, the OCR was estimated to be 10.7% with an absolute error of 5.4%. Other operational experience (Hoehne, 2007) with these compressors indicates an OCR of 5%. Therefore, the average of the two values, 7.9%, was used for compressor calculations. The compressor isentropic efficiency values were found to range from 56% to 67%, while the volumetric efficiency varied from 62% to 82%. Compressor electrical power measurements ranged from 1.33 to 1.90 kW. Using these data, regression analyses were performed to fit equations of a biquadratic form as a function of suction and discharge pressures. The isentropic efficiency model also explicitly accounted for the effect of superheat.

The models developed in this study enable fast, accurate calculation of inlet and outlet conditions of each component, and easy integration into system level models for determining system COP, heat duties and other performance data under varying conditions. The compact, microchannel heat exchangers and low cost reciprocating compressor utilized in the current study represent the most likely configuration required for the implementation of natural refrigerant heat pump technology for water heating and space-conditioning. Thus, a system model developed from these component models can provide a more realistic assessment of the applicability of CO₂ heat pump water heating systems on an energy consumption and component size basis compared to either synthetic refrigerant heat pump systems or other conventional water heating technologies. The second part of this paper presents the results of a system level model utilizing the component models developed here, along with a comprehensive parametric analysis of heat pump performance under varying input conditions, water heating strategies and with and without a SLHX.

NOMENCLATURE

			Subscripts	
A	area	(m ²)	d	displacement
C	thermal capacitance rate	(kJ K ⁻¹ s ⁻¹)	dis	discharge
D	diameter	(m)	high	high pressure
η	efficiency	(-)	low	low pressure
h	heat transfer coefficient	(kW m ⁻² K ⁻¹)	ref	refrigerant-side
i	specific enthalpy	(kJ kg ⁻¹)	s	isentropic
j	Colburn factor	(-)	suc	suction
k	thermal conductivity	(kW m ⁻¹ K ⁻¹)	vol	volumetric
$\dot{m}$	mass flow rate	(kg s ⁻¹)		
N	compressor speed	(rad s ⁻¹)		
P	pressure	(kPa)		
$\dot{Q}$	heat duty	(kW)		
Re	Reynolds number	(-)		
t	thickness	(m)		
T	temperature	(°C)		
V	volume	(m ³)		
$\dot{W}$	power	(kW)		

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