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Transcritical carbon dioxide microchannel heat pump water heaters: Part I – validated component simulation modules

Christopher Goodman [a], *Brian M. Fronk* [b], *Srinivas Garimella* [b,*]

[a] *United Technologies Research Center (UTRC), East Hartford, CT, USA*
[b] *Sustainable Thermal Systems Laboratory, George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, GA 30332, USA*

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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 CO_2 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%–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).

Systèmes de chauffage d'eau sanitaire à pompe à chaleur au dioxyde de carbone transcritique: Partie I – Modules de simulation des composants validés

Mots clés : Refroidisseur à gaz ; Microcanal ; Pompe à chaleur ; Transcritique cycle ; Dioxyde de carbone

1. Introduction

A desire to utilize low global warming potential (GWP) fluids in HVAC&R equipment has generated much interest in carbon dioxide (CO_2) due to its low GWP, favorable thermophysical properties, non-toxicity and low cost. CO_2 was once a widely used refrigerant, but fell out of favor with the advent of halocarbons, before being reintroduced as a viable refrigerant by Lorentzen and Pettersen (1993). Perhaps one of the most attractive applications of CO_2 transcritical systems is for water heating applications. Compared to synthetic refrigerants (e.g. R134a), CO_2 exhibits a distinct advantage in water heating

* *Corresponding author*. Tel.: +1 404 894 7479.
E-mail address: sgarimella@gatech.edu (S. Garimella).
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Nomenclature

A	area (m^2)
C	thermal capacitance rate ($kJ\ K^{-1}\ s^{-1}$)
D	diameter (m)
η	efficiency (–)
h	heat transfer coefficient ($kW\ m^{-2}\ K^{-1}$)
i	specific enthalpy ($kJ\ kg^{-1}$)
j	Colburn factor (–)
k	thermal conductivity ($kW\ m^{-1}\ K^{-1}$)
$\dot{m}$	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)
T*	temperature assuming pure refrigerant (°C)
V	volume (m^3)
$\dot{W}$	power (kW)

Subscripts

d	displacement
dis	discharge
high	high pressure
low	low pressure
ref	refrigerant-side
s	isentropic
suc	suction
T	total
vol	volumetric

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 CO_2 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, CO_2 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 3–4 (Kim et al., 2004). Many researchers (Cecchinato et al., 2005; Kim et al., 2005; Laipradit et al., 2008; Neksa et al., 1998; Sarkar et al., 2006, 2009; Stene, 2005) have conducted experimental studies and developed analytical models of systems for providing hot water, space heating, air conditioning, or a combination of all three. Nearly all prior investigations of CO_2 water heating systems utilize concentric tube-in-tube counter flow heat exchangers for both the evaporator and gas cooler (Cecchinato et al., 2005; Kim et al., 2005; Neksa et al., 1998; 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 CO_2 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 CO_2 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 CO_2. Little research has been published on CO_2 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 CO_2 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 CO_2 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.

2. Experimental approach

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 were controlled by adjusting the flow rate and temperature of the closed water loops. Two hermetic, reciprocating compressors, each with a swept volume of 2.46 cm^3 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-counterflow 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,