

Experimentally Validated Model of Refrigerant Distribution in a Parallel Microchannel Evaporator

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ABSTRACT

This paper develops a model for a parallel microchannel evaporator that incorporates quality variation at the tube inlets and variable mass flow rates among tubes. The flow distribution is based on the equal pressure drop along each flow path containing headers and tubes. The prediction of pressure drop, cooling capacity, and exit superheat strongly agree with 48 different experimental results obtained in four configurations using R134a. Predicted temperature profiles are very close to infrared images of actual evaporator surface. When compared to the uniform distribution model (that assumes uniform distribution of refrigerant mass flow rate and quality) results from the new model indicate superior prediction of cooling capacity, and exit superheat. Model results indicate maldistribution of refrigerant mass flow rate among the parallel tubes, caused primarily by pressure drop in the outlet header.

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INTRODUCTION

Because of the superior performance over the conventional round tube plate-fin heat exchangers, compact cross-flow heat exchangers with parallel flat multi-port microchannel tubes have been increasingly attractive in residential and commercial-scale air-conditioners and heat pumps. In microchannel (MC) condensers, due to only superheated vapor entering the inlet header, near-uniform vapor distribution is easily achieved among hundreds of parallel ports. However, in MC evaporators, the challenge is to distribute the liquid evenly among the microchannel ports, because the entering fluid is usually a saturated vapor-liquid mixture. This is also true for a reverse system in heating mode in which case the microchannel condenser functions as the evaporator. The thermalphysical properties of vapor and liquid phases are significantly different, usually resulting in flow maldistribution of refrigerant. This poor distribution significantly deteriorates the evaporator thermal performance and system COP.

Many studies have been performed to understand distribution of refrigerant in header-microchannel tube configuration, its effects, and the way it is exacerbated (Bowers et al., 2006; Hwang et al., 2007; Tompkins et al., 2002; Zou and Hrnjak, 2010). So far, to solve this problem,

primary approaches consist of using flow remedies to prevent separated flow along the inlet header (Jiang et al., 2010), developing special nozzles to create a Stokes flow (Fei et al., 2002), and adding a valve to bypass the flash gas for residential and mobile A/C systems (Beaver et al., 1999; Tuo et al., 2011).

Numerical modeling of microchannel heat exchangers is essential to save experimental cost and shorten design time of heat exchanger. However, most models available in open literature were developed on the simplification of simulating a single microchannel tube and thus refrigerant distribution was not taken into account (Yin et al., 2001; Kim & Bullard, 2001; Yun et al., 2007; Jin et al., 2011). Although these models gave reasonable predictions of heat exchanger capacity against experimental data, this does not necessarily indicate that the impact of maldistribution can be ignored. The maldistribution of refrigerant primarily impacts the superheat of the evaporator, not capacity calculation; since the superheat is assumed to be a certain value when designing an A/C system, it is of utmost importance to predict both superheat and capacity well in a system design.

Brix et al. (2009) developed a simplified model of two microchannel tubes and a distribution factor was introduced to quantify the effects of maldistribution on the heat

exchanger performance to some extent. Results indicated that at a fixed evaporator exit superheat, predicted evaporator performance significantly deteriorates with an increase in degree of refrigerant maldistribution. Based on a similar approach proposed by [Zietlow and Pedersen \(1995\)](#), [Litch and Hrnjak \(1999\)](#) modeled a microchannel ammonia condenser, and maldistribution in the intermediate header was considered by imposing a quality distribution profile at the inlet of tubes on the second. [Shao et al. \(2008\)](#) used the microchannel evaporator manifold distribution model developed by [Jin \(2006\)](#) to simulate the performance of a heat pump system with a microchannel evaporator. The maldistribution model showed good agreement with the test results, whereas if the uniform distribution model was used, heating time was overpredicted by 100%. [Stott et al. \(1999\)](#) compared simulated cooling capacity of a microchannel evaporator for a R410A split unit based on their uniform distribution model to experimental results. Overprediction of about 15% for most test cases was found and primarily was attributed to refrigerant distribution. [Yin et al. \(2002\)](#) characterized individual pressure loss coefficients in a microchannel heat exchanger by using single-phase nitrogen flow test. [Kulkarni et al. \(2004\)](#) discussed the header design concerning refrigerant distribution in a microchannel evaporator, and proposed that header diameter had a strong effect on gravity-induced and pressure-induced refrigerant maldistribution. [Ablanque et al. \(2010\)](#) created a numerical simulation model of two-phase flow distribution in a system with multiple T-junction branches.

There is no experimentally validated model of a microchannel evaporator that comprehensively considers refrigerant-side distribution and its effect on the performance. Although some models ([Jin et al., 2011](#); [Yin et al. 2001](#)) calculated pressure losses in the headers, these losses were not considered in refrigerant distribution; they were only considered to increase accuracy of pressure drop prediction.

OBJECTIVE

There are two types of maldistribution seen in a typical microchannel heat exchanger: maldistribution due to different flow resistances of the tubes, and maldistribution due to uneven quality distribution. In order to reduce the effect of uneven quality distribution and investigate the flow resistance effect independently, a flash gas bypass (FGB) separator is used in the FGB mode to remove the flash gas generated in the expansion valve. The evaporator is supplied with only single phase liquid refrigerant in this mode, rather than a two-phase mixture. This will be referred as FGB mode.

The system is also run in direct expansion (DX) mode, in order to see the effect of uneven quality distribution combined with flow-resistance induced maldistribution. DX mode is operated by closing the valve on the bypass line of the vapor-liquid separator.

This paper presents a model of refrigerant distribution in a microchannel evaporator for R134a. Each flow path through

the evaporator is comprised of various pressure drops, which sum to the same overall pressure drop when convergence is obtained. Maldistribution of refrigerant occurs because flow rate has to self-adjust to provide same pressure drop in all flow paths.

Quality at the inlet to the tubes differentiates the FGB mode (liquid only) and the DX mode in which each tube gets its own quality. In the model a quality distribution function is assigned to the inlet header to simulate the actual quality distribution, in addition to describing local pressure drops. The model was written using Engineering Equation Solver (EES).

Twelve data points for every configuration (same side (SS) or opposite side (OS) inlet/outlet locations) and mode (DX or FGB) were taken in the experimental facility, for a total of 48 data points. The model results are verified through infrared images of the evaporator surface and the experimentally obtained measurements of pressure drop, cooling capacity, and exit superheat.

EXPERIMENT DESCRIPTION

The experiments are carried out in a mobile A/C system facility. Schematic of test facility is shown in [Fig. 1](#). The modeled microchannel heat exchanger is in a single-pass and single-slab design with total 34 parallel microchannel tubes. Although this microchannel evaporator is undersized compared to other components in this MAC system and correspondingly refrigerant mass flow rate and cooling capacity is smaller than that for full size (multi-pass and multi-slab) evaporator, its simple flow pass design facilitates to study and understand two phase flow distribution among the parallel tubes, since the refrigerant distribution is completed in the inlet header. And, the finding and modeling results are applicable for other cases. In order to validate this distribution model for different refrigerant supply cases, 4 different cases of refrigerant supply to the microchannel evaporator are explored. They are 2 operation modes - direct expansion (DX) and flash gas bypass (FGB), 2 evaporator inlet/outlet location - same side and opposite side, as shown in [Fig. 2](#). Data were recorded at total 48 points for 3 indoor/outdoor air temperatures (25 °C /35 °C, 35 °C /35 °C, 40 °C /40 °C). The data range from 299 to 387 kPa for the absolute refrigerant evaporation pressure, 9.8~29 g/s for the mass flow rate of R134a refrigerant. Superheat at the compressor inlet is controlled in the range of 5~20 °C by a manually controlled electronic expansion valve. Air volumetric flow rate is fixed at about 0.21m³/s for all cases. The uncertainties for pressure, temperature and mass flow rate measurements are about ± 3.56 kPa, ± 0.5 °C and $\pm 0.1\%$, respectively. The resulting uncertainty for the capacity measurement is about 3.3%.

MODEL DESCRIPTION

In order to model refrigerant distribution among parallel microchannel tubes and overall performance of the

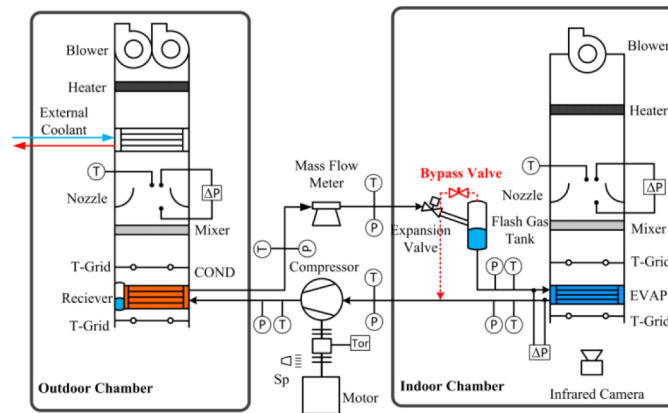


Figure 1. Schematic of test facility

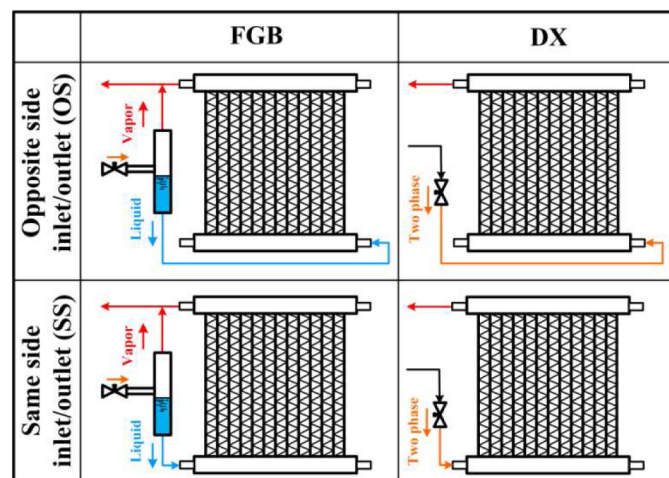


Figure 2. Four cases of refrigerant supply into the microchannel evaporator

evaporator, a discretized pseudo 2D-model of multiple microchannel tubes with inlet and outlet headers is built. The discretization schemes have been shown in Fig. 3(a) (b). Following proper assumptions are made to simplify the model without losing generality: 1) No heat conduction is considered along the tube and between tubes through fins; 2) Inlet and outlet headers are adiabatic; 3) Refrigerant distribution is uniform among parallel channels of each tube; 4) Incoming air has uniform temperature and velocity profile; 4) In FGB mode, quality at the inlet of each channel is zero.

1. EQUAL PRESSURE DROP ALONG EACH FLOW PATH

In this model, a path starting from the evaporator inlet and ending with the evaporator outlet through any microchannel tube in the evaporator is considered as a unique flow path. Based on fluid mechanics principle, regardless of the flow path the refrigerant takes, the various pressure drops along each flow path must add up to the same overall pressure drop.

Mass flow rate is different at each location along the inlet and outlet headers due to refrigerant branching into the

microchannel tubes. These different mass flow rates cause different local minor pressure drops in the headers. Due to different minor pressure drops in the headers, the stipulation that all unique flow paths must have the same pressure drop is met by changing mass flow rates among the microchannel tubes. The model reaches convergence when the pressure drop across each flow path is the same. A description of the model and various pressure drops considered in the model is shown in Fig. 3.

The overall pressure drop for each unique flow path can be expressed as:

$$\Delta P_{i, \text{flow_path}} = \Delta P_{in} + \sum_1^i \Delta P_{i, \text{ihd}} + \Delta P_{i, \text{tub}} + \sum_i^N \Delta P_{i, \text{ohd}} + \Delta P_{out} \quad (1)$$

The pressure drop in a tube includes: frictional, momentum, gravitational pressure drops, and sudden contraction as well as sudden expansion as following: