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Simulation of a Microchannel Heat Exchanger Considering Transverse Conduction

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Abstract – The use of microchannel heat exchangers (MCHX) is increasing in stationary air conditioning applications due to their high compactness and smaller liquid hold-up compared to traditional tube and fin heat exchangers (TFHX). Different approaches have been used in the literature to simulate the performance of MCHX. Three approaches including a method that considers the transverse heat conduction between the different refrigerant passes, are compared in this work. The methods are validated using experimental data available in the literature. The results of our study are presented in this article.

Keywords: MCHX, Condenser, HVACR, Rating

1. Introduction

Microchannel heat exchangers (MCHX) have been used in the automobile industry as radiators as well as in HVAC systems for several decades because of their compactness and low weight. MCHXs are ideal for use with flammable refrigerants since the liquid hold-up is much smaller due to their high compactness. MCHXs are therefore finding increasing use in stationary HVACR applications operating with low Global Warming Potential (GWP) refrigerants that are flammable.

Several approaches have been used to simulate the performance of an MCHX. Many authors [1-6] had discretized the microchannel tubes along the refrigerant flow direction. The authors assumed each cell as a crossflow heat exchanger having an Unmixed-Unmixed type of flow condition. The heat transfer rate is estimated using the ε-NTU formula of the Unmixed-Unmixed flow configuration. Authors [7-9] used either the AMTD or LMTD method to rate the MCHX. The AMTD or LMTD-based approaches need iterative procedures to reach the final solution as the outlet temperatures of any cell are not known in the first iteration. All these authors treated the individual cell as a counterflow heat exchanger. Such an assumption is not strictly valid in the desuperheating and subcooling regions. Besides the above rating techniques based on the standard methods like ε-NTU or LMTD and AMTD, some studies [10-12] are also available in the literature in which the authors had selected a computational domain and solved the energy, momentum, and continuity equations for each cell in the domain, using a suitable numerical technique. Glazar et al. [13] performed a detailed CFD analysis on a small computational domain of the MCHX to study the heat transfer phenomenon in detail. The drawback of the last two approaches is that they focus on only a particular part of the MCHX.

Some researchers have also used different techniques to incorporate transverse heat conduction between the passes (through the fins on the air side) in their rating models. Yin et al. [14] used the Fourier heat conduction equation with the ε-NTU method and solved the energy balance equations for each cell. Martínez-Ballester et al. [12] numerically solved the fundamental differential equation for fins along with the other governing equations for fluids and tube walls. Tube wall temperatures were used as boundary conditions for the fin differential equations. Prasad [15] presented a novel approach for taking into account the variation of transverse temperature of parting sheets of large multi-stream counterflow plate-fin heat exchangers used in air separation plants. In that approach, the fin is not considered adiabatic at half the height, and its location was allowed to vary even beyond the length of the fin. Thus, the approach of Prasad [15] allows conduction across the fins, not considered normally by other researchers studying the performance of

MCHX. This approach has been used to simulate the performance of an MCHX with transverse heat conduction between the passes in this work. The following methods have been implemented in our program:

1. ε-NTU method with Mixed-Unmixed flow condition, ignoring transverse conduction.
2. ε-NTU method with Unmixed-Unmixed flow condition, ignoring transverse conduction.
3. AMTD method of Huang et al. [8] combined with that of Prasad [15] to consider transverse conduction.

An experimental dataset of Li [16] for a condenser working with R134a has been used for validation. The performance estimated with the three different methods is compared in the following pages.

2. Methodology

2.1 Geometry details

An MCHX used in HVACR applications is an air-to-refrigerant, crossflow heat exchanger. The MCHX consists of stacks of flat tubes, called slabs. Fig. 1 shows a typical single slab, two-pass MCHX. The number of slabs used in an MCHX depends on the heat capacity required. The flat tubes are normally made by extrusion and consist of multiple parallel channels inside. The channels usually have a hydraulic diameter less than 1 mm, hence the name microchannel heat exchanger. The flat tubes are connected to headers at both ends. The number of headers depends on the number of passes required. Fins are brazed in the gaps present between the consecutive flat tubes (in the same or adjacent passes). Air flows through the gaps and over the fins, while the refrigerant flows through the microchannels in the flat tubes. Louvered fins are most common in automobile applications, while offset strip fins are preferred in aerospace or air separation applications. Either can be used in stationary HVACR applications.

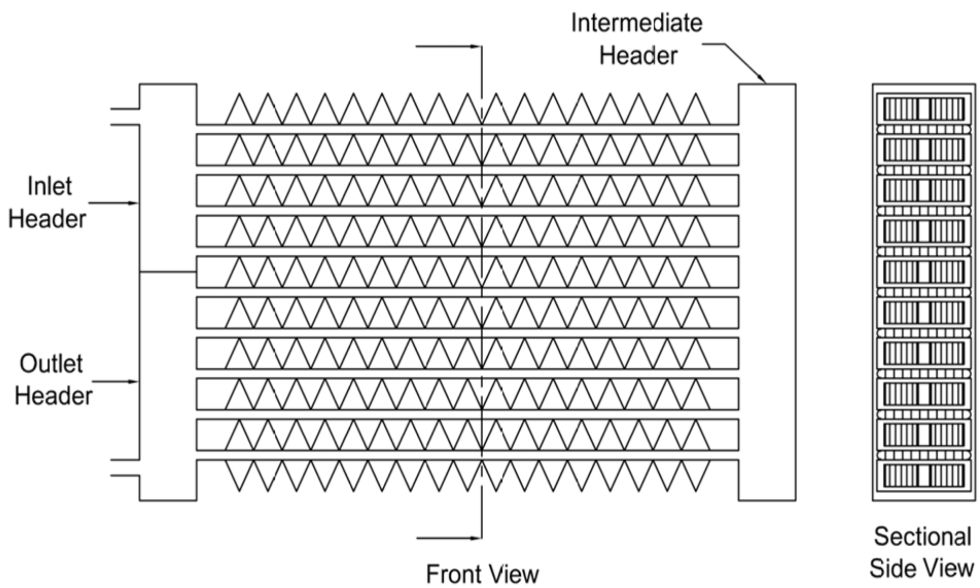

Fig. 1: A typical two-pass MCHX geometry

2.2 The MCHX rating algorithm

The discretization scheme employed for all three methods is shown in Fig. 2. The grid lines are shown by dotted lines while the actual boundaries of the microchannel tubes are shown by continuous lines. Air and refrigerant flow directions are also shown in the figure. All microchannel tubes are discretized into ‘n’ parts along the refrigerant flow direction. Each control volume (cell) formed after discretization has an elementary length dx and width equal to that of the flat tube. Each elementary cell is solved in the refrigerant flow direction to evaluate the heat transfer and pressure drop in the cell.

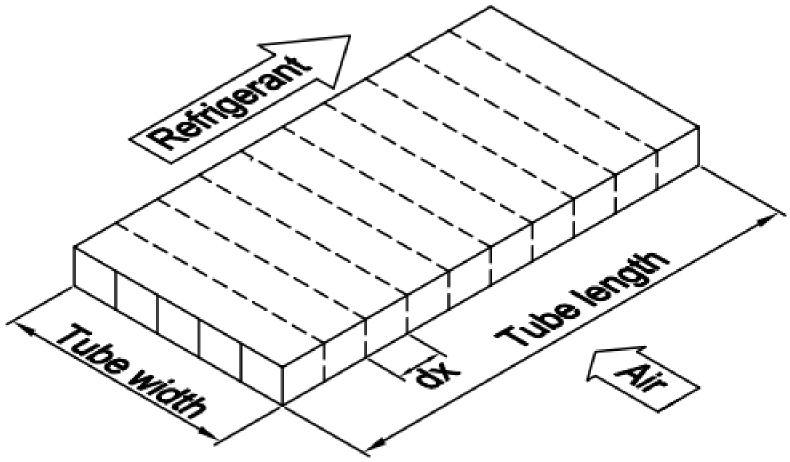

Fig. 2: Discretization scheme

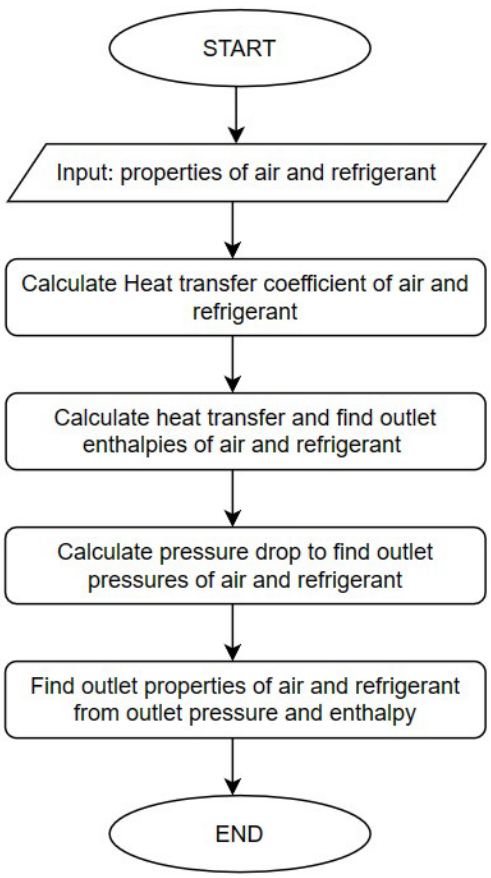

Fig. 3: Solution methodology

The main steps used to solve each cell by all three methods are presented in Fig. 3. All the thermophysical properties of refrigerant and dry air are determined using NIST REFPROP[17]. The humid air properties such as specific humidity, enthalpy, and relative humidity are determined using the methods specified by ASHRAE [18]. The heat transfer coefficient and friction factor are calculated from the correlations available in the open literature. The correlations developed by Shah [19] are used to calculate the heat transfer coefficient and that of Jige et al. [20] for the friction factor of the refrigerant flowing in microchannels. The correlations for heat transfer coefficient and friction factor for airflow are taken from Chang and Wang[21] and Chang et al. [22] respectively.

3. Results

3.1 Validation of the program

The rating program developed in this work has been validated using the experimental results of Li [16]. The author [16] performed experiments on an MCHX condenser with R134a at three different operating conditions. The author [16] was kind enough to share all the operating conditions, which were used to perform simulations with our rating program. The validation results are presented in Figs. 4 and 5.

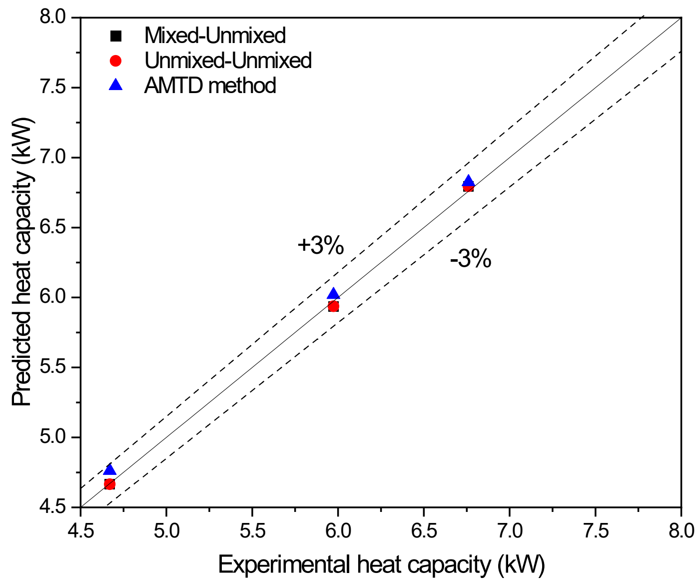

Fig. 4: Comparison of condenser heat capacity with experimental data

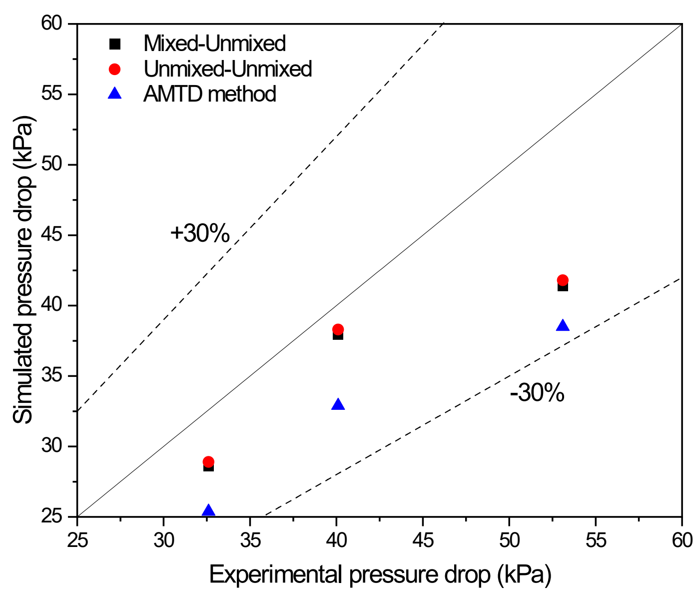

Fig. 5: Comparison of condenser pressure drop with experimental data

It is evident from Figs. 4 and 5 that all three methods predict the heat capacity to an accuracy of ±3%. On the other hand, all the methods underpredict the pressure drop in refrigerant significantly. Our method did not take into account the pressure drop in the headers. A change of direction of the refrigerant in the header is normally not negligible in multipass heat exchangers such as the one studied in this work. Alternately, the two-phase friction factor correlations used in this work may need to be replaced. These studies are currently being undertaken.

3.2 Heat transfer prediction considering transverse heat conduction

Studies were performed using the AMTD approach [8] with and without considering transverse heat conduction between the flat tubes through the air-side fins. The method of Prasad [15] was used to account for heat conduction between flat tubes. Fig. 6 shows the results of our study.

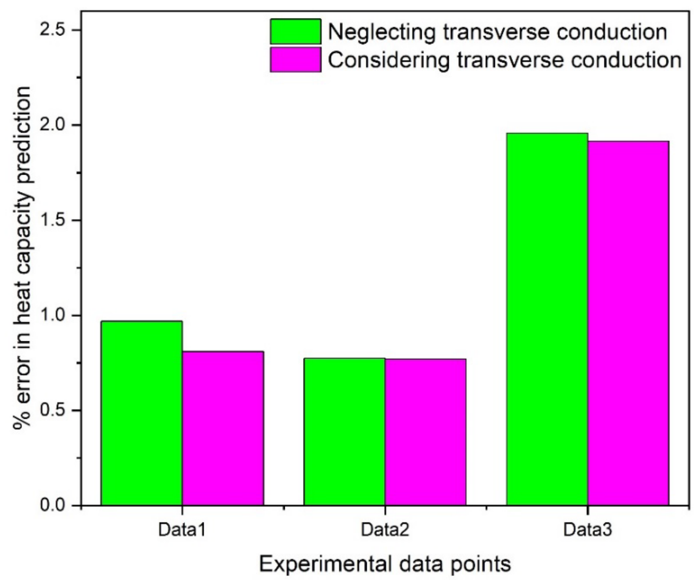

Fig. 6: Heat capacity prediction by considering and neglecting the transverse heat conduction

It is evident from Fig. 6 that the accuracy of the simulation does not improve significantly when the transverse heat conduction is considered. The study thus reconfirms that transverse heat conduction may be safely neglected in the type of MCHX studied in this work.

4. Conclusions

- A rating program has been developed for the simulation of an MCHX taking into account the conduction heat transfer due to temperature variation between the different tubes (passes). The method of Prasad [15], originally developed for multistream counter flow heat exchangers to account for transverse conduction has been used in this study. The method is capable of handling large temperature differences between adjacent tubes for example in a gas cooler of CO_2 based HVAC system.
- A comparison of the simulated heat load with those obtained by Li [16] shows that the uncertainty in the predictions is quite small (less than ±3%). The pressure drops predicted, however, are underpredicted by all the methods studied (up to ±30%), probably due to wrong friction factor correlation or neglect of the pressure drop in the headers, or both.
- The results presented reconfirm that transverse conduction can safely be neglected in the simulation of MCHX of the type studied in this work, with traditional refrigerants.

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