A Literature Review of Numerical Modeling Techniques for Vapor Compression Systems with Focus on Heat Exchanger Modeling

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ABSTRACT

Vapor compression systems are the most widely used system type in heating, ventilation, air-conditioning, and refrigeration (HVAC&R) applications. Experimental and numerical modeling techniques are used to analyze the performance of the vapor compression systems. With the introduction of high-performance computers, numerical modeling techniques are used extensively to develop cost-effective and efficient HVAC&R equipment. Experimental iterations on the design of vapor compression systems are costly; however, numerical techniques can reduce the number of experimental iterations, substantially decreasing the development cost and time. Because of the benefits associated with the numerical simulation, many researchers working in the HVAC&R field have attempted to develop efficient, robust, and accurate simulation models. This paper provides an in-depth review of heat exchanger modeling techniques as well as integration strategies to develop holistic system models.

1. INTRODUCTION

The four basic components of the vapor compression systems are evaporator, condenser, compressor, and expansion device. Various, sophisticated, numerical modeling techniques have been developed to examine the performance of vapor compression systems. Numerical models, once developed, can be run at very low cost compared to physical prototype development. A reduction of experimental iterations during new product development can be achieved by tuning the simulation models with limited experimental data. A heat exchanger model has substantial influence on the overall accuracy and fidelity of the vapor compression system model. As a result, these models warrant particular attention to ensure appropriately representative and accurate results from the accompanying system model. This study will examine heat exchanger modeling techniques in this context.

2. HEAT EXCHANGER MODELS

A vapor compression system has two distinct heat exchangers, one operating as a condenser and the other one as an evaporator. Heat exchanger modeling can be divided into four broad categories based on accuracy and computational time: lumped parameter models, moving boundary models, tube-by-tube models, and segment-by-segment (distributed parameter) models. In the following sections, this work will describe each of these modeling approaches, where they have been applied, and for what reasons. Additionally, this section will explore various limitations to these modeling techniques.

2.1 Lumped Modeling Approaches

The lumped modeling approach is the simplest approach amongst the four main modeling approaches. In this modeling approach, the whole heat exchanger is considered a control volume and an overall heat conductance (UA) value is used to analyze the performance of the heat exchanger. Either the log mean temperature difference (LMTD) or ε-NTU method is used to calculate the capacity of the heat exchanger. This model doesn't take into account the phase change transition inside the heat exchanger.

Parise (1986) developed a heat pump simulation model using simple component models to predict the overall system performance. The condenser and evaporator were both modeled based on lumped modeling approach using constant overall heat transfer coefficients based on the arithmetic overall temperature difference. The overall heat transfer coefficients for condenser and evaporator, which were predetermined empirical parameters to the model, considered both two-phase and superheated regions.

Braun (1988) presented general method to design, retrofit, and control equipment in large chilled water systems. For his chiller model, he developed simplified shell and tube condenser and evaporator model using lumped modeling approach (*i.e.* overall heat conductance and LMTD). He used condensing temperature to determine the LMTD for the entire condenser ignoring the superheated refrigerant at the condenser inlet. The smaller temperature difference was approximately compensated for by the higher heat transfer coefficient by considering the condensing temperature for the entire condenser. A similar approach was used for the evaporator. The simplified lumped models in comparison to more detailed models (*e.g.* tube-by-tube models) for condenser and evaporator were used to reduce the overall simulation time associated with the optimal control of the large chilled water system involving multiple components.

Jin and Spitler (2002) developed a simulation model for water-to-water vapor compression heat pumps based on parameter estimation for use in energy calculations and building simulation programs. The model contained evaporator and condenser models based on the lumped approach. A constant value of overall heat transfer coefficient (UA) was used in evaporator and condenser independent of fluid temperatures and flowrates. Also, refrigerant superheating was ignored in the evaporator while refrigerant superheating and subcooling were ignored in the condenser. The UA values in condenser and evaporator were estimated based on manufacturer's data. Therefore, the under-prediction of heat transfer in evaporator due to neglect of the superheated region was approximately compensated for by the estimated UA value. Similarly, neglecting superheated and subcooled region in condenser was approximately compensated for by the estimated UA value.

From the literature, it is found that for entire building level simulation containing multiple pieces of equipment, using a lumped approach for heat exchanger models can reduce the overall simulation time (Jin and Spitler, 2002). Also, models based on parameter estimation make use of lumped approach to develop simplified models of condenser and evaporator. However, these models may not provide accurate results (*i.e.* coil capacity etc.) as they don't take into account the phase transition and change in local refrigerant properties. Also, lumped models based on parameter estimation require tuning of parameters with the heat exchanger's performance data, which in some cases is unavailable.

2.2 Moving Boundary Modeling Approaches

The moving boundary modeling approach is more detailed than the lumped modeling approaches because it takes refrigerant phase change transition into account. On the air side, it considers transition from dry to wet section (if air dehumidification occurs). In this modeling approach, the heat exchanger is divided into single-phase and two-phase zones. Each zone is then solved using a lumped approach.

Braun (1988) developed cooling coil model for optimization of the chiller plant that considered transition from dry to wet coil section on the airside. Stefanuk *et al.* (1992) modeled tube in tube condensers as three separate heat exchangers connected in series to take into account three refrigerant phases (superheated, two-phase, and subcooled). They didn't consider airside transition from dry to wet section in their model.

Bell (2012) developed air conditioning and heat pump simulation model, ACHP, which uses a moving boundary fin-tube heat exchanger model. The ε-NTU method was used to analyze the performance of heat exchanger, which assumed constant specific heat values for both the fluids (refrigerant and air or cooling fluid). The average heat transfer coefficient on the air and refrigerant side (for each phase) were used for the analysis. The transition from dry to wet section was considered on the airside.

It is found from the literature that cooling coil models based on the moving boundary approach take into account the airside dry to wet section transition (Braun (1988), Bell (2012)). Evaporator and condenser models based on the moving boundary approach may or may not consider airside transition from dry to wet section. The moving boundary models can be more accurate than the lumped models while maintaining favorable computational speed. However, a comparison between the accuracy and solving time of models based on moving boundary approach and lumped approach was not found in open literature. The moving boundary modeling approach cannot take refrigerant or airside maldistribution into account.

2.3 Tube-by-Tube Modeling Approaches

The tube-by-tube modeling approaches include more detail than the moving boundary and lumped modeling approaches. In this approach, each tube of the heat exchanger is considered a control volume and lumped approach is used to solve each control volume (tube) individually to predict the heat exchanger performance. This approach can take into account the refrigerant side maldistribution by using the circuitry information and one dimensional airside maldistribution by using a 1D air profile. Transition from single-phase to two-phase or two-phase to single-phase can be taken into account by dividing the control volume *i.e.* heat exchanger tube into smaller control volumes where phase transition occurs. Alternatively moving boundary approach can be applied to each individual tube (Bach *et al.*, 2014a).

The tube-by-tube approach has been used in many models developed to analyze the heat exchanger performance. A software package originally developed by Domanski and Didion (1983), EVAP-COND, contains simulation models of fin-and-tube heat exchangers based on tube-by-tube approach. EVAP-COND is currently maintained by National Institute of Science and Technology (NIST); the model predicts the performance of each tube separately by assuming uniform refrigerant temperature in each tube and an average air temperature for all the tubes in a given row. However, the original model by Domanski and Didion did not consider non-uniform air distribution. Domanski (1989) later developed the evaporator simulation model EVSIM based on tube-by-tube approach. Unlike EVAP-COND, EVSIM accounted for both the air and refrigerant side distribution. To take into account the refrigerant side distribution, simulation starts with the refrigerant inlet tube of the given circuit and follows the consecutive tubes until the exit tube is reached. If there is a split in circuit, then the model solves one branch of the circuit first and then comes back to the split point to solve the remaining branches of same circuit. The same process is repeated for all other circuits of heat exchanger.

Liu *et al.* (2004) developed a fin-and-tube heat exchanger model based on graph theory using tube-by-tube modeling approach. The control volume included refrigerant inside the tube, fin and tube, and air outside the tube. This model was based on the application of energy conservation on each control volume. The refrigerant and air outlet states and wall temperature were obtained for each control volume by applying energy conservation on each control volume in an iterative method. An adjacency graph and corresponding adjacency matrix in graph theory were developed to explain the connections amongst each tube for solving complex refrigerant circuitry. The adjacency graph is a form of conceptualized hierarchy represented by vertices connected by edges. Each vertex represents a heat exchanger tube while edge shows the relationship between two tubes and contains the flow direction. The mathematical form of the adjacency graph is adjacency matrix. The longitudinal tube conduction and airside pressure drop were neglected in the model and refrigerant was assumed to have one-dimensional (1D) axial flow. An overall iterative algorithm based on graph-based traversal method was developed to solve the heat exchanger model. Energy and momentum equations were decoupled by solving the energy equations separately from the momentum equations. As a result, computational time of the new method was 1/40 to 1/60 of the previous methods to solve same heat exchanger model with similar level of detail.

Richardson (2006) developed the heat pump simulation model based on the object-oriented scheme. The heat exchanger model in the heat pump simulation model was modeled as a cross-flow fin and tube heat exchanger. Tube-by-tube approach was used to model the heat exchanger. To simplify the heat exchanger model, many tube passes were reduced into single pass**Error! Reference source not found.**.

A single segment was used to represent a long tube length with lumped pressure drop and heat transfer correlations. Due to above-mentioned assumptions, corrections factors were used to tune model capacity with the experimentally calculated capacity. A simultaneous solver was used to solve the hydraulic equations, while a successive substitution routine was used to solve the thermal equations.

The tube-by-tube modeling approach has been used by various researchers to develop detailed heat exchanger models which unlike lumped or moving boundary approach can take into account the refrigerant and airside (1D) maldistribution. These models can be more accurate than lumped and moving boundary models but they also require more computational effort. The heat exchanger models based on tube-by-tube approach were used for steady state simulations of vapor compression system (VapCyc (Richardson (2006)). Although detailed, these models cannot take into account 2D airside maldistribution.

2.4 Segment-By-Segment Modeling Approach

The tube-by-tube modeling approach can consider both refrigerant and airside 1D maldistribution; however, it cannot take into account airside 2D maldistribution. Segment-by-segment approach considers 2D maldistribution on airside by dividing heat exchanger tube into number of segments where each segment is treated as a control volume. Figure 1 shows the airside 1D maldistribution in tube-by-tube approach and 2D maldistribution in segment-by-segment approach.

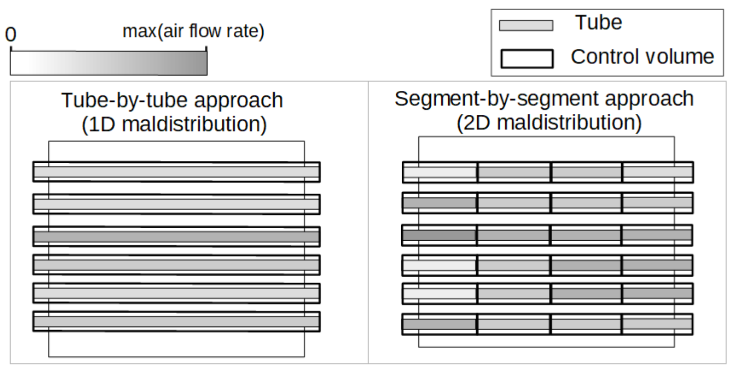

Figure 1: Illustration of airside 1D maldistribution in tube-by-tube modeling approach and 2D maldistribution in segment-by-segment modeling approach (tube return bends are not shown)

Similar to tube-by-tube approach, a segment can be further divided into sub segments if there is a phase transition in a segment. The control volume for different modeling approaches is shown in Figure 2.

Jiang *et al.* (2006) developed a heat exchanger simulation model, CoilDesigner based on segment by segment modeling approach. Each finned-tube macro-volume was divided into small segments to take into account 2D maldistribution on airside and the changing transport and thermal properties of the fluids. The segments were numbered in the direction of refrigerant flow. Each segment was considered a discrete unit of heat transfer and the ε-NTU approach was used to solve each segment. A junction tube connectivity matrix was used by Jiang *et al.* to incorporate the circuitry information for considering refrigerant side distribution.

Rossi (1995) developed heat pump simulation model, ACMODEL that included heat exchanger model based on segment-by-segment modeling approach. The capacity of the heat exchanger was found using ε-NTU approach on each segment. The original ACMODEL developed by Rossi didn't contain many modeling options e.g. the refrigerant side heat transfer correlation was only for smooth tube while the airside correlation was for louvered fins only. Also, it didn't consider refrigerant and air side maldistribution. Shen (2006) updated ACMODEL to include more modeling options. The heat exchanger model in the modified ACMODEL considered refrigerant side distribution by incorporating circuitry information.

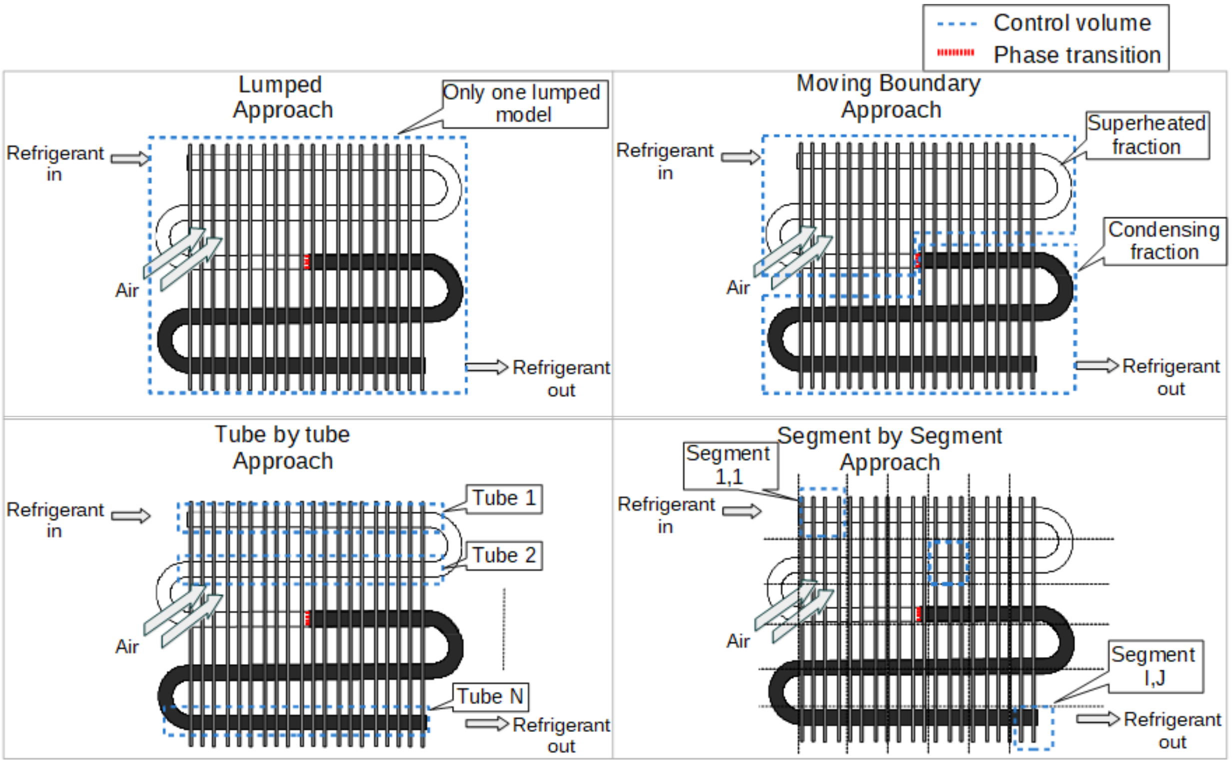

Figure 2: Control volume for different modeling approaches

Iu (2007) developed a heat exchanger model based on segment-by-segment modeling approach. He developed a flexible circuitry algorithm to account for refrigerant side distribution. He performed the experiments to calculate the local airside heat transfer coefficient as the airside heat transfer coefficient changes from row-to-row. The solution algorithm of heat exchanger involved several iterative procedures. The algorithm started by solving all the elements in the direction of refrigerant flow using ε-NTU method. After all elements were solved, the air properties were updated for each element. After the airside convergence, the refrigerant side distribution was incorporated in to model by providing the circuitry information.

Singh *et al.* (2008) developed a heat exchanger model using a segment-by-segment approach. They used the generalized circuitry algorithm developed by Jiang *et al.* (2006) to take into account the refrigerant distribution. Each finite segment was treated as a control volume and these segments were numbered in the direction of refrigerant flow. Equations for conservation of energy and log mean temperature difference (LMTD) methods were applied to take into account the tube-to-tube conduction through fins. They introduced two models to incorporate tube-to-tube conduction into the energy conservation equations. The first model was a resistance model in which Fourier law of heat conduction was used to obtain the heat transfer between the tubes. However, the effect of airside heat transfer coefficient was ignored. The second model was the conductance model in which two-dimensional heat diffusion equation was solved to calculate the amount of heat conducted between the tubes through fins. The conduction model was more accurate in the prediction of effect of conduction through fins on overall coil performance than the resistance model; however, it was computationally intensive. The accuracy of the resistance model was improved using the multipliers; however, determination of multipliers required tuning with experimental data.

The segment-by-segment approach is the most detailed heat exchanger modeling approach of all the modeling approaches as it considers refrigerant and 2D airside maldistribution, local refrigerant properties, and phase transition. This modeling approach may be more accurate than other modeling approaches but it requires significantly more computational effort than the tube-by-tube approach. A comparison of different modeling approaches in terms of computational time and accuracy was not found in literature.

2.5 Tube-to-Tube (Cross-Fin) Conduction

The tube-to-tube conduction through heat exchanger fins can degrade or increase (if multiple vapor compression systems on same heat exchangers) the performance of heat exchanger and needs to be incorporated in the heat exchanger simulation model for accurate prediction of its performance. Figure 3 shows the tube-to-tube or cross-fin conduction in a fin and tube heat exchanger.

Heun and Crawford (1994) developed an analytical model of cross-counter flow fin-and-tube heat exchanger to study cross-fin conduction. They compared capacity of same heat exchanger with continuous and split fins and found a maximum capacity degradation of 40% in continuous fin heat exchangers in comparison to split fin heat exchangers.

Romero-Méndez *et al.* (1997) developed an analytical fin and tube heat exchanger model to quantify heat conduction between neighboring tubes. They found that tube-to-tube conduction could result in capacity degradation of up to 20% by decreasing heat transfer from in-tube to over-tube fluid. Heun and Crawford (1994) and Romero-Méndez *et al.* (1997) found negligible effect of longitudinal tube conduction on heat exchanger performance.

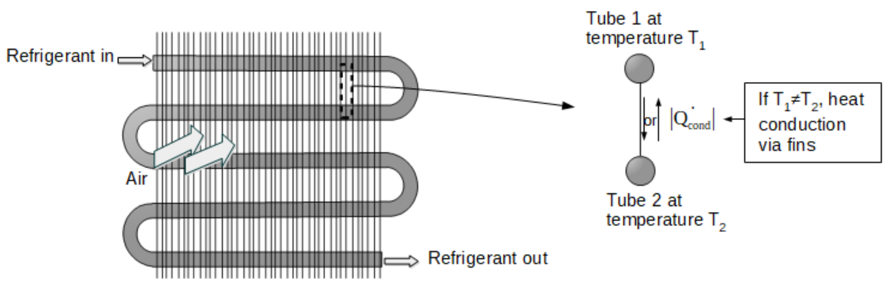

Figure 3: Tube to tube (or cross fin) conduction

Domanski *et al.* (2007) experimentally analyzed the effect of tube-to-tube conduction on the heat exchanger performance using 5 kW fin and tube heat exchanger. They performed experiments with the slit and slit cut fins to study the effect of cross fin conduction on heat exchanger performance. They found, experimentally, a capacity degradation of 20% in slit fin heat exchanger compared to slit split fin (tube depth rows separated by a cut in fins) heat exchanger for a superheat of 16.7°C. They found a negligible (one twentieth of a percent) loss in heat exchanger capacity due to longitudinal tube conduction.

Table 1 summarizes the heat exchanger models developed by some research groups. It includes the information on whether the tube-to-tube conduction was considered in the model. The effect of considering tube-to-tube conduction on examining heat exchanger performance is not provided by most of the researchers in Table *1*. Table 1 shows that models based on tube-by-tube or segment-by-segment modeling approach are required to analyze tube-to-tube conduction in heat exchangers.

Table 1: Summary of heat exchanger models developed by different groups

Model/Researcher	**Modeling Approach**	**Method used**	**Tube-to-tube conduction considered?**	**Refrigerant side and airside distribution considered?**
EVAP-COND (Domanski and Didion, 1983),	Tube-by-tube	ε-NTU	No	No
EVSIM (Domanski, 1989)	Tube-by-tube	ε-NTU	No	Yes (2D air distribution not considered)
EVAP5M (Lee and Domanski, 1997)	Tube-by-tube	ε-NTU	Yes	Yes (2D air distribution not considered)
Liu *et al.* (2004)	Tube-by-tube	Energy conservation	Yes	Yes (2D air distribution not considered)

Model/Researcher	Modeling Approach	Method used	Tube-to-tube conduction considered?	Refrigerant side and airside distribution considered?
Richardson (2006)	Tube-by-tube	ε-NTU	No	Yes (2D air distribution not considered)
CoilDesigner (Jiang *et al.*, 2006)	Segment-by-segment	ε-NTU	No	Yes
Rossi (1995), Bo Shen (2006)	Segment-by-segment	ε-NTU	No	Yes
Iu (2007)	Segment-by-segment	ε-NTU	No	Yes
ACHP (Bell, 2012)	Moving boundary	ε-NTU	No	No
Heun and Crawford (1994)	Analytical	Energy conservation	Yes	No
Romero-Méndez *et al.* (1997)	Analytical	Energy conservation	Yes	No
Singh *et al.* (2008)	Segment-by-segment	Energy conservation and LMTD	Yes	Yes

2.6 Air and Refrigerant Side Maldistribution

Both air and refrigerant side maldistribution can result in significant degradation of heat exchanger performance. Different researchers studied these flow maldistributions and presented different solutions to minimize both the air and refrigerant side maldistributions.

Domanski *et al.* (2004) developed a refrigerant circuitry optimization algorithm, ISHED1 (Intelligent System for Heat Exchanger Design), based on the non-Darwinian evolutionary computation method (Michalski, 2000). This algorithm optimized the refrigerant circuitry for both the uniform and non-uniform airflow profiles. The different designs were simulated using EVAP heat exchanger simulation model. However, refrigerant circuitry developed by ISHED1 required some post processing to make certain that the proposed circuitry can be manufactured.

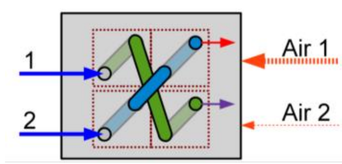

Figure 4: Refrigerant circuitry interleaved (Bach *et al.* 2014b)

Bach *et al.* (2014b) developed an interleaved circuitry approach (see Figure 4) and an active refrigerant flow control to accommodate for varying airside maldistribution. They compared the hybrid (active) approach (both refrigerant circuitry and active refrigerant flow control considered) with the passive approach (only refrigerant circuitry considered) to mitigate the airside maldistribution. They found the passive approach to be less effective than the active approach to recover the performance losses for varying airflow maldistribution. For example, for an airflow maldistribution of 50%, the capacity dropped by 65% of its original value for the standard evaporator. However, using passive and hybrid control scheme, the capacity was within 81% and 93% of its original value respectively. They considered implementation cost and reliability of passive approach better than for the active one.

Various numerical and experimental studies can be found in literature to analyze and reduce the airside and refrigerant side maldistribution. The models based on tube-by-tube or segment-by-segment modeling approach are required to study the air and refrigerant side maldistribution numerically. The passive approach is commonly used in the development of commercial coils e.g. interleaved circuitry to combat refrigerant and airside maldistribution due to lower cost and better reliability as found by Bach *et al.* (2014b).

2.7 Fin-Tube Contact Resistance

The tubes are inserted into the holes printed into a stack of fins to manufacture fin-and-tube heat exchangers. The tubes are then expanded to make tube-to-fin pressure contacts. The interference fit results in some contact resistance at the fin-tube attachment. This resistance is very small as compared to the total airside thermal resistance of the fin and tube heat exchanger. However, some studies have shown that there may be situations where this contact resistance cannot be neglected.

Sheffield *et al.* (1989) showed that with the increase in fin spacing or thickness, the contact resistance can be as much as 16% of the total airside thermal resistance. The study also developed a correlation that can estimate the thermal contact resistance in fin and tube heat exchangers based on fin thickness, fin spacing, tube diameter, and tube spacing. EVAP5M (Lee and Domanski, 1997), a fin and tube heat exchanger model, incorporated the fin-tube contact resistance by including contact resistance correlation developed by Sheffield *et al.* (1989) in the overall heat transfer coefficient calculation.

Jeong *et al.* (2006) used an experimental-numerical method (a numerical scheme using experimental data) to estimate the thermal contact conductance in 22 different heat exchangers with 7 mm tube and developed a correlation for it. They found a significant effect of fin type, tube-manufacturing type etc. on the thermal contact conductance in the heat exchanger. They performed the experiments in the vacuum chamber that improved the accuracy of numerical procedure for thermal contact resistance estimation. However, the heat transfer in the gap between the fin and tube is different in vacuum than the actual conditions encountered in industrial settings.

Taler (2007) estimated the thermal contact resistance at the fin tube interface based on the condition that the Colburn j-factors obtained using the experimental data and the CFD simulation of heat transfer in heat exchanger must be in good agreement to estimate the thermal contact resistance at fin-tube interface. Taler and Ocłoń (2014) studied the effect of thermal contact resistance using CFD simulations. They presented a method to calculate fin efficiency numerically as a function of mean thermal contact resistance using mixed finite element and finite volume method. The thermal contact resistance lowered fin efficiency from 0.64 to 0.4 for the same airside heat transfer coefficient.

From the literature, the fin-tube contact resistance is found to cause a significant change in airside heat transfer coefficient. Although important, it is not evident from the review of different heat exchanger models whether fin-contact resistance was included while determining overall heat transfer coefficient on the airside.

3. SYSTEM MODELS

The main focus of this paper is the heat exchanger modeling, however, this section provides a brief overview of the commonly used system models. A model for a vapor compression system is developed by integrating its component models. A solution scheme is required to solve the unknown variables (*e.g.* state points containing fluid information, in the system model to predict system performance). The solution scheme performs the steady state simulations that are important for designing the system and predicting its performance. The solution scheme should be inherently efficient, accurate, and robust. Two approaches are commonly used to solve unknown variables. The first one is successive substitution approach in which the system components are solved one-by-one in the refrigerant flow direction. The second approach is simultaneous approach that solves all the unknown variables of different system components simultaneously by using non-linear equation solvers such as Newton-Raphson or Broyden's method (quasi-Newton solver).

The successive substitution approach uses more than one nested loop to solve the system model. It is easier to develop an efficient iteration scheme, which is both robust and accurate, based on successive substitution approach for simple system configurations. However, once the system gets complicated (*e.g.* system with multiple compressors or a multistage system) it is difficult to develop an efficient scheme based on this approach. Furthermore, this approach is not very flexible: addition of any component to the system requires significant changes in the system solver.

Unlike the successive substitution approach, the simultaneous approach provides the flexibility and generality to the system solution scheme. This approach allows the decoupling of system solver from the component models i.e. component models appear black box to the system solver. The component models solve their equations based on the boundary conditions provided by the system model and the resulting outputs from the component models are fed back to the system solver. However, the simultaneous method is less robust than the successive substitution method because more initial values are required to solve the system and reasonable initial guesses are not always possible to specify. Table 2 lists the approaches used by different researchers to solve their system model.

Table 2: Approach to solve the system model by different researchers

System solver	**Model and/or Citation**
Successive substitution approach	Hiller and Glicksman (1976), Ellison and Crewick (1978), EVAP-COND (Domanski and Didion, 1983), EVSIM (Domanski, 1989), EVAP5M (Lee and Domanski, 1997), Fischer (1999), Tandon (1999), Robinson and Groll (2000), Sarkar *et al.* (2006), Iu (2007)
Simultaneous approach	Parise (1986), Jolly *et al.* (1990), ACMODEL (Rossi, 1995), Hwang and Radermacher (1998), VapCyc (Richardson, 2006), CoilDesigner (Jiang *et al.*, 2006), Shao *et al.* (2008)

4. CONCLUSIONS

During the review of heat exchanger models, only few references are found that considered cross-fin conduction in their heat exchanger model. Cross-fin conduction can have significant effect on the performance of heat exchanger if the neighboring tubes have refrigerant at different temperatures due to refrigerant or airside maldistribution.

Different researchers included 1D or 2D airside maldistribution in their heat exchanger models by using tube-by-tube or segment-by-segment modeling approaches. However, no detailed comparison was found in the literature between considering the effect of 1D or 2D airside maldistribution on the heat exchanger performance.

Fin-to-tube contact resistance can have significant effect on heat exchanger performance. Few heat transfer coefficient correlations take into account the fin-to-tube contact resistance. However, it is not clear from the literature of most of the heat exchanger models on whether or not contact resistance is taken into account.

Different solvers are developed by researchers to solve the heat exchanger and vapor compression system models. However, there is still a need to develop a solver that can solve both simple and complex system configurations efficiently while maintaining robustness. The solver should also be able to address the discontinuities in property functions, correlations, and sub models.

Despite fin and tube heat exchangers being a well-established technology there is still a substantial room for improvement of modeling tools.

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