Simulation and optimisation of miniature bare tube heat exchangers

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SIMULATION AND OPTIMISATION OF MINIATURE BARE TUBE HEAT EXCHANGERS

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2005

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A thesis submitted to Nanyang Technological University
in fulfillment of the requirement for the degree of
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2005

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SUMMARY

In this project, new miniature bare tube heat exchangers, i.e., the evaporator and condenser for miniature vapor compression cooling system have been introduced. Miniature bare tubes have been applied for the miniature heat exchangers. A simple approach to simulate miniature bare tube heat exchangers with simple refrigerant circuitry has been developed. In contrast to the existing models in the literature, the proposed model leads to a better insight into the processes taking place in the bare tube heat exchangers. A mathematical model, namely the distributed model, has been developed to analyze the operational behavior of heat exchangers, which includes single and two-phase flow heat transfer. Matching with the uneven heat transfer characteristics of the heat exchangers, this model is based on an arbitrarily size control volume, where the local information on the heat transfer and fluid flow can be provided for the performance evaluation in the miniature heat exchanger design.

The heat transfer characteristic and pressure drop behaviors have been certified with measured results. The model is then applied to design the coils of miniature evaporator and condenser. The model has been linked with an optimization algorithm to search for a combination of geometrical dimensions that produces optimum heat exchanger performance, i.e., maximum heat transfer at minimum pressure drop. In the optimization design of the miniature heat exchangers, the BOX complex optimization method is used. A feasible miniature bare tube heat exchanger (3cm × 3cm × 1cm) has been obtained.

Based on the proposed computer simulation algorithm, the coil with arbitrary complex refrigerant circuitry can be simulated by the model for the future study.

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Table 4-4 Implicit constraints.

NOMENCLATURE

A	effective heat transfer area of (m^2)
Bo	Boiling number
C	mass transfer parameter defined by Eq. (3-1c)
C_n	constant for a heat exchanger with a row number rather than 4
$C_{p, a}$	isobaric specific heat for air ($\text{kJ/kg} \cdot ^\circ\text{C}$)
$C_{p, w}$	isobaric specific heat for water ($\text{kJ/kg} \cdot ^\circ\text{C}$)
d_i	tube inner diameter (m)
d_o	tube outer diameter (m)
D_h	Hydraulic diameter (m)
f	friction factor
F_{obj}	objective function used in optimization studies
g	gravitational acceleration
G	mass velocity ($\text{kg/m}^2 \cdot \text{s}$)
H	duct height (m)
H_t	height of heat exchanger (m)
h_{a1}	inlet enthalpy of air (kJ/kg)
h_{a2}	outlet enthalpy of air (kJ/kg)
h_{r1}	inlet enthalpy of refrigerant (kJ/kg)
h_{r2}	outlet enthalpy of refrigerant (kJ/kg)
i_{fg}	latent heat of moisture condensation (kJ/kg)
j	colburn factor
k	thermal conductivity (kW/m.K)
L	length of heat exchanger (m)
Le	Lewis number
M	mass (kg)
m_a	mass flow rate of air (kg/s)
m_r	mass flow rate of refrigerant (kg/s)
N_r	number of tube row

N_t	number of tube transverse
N_u	Nusselt number
P	pressure (Pa)
P_r	Prandtl number
Q	heat exchange (kW)
δq	heat exchange rate (kW)
S_l	tube longitudinal space (m)
S_t	tube transverse space (m)
St	Stanton number
T	temperature (°C)
U	velocity (m/s)
U_o	overall heat transfer coefficient
V	volume (m ³)
W_t	width of heat exchanger (m)
W_a	air specific humidity (kg/kg)
W_s	specific humidity of saturation air (kg/kg)
X	vapor quality
Y_f	fin thickness (m)
Δ	difference element
ΔP	pressure drop (Pa)
ΔP_f	friction pressure drop (Pa)
ΔP_v	vapor pressure drop on refrigerant side (Pa)
ΔP_l	liquid pressure drop on refrigerant side (Pa)
Re	Reynolds number

Greek symbols

V	specific volume (m ³ /kg)
α	heat transfer coefficient (kW/m ² .K), void fraction
α_{sen}	air sensible heat transfer coefficient (kW/m ² .°C)
α_{lat}	air latent heat transfer coefficient (kW/m ² .°C)

α_m	mass transfer coefficient (kg/m ² .s)
β	parameter defined by Eq.2-35b, nozzle diameter ratio
λ	friction coefficient
δ	incremental element
ϕ	fin efficiency
ρ	density (kg/m ³)
ρ_f	fluid density (kg/m ³)
ρ_l	liquid refrigerant density (kg/m ³)
ρ_v	vapor refrigerant density (kg/m ³)
δz	length of one control volume (m)
ψ	friction multiplier

Subscripts

1	inlet
2	outlet
a	air
act	actual
c	condensation
cec	convective coefficient
cf	coil face
d	dry
e	evaporation
eq	equivalent
f	friction, fin
i	inner
l	liquid
lat	latent
m	momentum, mean
o	outer
p	predicted

<i>r</i>	refrigerant
<i>s</i>	single-phase flow
<i>sub</i>	subcooled
<i>sup</i>	superheated
<i>t</i>	tube
<i>TP</i>	two-phase flow
<i>v</i>	vapor

Chapter 1

Introduction

Chapter 1

Introduction

1.1 Background

Microelectronic devices including the central processing units have been a significant decrease in physical size and have provided improved function at a reduced cost in the past thirty years. This is the unique factor that has made the sustained and successful growth of the semiconductor industry since the early 1970s. The miniaturization of electronic components and the rapid increase in power density of advanced microprocessors and electronic components have created a crucial need for improved cooling technologies to achieve high heat-dissipation rates. The science of heat transfer is exceeded to the designer of electronic components and devices; it can lead to accurate predictable engineering designs before construction if sufficient details of the fluid-flow and thermal flow paths are available. In many applications, the absence of compact cooling techniques has challenged the viability of the future high-powered high performance electronics. Proper use of cooling technologies can also lead to important gains in efficiency and performance. Conventional air-cooling with heat sinks was the only accepted mode, today; however even very traditional industries are now considering the liquid cooling. Those high heat fluxes cannot be easily dissipated using existing cooling technologies although electronics cooling technology has been developed vastly in the past few years. Therefore, more effective, high-performance and compact cooling approaches are needed.

Chapter 1**Introduction****1.1.1 Comparison of existing miniature refrigeration systems**

While the development of fans with high pressure and low acoustic emission is under the way, the development of low price, compact and reliable water-cooling technologies as well as the application of other refrigeration systems such as thermoelectric, vapor compression, Stirling, pulse tube, sorption and reverse Brayton are seen as offering great potential for cooling technologies.

A succinct comparison of the most promising miniature and miniature refrigeration systems, including thermoelectric, vapor compression, Stirling, pulse tube, sorption, and reverse Brayton for electronic packaging is presented in Table 1-1 (Phelan *et al*, 2002). Some explanation of the terms in the table is in order to aid its comprehension. The maximum cooling power, in [Watt], is that refrigeration capacity achieved for a minimal temperature reduction. The minimum cold-end temperature, in [degree Celsius], is the minimum achievable temperature for a cooling power of ~ 0 Watt, or some very small cooling power. Note that the maximum cooling power and minimum cold-end temperature cannot be achieved simultaneously. The COP represents the coefficient of performance, a measure of the refrigerator's efficiency, and is defined from

$$COP = \frac{Q}{W} \quad (1-1)$$

where Q is the refrigeration capacity, or cooling power, in [Watt], and W is the work input to the refrigerator, also in [Watt]. Note that the COP can be greater than one, and is limited only by the Carnot COP. The remaining columns in Table 1-1,

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reliability, cost, size, and comments, are all qualitative evaluations. The estimation of reliability takes into account the number of moving (mechanical) parts in the refrigerator, such that the refrigerators with the fewest number of moving parts (thermoelectric, pulse tube and sorption) are rated as having the highest reliability. The cost is evaluated relative to the cost of an ArtiCoolerTM Model CA fan, manufactured by Agilent, which is reported to have a unit cost of about \$13. Refrigerators with estimated (or actual) unit costs well in excess of this amount are considered to be ‘high cost’, while those with comparable unit costs are considered to be ‘medium cost’, and finally those with unit costs less than this amount are considered to be ‘low cost’ (Phelan *et al*, 2002). The author anticipated that by 2010, the miniature vapor compression system may be available and it is clearly shown in the table 1-1, that it produces a highest cooling power capacity at a high COP.

Chapter 1**Introduction****Table 1-1** Comparison of miniature active cooling technologies (Assumes heat rejection temperature of 25 °C) (Phelan *et al*, 2002)

		Maximum Cooling (W)	Minimum Tem. (K)	COP	Reliability	Cost	Size	Comments
Thermo-electric (single-stage)	Present (2000)	125	228 (-45 °C)	0.3 @ 0 °C	High	High \$43/unit	Scaleable to micron level	Solid-state device; easily controlled
	Future (2010)	250	163 (-110 °C)	0.4 @ 0 °C	High	High	Scaleable to micron level	Solid-state device; easily controlled
Thermo-electric (multi-stage)	Present (2000)	60	165 (-108 °C)	0.3 @ 0 °C	High	High	Scaleable to micron level	Solid-state device; easily controlled
	Future (2010)	120	118 (-155 °C)	0.4 @ 0 °C	High	High	Scaleable to micron level	Solid-state device; easily controlled
Vapor Compression	Present (2000)							No available Small systems
	Future (2010)	350	285 (12 °C)	3-6 @ 0 °C	Medium	Medium	width: 0(cm) thick: 0(mm)	Utilizes R-134a
Stirling	Present (2000)							
	Future (2010)	270 (5cm long)	285 (12 °C)	~7 @ 0 °C	Low	High	0(cm)	Two moving parts (piston & displacer)
Pulse Tube	Present (2000)							
	Future (2010)	85 (5cm long)	20 (-253 °C)	0.1 @ 0 °C	High	Medium	0(cm)	Only one moving part, which is at high temp.
Sorption	Present (2000)							
	Future (2010)	< 1	80 (-193 °C)	<0.005 @ 80K	High	Low	0(cm)	Sorption compressor JT cooler
Reverse Brayton	Present (2000)							
	Future (2010)		80 (-193 °C)	0.2 @ 80K	Medium	High	~7.5cm	Requires miniature turbine & compressor

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1.1.2 Prospect of new cooling technology

Electronics cooling technology has been developed vastly in the past few years mainly due to the requirement to remove high heat fluxes from ever increasingly smaller microelectronic systems. The comparison of existing and predicted future miniature refrigeration systems reveals that the most mature, effective and the most efficient cooling system available today is the vapor compression refrigeration system. It is anticipated that, ten years from now, it is likely that several types of miniature refrigerators will become available, including the Stirling, pulse tube, reverse Brayton and sorption refrigerators. But they are limited in the electronic cooling applications, in part due to their low efficiencies or low reliability. Vapor compression systems are reliable and relatively inexpensive (Schmidt and Roger, 2000). Presently, vapor compression refrigeration is being adapted to cool computers and telecommunications equipments in a limited number of high performance applications. The vapor compression refrigeration system is unsuitable to be used in the microelectronics cooling due to the unavailability of miniature heat exchanger and small-scale refrigeration compressor. Currently, the traditional finned tube heat exchanger is too large to make the miniature vapor refrigeration system. Obviously, there is a need to reinvent the vapor compression cycle to handle cooling in microelectronics, which may involve miniaturization of heat exchanger, compressor and finally the whole system. Miniaturization of the vapor compression system may not be difficult to achieve technologically, but looking at the current vapor compression systems, the shrinking in physical size alone will not make the system viable as the cost factor may go beyond the feasible threshold. It is thus believed that breakthrough innovation in

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transforming today's vapor compression technology into tomorrow's effective miniaturization cooling technology is much need.

In this project, new miniature bare tube heat exchangers, evaporator and condenser for miniature vapor compression cooling system will be introduced. The bare tube heat exchangers will be systematically modeled, numerically and experimentally studied.

1.1.3 General Description of Heat Exchanger

The most common evaporators and condensers use air as the heat transfer medium, where air flows over a straight-tube bank and refrigerant flows inside the tube. To enhance the heat transfer in the airside, the straight tubes are externally finned with continuous plates. To control the refrigerant flow, the finned tubes are connected to form a coil with an appropriate number of refrigerant paths. In this study, miniature bare tubes are applied for the heat exchanger.

The performance study of evaporator and condenser is complicated by a number of factors. One of the factors is that the flowing refrigerant inside the tubes undergoes phase change, i.e. evaporating process or condensing process, which results in significant variations in heat transfer and fluid flow characteristics along the refrigerant path. Another factor is that there are numerous possible refrigerant circuitry arrangements in coil design. A complex refrigerant circuitry of the heat exchangers contributes to the uneven heat transfer. As far as the airside is concerned, a variety of air flow velocity are being used and the nature of air flow and heat transfer for different positions of the coil is hard to be correlated in a general way. All

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these factors lead the heat exchangers to a new modeling approach that is capable of considering all the variations of the parameters rather than the classical heat exchanger analysis techniques, such as LMTD and ζ -NTU approaches.

1.1.4 Research Needs

A literature survey, which will be presented separately in the relevant chapters according to the subject classification used in this thesis, has revealed substantial deficiencies in the knowledge related to the heat exchanger performance. The following is the summery of these research needs:

- ◆ The plate-fin-tube is commonly used for a heat exchanger. However, the bare tubes applied for the miniature heat exchanger has been scarcely reported.
- ◆ Moisture condensation on the air side results in the simultaneous heat and mass transfer. The effect of mass transfer on the heat exchanger performance has not been fully investigated. Little experimental data on coil performance in high air humidity have been published.
- ◆ The increasing interests in the application of new environment friendly working fluids have created a need for a detailed analysis of the heat exchangers with R134a as a refrigerant. Although extensive research related to the heat transfer characteristics of R134a has been done, the experimental study of heat exchangers using R134a as refrigerant is still lacking.
- ◆ Although an existing condenser model is capable of simulating a condenser coil with a specified refrigerant circuitry (Ellison *et al* 1981), it is based on a tube-to-tube computation approach. There is no existing simulation model which is capable of investigating the complex refrigerant circuitry of heat exchangers

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based on any number of control volumes. Furthermore, the optimization of the coil performance using the complex refrigerant circuitry has been scarcely documented both on the theoretical analysis and on the experimental study.

- ◆ For the optimization design of heat exchanger, Complex Box Method has been commonly accepted. However, little work has been reported on the combination of coil modeling and geometrical optimization design based on the Complex Box Method.

The deficiencies in the knowledge have been the obstacles for the performance analysis and improvement. This makes up a pressing need for an accurate approach to the heat exchanger performance study.

1.2 Objectives and Scopes

Based on the awareness of the deficiencies in the knowledge, the objectives of this research are set out as follows:

- 1) To model bare tube evaporator and condenser coils mathematically,
- 2) To develop a general computer algorithm to simulate coils with complex refrigerant circuitry,
- 3) To design and set up an experimental test rig to validate the mathematical model,
- 4) To improve the coil performance using complex refrigerant circuitry,
- 5) To geometrical optimization design for the bare tube heat exchanger performance using Complex Box Method,
- 6) To identify the parameters influencing the coil performance in a humid environment.

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To achieve the above-mentioned objectives, the approaches are to combine modern mathematical and computational methods with experimental techniques. A bare tube heat exchanger with a total physical size of around 3 cm by 3 cm by 1 cm has been developed. The mathematical model has been developed to analyze the operational behavior of this heat exchanger, which includes single and two-phase flow heat transfer. The heat transfer characteristics and pressure drop behavior are established.

The heat exchanger model has been linked with an optimization algorithm to carry out optimum design of the heat exchanger, in which the design operation at the maximum heat transfer rate, at a smallest physical volume and pressure drop has been developed. Test apparatus is being designed and fabricated to verify the performance of this heat exchanger.

1.3 Outlines of the Thesis

This thesis presents a detailed study of the miniature bare tube heat exchanger performance. Particular attention is given to those conditions where sensible and latent heat transfer takes place simultaneously outside the bare tube evaporators. The organization of the thesis is shown by the followings:

In chapter 1, the advantages and disadvantages of current miniature refrigeration cooling technologies for high power microelectronics are compared. The background of the study is introduced. The latest development in microelectronics

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cooling technology and applications is presented. The objectives of this study and the organization of the thesis are outlined.

In chapter 2, a review of the studies concerning the miniature heat exchanger, the heat transfer and fluid flow of the heat exchanger is presented.

Chapter 3 mainly describes modeling of the miniature bare tube heat exchanger. A heat exchanger model based on numerous control volumes is proposed. The computer simulation technique is used to link all the control volumes for a heat exchanger with an arbitrarily complex refrigerant circuitry.

In chapter 4, the geometrical optimization algorithm has been clearly presented. The major results and findings from the geometrical optimization design are concluded.

In chapter 5, the experimental setup and the operating procedures are presented. In addition, the experimental uncertainty and data reduction are discussed.

At last, the main conclusions from the study and experiments are drawn along with some recommendations for the future work in the miniature bare tube heat exchanger coil performance study in chapter 6.

Chapter 2

Literature Review

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Literature Review

2.1 Introduction

Compact heat exchangers are becoming increasingly important elements in many industrial processes, both in their original roles as contributors to increase energy efficiency, and more recently as the basis for novel intensified unit operations, such as compact evaporator in miniature cooler for the next generation high power electronic systems cooling. Reay (2002) described the feature of compact heat exchangers and gave a general comparison picture of the area density and typical hydraulic diameters of a range of conventional and compact heat exchangers as figure 2-1.

Fig. 2-1 Heat exchanger area densities and hydraulic diameters: S&THX, shell and tube heat exchanger; PHE, plate heat exchanger; PFHE, plate-fin heat exchanger; PCHE, printed circuit heat exchanger (Reay, 2002).

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Literature Review

The feature of compact heat exchangers which sets them different from other types, most specifically the conventional shell and tube heat exchanger used in very large numbers in chillers and the larger absorption cycle heat pumps and heat transformers, is their high area density. Area density is the ratio of heat transfer surface to heat exchanger volume, with the unit of m^2/m^3 . As compared with shell-and-tube heat exchangers, the distinguishing characteristics of compact heat exchangers are as follows:

1. They are very flexible, and it is easy to apply different area densities on the hot and cold sides separately as desired.
2. Fluid must be clean and relatively non-corrosive because of small flow passages used to achieve high surface area density, and one of the fluids is typically a gas.
3. Operating pressure and temperature are limited because of construction features of brazing joints.
4. The fluid pumping power (i.e., pressure drop) is often of equal importance to the heat transfer rate.
5. The header design of a compact heat exchanger is much more difficult and is also important for uniform flow distribution.

Brooks *et al* (1999) developed a miniature combustor & evaporator that provides a lightweight and compact source of heating, cooling, or energy generation for both man-portable and stationary applications. The device used microscale flow channels that increased the available surface area for heat transfer and reduced the fluid boundary layer.

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Sawat (2000) developed an optimum design tool based on the simulated annealing technique for a compact heat exchanger. Only small diameter tubes without conventional fins were employed in the compact heat exchanger. The range of tubes diameter in the optimum design was 0.3-0.5 mm. By experimental comparison with the commercially available compact heat exchanger, the compact bare tube heat exchanger design offered a significant degree of improvement in terms of the pumping power, the heat transmission efficiency and the core volume size. With large core dimensions, the improvement of micro bare tube heat exchanger performance was laid on compact heat exchangers for automobile air-conditioning system.

Miniaturization of heat exchangers opens up the way to a considerable increase of their volumetric characteristics. Rachkoskij *et al* (1998) presented a “checkmate” design of heat exchanger that allowed combining a number of micro cross-flow heat exchange modules. The changes in heat transfer due to decreasing tube size and relative length was examined. An estimated volumetric heat transfer coefficient of $4.3 \text{ MW/m}^3\text{K}$ was obtained with a single symmetric air-air module of 50 mm^3 volume and 6 mm height at 260 Pa pressure drop, temperature difference of 5 K, pumping power comprising 4.3 % of the thermal power, and micro tube size of $128 \times 1200 \text{ }\mu\text{m}$.

Saji *et al* (2001) presented the design concept and manufacturing of a new compact laminar flow heat exchanger with stainless steel micro-tubes for helium refrigerators. A micro-tubes strip counter flow type heat exchanger that consisted of 12 elements with a total of 4800 stainless steel micro-tubes was developed. Each element was formed with 400 tubes and a newly developed vacuum brazing method was applied

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for the bonding to the side plate. Each tube had an inner diameter of 0.5 mm, an outer diameter of 0.7 mm and is 310 mm long. In aerodynamic and thermal design of the element, the laminar flow conditions were adopted for the flows of inner and outer tubes to keep a high transfer rate and a low pressure drop.

Jiang *et al* (2002) developed a microchannel heat exchanger through a numbers of multichannels with shared inlet and outlet manifolds. The design of the microchannel heat exchanger aimed to achieve the lowest possible thermal resistance with a reasonable pressure drop for a specified range of heat generation and liquid flow rate. The heat exchanger, with 40 microchannels of 100 μm hydraulic diameters, was predicted to be able to achieve a junction-fluid thermal resistance of less than 0.1 K/W and a heat exchanger capable of dissipating 100 W from a 1 cm $\times$ 1 cm thermal chip with a pressure drop of about 160 kPa. The simulations were effective but there were not enough experimental data to verify this tool for design and optimization of heat exchangers.

The plate-fin and tube geometry is a common configuration in heat exchangers. The fin spacing and fin thickness on the over-tube side of a heat exchanger play great role on the performance of heat exchanger. Through flow visualization and numerical computation, Richardo *et al* (2000) examined the influence of fin spacing on the over-side of a single-row fin-tube heat exchanger. One dimensionless parameter, the distance between fins divided by the tube diameter had been applied in the study. When this parameter was small, the flow was Hele-Shaw; as it increased, a horseshoe vortex was formed just upstream of the tube; a separation region was then developed behind the tube; this became larger and eventually communicated with the fluid

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downstream of the heat exchanger. The fins in a plate-fin and tube heat exchanger formed a series of channel that created narrow passages for the external flow and acted as extended surfaces to increase external heat transfer from the tubes. In this and other similar heat exchangers, thermal conduction through the tube wall and through the fins played a crucial role in the overall heat transfer rate. The thermal conductance of the wall was not a problem since it was usually high. Conduction in the fins, however, was a more complicated matter: on the one hand the fins were the extended surfaces which enhanced convective heat transfer to the over-tube fluid and on the other hand they increased tube-to-tube conduction. The former was desirable while the latter degraded the performance of the heat exchanger.

Richardo *et al* (1997) studied the effect of heat transfer between the tubes of a plate-fin and tube heat exchanger by conduction through the fins. In normal scale, fin conduction not only enhanced heat transfer to the over-tube fluid, but also increased tube-to-tube conduction. With a continuum model, the conduction was evaluated and the effect of parametric variation on the performance of a single-row plate-fin and tube heat exchanger was considered. Tube-to-tube conduction was found to degrade the performance of the heat exchanger. Results obtained showed that the same heat transfer area distributed in a different number of tubes affected its performance.

Heat exchangers are devices through which heat is transferred from one fluid to another. The principal heat exchangers in a vapor-compression refrigeration system are evaporator and condenser.

Chapter 2

Literature Review

In the literatures, there have been many studies made concerning the evaporator and condenser coils. The literature on the finned-tube evaporator and condenser coils is extensive in surveys published by Kays and London (1984), Pate (1988), Zukauskas *et al* (1988), Kandlikar (1990), Gidwani *et al* (1998), etc. Because of the diverse problems investigated in the present study, it is unrealistic to attempt a complete survey of the finned-tube heat exchangers. However, a review will be made of some of the commonly used methods and correlations. Special attention is given to the existing correlations for the air-side sensible and latent heat transfer coefficients, refrigerant-side condensing and evaporating heat transfer coefficients and refrigerant-side pressure drop in two-phase flow region.

2.2 Air Side

To make the heat exchanger more compact and efficient, extended surface such as continuous plate fins made of aluminum or copper are added to the tube surface on the air side. The addition of fins to the tubes greatly increases the outer surface area but at an expense of decreasing the mean temperature difference between the surface and the air stream. Thus, fin efficiency is introduced to evaluate the thermal resistance to overall heat transfer of extended surface. The fins can be smooth flat fin, wavy or louvered, usually of a rectangular plate shape. The air-side area of a finned condenser or evaporator is composed of two portions, the primary area and extended area. The primary area A_t is that of the tube between of the fins and the extended area A_f is that of the fin surface.

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For tube arrangement, the tube rows of a bank are either staggered or aligned in the direction of the air flow. The configuration is characterized by the tube diameter and by the transverse pitch S_t and longitudinal pitch S_l measured between tube centers, shown in Figure 2-2. It is noted that staggered tube arrangement is commonly used. As air flows over the tubes, due to the effect of adverse pressure gradient, boundary layer separation occurs. The position of the separation point depends on the Reynolds number.

As the airflow is dominated by boundary layer separation effect and wake interactions, the airflow in a finned-tube heat exchanger is rather complex. Little progress has been made to predict the heat transfer coefficient and pressure drop theoretically (Webb 1994). As such, the heat transfer coefficient and pressure drop are usually correlated experimentally using multiple regression technique.

The ensuing sections are a survey of the existing correlations for the air-side heat transfer coefficient and pressure drop of finned-tube heat exchangers.

2.2.1 Dry Surface Heat Transfer Coefficient

Because of the great variety of geometry used to enhance the air-side heat transfer performance in finned tube heat exchangers, any correlation to be used should include terms to adjust for effect such as the number of tube rows in the air flow direction, the fin spacing, the tube spacing and the type of fins.

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Generally, the air-side heat transfer coefficient is correlated in term of Colburn factor j , which is defined using Stanton number St and Prandtl number Pr :

$$j = St \cdot Pr^{\frac{2}{3}} \quad (2-1)$$

$$St = \frac{\alpha_a}{C_{p,a} G_{\max}} \quad (2-2)$$

$$Pr = \frac{C_{p,a} \mu}{k} \quad (2-3)$$

where $C_{p,a}$ is air specific heat at constant pressure, G is the mass velocity, α is the side heat transfer coefficient, k is the air thermal conductivity, and μ is air dynamic viscosity, and subscripts a and max denote air and maximum respectively.

Substituting Equations (2-2) and (2-3) into Equation (2-1), the air-side heat transfer coefficient α_a is given by:

$$\alpha_a = \frac{j \cdot C_p G_{\max}}{Pr^{\frac{2}{3}}} \quad (2-4a)$$

It is noted that most of the correlations were developed for the heat exchangers with smooth flat fins. Thus, a more general equation is commonly used to take into account of the effects of the number of tube rows in the air flow direction and the type of fins (e.g. Fisher et al. 1983, Liu 1996)

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$$\alpha_a = C_o C_n \cdot j_4 \cdot \frac{C_{p,a} G_{\max}}{\text{Pr}^{\frac{2}{3}}} \quad (2-4b)$$

where j_4 is the Colburn factor for a 4-row heat exchanger with smooth flat fins, C_n is a constant for the heat exchanger with a row number rather than 4, and C_o is a constant for the heat exchanger with the fins rather than smooth flat fins.

2.2.1.1 Colburn Factor

The Colburn factor is often expressed in term of air flow Reynolds number and coil geometric parameters. Elmahdy and Biggs (1979) presented the correlation based on the previously published test data of twenty different heat exchangers. The correlation takes into account of the various coil geometric parameters and air flow Reynolds number.

The Colburn factor in dry-zone for flat fin heat exchanger is expressed as:

$$j_{a,D} = C_1 (\text{Re}_{D_h})^{C_2} \quad (2-5a)$$

$$\text{Re}_{D_h} = \frac{G_{\max} D_h}{\mu} \quad (2-5b)$$

$$C_1 = 0.159 \left(\frac{D_h}{Y_f} \right)^{0.065} \left(\frac{Y_f}{F_h} \right)^{0.141} \quad (2-5c)$$

$$C_2 = -0.323 \left(\frac{Y_f}{F_h} \right)^{0.049} \left(\frac{F_h}{S_l} \right)^{0.549} \left(\frac{S_f}{Y_f} \right)^{0.077} \quad (2-5d)$$

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where D_h is the coil hydraulic diameter, F_h is the fin height, S_f is the fin spacing, S_l is the tube longitudinal spacing, Y_f is the fin thickness, and subscript D denotes dry. It is noted that the Reynolds number is based on the coil hydraulic diameter.

McQuiston (1981) presented a correlation in a graph form for a four-row flat-fin dry coil based on his experimental results and the data presented by Rich (1975). This correlation can be represented in the following equation, which is widely used for the study of the finned heat exchangers.

$$j_4 = 0.0014 + 0.2618 \cdot JP \quad (2-6a)$$

$$JP = \text{Re}_{d_o}^{-0.4} \left(\frac{A_o}{A_t} \right)^{-0.15} \quad (2-6b)$$

$$\text{Re}_{d_o} = \frac{G_{\max} d_o}{\mu} \quad (2-6c)$$

where A_o is the total surface area, and A_t is the primary surface area. It is noted that the Reynolds number is based on tube outer diameter.

Gray and Webb (1986) proposed a correlation for four-row flat-fin heat exchangers using the data from McQuiston and other published experimental data from the literature. The expression is:

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$$j_4 = 0.14 \text{Re}_{d_o}^{-0.328} \left(\frac{S_t}{S_l} \right)^{-0.502} \left(\frac{F_h}{d_o} \right)^{0.0312} \quad (2-7)$$

where Reynolds number is defined using the tube diameter as the characteristic dimension as Equation (2-6c). A comparison with the experimental data results in a root-mean-square deviation of 7.3%, compared to a root-mean-square deviation of 9.5% when the McQuiston correlation is compared to the same data. It is reported that by Webb (1994) that the McQuiston and the Gray and Webb heat transfer correlations are comparable in accuracy.

2.2.1.2 C_n Constant

The tube rows affect the air-side heat transfer. The heat transfer coefficient associated with a tube is determined by its position in the bank. The coefficient for a tube in the first row is approximately equal to that for a single tube in cross flow, whereas larger heat transfer coefficients are associated with tubes of the inner rows. The tubes of the first few rows acts as a turbulence grid, which increases the heat transfer coefficient for tubes in the following rows. However, heat transfer conditions stabilize, such that little change occurs in the convection coefficient for a tube beyond the fourth or fifth row.

The work of Rich (1975) made it possible to determine the heat transfer coefficient for coils with a number of rows other than four. An equation for this is given by McQuiston (1981):

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$$C_n = \frac{j_n}{j_4} = \frac{1 - n \cdot 1280 \cdot \text{Re}_l^{-1.2}}{1 - (4 \cdot 1280) \cdot \text{Re}_l^{-1.2}} \quad (2-8a)$$

$$\text{Re}_l = \frac{G_{\max} \cdot S_l}{\mu} \quad (2-8b)$$

where n is tube row number.

Another equation is provided by Gray and Webb (1986) for coils with less than four rows.

$$C_n = \frac{j_n}{j_4} = 0.922 \left[2.24 \text{Re}_{d_o}^{-0.092} \left(\frac{n}{4} \right)^{-0.031} \right]^{0.6074(4-n)} \quad n < 4 \quad (2-9a)$$

If the number of tube row is greater than 4, no change in the j-factor has been verified by them.

$$C_n = 1 \quad n > 4 \quad (2-9b)$$

2.2.1.3 C_0 Constant

To enhance heat transfer, complex fin configurations such as wavy or louvered are being used, which provide a better compromise among the heat transfer rate, manufacture cost and fan power consumption associated with air-side pressure drop. The smooth flat fins can be modified by metal stamping processes to form wavy fin patterns, also called corrugated or ripple fin patterns. Both sine-wavy and triangle-shaped (also called wedge-shaped) fins are commonly used. As the coil manufacture may be different among the manufacturers, the availability of general

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correlations that account for the geometric parameters is scarce. Beecher and Fagan (1987) investigated the fin geometric parameters on the wedge-shaped fins. The parameters were tube spacing, fin spacing, wave height, wave length, collar diameter and the number of tube rows. Because of the difficulties with correlating all geometric parameters into a single equation for wavy fins, a series of equations were used to calculate Nusselt numbers. However, Webb (1990) developed a general correlation for wavy-fin coils using the data obtained by Beecher and Fagan. Since the data and the relevant correlation are presented in term of Nusselt number based on the arithmetic mean temperature difference, the application is somewhat restricted.

Due to the complexity of the fluid flow and heat transfer characteristics in wavy and louvered fins, limited experimental data are available in the literature. Fisher and Rice (1981) introduced a constant C_o to account for increase in the heat transfer coefficient with the use of wavy or louvered fins. The value of C_o equals 1.0, 1.45 or 1.75, depending on whether the fins are smooth, wavy or louvered respectively.

Louvered fins (also referred to as strip fins, slot fins, and offset strip fins) can achieve a higher heat transfer performance than either smooth flat or wavy fins. The trade-off, however, is higher pressure drop and the potential contamination by foreign matter.

It is noted that the general approach in correlating air-side heat transfer coefficient depends on the data collection, the formulation of a correlation and its validation with experimental data to see the deviation. The possible uncertainty associated with each of these steps should be considered when applying the correlations. Firstly, there are

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experimental errors associated with the collections of the original data. Secondly, additional errors may be introduced in data reduction such as in the fin efficiency calculation or in the separation of latent heat transfer and sensible heat transfer. Finally, the fin geometry and the coil manufacture process may be different among manufacturers. These uncertainties may introduce errors for the application of the correlations. Therefore, it is necessary to refer to the respective original documents to get the confidence for the applicability of the correlations.

2.2.2 Wet-Surface Heat Transfer

As the surface temperature of the evaporator coil is below the air dew point, heat and mass occurs simultaneously with the condensation of moisture. The effect of air moisture condensation is to increase in the heat transfer. The ensuing paragraphs are a survey of the problems associated with the simultaneous heat and mass transfer.

2.2.2.1 Single-potential and dual-potential methods

Since condensation is not dependent upon temperature difference alone, an effective method is required to account for the simultaneous heat and mass transfer. Generally, the heat and mass analogy is employed based on Lewis number Le . The mass transfer coefficient in the wet zone is calculated from the sensible heat transfer coefficient (Webb 1991).

$$\alpha_m = \frac{\alpha_a}{C_{p,a} \cdot Le^{1-n}} \quad (2-10)$$

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where α_m is mass transfer coefficient. Based on experimental evidence, the exponent $n = 1/3$ is typically employed (Webb 1991).

There are two approaches to determine the total heat transfer rate for simultaneous heat and mass transfer. One approach uses an enthalpy difference as the sole driving force (Threlkeld 1970, Elmahdy 1975).

$$dQ \cong \frac{\alpha_{a,sen} \cdot dA}{C_{p,a}} (h_a - h_{s,w}) \quad (2-11)$$

where h is enthalpy, Q is the heat transfer rate, subscripts s , sen and w denote saturated, sensible and water, and $h_{s,w}$ the enthalpy of the saturated air at the water surface temperature.

This method is sometimes referred as a single-potential method. The disadvantage of the single-potential method is that the heat and mass transfer coefficients cannot be determined separately while analyzing experimental data.

Another approach is a dual-potential method which calculates the heat and mass transfer separately. The two driving forces are temperature difference (which drives the sensible heat transfer) and specific humidity difference (which drives the latent heat transfer).

$$dQ = \frac{\alpha_{a,sen} \cdot dA}{C_{p,a}} \left[C_{p,a} (T_a - T_{s,w}) + \frac{i_{fg} \cdot (W_a - W_{s,w})}{Le^{1-n}} \right] \quad (2-12)$$

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where i_{fg} is latent heat of condensation of moisture, T temperature, and W air specific humidity.

Theoretically, Equation (2-11) can be taken as a simplified form of Equation (2-12). However, for a finned-tube heat exchanger, both the single-potential and the dual-potential methods require different fin efficiency calculation methods (Threlkeld 1970, McQuinston 1975). Furthermore, parameters h_a , T_a and W_a in the above equations are referred to the air average properties which require different iterative procedures to obtain for these two methods (Elmahdy 1975, Oskarsson et al. 1990a, Hill and Jeter 1991). These additional factors commonly exaggerate the difference between the single-potential and dual-potential methods.

In deducing heat transfer coefficient to the primary surface from experimental data in coil tests, Mirth and Ramadhyani (1993b) showed that the deduced Nusselt numbers are strongly depend upon which heat and mass transfer method is used, with a difference in Nusselt number values up to 25%. Another aspect is in the heat exchanger performance prediction. The prediction results on heat transfer rate are also different using single-potential method and dual-potential method by 1% to 2% (Mirth and Ramadhyani 1993b).

Besides adding a latent load to the coil by the mass transfer, the moisture condensation on the coil surface results in some effects on the sensible heat transfer,

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i.e. the thermal resistance of water film, the change of the heat transfer area and the heat transfer enhancement due to the roughness of the condensate surface.

2.2.2.2 Thermal resistance of condensate

The thermal resistance of the condensate depends on the thickness. The prediction of the falling film thickness for a finned dehumidifying coil is complex due to the complexity of the air flow pattern and the film flow pattern. It is dominated by the gravity, wall shear force, and the surface share force exerted by the airflow.

Luxton and Shaw (1991) showed from a direct observation that the condensation process, at face velocities below 2.1 m/s at least, is drop-wise rather film-wise. Due to the difficulty of identifying the thickness of the condensate on the coil surface, a uniform film thickness of 0.1 mm was assumed by Elmahdy (1975) despite of the different coil operating conditions.

2.2.2.3 Heat Transfer Coefficient

Elmahdy (1975) reported based on his experimental data that the average j factor of the wet surface coil was higher than that of a dry surface at the same airflow Reynolds number. It was explained that the surface of the water film would not be perfectly smooth, even if complete film-wise condensation occurred on the surface. Furthermore, due to the occurrence of the combination of drop-wise and film-wise condensation, the rougher surface would probably promote additional turbulence in the air stream, which resulted in an increase in sensible heat transfer coefficient and hence the average j factor.

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$$j_{a,w} = j_{a,D}(C_j) \quad (2-13a)$$

$$C_j = 1.425 - 0.51 \left(\frac{\text{Re}_{D_h}}{1000} \right) + 0.263 \left(\frac{\text{Re}_{D_h}}{1000} \right)^2 \quad (2-13b)$$

where C_j is a factor taking account of the effect of dehumidification, and subscript w denotes wet.

McQuiston (1978) noted that there were two different condensation forms: film condensation and drop-wise condensation. The j factors for film and drop condensation were in close agreement with a tendency for drop-wise to be somewhat larger. For a wet coil, Equation (2-6a) is modified by introducing a parameter JS :

$$j_{4,w} = 0.0014 + 0.2618 \cdot JP \cdot JS \quad (2-14a)$$

in which

$$JS = \left(0.95 + 0.4 \times 10^{-5} \text{Re}_l^{1.25} \right) \cdot Fs \quad (2-14b)$$

$$Fs = \frac{S_f}{S_f - Y_f} \quad (2-14b)$$

Mirth *et al* (1993a) compared the experimental data under both dry and wet conditions and concluded that wet-surface heat transfer performance may be accurately predicted using dry-surface heat transfer correlations. The heat transfer predictions using dry-surface Nusselt number were with $\pm 5\%$ of the wet-surface experimental results. Considering the insensitivity of the heat transfer prediction to the variation in air-side heat transfer correlation, the results of Mirth *et al* (1993a) were acceptable. In general,

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the net effect of the condensate formation on the sensible heat transfer is related to the heat exchanger geometry and air-side Reynolds number.

In coil modeling, due to the lack of sensitivity of the heat transfer prediction to the variation in air-side heat transfer correlation, whether using the wet j factor correlations (Elmahdy 1975, McQuiston 1981, Turaga 1988) or using the dry j factor correlations to predict the wet coil heat transfer may not be critical. However, further study may be required to obtain the information of the film thickness distribution and its effect on the heat transfer in order for the better understanding of the coil performance in wet conditions.

2.2.3 Pressure Drop

For finned tube heat exchangers, air-side pressure drop associated with flow across a tube bank has little effect on the overall heat transfer rate. The determination of air-side pressure drop is required to estimate the fan power consumption which is an important performance parameter for the heat exchanger. The pressure drop correlations for the dry and wet coils with smooth flat fins have been proposed by McQuiston (1981) and Gray and Webb (1986). The McQuiston friction correlation gives an error range from +167% to -21% when compared with experimental data (Webb 1994). Gray and Webb friction correlation assumes that the pressure drop is composed of two terms. The first term accounts for the drag force on the fins, and the second term accounts for the drag force on the tubes. It is reported that the correlation correlates 95% of the data for 19 heat exchangers within $\pm 13\%$. However, it is noted that the calculation of air-side pressure drop using Gray and Webb calculation is complex.

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Turaga *et al* (1988) proposed a pressure drop factor correlation for dry and wet coils with smooth flat fins. The correlation is expressed as:

$$f_{a,D} = 0.589 \left(\frac{A_o}{A_p} \right)^{-0.28} (\text{Re}_{D_h})^{-0.27} \quad (2-15a)$$

$$f_{a,w} = 0.318 (f_{a,D})^{-0.94} \left(\frac{S_f}{Y_f} \right)^{1.15} (\text{Re}_{D_h})^{-0.92} \quad (2-15b)$$

where f is friction factor.

Comparison of the correlation with McQuiston friction correlation shows that Turaga *et al.* friction correlation gives a better prediction (Turaga *et al* 1988).

As the correlations available for coil air-side pressure drops are not widely verified or well correlated, pressure drops for wavy and louver fins are calculated by multiplying the friction factor for smooth fat fins with a correction constant. The correction constant is the same as for the calculation of heat transfer coefficient (Fisher and Rice 1983).

2.3 Refrigerant Side

For evaporators, there are two heat transfer zones inside the tubes and they are two-phase evaporation zone and single-phase superheating zone. For condensers,

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there are three heat transfer zones and they are single-phase de-superheating, two-phase condensation and single-phase subcooling zone.

2.3.1 Single Phase

In the case of single-phase flow for both subcooled and superheated refrigerant, Dittus-Boelter equation is commonly used to compute the heat transfer coefficient.

$$Nu = 0.023 Re^{0.8} Pr^n \quad (2-16)$$

in which

$$Re = \frac{G_r d_i}{\mu_s} \quad (2-17)$$

$$Pr = \frac{C_{p,s} \mu_s}{k_s} \quad (2-18)$$

where Nu is Nusselt number, subscripts i and s denote inner and single-phase respectively, and n equals to 0.4 and 0.3 for cooling and heating respectively.

Another working correlation for single-phase heat transfer of turbulent flow in tubes was presented by Petukhov (1970).

$$Nu = \frac{(f/8) Re \cdot Pr}{1.07 + 1.27(f/8)^{0.5} (Pr-1)^{0.67}} \quad (2-19a)$$

where

$$f = (1.82 \log_{10} Re - 1.64)^{-2} \quad (2-19b)$$

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Comparison of Dittus-Boelter correlation with Petukhov correlation for working fluids R12 and R134a shows that the difference between the predicted heat transfer coefficients using the two correlations is within 10% (Eckels and Pate 1990).

2.3.2 In-tube Refrigerant Condensation

In most condenser coils, film condensation occurs inside tubes, and liquid condensate covers the condensing surface with a continuous film. The rate of heat transfer depends on condensate film thickness, rate of condensate removal and system pressure. The film flow is usually laminar when condensation takes place on the horizontal tubes. Several correlations have been successfully used to predict in-tube condensation for refrigerants (e.g. Pate 1988, Eckels and Pate 1990, Judge and Radermacher 1997). The correlation proposed by Travis *et al* (1973) for condensing heat transfer coefficient was originally based on an analytical derivation, assuming annular flow in a tube with the von Karman universal velocity distribution. Experimental data obtained for R12 and R22 were found to be in good agreement with the analytical prediction. The correlation for local condensation heat transfer coefficient is used as follows:

$$\alpha_{r,TP} = k_l \text{Pr}_l \text{Re}_l^{0.9} \frac{F(X_{tt})}{d_i F_2} \quad F(X_{tt}) < 1 \quad (2-20a)$$

$$\alpha_{r,TP} = k_l \text{Pr}_l \text{Re}_l^{0.9} \frac{F(X_{tt})^{1.15}}{d_i F_2} \quad 1 \leq F(X_{tt}) \leq 15 \quad (2-20b)$$

in which

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_v} \right)^{0.1} \quad (2-20c)$$

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$$F(X_{tt}) = 0.15 \left(\frac{1}{X_{tt}} + 2.85 X_{tt}^{-0.476} \right) \quad (2-20d)$$

$$F_2 = 0.707 \text{Pr}_l \text{Re}_l^{0.5} \quad \text{Re}_l < 50 \quad (2-20e)$$

$$F_2 = 5.0 \text{Pr}_l + 5.0 \ln[1.0 + \text{Pr}_l (0.09636 \text{Re}_l^{0.585} - 1.0)] \quad 50 \leq \text{Re}_l \leq 1125 \quad (2-20f)$$

$$F_2 = 5.0 \text{Pr}_l + 5.0 \ln[1.0 + 5.0 \text{Pr}_l] + 2.5 \ln[0.00313 \text{Re}_l^{0.812}] \quad \text{Re}_l > 1125 \quad (2-20g)$$

$$\text{Re}_l = \frac{G_r(1-x)d_i}{\mu_l} \quad (2-20h)$$

where ρ is density, x is vapor quality, and subscripts l , r , TP and v denote liquid, refrigerant, two-phase and vapor respectively.

Another general empirical correlation of the condensing heat transfer coefficient inside tubes is given by Cavallini and Zecchin (1974). Data for several refrigerants, including R11, R12, R21, R22, R113, and R114, were used to derived and verify the correlation. The form is very similar to the single-phase Dittus-Boelter equation. The use of this correlation is recommended for the equivalent Reynolds number in the range from 7000 to 53000.

$$Nu = 0.05 \text{Re}_{eq}^{0.8} \text{Pr}^{0.23} \quad (2-21a)$$

where

$$\text{Re}_{eq} = \text{Re}_l + \text{Re}_v \left(\frac{\mu_v}{\mu_l} \right) \left(\frac{\rho_l}{\rho_v} \right)^{0.5} \quad (2-21b)$$

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The correlation in term of α_l by Shah (1979) was developed from a large group of fluids. The applicability of this correlation is recommended for the liquid Reynolds number in the range from 350 to 35000.

$$\alpha_{r,TP} = \alpha_l \left(1 + \frac{3.8}{Z^{0.95}} \right) \quad (2-22a)$$

where

$$\alpha_l = 0.023 \text{Re}^{0.8} \text{Pr}^{0.4} \quad (2-22b)$$

$$Z = \left(\frac{1-x}{x} \right)^{0.8} \text{Pr}^{0.4} \quad (2-22c)$$

Comparison of the above three condensing heat transfer correlations for R12 and R134a was reported by Eckels and Pate (1990). The results show that the predicted condensing heat transfer coefficient are fairly different when using the above three correlations. Under the comparison conditions, for R12 at low qualities, Cavallini-Zecchin correlation predicts about 40% higher than the Shah correlation at lower qualities and while at high qualities, it is about 27% higher. At low qualities, Traviss et al. correlation compares well with Shah Correlation with a difference within 10%, while at high qualities, the difference can be as much as 59%. The results show that Shah Correlation gives the lowest heat transfer coefficient, whereas Cavallini-Zecchin and Traviss et al. correlations give comparable predictions.

The correlations discussed above are applicable to the annular flow pattern and they have been widely tested for various halocarbon refrigerants such as R22 and R12. However, the above correlations are not widely verified for the alternative refrigerants such as R134a in a condenser coil. In addition, due to the higher condensation thermal

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resistance as compared to an evaporator, the selection of the condensing heat transfer correlation is very critical in condenser coil modeling.

2.3.3 In-Tube Refrigerant Evaporation

It seems that the development of heat transfer correlations for flow boiling inside tubes has enjoyed significant progress in recent years. Darabi *et al* (1995) made a comprehensive review of the available correlations for the prediction of flow boiling heat transfer in tubes. Four different kinds of models are used in the investigation of flow boiling. These are:

- 1) models based on dimensional analysis (Shah 1976, 1982),
- 2) models assuming that the heat transfer is additive of nucleate boiling and convective evaporation contribution (Chen 1966, Bjorge 1982, Gungor 1986),
- 3) models that utilize the greater of the nucleate and convective components with a boiling number (Klimenko 1988, 1990, Kandlikar 1990), and
- 4) asymptotical models that are based on the power-type addition of two boiling components (Liu 1991, Steiner 1992).

A comparison of the correlations was done by Darabi *et al* against the experimental data. It was noted that Kandlikar correlation overpredicted the heat transfer coefficient and Shah (1982) and Liu-Winterton (1991) correlations underpredicted the heat transfer coefficient.

In the following paragraphs, a commonly used correlation is discussed. Jung *et al.* based on Chen's supposition (1966) that the contributions of nucleate boiling and convective heat transfer could be superimposed, presented a correlation for predicting two-phase coefficient for various refrigerants.

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$$\alpha = \alpha_{nbc} + \alpha_{cec} = N \cdot \alpha_{sa} + F \cdot \alpha_{lo} \quad (2-23a)$$

where N , α_{sa} , F , and α_{lo} are, respectively, a boiling suppression factor, and a pool boiling heat transfer coefficient, a two-phase enhancement factor, and a single-phase heat transfer coefficient for liquid-only flow, as indicated by the $(1-x)$ term in α_{lo} obtained by the Dittus and Boelter correlation.

$$N = 4048 X_{tt}^{1.22} Bo^{1.13} \quad X_{tt} \leq 1 \quad (2-23b)$$

$$N = 2.0 - 0.1 X_{tt}^{-0.28} Bo^{-0.33} \quad 1 < X_{tt} \leq 5 \quad (2-23c)$$

$$Bo = \frac{q}{G \cdot i_{fg}} \quad (2-23d)$$

$$\alpha_{sa} = 207 \frac{k_l}{bd} \left(\frac{qbd}{k_l T_s} \right)^{0.745} \left(\frac{\rho_v}{\rho_l} \right)^{0.581} Pr_l^{0.533} \quad (2-23e)$$

$$bd = 0.0146 \beta [2\sigma / (g(\rho_l - \rho_v))]^{-0.5} \quad \text{contact angle } \beta = 35^\circ \quad (2-23f)$$

$$F = 2.37 \left(0.29 + \frac{1}{X_{tt}} \right)^{0.85} \quad (2-23g)$$

$$\alpha_{lo} = 0.023 \frac{k_l}{d} \left[\frac{G(1-x)d}{\mu_l} \right]^{0.8} \left(\frac{C_{p,l} \mu_l}{k_l} \right)^{0.4} \quad (2-23h)$$

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where q is heat flux, i_{fg} is latent heat of evaporation, X_{tt} is the Lockhart-Martinelli number, and σ is surface tension of refrigerant.

This correlation was developed based on the experimental data of refrigerants R22, R12, R152a, and R114. The correlation is further validated by comparing it with experimental data of refrigerants R11 and R134a (Jung and Radermacher 1991). The mean deviation is less than 7% for heat transfer of six pure refrigerants. Up to now, Jung's correlation may be the most suitable correlation for computing the refrigerant convective boiling heat transfer. Judge and Radermacher (1997) verified the correlation in evaporator modeling with a comparison of the model predictions with experimental results. A good agreement was obtained.

The correlations mentioned above are suitable for the oil-free refrigerant. Chaddock (1989) reported the measurements of local evaporative heat transfer coefficient for R22, R502, and R717 with and without a small fraction of oil present. The refrigerant-oil combinations and the temperature ranges used produced three different mixture characteristics: complete miscibility, partial miscibility and immiscibility. These differing characteristics led to conditions of enhancements of the local heat transfer coefficient for complete miscibility and to severe decreases in the performance for immiscibility and low evaporating temperature.

2.3.4 Pressure Drop on the Refrigerant Side

The pressure drop correlations on the refrigerant side are used for single-phase and two-phase flows according to the flow conditions of the coils. The pressure drop on the refrigerant side is attributed to three separate components, namely component due

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to the friction effect at the tube wall ΔP_f , component due to acceleration for flow boiling in the evaporator or deceleration for flow condensation in the condenser ΔP_m , and component due to return bends ΔP_{rb} .

$$\Delta P = \Delta P_f + \Delta P_m + \Delta P_{rb} \quad (2-24)$$

2.3.4.1 Single-Phase Flow

For single-phase flow, the friction effect is computed by means of Darcy-Weissbach formula (Paliwoda 1989):

$$\Delta P_{f,s} = \lambda \frac{L}{2d_i} \frac{G_r^2}{\rho_s} \quad (2-25)$$

where L is tube length, λ is friction coefficient, and subscript f denotes friction.

The friction coefficient λ is calculated according to the Reynolds number:

$$\lambda = \frac{64}{\text{Re}} \quad \text{Re} \leq 1187 \quad (2-26a)$$

$$\lambda = \frac{0.3164}{\text{Re}^{0.25}} \quad \text{Re} > 1187 \quad (2-26b)$$

The momentum effect is often neglected for single-phase flow. Pressure drop at the return bends $\Delta P_{rb,s}$ is computed using a local pressure drop factor Ω_s :

$$\Delta P_{rb,s} = \Omega_s \cdot \frac{G_r^2}{\rho_s} \quad (2-27)$$

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2.3.4.2 Two-Phase Flow

Frictional Loss

For two-phase flow, the frictional loss is often calculated as a single-phase flow by means of a dimensionless two-phase multiplier. Elision et al. (1981) summarized the Lockhart-Martinelli method as follows:

$$\Delta P_{f,TP} = \phi_l^2 \cdot \Delta P_{film} \quad (2-28a)$$

where

$$\phi_l = 1.467 - 0.51346 \ln(X_{tt}) + 0.048789 [\ln(X_{tt})]^2 \quad (2-28b)$$

$$\Delta P_{film} = 2 \left(0.049 \text{Re}_{film}^{-0.2} \right) \frac{L}{d_i} \frac{G^2}{\rho_m} \quad (2-28c)$$

Soliman (1968) developed an empirical correlation for pressure drop in two-phase flow region. The correlation is expressed as:

$$\Delta P_{f,TP} = \phi_v^2 \cdot \Delta P_{v,f} \quad (2-29a)$$

where

$$\phi_v = 1 + 2.82 X_{tt}^{0.525} \quad (2-29b)$$

$$\Delta P_{v,f} = \lambda \frac{L(xG)^2}{2d \cdot \rho_v} \quad (2-29c)$$

Paliwoda (1989), based a statistical analysis, proposed a general method for the pressure drop with boiling and condensing refrigerants within the entire saturation zone. The two-phase frictional loss is calculated as:

$$\Delta P_{f,TP} = \left[\Delta P_{l,f} + 2x(\Delta P_{v,f} - \Delta P_{l,f}) \right] \cdot (1-x)^{1/3} + x^3 \cdot \Delta P_{v,f} \quad (2-30a)$$

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in which

$$\Delta P_{v,f} = \lambda \frac{L}{2d} \cdot \frac{G^2}{\rho_v} \quad (2-30b)$$

$$\Delta P_{l,f} = \lambda \frac{L}{2d} \cdot \frac{G^2}{\rho_l} \quad (2-30c)$$

Momentum Effect

The momentum component ΔP_m can reach significant values with fully developed boiling or condensing refrigerants at a high density of heat flux on the external heat transfer area. In such cases, the value of ΔP_m can approach 15% of the total pressure gradient (Paliwoda 1989). At typical operational conditions of refrigeration systems, the error resulting from neglecting ΔP_m usually does not exceed 5%.

For the two-phase flow region, ΔP_m can be computed by (Liu 1996):

$$\Delta P_m = G_r^2 (v_2 - v_1) \quad (2-31)$$

where v is specific volume, and subscripts 1 and 2 denote inlet and outlet of the two-phase flow region.

Return Bends

For the local loss at the return bends in two-phase flow region, Ellison et al. (1981) recommended the method by Pierre (1964). By equation fitting, the local pressure loss factor Ω_{TP} is evaluated by the following equation.

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$$\Delta P_{rb,TP} = \Omega_{TP} \cdot \frac{G_r^2}{2\rho_m} \quad (2-32a)$$

$$\Omega_{TP} = 0.696 + 0.2216x + 2.3411x^2 - 2.662x^3 \quad (2-32b)$$

where subscript m denotes mean.

Paliwoda (1992) recommended a two-phase multiplier for the local resistance coefficient for the return bend pressure drop.

$$\Delta P_{rb,TP} = \frac{G_r^2}{2\rho_v} \cdot \Omega_s \cdot \beta \quad (2-33a)$$

$$\beta = \left\{ \left(\frac{v_l}{v_v} \right) \left(\frac{\mu_l}{\mu_v} \right)^{0.25} + 3 \left[1 - \left(\frac{v_l}{v_v} \right) \left(\frac{\mu_l}{\mu_v} \right)^{0.25} \right] \cdot x \right\} \cdot (1-x)^{0.333} + x^{2.276} \quad (2-33b)$$

where Ω_s is local pressure drop factor for single-phase flow.

In heat exchanger modeling, since the pressure drop has an effect on the condensing and evaporating temperature, the computation of the refrigerant side pressure drop is more critical than that of the air side. Perhaps the best approach for the selection of the correlations is to test them against the experimental data.

2.3.5 Void Fraction

In heat exchanger performance study, the analysis of the refrigerant inventory of the coils is of considerable importance mainly for two reasons. Firstly, the heat exchanger performance is determined, to some degree, by the amount of the refrigerant charge in the system. Finding the charge which gives good operation at both design and

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off-design conditions is valuable in the search of an optimal operating condition for the heat exchanger itself and the related system. Secondly, in simulating the coil transient performance, the refrigerant mass distribution along the coil as the initial condition of the problem is essentially required. The main difficulty in inventory analysis lies on the prediction of the refrigerant mass in the two-phase regions of the condenser and evaporator coils, as the calculation of the refrigerant mass in single-phase regions is straightforward. In order for the prediction of refrigerant inventory, the void fraction throughout the two-phase region is required, which can be seen from the following equation.

$$M_r = M_{r,v} + M_{r,l} = A \int_0^L [\alpha \rho_v + (1 - \alpha) \rho_l] dz \quad (2-34)$$

where M is mass, and α is void fraction.

Therefore, the inventory problem is closely related to the prediction of void fraction, which is generally achieved by the use of a void fraction correlation.

Rice (1987) presents a comprehensive survey on the void fraction correlations. The void fraction correlations were compared for the mass inventory predictions for condensers and evaporators. In his conclusion, the correlations of Hugmark(1962), Premoli (1971), Tandon (1985) and Baroczy (1966) were found to give the highest predictions and closest agreement to measured total system charge. Damasceeno *et al* (1991) compared test data with the refrigerant inventory prediction of a computer heat pump model. Several void fraction correlations were incorporated for the predictions. Hugmark's correlation led to a good agreement between measured and predicted dependence of heat pump capacity on charge. Farzad and O'neal (1994) used eight

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void fraction correlations to study the impact on the estimation of system variables. Results indicated that the Hughmark's correlation appeared to provide the best comparison to the measured data. Here, the Hughmark's correlation is given in the following equations.

$$\alpha = K_H \cdot \beta = \frac{K_H}{1 + \left(\frac{1-x}{x} \right) \cdot \frac{\rho_v}{\rho_l}} \quad (2-35a)$$

$$\beta = \frac{1}{1 + \left(\frac{1-x}{x} \right) \cdot \frac{\rho_v}{\rho_l}} \quad (2-35b)$$

$$K_H = 0.20622 - 0.30545 \ln Z + 1.08868 (\ln Z)^2 - 0.71459 (\ln Z)^3 + 0.20883 (\ln Z)^4 - 0.02878 (\ln Z)^5 + 0.00153 (\ln Z)^6 \quad (2-35c)$$

$$Z = \left[\frac{d_i G}{\mu_l + \alpha(\mu_v - \mu_l)} \right]^{\frac{1}{6}} \left[\frac{1}{g d_i} \left(\frac{G x}{\rho_v \beta (1 - \beta)} \right)^2 \right]^{\frac{1}{8}} \quad (2-35d)$$

Figure 2.2 A plate-fin-tube heat exchanger

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Utilizing refrigeration may provide the only means by which future high-performance electronic chips can be maintained below predicted maximum temperature limits. Widespread application of refrigeration in electronic packaging will remain limited, until the refrigerators can be made sufficiently small so that they can be easily incorporated within the packaging.

Vapor compression refrigeration offers several important advantages. These include low mass flow rate, high COP, low cold plate temperature and the ability to transport heat away from its source. The following is a more detailed look at vapor compression refrigeration of high performance electronics.

Figure 3-1 is a typical schematic representation of the vapor compression cycle. In the refrigeration system, heat is introduced to the system by a miniature evaporator, i.e., the miniature heat exchanger. This heat vaporizes liquid refrigerant in the evaporative cold plate. This vapor is subsequently carried through the suction tube to the compressor. Work is supplied to compress the warm vapor into a hot, high-pressure vapor that is passed to the miniature condenser.

The hot high-pressure vapor releases its heat to the air stream across the condenser as it is condensed into a warm liquid. Warm liquid is pumped from the bottom of the

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condenser through an expansion device where its pressure and temperature drop significantly, creating the refrigeration effect. The cycle completes as the cold fluid passes to the cold plate.

Reliable vapor compression driven cooling subsystems can be designed and manufactured for use in high performance electronic applications. Vapor compression cooling can be used for thermal management and performance enhancement. Today a small fraction of all computers are equipped with vapor compression coolers. Broader use of this powerful cooling technology depends on several factors. First, the cooling technology will evolve to become a better fit for computer and telecommunication applications. Second, programs are underway to reduce the size and weight of vapor compressors and to build-in capacity control features that can interface seamlessly with computing. The individual components of the refrigeration system will first be developed and these components will be optimized and linked to form the complete miniature refrigeration cycle. Among the components of a refrigeration system, compressor plays the most critical role. In this project, a newly and simple approach has been studied and used in the new refrigeration system. In this chapter, the mathematical and numerical models of the miniature bare tube heat exchanger have been developed.

3.1 Modeling of Miniature Bare Tube Heat Exchanger

Existing Coil Simulation Models

In the study of evaporator and condenser coils, numerical modeling approaches are commonly used. The numerical models can be classified into three categories:

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- 1) Two-zone or three-zone models,
- 2) Tube-by-tube models, and
- 3) Distributed models.

In evaporator modeling, the two-zone models divide the evaporator into a two-phase flow region and a superheated single-phase flow region. While in condenser modeling, the three zone models divide the condenser into a superheated single-phase flow region, a two-phase flow region and a subcooled single-phase flow region.

Tube-by-tube models involve dividing the heat exchanger into a number of straight tubes and performing computation on each tube.

Distributed models involve dividing each tube of the heat exchanger into a number of control volumes and performing heat transfer and mass balance analysis on each control volume.

What kind of modeling methodology is used in the study of coils mainly depends on the compromise one is willing to make between the accuracy (or flexibility) and complexity of the model to be used. Some of the most widely accepted models are summarized below.

3.1.1 Two-Zone or Three-zone Models

Goldstein (1983) presented a detailed two-zone steady-state model for a direct-expansion evaporator; taking into account the refrigerant two-phase flow

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characteristic and the mass transfer effect on the air side. The model assumed that latent heat transfer occurs at the two-phase region of coil and moisture removal in the superheated portion of the coil was neglected. The model used the log mean enthalpy difference as the sole driving potential to calculate the heat and mass transfer to the coil surface. The convergence logic was the matching of air-side and refrigerant-side capacities. Predictions of evaporator performance to within $\pm 10\%$ of actual results were reported.

Fischer and Rice (1983) presented a comprehensive heat pump model, which has been widely used as a standard reference tool in heating, ventilation and air-conditioning industry. In modeling, the condenser was divided into three regions and evaporator was divided into two-regions. In the two-phase flow region, the two-phase heat transfer coefficient was obtained by integrating local values of concerned correlation over the two-phase region. The integration was performed over the range of refrigerant quality by assuming that the temperature difference of heat transfer remained constant along the two-phase region. The fractions of the coil for superheated and subcooled regions were determined by dividing the partial air mass flow rate necessary to achieve the heat transfer rate of concerned region by the total air mass flow rate. The partial air mass flow rate was obtained by using ζ -NTU method. The model took into account of the local variation of heat transfer coefficient, while its computation time was largely shortened in comparison with the distributed parameter method. The shortcoming of this model was due to the assumption that the temperature difference of two-phase heat transfer was constant and the model is not suitable for the coils with complex tube circuitry.

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Kempiak and Crawford (1992) developed a three-zone condenser model for a mobile air-conditioning system simulation. The condenser was divided to three sections: the de-superheating, condensing and subcooling regimes. The overall heat transfer coefficient for each of the three sections was expressed in term of the refrigerant mass flow rate and the volumetric air flow rate by analyzing the existing heat transfer correlations of the refrigerant side. Based upon the energy equations and the fact that the sum of the fractions of the heat exchanger in each of the three sections should equal to one, a final equation for the whole heat exchanger was developed. The experimental data were used to determine the values of the constants in the equation through the least-square analysis. Predictions of condenser performance within 2% of the overall heat transfer rates and 20% of the overall pressure drop were reported. It is noted the model was developed based on experimental data. Hence it was an empirical simulation model.

3.1.2 Tube-by-Tube Models

Ellison *et al* (1981) presented a rigorous computer model for air-cooled condensers with specified circuiting. Intended for use in detailed design analysis of heat exchangers, the model relies on a tube-by-tube computational approach. The thermal and fluid-flow performances of each tube (the length of tube between two return bends) were computed individually, using local temperature and heat transfer coefficients. In order to accommodate complex refrigerant circuiting, the program traced the flow of refrigerant in each circuit, taking proper account of the joining or branching of parallel circuits. Coils of different configuration were tested in the laboratory. The comparison between the predicted and the measured wall temperature

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at the return bends showed that the model was reasonably good. However, no detailed information on the computation was given when the separated points of two different heat transfer regions occurred between two return bends of the tubes. There is no return bends in the miniature heat exchanger coil of this study.

Domanski (1991) also used the tube-by-tube computation approach while developing an evaporator model to study the effect of nonuniform air distribution on the performance of the heat exchanger. The results conformed reasonably well to the experimental data.

3.1.3 Distributed Models

Anand and Tree (1982) modeled a single tube-finned condenser using distributed-parameter method with the following assumptions:

- 1) the air-side heat transfer is constant,
- 2) the two-phase region is a homogeneous mixture of liquid and vapor,
- 3) the subcooled properties of the refrigerant is approximated to saturated properties, and
- 4) conduction heat transfer in the axial direction of the tube and kinetic energy associated with the flow of refrigerant are neglected.

The model took into consideration the variations of fluid properties, friction factor and heat transfer coefficient due to the change of phase during condensation. The laws of conservation were applied to the refrigerant, tube and air control volumes and the resultant matrix was solved using a finite difference method. Results of simulation

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were not validated with experiments. As the condenser was simplified as a single finned tube, this limited the applications of the model to the complex tube circuitry arrangements.

Judge and Radermacher (1997) developed a heat exchanger model for transient and steady state performance simulations of mixtures and pure refrigerants. Since the heat exchanger simulation would be applied to cycle simulation, the main concern was to predict the refrigerant outlet conditions given the refrigerant and air inlet conditions. With this in mind, an inclined multi-circuit heat exchanger was treated as one tube in cross flow relative to the air. The governing equations of momentum, continuity, species and energy equations were solved and the steady state results were verified experimentally in residential heat pump systems.

Oskarsson *et al* (1990) presented three models for evaporators, namely a finite element model, a three-zone model and a parametric model. The proposed models simulated the finned coils operating in dry, wet and frosted conditions. A finite element model was used to study the local behavior of heat transfer of fluids and the effect of refrigerant-side pressure drop on heat transfer performance. The method used is identical to represent the whole evaporator, this model is not suitable row of the multi-row was selected to represent the whole evaporator; this model is not suitable for the evaporators having complex circuits. Three-region model divided an evaporator into three regions: two-phase region, the transient region and the superheated region, assuming that the pressure drop on the refrigerant side can be neglected and the superheated region was dry. The procedure of computation was

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similar to the finite element model. Except that the three regions were computed separately. The parametric model introduced three empirical equations for three parameters: the coil characteristic, the coil enthalpy effectiveness, and the coil temperature effectiveness. The leaving air conditions as well as the heat transfer were determined by solving the empirical equations. As the empirical equations were developed by a regression analysis using the computation results of the three-region model based on a given coil, the application of the model is strictly limited to the given coil. The author didn't compare the finite element model with experimental results. However, the comparisons of three-region model with finite element model showed that both models were in good agreement. Three region model and parametric model were verified by experimental results.

It is noted that the model proposed by Ellison *et al* (1981) is capable of coping with the heat exchanger with complex refrigerant circuitry. It is a tube-by-tube model, whose accuracy may be low compared to the distributed models while the number of the straight tubes in a refrigerant path is small. However, the existing distributed models all simplify the coils and do not take into account the effect of refrigerant circuitry. In addition, it is noted that the efficiency of a wet fin where simultaneous heat and mass transfer occurs has not been looked into a very detail by the existing models. To increase the model capability, more attention may be needed to the wet fin performance in the evaporator coil modeling.

What kind of modeling methodology is used in the study of coils mainly depends on the compromise one is willing to make between the accuracy (or flexibility) and complexity of the model to be used.

3.2 Modeling Development

A numerical model for studying heat exchanger design and performance has been developed. The model is intended to provide a complete insight into the local air and refrigerant conditions as well as the overall coil behavior. The model is based on a distributed model for bare tube heat exchangers. To reduce the computational resource requirement needed by the optimization study, the proposed model is kept simple yet able to identify significant aspects of any design. So that geometrical, physical, both heat transfer and fluid flow effects may be studied with comparative ease.

There are three elements for a heat exchanger in this model: branch, tube and control volume. A heat exchanger may consist of a number of separated branches. A branch is comprised of a number of straight tubes, inside with the refrigerant flows continuously without joining or splitting. Further, a straight tube is subdivided into several control volumes of equal length. The basic element for computation in the model is the control volume, where the air and refrigerant flow outside and inside the tube respectively and heat transfer occurs between. The bare tube is applied to the miniature heat exchanger in this study. Each tube is a branch in the heat exchanger and there is no branch joining or splitting among the tubes. There are no return bends among the bare tubes. The refrigerant flows from one end of the tube and exits from another end of the tubes, shown in Figure 3.3. Every straight tube is subdivided into many control volumes of equal length.

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The type of heat exchanger studied in this study is a single-pass air-R134A bare tube crossflow heat exchanger with stagger arrangement. Although the numerical models of condenser and evaporator are not the same, there are many similarities in the main structure of modeling. The modeling procedures described in computation between the condenser and evaporator coils. The differences in computation between the condenser and evaporator coils are discussed in the following Sections. The proposed numerical model of heat exchanger mainly comprises of the following parts:

- 1) The development of the governing equations for a control volume,
- 2) The selection of the concerned correlations for the air-side and the refrigerant side,
- 3) The computation technique for the governing equations,
- 4) The approach to link all of the control volumes together in the heat exchanger simulation.

3.2.1 Governing Equations for a Control Volume

The governing equations for the airside and refrigerant side are established based on a control volume shown in Figure 3-2.

Before the equations are developed, the assumptions made are as follows:

- 1) Refrigerant flows inside tube is one-dimensional,
- 2) Compared to the refrigerant internal energy, the dynamic energy is neglected,
- 3) The pressures of liquid phase and vapor phase are the same in any cross section,
- 4) The refrigerant is equably and stably distributed into the tubes,
- 5) The effect of the pressure drop along the tube direction on the saturated phase (vapor or liquid) enthalpy could be omitted, and

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6) Gravity is neglected for horizontal tubes.

In a heat exchanger, inside the tubes, flowing refrigerant undergoes different phase regions, namely single-phase and two-phase flow regions. There are three possible heat transfer zones for an evaporator which corresponding to evaporation, transition and superheating of refrigerants. The condenser is divided into a superheated single-phase flow region, a two-phase flow region and a subcooled single-phase flow region. On the air side, the heat transfer mechanisms for dry surface and wet surface of the coils are different depending on the dew point of the air and tube outer surface temperature. For dry surface, only the energy conservation equation is needed. For wet surface, since simultaneous heat and mass transfer occurs, both the energy and mass conservation equations are needed.

3.2.1.1 Air Side

The governing equations on the dry surface and the wet surface are separately developed due to the difference of the heat transfer mechanism.

Dry Surface***Governing equation***

The air side governing equation is established along the airflow direction. It is noted that the heat transfer on the air side actually is two-dimensional in a control volume shown in Figure 3.2, when air flows across the bare tubes. The heat transfer both in the airflow direction and in the perpendicular direction varies. For a differential heat transfer area where the relevant properties are averaged along the perpendicular

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direction, the heat balance equation of the air side along the airflow direction can be written as follows:

$$\delta m_a \cdot C_{p,a} dT_a = q_o dA_o \quad \text{for condenser} \quad (3-1a)$$

$$\delta m_a \cdot C_{p,a} dT_a = -q_o dA_o \quad \text{for dry-zone of evaporator} \quad (3-1b)$$

Determination of Heat Transfer Rate

The use of spatially averaged properties in the direction perpendicular to airflow allows a preservation of the governing equations of the airside in a one-dimensional differential level. Although the above governing equations can be readily discretized for calculation, the solution is complicated by the dependency on the heat transfer rate. Here, the determination of the heat transfer rate for a control volume is discussed.

For a control volume, the heat flux q_o is determined using the following energy balance equation:

$$\delta Q = U_o \delta A_o \cdot \Delta T_m = \delta m_a \cdot C_{p,a} (T_{a2} - T_{a1}) = q_o \delta A_o \quad \text{for condenser} \quad (3-2a)$$

$$\delta Q = U_o \delta A_o \cdot \Delta T_m = -\delta m_a \cdot C_{p,a} (T_{a2} - T_{a1}) = -q_o \delta A_o \quad \text{for dry-zone of evaporator} \quad (3-2b)$$

where U_o is overall heat transfer coefficient, and ΔT_m is the log mean temperature difference.

The main difficulty to solve the above heat balance equations is to determine the value of the overall heat transfer coefficient U_o and the log mean temperature

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difference ΔT_m . According to the thermal resistance existing in the heat transfer process, U_o can be expressed as:

$$\frac{1}{U_o} = R_a + R_t + R_r = \frac{1}{\alpha_{a, sen}} + \frac{\delta A_o \cdot (d_o - d_i)}{\pi(d_o + d_i)\delta z \cdot k_t} + \frac{\delta A_o}{\delta A_i \alpha_r} \quad (3-3)$$

where subscript t denotes tube.

The log mean temperature difference is expressed as:

$$\Delta T_m = \frac{(T_{r1} - T_{a1}) - (T_{r2} - T_{a2})}{\ln \left[\frac{T_{r1} - T_{a1}}{T_{r2} - T_{a2}} \right]} \quad \text{for condenser} \quad (3-4a)$$

$$\Delta T_m = \frac{(T_{a1} - T_{r1}) - (T_{a2} - T_{r2})}{\ln \left[\frac{T_{a1} - T_{r1}}{T_{a2} - T_{r2}} \right]} \quad \text{for dry-zone of evaporator} \quad (3-4b)$$

If the local air-side heat transfer coefficient $\alpha_{a, sen}$ and refrigerant-side heat transfer coefficient α_r are known, U_o can be readily determined using Equation (3-3).

ΔT_m is determined by an iterative procedure to calculate the outlet temperatures of refrigerant side and air side. After U_o and ΔT_m are determined, the heat transfer rate δQ of the control volume is obtained, thus the air outlet temperature T_{a2} of the control volume can be obtained with the above heat balance equation. The outlet state of the air side is determined when the air outlet temperature is obtained, since the moisture content is unchanged in the process.

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Wet Surface

Governing equation

Like the dry-surface, for a differential heat transfer area where the relevant properties are averaged along the perpendicular direction, the heat balance equation of the airside is as follows:

$$m_a(dh_a - h_w dW_a) = -q_o dA_o \quad (3-5)$$

where the term $h_w dW_a$ represents the energy taken away by the condensate.

The mass conservation equation is as follows:

$$-m_a dW_a = \alpha_m (W_a - W_s) dA_o \quad (3-6)$$

Equations (3-5) and (3-6) contain four unknowns: air enthalpy h_a , mass transfer coefficient α_m , air specific humidity W_a and air saturated specific humidity W_s at condensate surface temperature, provided that the outer heat transfer flux q_o is known. Obviously, an additional equation is required to determine W_s .

Determination of Heat Transfer Rate

For a control volume, Equation (3-4) takes the following form:

$$\delta Q_{tot} = \delta Q_{sen} + \delta Q_{lat} = U_o \delta A_o \cdot \Delta T_m \quad (3-7a)$$

$$\delta Q_{tot} = \delta m_a (h_{a1} - h_{a2}) - h_w \cdot \delta m_a \cdot (W_{a2} - W_{a1}) \quad (3-7b)$$

The detailed heat transfer rate is expressed as:

$$\delta Q_{tot} = \alpha_{sen} \delta A_t \cdot \left[(T_a - T_{t,o}) + \frac{i_{fg}}{Le^{1-n} \cdot C_{p,a}} (W_a - W_{t,o}) \right] \quad (3-8)$$

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Introducing a mass transfer factor C^* ,

$$C^* = \frac{W_a - W_{t,o}}{T_a - T_{t,o}} \quad (3-9)$$

Then, Equation (3-8) becomes:

$$\delta Q_{tot} = \delta A_t \cdot \left(\alpha_{sen} + \frac{i_{fg} \cdot \alpha_{sen}}{Le^{1-n} \cdot C_{p,a}} C^* \right) \cdot (T_a - T_{t,o}) \quad (3-10)$$

Hence, for wet surface, the overall air-side heat transfer coefficient $\alpha_{a,w}$ can be expressed as:

$$\alpha_{a,w} = \alpha_{sen} \cdot \left(1 + \frac{i_{fg} \cdot C^*}{Le^{1-n} \cdot C_{p,a}} \right) \quad (3-11)$$

Therefore, the overall heat transfer coefficient U_o in Equation (3-7a) can be given as:

$$\frac{1}{U_o} = R_a + R_t + R_r = \frac{1}{\alpha_{a,sen}} + \frac{\delta A_o \cdot (d_o - d_i)}{\pi(d_o + d_i)\delta z \cdot k_t} + \frac{\delta A_o}{\delta A_i \alpha_r} \quad (3-12)$$

It is noted that the air-side heat transfer coefficient in Equation (3-12) is different from that in Equation (3-8).

From Equations (3-7) and (3-8), the energy balance equation of the air side for a control volume is obtained:

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$$\delta m_a h_{a2} = \delta m_a h_{a1} - \alpha_{a, \text{sen}} \cdot \delta A_t \cdot \left[(T_a - T_{t,o}) + \frac{i_{fg}}{Le^{1-n} \cdot C_{p,a}} (W_a - W_{t,o}) \right] + C_{p,w} T_{t,o} \delta m_a (W_{a1} - W_{a2}) \quad (3-13)$$

At steady state, the heat balance between the airside and the refrigerant inside the tube is:

$$\delta Q_{tot} = \alpha_c \delta A_i (T_{t,o} - T_r) \quad (3-14)$$

where α_c is the heat transfer coefficient from the tube outer surface to the refrigerant.

If α_c is known, $T_{t,o}$ can be obtained from Equation (3-14). The following heat transfer equation is used to determine α_c .

$$\alpha_c \delta A_i (T_{t,o} - T_r) = \alpha_r \cdot \delta A_i (T_{i,t} - T_r) = \frac{\pi(d_o + d_i) \cdot k_t \cdot \delta z}{(d_o - d_i)} (T_{i,o} - T_{i,t}) \quad (3-15)$$

The expression of α_c can be obtained:

$$\alpha_c = \frac{1}{\frac{1}{\alpha_r} + \frac{\delta A_i (d_o - d_i)}{\pi(d_o + d_i) k_t \delta z}} \quad (3-16)$$

An iteration procedure is needed to determine $T_{t,o}$, since the calculation requires the value of heat transfer rate.

Chapter 3**Modeling of the Heat Exchanger****Pressure drop of the airside**

For a heat exchanger, airside pressure drop associated with flow across the bare tube bank has little effect on the overall heat transfer rate. The determination of airside pressure drop is required to estimate the fan power consumption which is an important performance parameter for the miniature heat exchanger. Turaga *et al* (1988) proposed a friction factor correlation for dry and wet coils with flat fins. The air-side pressure drop can be determined by the modified friction factor using the following expression:

$$\Delta P_a = f \frac{A_0}{A_{cf}} \cdot \frac{G_{\max}^2}{2\rho_a} \quad (3-17a)$$

in which

$$f_{a,D} = 0.589(\text{Re}_{D_h})^{-0.27} \quad \text{for dry surface} \quad 3-17b$$

$$f_{a,w} = 0.318(f_{a,D})^{-0.94}(\text{Re}_{D_h})^{-0.92} \quad \text{for wet surface} \quad (3-17c)$$

where A_{cf} is coil face area and f is friction factor.

3.2.1.2 Refrigerant Side

The differential forms of governing equations are developed by applying conservation laws to the refrigerant side. The governing equations in the two-phase and single-phase flow regions are separately considered. In order to adapt the model to the detailed analysis on the refrigerant side, such as the effect of refrigerant inventory on

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the heat exchanger performance, a separated-flow model is used in the two-phase flow region.

Two-Phase Flow Region

There are two theoretical flow models which are commonly used to simulate two-phase pipe flow, namely homogeneous flow model and separated flow model. The homogeneous flow model ignores the slip effect between the vapor phase and the liquid phase and considers the two phases to be intimately mixed together. Generally, the homogeneous flow model is applied to the dispersed flow pattern. In contrast to the homogeneous flow model, the separated model considers the two phases to be separated with different properties and different mean velocities. As the phase velocities differ, slip exists between the phases and hence when the conservation equations are applied to the vapor phase and liquid phase, extra variables are introduced, namely void fraction and slip ratio. Despite of the complexity, the separated flow model is believed to give more accurate predictions than the homogeneous flow model. In this model, the separated flow model is incorporated in the refrigerant-side simulation.

In a separated flow model, conservation equations may be applied to each phase allowing for the interactions which take place between the phases. However, the respective conservation equations for vapor phase and liquid phase can be added together to give an overall balance equation for the two-phase flow. The following governing equations are developed for the two-phase flow. It is noted that the Z coordinate is along the refrigerant flow direction.

Chapter 3**Modeling of the Heat Exchanger****Continuity equation**

$$\frac{dm_r}{dz} = \frac{d}{dz}(m_{r,v} + m_{r,l}) = 0 \quad (3-18a)$$

$$m_{r,v} = x \cdot m_r = \alpha A \rho_v U_v \quad (3-18b)$$

$$m_{r,l} = (1 - x) \cdot m_r = (1 - \alpha) A \rho_l U_l \quad (3-18c)$$

where A is cross section area, and U is velocity.

Momentum conservation equation

$$\frac{dP_r}{dz} = \left. \frac{dP_r}{dz} \right|_m + \left. \frac{dP_r}{dz} \right|_f \quad (3-19)$$

The friction pressure gradient is obtained by using two-phase flow friction multiplier, whereas the momentum pressure gradient is given by:

$$\left. \frac{dP_r}{dz} \right|_m = \frac{1}{A} \frac{d}{dz} (m_{r,v} U_v + m_{r,l} U_l) = \frac{m_r}{A} \frac{d}{dz} [x U_v + (1 - x) \cdot U_l] \quad (3-20a)$$

$$\left. \frac{dP_r}{dz} \right|_m = \frac{m_r}{A} \left[x \frac{dU_v}{dz} + (1 - x) \frac{dU_l}{dz} + (U_v - U_l) \frac{dx}{dz} \right] \quad (3-20b)$$

Energy conservation equation

$$\frac{d}{dz} [m_{r,v} h_v + m_{r,l} h_l] = m_r \frac{d}{dz} [x \cdot h_v + (1 - x) \cdot h_l] = \pi d_i q_i \quad (3-21)$$

For simplicity, it is further assumed that the effect of the pressure drop on the saturated vapor enthalpy and liquid enthalpy is negligible for condenser and evaporator coils.

$$\frac{dh_v}{dP_r} \frac{dP_r}{dz} = 0, \quad \frac{dh_l}{dP_r} \frac{dP_r}{dz} = 0 \quad (3-22)$$

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Thus

$$m_r h_{fg} \cdot \frac{dx}{dz} = \pi d_i q_i \quad (3-23)$$

For a control volume with a specified length, shown in Figure 3.2, the above governing equations are discretized for computation.

From Equations (3-18b), (3-18c), there are

$$x_2 \cdot \delta m_r = A \rho_v \alpha_2 U_{v2} \quad (3-24a)$$

$$(1 - x_2) \cdot \delta m_r = A \rho_l (1 - \alpha_2) U_{l2} \quad (3-24b)$$

Equation (3-19) is modified to include the pressure drop of friction and momentum factors. Thus, the pressure drop for a control volume is computed as the sum of friction pressure drop and momentum pressure drop.

$$P_{r1} - P_{r2} = \delta P_r|_f + \delta P_r|m \quad (3-25)$$

where $\delta P_r|m$ can be obtained from Equation (3-20b):

$$\delta P_r|m = \frac{\delta m_r}{A} \left[\frac{x_1 + x_2}{2} (U_{v2} - U_{v1}) + \left(1 - \frac{x_1 + x_2}{2} \right) (U_{l2} - U_{l1}) + \left(\frac{U_{v1} + U_{v2}}{2} - \frac{U_{l1} + U_{l2}}{2} \right) (x_2 - x_1) \right] \quad (3-26)$$

Using the friction multiplier and local pressure loss factor for two-phase flow $\delta P_r|_f$

can be obtained:

$$\delta P_r|_f = \psi_s^2 \cdot \delta P_s \quad (3-27)$$

It is noted that subscript s denotes single phase, which can be either a liquid phase or a vapor phase depending on which correlations are employed to calculate ψ_s .

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The pressure drop at the refrigerant side is computed as the sum of two separate components, namely the component due to the friction effect at the tube wall ΔP_f and the component due to acceleration for flow boiling in the evaporator or deceleration for flow condensation in the condenser, i.e., momentum pressure drop ΔP_m .

For single-phase flow, the friction effect is computed by means of the following formula:

$$\Delta P_{f,s} = \lambda \cdot \frac{L}{2d_i} \frac{G_r^2}{\rho_s} \quad (3-28)$$

where L is tube length, λ is friction coefficient, and subscript f denotes friction.

The friction coefficient λ is calculated according to the Reynolds number (Paliwoda, 1989):

$$\lambda = \frac{64}{\text{Re}} \quad \text{Re} < 1187 \quad (3-29)$$

$$\lambda = \frac{0.3164}{\text{Re}^{0.25}} \quad \text{Re} > 1187 \quad (3-30)$$

The momentum effect of pressure drop is often neglected for single-phase flow.

For two-phase flow, the frictional loss is often calculated as a single-phase flow by means of a dimensionless two-phase multiplier. Liang *et al* (2000) recommended the correlation for two-phase flow pressure drop on refrigerant side as follows:

$$\Delta P_{f,tp} = \left[\Delta P_{l,f} + 2x(\Delta P_{v,f} - \Delta P_{l,f}) \right] \cdot (1-x)^{1/3} + x^2 \cdot \Delta P_{v,f} \quad (3-31)$$

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in which

$$\Delta P_{v,f} = \lambda \cdot \frac{L}{2d} \frac{G^2}{\rho_v} \quad (3-32)$$

$$\Delta P_{l,f} = \lambda \cdot \frac{L}{2d} \frac{G^2}{\rho_l} \quad (3-33)$$

The momentum component pressure drop ΔP_m will be significant for case of fully developed boiling or condensing refrigerant at high density of heat flux on the external heat transfer area. In such case, the value of ΔP_m can approach 15% of the total pressure gradient (Paliwoda, 1989). At typical operational conditions of refrigeration systems, the error resulting from neglecting ΔP_m usually does not exceed 5%.

In the two-phase flow region, ΔP_m can be computed by:

$$\Delta P_m = G_r^2 (v_2 - v_1) \quad (3-34)$$

where v is specific volume, subscript m denotes momentum, and subscripts 1 and 2 denote inlet and outlet of the two-phase flow region.

Discretizing Equation (3-23) gives:

$$\delta m_r h_{fg} \cdot (x_2 - x_1) = \pi d_i q_i \cdot \delta z \quad (3-35)$$

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If the inlet properties is given, the outlet properties of the refrigerant side are calculated through the simultaneous solution of Equations (3-24a), (3-24b), (3-25), and (3-35). These equations contain five unknowns, i.e. the outlet vapor quality x_2 , void fraction α_2 , vapor velocity U_{v2} , liquid velocity U_{l2} , and refrigerant pressure P_{r2} , provided that the inner heat transfer flux q_i is known. Therefore, one additional equation is required, namely the correlation of void fraction.

Single-Phase Flow Region

In single-phase region, as the void fraction α equals zero or unit and only one velocity of either the vapor or the liquid exists, the governing equations for single-phase flow are given as follows:

$$\frac{d}{dz}(\rho U) = 0 \quad (3-36)$$

$$\frac{dP_r}{dz} = -\frac{dP_r}{dz} \Big|_f \quad (3-37)$$

$$\delta m_r \frac{d}{dz}(C_p T_r) = \pi d_i q_i \quad (3-38)$$

It is noted the momentum pressure gradient for the single-phase flow is neglected.

For a control volume, the above equations are modified in the followings:

$$U_s = \frac{m_r}{A \rho_s} \quad (3-39)$$

$$P_{r1} - P_{r2} = \lambda \frac{\delta z}{2d_i} \frac{\delta m_r^2}{\rho_s A^2} \quad (3-40)$$

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$$\delta m_r C_{p,s} (T_{r1} - T_{r2}) = \pi d_i q_i \cdot \delta z \quad \text{for condenser} \quad (3-41a)$$

$$\delta m_r C_{p,s} (T_{r2} - T_{r1}) = \pi d_i q_i \cdot \delta z \quad \text{for evaporator} \quad (3-41b)$$

Since the outlet pressure P_{r2} and temperature T_{r2} of the control volume can be determined directly from Equations (3-40) and (3-41), the computation in single-phase flow region is much easier compared to the two-phase flow region.

3.2.1.3 Computer Simulation

There are three main elements in the model: i.e. branch, tube and control volume. The basic element for computation is the control volume. Therefore, the computation of a control volume is the core of the model. In order to simulate a coil with complex refrigerant circuitry, an approach is proposed to link all the control volumes during the computation. The coil is firstly divided into a large number of volumes of equal length. The number of control volume depends on the number and length of the refrigerant flow paths. For a control volume of length δz , shown in Figure 2, the geometric parameters of the tube and the spaces of between tubes to calculate tube inner surface area δA_i and tube surface area δA_o .

$$\delta A_i = \pi d_i \cdot \delta z \quad (3-42)$$

$$\delta A_o = \pi d_o \cdot \delta z \quad (3-43)$$

The circuitry is relatively complex with branch joining and splitting. Although the refrigerant circuitry may be complex, the branch network usually is quite simple. In this way, the whole coil can be regarded as a network consisting of many separated

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branches and each tube is subdivided into several control volumes. To simulate a coil, a hierarchical system is used, which follows the procedure given below:

- First level: to determine the computation sequence for all branches in the coil and to carry out computation branch-by-branch,
- Second level: to determine the computation sequence for all tubes in the branch and to carry out computation tube-by-tube, and
- Third level: to determine the computation sequence for all control volumes in the tube and to carry out computation control volume-by-control volume.

An iterative procedure is required to organize the complex computation. When the air and refrigerant properties at the inlet (e.g. δm_a , h_{a1} , T_{a1} , W_{a1} , δm_r , h_{r1} , T_{r1} , P_{r1} , U_{v1} , U_{l1} , α_1, x_1) are given, the governing equations, the correlations for heat and mass transfer coefficients and pressure losses are applied. The heat transfer rate and the properties of the air and the refrigerant at the exit of the control volume are determined.

With the required inlet parameters, the computation sequence of a control volume can be summarized as three steps: namely, estimating the values of the unknown parameters, applying the governing equations and updating the values for the unknown parameters.

The unknown parameters that need to be estimated include: the air outlet dry-bulb temperature T_{a2} , air outlet specific humidity W_{a2} , and heat transfer rate δQ . After the estimation, the mean values of the variables of the airside and the refrigerant side for the control volume can be calculated.

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After applying the governing equations to the control volume, the values of the unknown parameters can be updated. The procedure is repeated until the unknown parameters converge.

During computation, the vapor quality is checked to determine the flow transitions from single-phase superheated region to two-phase flow region or from two-phase flow region to single-phase subcooled region. On the airside, the tube wall temperature is computed and updated in the iteration procedure. If the tube wall temperature is lower than the mean dew point temperature, the coil surface is considered wet. Otherwise, the coil surface is dry.

The simulation study allows the performance of a heat exchanger to be assessed and the effects of changing design parameters on the performance to be observed i.e. the total pressure drop and the capacity of heat exchanger coil. Figure 3-4 shows the comparison of refrigerant-side pressure drop of the predicted and measured results on one heat exchanger. Figure 3-5 shows the typical comparison of the coil capacity of the predicted and measured results on the same heat exchanger. The figures show that, within the range of comparison, good agreement has been obtained.

Figure 3-1 Schematic refrigeration cycle

Figure 3-2 A control volume along a tube

Figure 3-3 Schematic structure of the tube heat exchanger

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Figure 3-4 Comparison of predicted refrigerant-side pressure drop with the experimental measurement (Liang, 2000)

Figure 3-5 Comparison of predicted coil capacity with the experimental measurement (Liang, 2000)

Chapter 4

Geometrical Optimization Design of the Heat Exchanger

By linking the previous simulation model of the bare tube heat exchanger with an optimization algorithm, it is possible to allow the proper selection of geometrical design parameters to achieve an overall optimum performance of a bare tube heat exchanger under prescribed working conditions.

In this study, the Box's COMPLEX optimization technique is used to geometrical optimization design of the miniature bare tube heat exchanger. Results of the optimization are presented later in this section.

4.1 Box Complex Method

In general, optimization techniques can be classified into unconstrained methods, linear and non-linear constrained methods. Non-linear constrained methods include two categories, i.e. direct search method and gradient method. In a direct search method, only the objective function values are required while that for the gradient method, both the derivatives and the values of the function are required. Gradient method is found to be superior in convergence rates compared with direct search method. However, this method is not practical if the derivatives of the objective function are difficult or impossible to be obtained.

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The objective function involved in the current study is highly non-linear and may not be differentiable. Therefore the suitable optimization technique should be a direct search method. In the current optimization study, the COMPLEX optimization method developed by M. J. Box (M. J. Box, 1965). The details about this technique are introduced in the following section.

This method is a sequential technique that has been proven effective in solving problems with non-linear objective functions. This procedure was claimed that it tends to find the global optima.

In general, an optimization study may be mathematically represented by:

$$\text{Maximize/Minimize: } F(x) = f(x_1, x_2, \dots, x_n) \quad (4-1)$$

Subject to:

$$(i) \quad L_E(x)_i \leq E(x)_i \leq H_E(x)_i, \quad i = 1, 2, \dots, N \quad (4-1a)$$

$$(ii) \quad L_G(x)_j \leq G(x)_j \leq H_G(x)_j, \quad j = 1, 2, \dots, M \quad (4-1b)$$

$$(iii) \quad L_I(x)_k \leq I(x)_k \leq H_I(x)_k, \quad k = 1, 2, \dots, L \quad (4-1c)$$

where E , G and I stand for the explicit constraints, geometrical constraints and implicit constraints respectively, L and H represent the lower limit and higher limit, i , j and k are the number of explicit, geometrical and implicit constraints respectively.

The Box COMPLEX method is implemented in the following steps:

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1. A feasible starting point is picked that satisfies all geometrical, explicit and implicit constraints. $K-1$ additional points are generated from pseudo-random numbers and constraints for each of the independent variables:

$$X_{i,j} = v_i + r_{i,j} \times (h_i - v_i) \quad i = 1, 2, \dots, N; j = 1, 2, \dots, K-1 \quad (4-2)$$

These additional points and the starting point together are called the first COMPLEX.

In the above, N is the number of the explicit free variable constraints and $K = N + 1$; v_i and h_i are the lower and upper constraints of the free variable respectively; $r_{i,j}$ is the pseudo-random numbers between 0 and 1. The selected points satisfy the explicit constraints of the free variables but may violate other constraints.

2. If any geometric or implicit constraint is violated, the trial point is moved one half of the distance to the centroid of the remaining points:

$$X_{i,j}(new) = (X_{i,j}(old) + \bar{X}_{i,c}) / 2 \quad i = 1, 2, \dots, N \quad (4-3)$$

where the coordinates of the centroid of the remaining points, $\bar{X}_{i,c}$ are defined by

$$\bar{X}_{i,c} = \frac{1}{K-1} \left[\sum_{j=1}^{K-1} X_{i,j} - X_{i,j}(old) \right] \quad i = 1, 2, \dots, N \quad (4-4)$$

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This process is repeated until all the constraints are satisfied. For a highly constrained optimization problem, especially when the constraints are non-linear, the setting up of feasible initial complexes may be very difficult.

3. Evaluate the objective function at each point. The point having the worst value is reflected by a reflection factor of α along the line linking the replaced point and the centroid of the remaining points. The resulted new point is:

$$X_{i,j}(new) = (1 + \alpha)\bar{X}_{i,c} - \alpha X_{i,j}(old) \quad (4-5)$$

Here $X_{i,j}(old)$ is the worst point and $\bar{X}_{i,c}$ is the calculated centroid of the remaining points. To take more advantage of information about the search area that has already been explored and to increase the effectiveness of the search, the centroid is calculated by weighing each point selected according to its objective function value. The following equation is used:

$$\bar{X}_{i,c} = \frac{1}{K-1} \left[\sum_{j=1}^k X_{i,j} \left(\frac{F_j - F_{worst}}{F_{best} - F_{worst}} \right) \right] \quad i = 1, 2, \dots, N \quad (4-6)$$

4. If after the reflection, an explicit constraint is violated, the point is moved inside the constraint boundary by a factor of δ .
5. The objective function is evaluated only if the points satisfy all the explicit and implicit constraints.
6. The newly obtained function value is then checked for any improvement. If the new point shows no improvement, it is moved halfway towards the centroid of the

remaining points. However, if 5 successive moves towards the centroid do not yield any improvement in the objective function value, the point is moved halfway towards the best complex. This usually gives an improvement.

7. In addition, the second worst point is moved towards the best complex.
8. If there is no improvement after 5 successive moves towards the best point, the solution is said to have converged. In practice, convergence is assumed when after a specified number of consecutive successful iterations, the objective function at a new point which satisfies all the constraints lies within a pre-set tolerance of the value of the objective function of the best point.
9. The search should then be repeated with a different random number “seed”. The obtained results should be compared with those of previously search to confirm the existence of a global optimum.

4.2 Optimization of the Bare tube Heat Exchanger

In this study, heat exchanger designs are optimized for their heat transfer and mechanical performance under prescribed operating conditions. In the optimization study, the objective is to obtain a heat exchanger which can transfer a maximum heat load and yet operate at a relative low possible pressure drop for a given overall dimensions under prescribed operating conditions. Hence, the objective function was chosen to be the total heat transfer divided by the total pressure drop at the refrigerant side.

$$F_{obj} = Q / \Delta P \quad (4-7)$$

Chapter 4**Geometrical Optimization Design**

In the design of a heat exchanger for specified operating conditions, some 9 geometrical parameters may be identified; these are W_t , H_t , d_o , d_i , N_r , N_t , S_t , S_l , and L .

The variables that are allowed to vary in the present optimization study are: d_o , d_i , L , S_t , and S_l . The number of tube row (N_r) and tube transverse (N_t) can clearly be varied but optimizations were only performed for specified tube row and transverse numbers to avoid difficulties encountered with the optimization algorithm in respect of integer variables. The height (H_t) and width (W_t) are set as given values.

The operating conditions of the heat exchanger in the present study are shown in Table 4-1. Detail of the various constraints applied during the present study is given in tables 4-2, 4-3 and 4-4. The various constraints are explained as follows:

Explicit constraints are shown in Table 4-2. The first constraint is set to ensure that the outer diameter (d_o) is greater than the inner diameter (d_i). The second constraint is set such that the transverse space (S_t) is greater than outer diameter (d_o). The last constraint is to ensure that the longitudinal space (S_l) is greater than the outer diameter (d_o).

All of the constraints in Table 4-3 are geometric parameters and hence have to be non-negative. The numbers of the tube row (N_r) and the tube transverse (N_t) have to be positive integer. Due to the width (W_t) of the heat exchanger is less than its

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height (H_t); therefore the longitudinal space (S_l) is usually greater than transverse space (S_t).

Two implicit constraints were used in the present study and they are shown in Table 4-4. Those are the total heat exchange (Q) and the total pressure drop (ΔP) at refrigerant side. This optimization has to obtain a maximum heat exchange (Q) with a given core volume and compact bare-tube heat exchanger at a relative low pressure drop on both the airside and the refrigerant side. The heat exchange (Q) and pressure drop (ΔP) are restricted to positive values.

In this study, the optimization searches have been carried out to search for optimum constraints of heat exchanger characteristics, i.e., d_o , d_i , S_t and S_l , under specified operating conditions. The operating conditions in this study are as follows:

The mass flow rate of refrigerant (M_r) is 0.8 g/s, the inlet temperature of refrigerant ($T_{r, in}$) is 5.0°C, the inlet temperature of air ($T_{a, in}$) is 24.0°C, the inlet velocity of air ($V_{a, in}$) is 15.0 m/s, the inlet relative humidity of air ($Rh_{a, in}$) is 80.0%, the condensing temperature of refrigerant (T_c) is 40°C, the evaporating temperature of refrigerant (T_e) is 5.0°C, both ΔT_{sub} and ΔT_{sup} are 5.0°C.

During the optimization searches, the free variables d_o , S_t , and S_l will vary. Since the tube wall thickness is set as 0.075mm, hence the inner tube diameter, d_i will be known where

Chapter 4**Geometrical Optimization Design**

$$d_i = d_o - 2t \quad (4-8)$$

And since the number of tube row N_r is related to W_t and S_l as,

$$N_r = \text{Integer} \left[\frac{W_t - 0.5S_l}{S_l} \right] \quad (4-9)$$

Similarly, for the number of tube transverse N_t

$$N_t = \text{Integer} \left[\frac{H_t - 0.5S_t}{S_t} \right] \quad (4-10)$$

Therefore, the number of tube row N_r and the number of tube transverse N_t will vary during the optimization searches.

The mathematical model together with the optimization algorithm has been implemented using ForTran programming language. The first feasible design point is set arbitrarily within the constraints ranges during the optimization search. Then the program runs and searches the feasible values of the variation.

Figure 4-1 shows the variation of the outer diameter during the optimization search. The program searches the feasible outer diameter in one relative large range at first, and then the search focuses on one very small range till to the set iteration times. The program iterates with the optimal values till to the set iteration times after it finds the optimal values. Figure 4-2 shows the variation of the tube inner diameter during the optimization search. Figures 4-3 and 4-4 show the variation of S_t and S_l during

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the optimization search. The first point value of S_t which is arbitrarily set is greater than the optimal value. The program searches the feasible S_t in one relative large range at first, and then the search focuses on one quickly decreasing small range till to the set iteration times. From Figure 4-3, we can see that the optimal S_t has been obtained faster than the optimal outer diameter. From figure 4-4, we can see the first point value of S_l which is arbitrarily set is less than the optimal value. The program searches the feasible S_l in one relative large and larger than the optimal value range at first, and then the search focuses on one quickly decreasing small range till to the set iteration times and the optimal S_l has been got quickly. In the program, the outer diameter constraint range is from 0.50mm to 0.80mm; tube thickness is set as 0.075mm. The final optimization design is obtained with the following dimensions:

$$d_o=0.60\text{mm}, d_i=0.45\text{mm}, S_t=0.65\text{mm}, S_l=2.00\text{mm}, L=30\text{mm}.$$

The schematic heat exchanger is shown in figures 4-8a and 4-8b. At these optimal dimensions, we obtain the heat exchange rate $Q = 125\text{W}$ and pressure drop at the refrigerant side $\Delta P = 28.80\text{kPa}$.

Figure 4-5 shows the variation of objective function during the optimization search. The result has been normalized by dividing it with the optimal value. The result shows that the value of the objective function increases from 33.2% of the first feasible design, which, in this study, is arbitrarily set in the constraint range, to 100%.

Fig. 4-1 Variation of the outer diameter during the optimization search

Fig. 4-2 Variation of the inner diameter during the optimization search

Fig. 4-3 Variation of S_t during the optimization search

Fig. 4-4 Variation of S_l during the optimization search

Fig. 4-5 Variation of F_{obj} during the optimization search

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Fig. 4-6a Schematic pitches of S_t , S_l and S_d

Fig. 4-6b Schematic structure of bare tube heat exchanger

Chapter 4**Geometrical Optimization Design****Table 4-1** Given operating conditions of heat exchangers

Definitions	$\dot{M}_r$ (g/s)	$T_{r,in}$ (°C)	$T_{a,in}$ (°C)	ΔT_{sup} (°C)	ΔT_{sub} (°C)	T_c (°C)	T_e (°C)	$Rh_{a,in}$ (%)	$V_{a,in}$ (m/s)
Heat exchanger	0.8	5.0	24.0	5.0	5.0	40.0	5.0	80.0	15.0

Table 4-2 Explicit constraints

i	L_{Ei}	E_i	H_{Ei}	Unit
1	0.50	$\leq d_o \leq$	0.80	mm
2	d_o	$< S_t \leq$	$0.15 * H_t$	mm
3	d_o	$< S_l \leq$	$0.50 * W_t$	mm

Table 4-3 Geometrical constraints

j	L_{Gj}	$G(x)_j$	H_{Gj}	Unit
1	0.0	$< d_i < d_o$		mm
2	d_o	$< S_t \leq 0.15 * H_t$		mm
3	$H_t/S_t - 1$	$\leq N_t \leq H_t/S_t + 1$		
4	$W_t/S_l - 1$	$\leq N_r \leq W_t/S_l + 1$		
5	$W_t=8$			mm
6	$H_t=30$			mm
7	0.0	$\leq L \leq 1000$		mm

Table 4-4 Implicit constraints

k	L_{Ik}	$I(x)_k$	H_{Ik}	Unit
1	0.0	$\leq Q \leq 1.0$		kW
2	0.0	$\leq \Delta P \leq 1 \times 10^3$		kPa

Chapter 5

Experimental Facilities

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5.1 Introduction

An experimental rig is designed in this project to validate the model predictions on the evaporator and condenser coil performance. This experimental test rig consists mainly of four systems, and they are wind tunnel, air property control system, refrigeration system and instrumentation and data acquisition system. The main features of the test rig are that the test heat exchanger coils can be tested as an evaporator or condenser configuration and the air humidity in the wind tunnel can be regulated in a wide range. The schematic diagram and the layout of the experimental test rig are shown in Figure 5.1.

The wind tunnel is an important test facility in the test rig. It serves to provide the required air flow conditions. The main requirements of this wind tunnel are as follows:

- 1) Uniform air speed at the inlet of test coil section.
- 2) The air face velocity of the test coils can be varied within a range.

Air property control system serves to provide the required air thermodynamics conditions for the test coils. The air properties including the dry-bulb temperature and

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relative humidity are valid in the tests so that the coil performance under various air conditions can be investigated.

Refrigeration system serves to provide the required refrigerant flow conditions for the test coils, which mainly include mass flow rate, condensing temperature, evaporating temperature, subcooling temperature difference or superheating temperature difference. The refrigeration system is required to be capable of varying its refrigerating capacity is a large range so as to match with the coil load variation under different testing conditions.

The instrumentation and data acquisition system collects all the data during the tests. The system consists of temperature measurements, pressure measurements, and low rate measurements on the air side and the refrigerant side. Accuracy and reliability of this system is very important for the success of the tests.

5.2 Wind Tunnel

The designed wind tunnel can be operated in opened circuit or closed circuit configurations by slight adjustment of the pipe section, as shown in Figure 5.1. If the removable part is installed, the wind tunnel will be in closed-circuit configuration. To change to an opened circuit configuration, the removable section is simple removed. When the test coil is to be used as an air-cooling evaporator, a closed circuit configuration is chosen such that variation in ambient conditions has no effect on the

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test. In addition, the test coil can serve as a dehumidifier to lower the air inlet humidity, as such no additional dehumidifier is needed.

During condenser coil testing, an opened circuit configuration is adopted and room air is used to cool the test coil. Since the heat given off by the test coil is rejected to the surroundings, there is no need for an additional air-cooler to lower the air temperature in the tunnel.

The design of the diverging section of the wind tunnel just prior to the test coil section is crucial. It ensures that the uniform air velocity is achieved through the whole cross-sectional area of the wind tunnel just before the test coil section. The approach proposed by Morel (1975) is used to determine the contour of the contraction, shown in Figure 5.2.

Air circulation is provided by two axial blowers (Model 9.5D12B by Woods) in series arrangement. The inverter-controlled blowers will be able to cater for a wide range of air velocity required during coil testing.

Two air mixers are installed at the exit of the test coils, as shown in Figures 5.1 and 5.3. The air mixer I serves to ensure thorough mixing of the outlet air before measurements are taken. The air mixer II serves to prevent radiative heat transfer to the measurement section from the heater element installed just after that section.

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The wind tunnel is tested to check for a uniform velocity and temperature distribution at the inlet of the test section using a hot-wire probe. A two-dimensional traveling mechanism is used to position the hot-wire probe. For the test carried out under various air velocities, it is noted that the readings of the hot wire are constant while the hot wire probe is moved around. It is believed that the designed contraction duct in the wind tunnel can meet the test requirement.

5.3 Air Property Control System

This system consists of two main components, a variable output steam humidifier and an electric heater. These two components will be able to control the air inlet thermodynamic properties to the required conditions.

The humidifier is an electric steam generator that injects steam through a distributor into the air tunnel to increase the air moisture content. A humidifier is also equipped with an air humidity sensor installed as the inlet of the test coil to control the steam injection rate automatically. It can also be manually regulated by an external power supply. The advantage of using a humidity sensor is that the steam injection rate is controlled according to the difference between the measured inlet air relative humidity of the test coil and the preset value. Thus, an external power supply is used to regulate the steam injection rate. A steam distributor is installed downstream the test coil in a closed circuit tunnel to ensure that the steam injected is fully absorbed by the flowing air.

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An electrical heater is installed just after the coil outlet measurement section of the wind tunnel to increase the air inlet temperature to the test coil. The heater power is controlled by a variable transformer that can be regulated to the required heating load.

5.4 Refrigeration Circuit

A refrigerant circuit has been designed for testing both condense and evaporator coils, as shown in Figure 5.3. The system is modified from the standard arrangement such that the test coil can be made to function as an evaporator coil or a condenser coil. This can be achieved by re-designing the refrigeration system circuitry to alter the flow path as required. For evaporator test, the shutoff valves v1, v3, v7 and v8 are opened; v2, v4, v5 and v6 are closed. While for condenser test, the shutoff valves v1, v3, v7 and v8 are closed; v2, v4, v5 and v6 are opened.

As the capacities of the test coils vary dramatically under different testing conditions, a main concern in the refrigeration system design is to match the system refrigerating capacity with the test coil load variation. A condensing unit with an open-type reciprocating compressor needs to be chosen due to its adaptability of a wide range of shaft speed. The compressor and the fan for the condenser are belt-driven by an inverter-controlled AC motor through a pulley system, whose speed can be regulated. Thus, the capacity of the compressor can be readily regulated to match with the coil load variation. Further, the fan of the evaporator coil is also controlled by an inverter during the condenser test.

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The needle-throttling valve installed in the liquid line serves as one of the means to regulate the condensing pressure of the system. This would vary the condensing temperature when the condenser coil is tested.

The auxiliary evaporator is used only when the test coil functions as a condenser coil since the original condenser coil on the condensing unit cannot be used as an evaporator. The auxiliary coil will bypass when the test coil functions as an evaporator coil.

The test coil is amounted in the test section normal to the horizontal air system. To validate the generality of the proposal model, several coils with different configurations are fabricated and tested as evaporators and condensers.

5.5 Instrumentation and Data Acquisition System

Conventional thermodynamics and fluid flow measurement techniques are used in this project. These include temperature, pressure, and flow rate measurements. The main instrumentation and computer-based data acquisition system are used in the test rig. The main components used in this system are described in this section.

5.5.1 Temperature Measurement

The refrigerant temperature measurements are conducted using K type thermocouples. In the calibration, the thermocouples are connected to the data acquisition system with the probes in a thermostatic bath, which is capable of maintaining a

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predetermined temperature within a very small tolerance and has a resolution of 0.05°C . The uncertainty of the temperature measurement after processing in the data acquisition system was estimated to be $\pm 0.2^{\circ}\text{C}$.

The air temperatures at the inlet and outlet of the test coils are measured using a Resistance Temperature Detector (RTD) probe that can be inserted into the air stream through measurement tapings at one side of the wind tunnel.

5.5.2 Humidity Measurement

The air humidity measurement is very critical in this project. A small reading error in the air relative humidity may have a relatively significant effect on the calculated coil capacity.

A microprocessor-based hygrometer unit is used in the test. The hygrometer is capable of measuring the air dew point, relative humidity and dry-bulb temperature with a high degree of accuracy. This unit consists of a control unit, a thermocouple probe (to measure the air-dry bulb temperature) and a dew point sensor with a built-in inert Rhodium mirror.

5.5.3 Flow Rate Measurement

A standard nozzle is installed in the wind tunnel to determine the air flow rate. The pressure difference between the upstream and downstream of the nozzle is measured using a manometer.

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Figure 5-1 Schematic diagram of the designed wind tunnel

Figure 5-2 Contour of the wind tunnel contracting duct

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Figure 5-3 Refrigeration circuit for heat exchanger test

Chapter 6

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6.1 Conclusions

New miniature bare tube heat exchangers, evaporator and condenser for miniature vapor compression cooling system have been introduced. Miniature bare tubes have been applied for the miniature heat exchangers. A new approach to simulated miniature bare tube heat exchangers with simple refrigerant circuitry has been developed. In contrast to the existing models in the literature, the proposed model leads a better insight to the processes taking place in the bare tube heat exchangers.

The main features of the model are summarized below:

1. A mathematical model has been developed to analyze the operational behavior of heat exchangers, which includes single and two-phase flow heat transfer. Matching with the uneven heat transfer characteristics of the heat exchangers, this distributed model is based on an arbitrarily small control volume, where the local information on the heat transfer and fluid flow can be provided for the performance evaluation in the miniature heat exchanger design.
2. Using the 1-D numerical model of the air flow, the sensible heat transfer and mass transfer on the air side are investigated separately for the bare tube heat exchangers.

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3. Based on the proposed computer simulation algorithm, the coil with arbitrary complex refrigerant circuitry can be simulated by the model.
4. The heat transfer characteristics and pressure drop behaviors have been developed. The model is then applied to design the coils of miniature evaporator and condenser. The model has been linked with an optimization algorithm to search for a combination of geometrical dimensions that produces optimum heat exchanger performance, i.e., maximum heat transfer at minimum pressure drop. In the optimization design of the miniature heat exchangers, the BOX Complex optimization method is used. A feasible miniature bare tube heat exchangers has been obtained.

6.2 Recommendations

This study has provided a sound basis for future simulation studies regarding the performance study of miniature bare tube evaporator and condenser coils. The following suggestions are made for future work in this field.

1. In the simulation study, the environment friendly working fluid, R134a is used. The experimental study of heat exchangers using R134a as refrigerant needs to be taken.
2. At a high Reynolds number, the boundary flow separation around the copper tubes results in the variation of heat transfer coefficient. Further study is needed to determine the function of the heat transfer coefficient that can be incorporated into the model to improve the model accuracy.

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3. The current study applied simple refrigerant circuitry for the miniature bare tube heat exchangers. Further study is required to investigate the different refrigerant circuitry coils for increasing the heat transfer capacity.

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