

Development and evaluation of an R-744 evaporator model

J. H. C. Potgieter
13088238

Dissertation submitted in partial fulfilment of the requirements for the degree
Magister in Mechanical Engineering at the Potchefstroom Campus of the North-West
University

Supervisor: Dr. M. van Eldik

September 2013

Abstract

In recent years carbon dioxide (CO₂, R-744) has moved to the foreground as an environmentally friendly alternative to commonly used CFCs and HFCs, which are being phased out due to its high ozone depleting and global warming potentials. R-744 is not only environmentally friendly but due to its unique properties, it is also ideally suited for the use in heat pump water heaters. High cycle efficiencies are achievable even at high hot water temperatures. The high cycle efficiency not only leads to energy and cost savings but also ties in with the drive for implementation of energy saving measures in South Africa. It is therefore paramount to continue development and implementation of R-744 in heat pump water heaters. Optimizing the cycle efficiency is only possible if detailed component simulation models, taking these unique properties of R-744 into account, are available.

The purpose of this study therefore was to develop a detail simulation model of a concentric tube-in-tube water-to-refrigerant evaporator, as well as a fin-and-tube air-to-refrigerant evaporator model.

Data from the North-West University R-744 heat pump test bench were used to verify the tube-in-tube evaporator simulation model. The discrepancies in the cooling capacity between the simulation and test bench can be attributed to the presence of lubricant in the system. The fin-and-tube model was verified by testing it against the NIST program EVAP-COND (NIST 2010). Overall there was good agreement between the results of the two programs, with EVAP-COND predicting a lower cooling capacity (6% to 14%) and a higher pressure refrigerant pressure drop (30% to 50%).

It was found that both the heat transfer correlation of Jung et al. (1989) and the pressure drop correlation of Choi et al. (1999) are able to predict the experimental values accurately and are valid for use in both the evaporator models developed.

To demonstrate the use of the detail evaporator fin-and-tube model, an evaluation of the different tube geometries, commercially available in South Africa, for use with R-744 fin-and-tube evaporators was done. For a fin-and-tube evaporator it was found that the most cost effective option is to use $\frac{3}{8}$ " (10.05 mm) copper tubes and the least effective is $\frac{1}{2}$ " (12.6 mm) stainless steel tubes.

Keywords: R-744; Carbon Dioxide; Evaporator; Heat pump; Heat transfer correlations; Pressure drop correlations; Simulation

Contents

List of Figures	v
List of Tables	vi
Nomenclature	vii
1 Introduction	1
1.1 Background	1
1.2 Objectives of this study	5
1.3 Method of Investigation	6
2 Literature survey	7
2.1 Heat pump water heating overview	7
2.2 Properties of carbon dioxide	10
2.2.1 Safety of R-744	11
2.2.2 Transcritical cycle overview	12
2.3 South African conditions	16
2.3.1 Ambient temperature range	16
2.3.2 Water supply temperature	17
2.3.3 Evaporation manufacturing technology	18
2.4 Evaporator modelling	21
2.4.1 Evaporator design pressures	21
2.4.2 Pressure drop and heat transfer considerations	24
2.4.3 Heat exchanger models	25
2.4.4 Fouling, contact and tube wall resistances	29
2.4.5 Water side correlations	30
2.4.6 Refrigerant side correlations	31
2.4.7 Simultaneous heat and mass transfer	37

2.4.8	Fin side correlations	38
2.4.9	Heat conduction between tubes	40
2.5	Summary	41
3	Theoretical background	42
3.1	Conservation equations - Tube side	42
3.1.1	Conservation of mass	43
3.1.2	Conservation of momentum	43
3.1.3	Conservation of energy	44
3.2	Conservation equations - fin side	44
3.2.1	Conservation of mass	45
3.2.2	Conservation of momentum	46
3.2.3	Conservation of energy	46
3.3	Element pressure drop	47
3.3.1	Single phase pressure drop	47
3.3.2	Two phase pressure drop	49
3.4	Element heat transfer	50
3.4.1	Overall heat transfer coefficient	51
3.4.2	Equivalent dry-bulb temperature method	52
3.4.3	Surface efficiency	54
3.4.4	Single phase heat transfer	55
3.4.5	Two phase heat transfer	55
3.4.6	Air side heat transfer correlations	57
3.4.7	Air side pressure drop	57
3.5	Summary	58
4	Simulation overview	59
4.1	Tube-in-tube evaporator	59
4.2	Finned tube evaporator	62
4.3	Summary	67
5	Evaporator model verification	68
5.1	Tube-in-tube heat exchanger	68
5.1.1	Evaporator description	68
5.1.2	Data reduction	69
5.1.3	Simulation results	71

5.1.4	Effect of number of elements	72
5.1.5	Heat transfer discussion	74
5.1.6	Pressure drop discussion	80
5.1.7	Conclusion	81
5.2	Fin-and-tube heat exchanger	82
5.2.1	Evaporator geometry	83
5.2.2	Evaporator operating conditions	83
5.2.3	Simulation results	85
5.2.4	Effect of number of elements	86
5.2.5	Conclusion	86
5.3	Coil geometry evaluation	87
5.3.1	Simulation inputs	87
5.3.2	Simulation Results	88
5.4	Summary	88
6	Conclusion	91
6.1	Tube-in-tube evaporator	91
6.2	Fin-and-tube evaporator	91
6.3	Evaluate coil geometries	92
6.4	Future studies	92
7	Tube-in-Tube Simulation Program	102
8	Finned Tube Simulation Program	113

List of Figures

1.1	Percentage use of main primary refrigerants in existing marine cargo installations classed by Lloyd's Register (Kim et al. 2004).	2
1.2	Number of papers on CO ₂ as a primary refrigerant presented at the IIR-Gustav Lorentzen Conference on Natural Working Fluids (Kim et al. 2004).	2
2.1	South African industrial heat pump installation utilizing R407c (van Eldik 2012).	8
2.2	R-744 heat pump for water heating, system and temperature-entropy diagram. (Kim et al. 2005)	13
2.3	R-134a heat pump with subcooler, condenser and desuperheater (Reulens 2009).	14
2.4	R-744 heat pump with single counter-flow gas cooler and optimised gas cooler pressure (Reulens 2009).	14
2.5	Monthly average cold water supply temperatures for the main metropolitan areas (Meyer & Greyvenstein 1992).	18
2.6	Typical fin-and-tube evaporator with hydrophilic coated wavy fins. (Courtesy of Booyco Engineering)	19
2.7	Discretization of heat exchanger tubes (Iu 2007).	26
3.1	Control volume - tube side	43
3.2	control volume, fin side	45
3.3	Cooling and dehumidifying of moist air over a flat plate (Wang & Hihara 2003)	53
4.1	Tube-in-tube evaporator overview.	59
4.2	Tube-in-tube element detail.	61
4.3	Finned tube evaporator overview.	63
4.4	Finned tube element detail (Ding et al. 2011).	65
4.5	Element airflow handling.	66

5.1	Temperature and pressure measurement points for the evaporator test bench at the North West University.	69
5.2	Results of simulated and experimental R-744 evaporator capacities.	71
5.3	Results of simulated and experimental R-744 pressure drop.	72
5.4	Capacity vs number of elements.	73
5.5	Pressure drop vs number of elements.	73
5.6	Heat transfer coefficients for different heat fluxes with $G = 800 \text{ kg/m}^2\text{s}$, $T_{sat} = 8$, $D_{eq} = 15.7\text{mm}$	76
5.7	Heat transfer coefficients for different tube diameters with $G = 800 \text{ kg/m}^2\text{s}$, $T_{sat} = 8$ and $q_{flux} = 80\text{kW/m}^2$	76
5.8	The effect of lubricant and heat flux on the heat transfer coefficient of R-744 (Wang et al. 2011).	77
5.9	Simulated capacity with heat transfer adjusted for the presence of lubricant. .	78
5.10	Detail temperature distribution for the refrigerant through the evaporator - Run 7.	79
5.11	Detail temperature distribution for the refrigerant through the evaporator - Run 10.	79
5.12	Simulated pressure drop vs calculated pressure drop.	80
5.13	Simulated pressure drop vs the experimental data of Bredensen et al. as presented by Cheng et al. (2008a) with $G = 400 \text{ kg/m}^2\text{s}$, $T_{sat} = -10$, $D_{eq} = 7\text{mm}$ and $q_{flux} = 9\text{kW/m}^2$	82
5.14	Cooling capacity comparison between the EES fin-and-tube simulation model and EVAP-COND.	85
5.15	Pressure drop comparison for R-744 between the EES fin-and-tube simulation model and EVAP-COND.	86
5.16	Coil weight vs Cooling capacity for different tube options.	89
5.17	Fan power vs Cooling capacity for different tube options.	89

List of Tables

2.1	Characteristics of refrigerants commonly used in heat pump applications. . .	10
2.2	Typical design temperatures and pressures for several large cities in South Africa.	16
2.3	Working pressures for different tube geometries as used in fin-and-tube heat exchangers.	20
2.4	Schedule 40 steel pipe working pressure (<i>ASHRAE Handbook, Systems and Equipment</i> 2000).	21
2.5	Saturated pressure for R-744 at different temperatures during evaporation. .	22
2.6	Optimum discharge temperature for various hot water temperatures by White et al. (2002).	22
2.7	Fin side correlation overview.	39
5.1	Datasets used from the tube-in-tube evaporator test bench of the North West University.	70
5.2	Maximum simulated heat flux for each run.	75
5.3	Fin-and-tube evaporator simulation operating conditions.	84

Nomenclature

A	area, m ²	h_{lv}	latent heat of vaporization, J/kg
A_c	cross sectional area, m ²	k	thermal conductivity, W/m·K
A_{ff}	free flow area, m ²	L	length, m
A_{fr}	frontal area, m ²	$LMTD$	Log Mean Temperature Difference
A_s	surface area, m ²	M	number of nodes, tubes, increments
A_{tot}	total surface area, m ²	m	mass, kg
Bo	Boiling number	$\dot{m}$	mass flow rate, kg/s
C	total heat capacity, J/kg·K	N	number of nodes, tubes, increments
c	specific heat capacity, J/kg·K	NTU	number of transfer units
c_p	specific heat capacity at constant pressure, J/kg·K	Nu	Nusselt number
c_v	specific heat capacity at constant volume, J/kg·K	p	pressure, Pa
D	diameter, m	P_w	wetted perimeter, m
D_e	Effective Diameter, m	Pr	Prandtl number
D_h	Hydraulic Diameter, m	Q	total energy transfer, J
e	surface relative roughness, m	$\dot{q}$	heat transfer rate, W
f	friction factor	R_{fin}	thermal resistance of a fin, K/W
FPI	fins per inch	R_f	fouling factor, m ² ·K/W
G	mass flux, kg/m ² ·s	R_{tot}	thermal resistance of finned surface, K/W
g	gravitational acceleration, m/s ²	R_t	thermal resistance, K/W
h	enthalpy, J/kg	Re	Reynolds number
h_c	convection heat transfer coefficient, W/m ² ·K	RH	relative humidity, %
		S_x	Longitudinal tube row pitch, m

Nomenclature

S_y	Transverse tube pitch, m	η_f	fin efficiency
T	temperature, K	η_o	overall finned surface efficiency
t	time, s	μ	viscosity, N·s/m ²
U	overall heat transfer coefficient, W/m ² ·K	π	3.146
V	volume, m ³ /s	ρ	density, kg/m ³
v	specific volume, m ³ /kg	σ	surface tension, N/m
$\dot{V}$	volume flow rate, m ³ /s		
w	specific humidity, kg _w /kg _a	<i>Subscripts</i>	
$\dot{W}$	rate of work transfer, W	a	dry air
W_t	wall thickness, m	e	exit, outlet
X	quality	h	homogeneous
X_{tt}	Lockhart Martinelli parameter	i	inlet
<i>Greek symbols</i>		i	inner
Δ_{fric}	frictional pressure drop, Pa	l	saturated liquid
Δ_{mom}	momentum pressure drop, Pa	m	mean value, position
Δ_{stat}	static pressure drop, Pa	o	outer
Δ_{tot}	total pressure drop, Pa	r	refrigerant
ϵ	heat exchanger effectiveness	v	vapour
η	efficiency	w	water

Chapter 1

Introduction

1.1 Background

Around the end of the 19th century carbon dioxide and ammonia were the most extensively used refrigerants. Since ammonia is both poisonous and flammable R-744 (carbon dioxide) was the refrigerant of choice where safety was essential. As a result, applications included air conditioning and refrigeration on ships, in hospitals and restaurants.

The rapid dwindling of R-744 in favour of chlorofluorocarbon(CFCs) refrigerants, such as R-12 from the early 1930's cannot be attributed to a single factor but rather a number of factors. One of the main contributing factors was the rapid loss in the capacity and coefficient of performance (COP) for a carbon dioxide refrigeration cycle with an increase in the cooling fluid temperature (Lorentzen 1995, Kim et al. 2004). This problem was further aggravated in marine applications, in tropical areas, where the required cooling on-board ships was high, with the cooling water temperature also very high. This also led to inefficient cycles where air cooling was implemented. The aggressive marketing of CFCs as safe refrigerants, led to the replacement of not only ammonia but also R-744 as an refrigerant.

For example, in the field of marine refrigeration carbon dioxide dominated as an refrigerant until the 1950's when it was overtaken by R-12 and R-22 as the refrigerants of choice (Nekså et al. 1998). This trend can clearly be seen from Figure 1.1.

Today R-744 is once again under consideration due to its limited environmental impact. Commonly used CFCs and HFCs are currently being phased out due to their

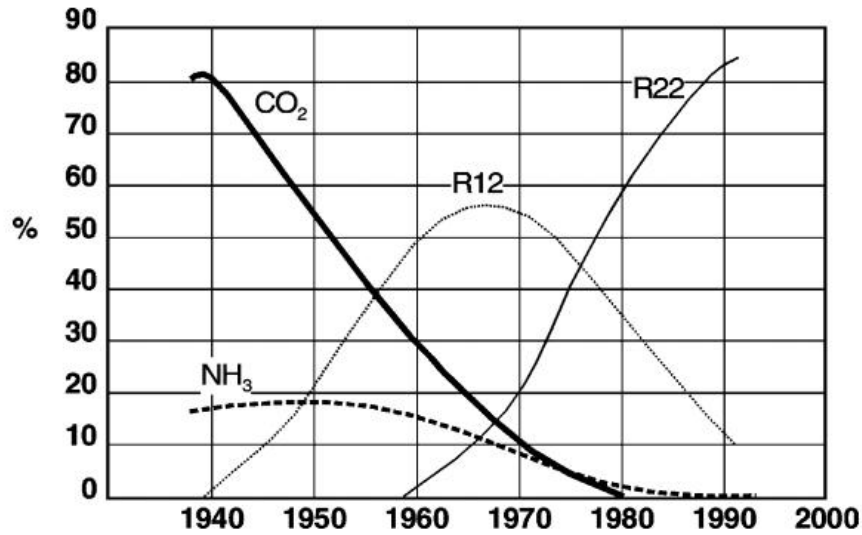


Figure 1.1: Percentage use of main primary refrigerants in existing marine cargo installations classed by Lloyd's Register (Kim et al. 2004).

high ozone depletion potential (ODP) (Mon 2000), or their high global warming potential (GWP) (Kyo 1998). Prof Gustav Lorentzen from Norway was proposing R-744 as an alternative refrigerant since the late eighties (Cecchinato et al. 2005). Others soon followed in his footsteps and since the 1990's there was a surge in research papers on this topic. Referring to Figure 1.2, it can be seen that the number of papers on carbon dioxide as a primary refrigerant at the IIR-Gustav Lorentzen Conference on Natural working fluids increased drastically in the period from 1994 to 2002.

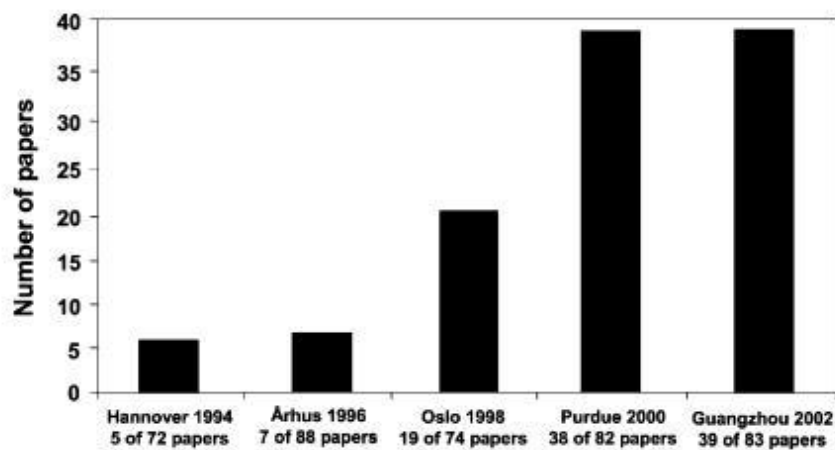


Figure 1.2: Number of papers on CO₂ as a primary refrigerant presented at the IIR-Gustav Lorentzen Conference on Natural Working Fluids (Kim et al. 2004).

The soaring popularity of R-744 in the academic circles is mirrored in commercial

applications and this can be seen on the internet. Dedicated websites such as www.R-744.com, provide a hub off information, not only for the academic but even more so for the technically inclined. Aside from these, perhaps the best indicator of the increased popularity of R-744 as working fluid for water heating heat pumps can be found in Japan. By October 2009 the cumulative number of EcoCute¹ heat pumps shipped was in excess of 2 million units (*Eco Cute sales on the rise* 2011). This is a major milestone and an indication of the success with which CO₂ heat pump water heating can be implemented. Implementation of R-744 however is not limited to heat pumps and the use of R-744 is widely studied for applications as diverse as vehicle air conditioners, supermarket refrigeration and cold rooms(Reulens 2009).

It is commonly accepted that by utilizing conventional heat pumps for water heating instead of direct electrical heating typical energy savings in the range of 66% are possible (Rankin et al. 2004, Rankin & van Eldik 2008, *Integrated Demand Management* 2012). Firstly these electrical energy savings translate into cost savings. Secondly the use of less energy translate into a reduction in carbon dioxide emissions and a reduced carbon footprint for individuals or companies.

In recent years, the implementation of energy saving measures has become a national priority in South Africa. Power management programs have been launched by Eskom to reduce the overall use of electricity in households, the commercial sector and industry (*Integrated Demand Management* 2012). A key aspect of this program is the promotion and implementation of energy efficient technologies. This includes but are not limited to the implementation of solar panels and heat pumps for water heating. Rankin et al. (2004) found that energy savings in the range of 61% to 71% were possible for commercial heat pump installations, implementing the correct layout. Furthermore, a reduction of 86% in the peak electricity demand was found for water heating. In the residential sector where a typical household utilize 30-50% of their total electricity consumption for hot water heating, heat pumps have been shown to have a payback period of between 2.3 - 4.7 years (Rankin & van Eldik 2008). This combined with the recent and projected power price hikes by Eskom, will further pave the way for greater implementation of heat pumps.

The implementation of R-744 as a refrigerant not only complies with the demand

¹EcoCute is a generic term for a range of domestic heat pumps manufactured in Japan.

for natural refrigerants but also help mitigate the energy crisis in South Africa. It is therefore paramount to continue development and implementation of R-744 in heat pump water heaters.

Due to the unique properties of R-744, high cycle efficiencies are possible for heat pumps (Lorentzen 1995, Neksa et al. 1998). Optimizing the cycle efficiency is only possible if detailed component models, taking these unique properties of R-744 into account, are developed (Reulens 2009, Mastrullo et al. 2010).

The main components for a heat pump system utilizing R-744 as operating fluid are:

- Compressor
- Gas cooler
- Expansion valve
- Evaporator
- Gas pre heater
- Receiver

Specially designed compressors, which are able to handle the high operational pressure of R-744, are required (Kim et al. 2004). As a result of many years of research, more and more compressors are being developed and refined by several manufacturers. The EcoCute heat pump, for example, includes compressors from Denso, Sanyo, Daikin, Matsushita and Hitachi (Reulens 2009).

According to Kim et al. (2004) several studies are also being done on gas coolers and gas heat exchangers for R-744. Research in this field is still an ongoing process, with a focus on establishing correlations for the Nusselt number for supercritical R-744 heat transfer. One such instance is the recent study by Venter (2010).

Several correlations are available in literature to calculate the heat transfer coefficients and pressure drop of evaporating refrigerants and R-744 in particular. These are discussed in detail in Chapter 2. With this said, limited information is available on completed fin-and-tube and tube-in-tube evaporator models.. For example, Kasap et al. (2011) did not divulge any detail about the correlations and methodology used

in their evaporator model. In the case of a system simulation used for comparative analyses the accuracy of the evaporator model was not evaluated in detail (Brown et al. 2002). In other cases where R-744 heat pump water heaters were tested experimentally, the heat source for the evaporators was glycol instead of air (Neksa et al. 1998, White et al. 2002).

Most development in heat pump technology has taken place in Europe, Japan and North-America. Naturally the focus are more orientated to their local conditions. Thus, the expected operational air temperature and municipal water supply temperature evaluated are lower than those in South Africa.

The purpose of this project therefore is to develop a detailed water-to-refrigerant simulation model (concentric tube-in-tube heat exchanger) as well as a detailed air-to-refrigerant (fin-and-tube heat exchanger) simulation model for the evaporator with special emphasis on application in South Africa. In the future these models can then be used to optimize the design of a R-744 evaporator used in hot water heat pumps.

1.2 Objectives of this study

The objectives of this study are as follows:

- Identify the operational conditions and constraints for an R-744 evaporator for South African conditions. This includes but are not limited to operational temperatures and local technology available for manufacturing evaporators.
- Evaluate the suitability of the different pressure drop and heat transfer correlations applicable for R-744 evaporation.
- Develop a detailed tube-in-tube evaporator simulation model.
- Verify the applicability of the correlations using the aforementioned simulation model and test data obtained from studies conducted at the North West University.
- Develop a detailed fin-and-tube evaporator simulation model.
- Validate the model using data from the literature as well as the results from other simulations programs.

- Use the fin-and-tube simulation model developed to evaluate the different local manufacturing options.

1.3 Method of Investigation

A literature survey is presented in chapter 2. In the survey the properties of R-744 in general were investigated as well as evaporator specific design constraints. The different correlations for pressure drop and heat transfer for both evaporating R-744 and the air-side, was investigated.

Chapter 3 presents the theoretical background for the models and establish the mathematical basis and correlations used in the development of both simulation models.

In order to simulate the heat exchangers, an elemental approach is followed. The methodology of how these models are implemented, using EES (Engineering Equation Solver, Klein (2011)), are discussed in Chapter 4.

In Chapter 5 the applicability of the chosen R-744 correlations are evaluated using the tube-in-tube simulation model and measured data. The complete fin-and-tube simulation model is verified against another simulation program. Using the fin-and-tube simulation model the different local coil manufacturing options are then investigated.

Chapter 6 provides a summary of the study, conclusions and suggestions for future work.

Chapter 2

Literature survey

This chapter will concentrate on the available literature on the design of a R-744 evaporator. This is not limited to the heat transfer and pressure drop correlations of R-744, but also extend to an overview of the complete heat pump system to determine the operational conditions for the evaporator.

2.1 Heat pump water heating overview

Heat pumps heat water by transferring energy from the ambient air to water, using a refrigeration cycle. Heat pumps generally have a COP in the order of 2 to 3 (Hepbasli & Kalinci 2009). This means that for a input of 1 kW electrical energy a heat pump can typically supply 2 to 3 kW of heating, compared to the 1 kW heating delivered by a resistance heater, for the same power input. Underfloor water heating, which is common in Europe, are rarely implemented in South Africa. Instead, heat pumps are typically used to provide higher temperature hot water for sanitary use. To date, heat pumps in South Africa have seen their greatest application in the commercial sector (hotels, hostels etc.) (Rankin & van Eldik 2008), where water heating is the fourth largest energy consumer. Figure 2.1 shows a typical industrial scale heat pump installation that would be found in South Africa. Another potential application source for heat pumps is in industrial processes, which require enormous amounts of process heat. Of this, about a quarter is low temperature process heat at a temperature of less than 130° (Reulens 2009).



Figure 2.1: South African industrial heat pump installation utilizing R407c (van Eldik 2012).

Neksà et al. (1998) reported on a 50 kW prototype R-744 heat pump system and found that for constant evaporation at 0°C , inlet water temperature of 8°C and hot water temperature of 60°C the COP of the system was 4.3. Increasing the hot water supply temperature to 80°C only reduced the COP slightly to a value of 3.6. Neksà et al. (1998) further pointed out that it is possible to increase the water temperature to 90°C without any operational problems. The expected COP at 90°C was not reported but from the data it can be estimated to be in the order of 3. It must be noted that this is for a very low water supply temperature. Increasing the supply water temperature to 20°C reduced the COP from 4.3 to 3.9 for a hot water supply temperature of 65°C .

White et al. (2002) constructed and tested a prototype heat pump to supply hot water at temperatures higher than 65°C while simultaneously providing refrigeration at 2°C and less. A simulation program was then developed based on the component performances in order to allow for parametric studies. For the prototype system supplying hot water at 65°C at an evaporation temperature of -6.4°C the COP was 3.12. Using the system simulation a COP of 2.46 is predicted for 120°C hot water. The water inlet temperature was not specified.

More recently Yamaguchi et al. (2011) experimented with a commercially available industrial model R-744 heat pump supplying hot water at a temperature of 90°C . Unlike the above units this was not a prototype but an existing commercial product. It was tested for cold water inlet temperatures ranging from 10°C to 40°C and air inlet

temperatures ranging from 13°C to 28°C. At 10°C inlet water temperature and 16°C air temperature the unit had a COP of 3.55. By increasing the water temperature to 40°C the COP was reduced to 2.6. If the water inlet temperature is kept constant at 20°C and the air temperature increase from 13°C to 28°C the COP increase from 3.2 to 3.6.

R-744 is not only able to provide high water temperatures, but also compares favourably with traditional heat pumps. Cecchinato et al. (2005) used a simulation study to compare a heat pump using R-744 with a heat pump using R-134a for supplying tap hot water. The operational conditions were varied to simulate a variety of conditions while the hot water supply temperature was kept steady at 45°C. It was found that the COPs of the systems were similar for typical winter conditions and that the R-744 heat pump outperformed the R-134a heat pump in summer conditions. This was for the case where full stratification took place, confirming that low water supply temperatures are required for high efficiency.

The Sanoy Eco Cute heat-pump was tested under a range of conditions, typically encountered in Sweden. This Swedish version is intended to supply both tap water and hot water for space heating. The unit also included a defrost cycle to enable it to operate at temperatures well below freezing. Chen et al. (n.d.) confirmed that the unit delivered the COPs specified in the technical manual. The COP was confirmed at 4.1 for an outlet water temperature of 50°C, inlet water temperature of 30°C and ambient temperature of 25°C. For the same water temperatures and an ambient of 7°C the COP is reduced to 3.1. In this study they also confirmed that the efficiency is reduced if the water inlet temperature increases or the ambient temperature decreases.

From the above studies the following trend can be observed for R-744 heat pumps with regards to the system COP:

- An increase in hot water supply temperature leads to a reduction in COP.
- Increasing the evaporation pressure leads to an increase in COP.
- Increasing the inlet water temperature reduces the COP of the system.

As discussed in Chapter 1, the rising energy costs are promoting the implementation of energy efficient solutions such as heat pumps in the residential, commercial and industrial sectors. R-744 heat pump systems are an attractive option as they can

deliver hot water at temperatures of up to 90° with high efficiency and compare very favourably with traditional heat pumps at lower temperatures. R-744 is therefore the ideal contender for the supply of hot water.

2.2 Properties of carbon dioxide

The properties of R-744 are very different to other refrigerants in general use. Table 2.1 provides an overview of the differences in properties of R-744 and other refrigerants. Until recently R22 was the most widely used gas in heat pump applications. Due to its systematic phase out this is no longer the case (Kyo 1998). It is however still included in the table to give an overview of the fluids used historically and the current range of fluids.

Table 2.1: Characteristics of refrigerants commonly used in heat pump applications.

	R-22	R-134A	R-407C	R-410A	R-744
Ozone Depletion Potential	0.04	0	0	0	0
Global Warming Potential	1500	1300	1530	1730	1
Natural refrigerant	No	No	No	No	Yes
Atmospheric life in years	12.1	14.6	<32.6	<32.6	N/A
Density ratio - liquid to gas - at 0°C	60.3	89.7	65	38.4	9.5
Critical pressure [kPa]	4989	4059	4597	4925	7377
Critical temperature [°C]	96.13	101	86.79	72.13	30.98
Volumetric refrigerant capacity [0°C]	4353	2868	3992	6833	22545
Maximum hot water temperature [°C]	55-60	60-65	55-60	55-60	80-100
Relative price per kg (South Africa)	1.0	2.3	2.9	3.0	0.58
First commercial use as refrigerant	1936	1990	1998	1998	1869
Phase out date	2020	TBD	TBD	TBD	N/A
Manufactured in South Africa	No	No	No	No	Yes

Although R-744 is a greenhouse gas, it has the lowest GWP of all the refrigerants that are currently in use. According to Lorentzen (1995) R-744 as a refrigerant has the following advantages:

- Reduced pressure ratio leading to smaller pressure differences over the compressor.
- Available globally without any supply monopoly. In South Africa both of the major gas suppliers (Afrox and Air Liquid) can supply R-744. As seen in Table 2.1 R-744 is also the most cost effective option.
- No need to be recycled as with other refrigerants. Greenhouse gasses needs to be recycled and leakages monitored. This adds additional logistics cost, increasing the life cycle cost of the heat pump.
- No requirement for special lubricants or materials required.
- Negligible global warming potential.
- No ozone depletion potential.

The main disadvantage of R-744 is its higher operating pressure. As indicated by Kim et al. (2004) and Pettersen et al. (1998) the operational pressure for R-744 is up to 10 times higher than the other refrigerants currently in use. This places restraint on the use of current commercially available compressors and components. In this study, the use of commercially available evaporators will be fully investigated.

2.2.1 Safety of R-744

Although this study will not focus on these aspects it is of value to investigate the safety aspects of R-744 and the ramifications this may have on the evaporator design.

There are four areas of safety concerns for any refrigerant:

- Toxicity
- Flammability
- Explosion hazard
- Bursting hazard

In terms of flammability R-744 is an ideal refrigerant as it is not only non-flammable but serves as a fire inhibitor. R-744 is also non-explosive in any ratio with air.

As the R-744 trans-critical cycle operates at such a high pressure accidental pipe bursting is perceived as a safety problem (Kim et al. 2004). Although R-744 operates at very high pressures, the system volume very small. The total energy released by a instantaneous accidental release is approximately the same for systems of the same capacity regardless of the refrigerant used (Lorentzen 1995).

From a toxicity perspective, one of the disadvantages of R-744 is that it can lead to death in concentrations of more than 10000 ppm in air. The normal level for R-744 in air is about 400 ppm. Air with concentrations of 1200 ppm of R-744 or more is usually perceived to be of poor quality and can lead to reduced concentration and headaches in human operators. Kim et al. (2004) refers to a Figure of 5% to be the upper limit for the concentration of R-744 in the air after accidental release, and a limit of 2% for slow release through a gas leak. This is to ensure that the reactions of maintenance personnel or machinery operators are not degraded. As a rule South African heat pumps are installed outside or in very well ventilate areas. From the afore-mentioned it follows logically that in almost all instances the accidental release of R-744 will not present any danger.

However, since there are some danger, safety measures must be implemented in the design of the system to prevent leakage and or pipe bursting. This will require any evaporator design to withstand not only the operational pressure, but any design pressures as specified by legislation.

2.2.2 Transcritical cycle overview

In a traditional system, the high side pressure is determined by the saturated pressure of the condensing fluid and the low side pressure by the saturated pressure of the evaporating fluid. The condensing and evaporating temperatures (and therefore pressures) typically depended on the heat exchanger efficiency and the temperature of the external fluid. Since the critical temperature of R-744 is 31.06°C heat rejection for water heating takes place in the supercritical region. Figure 2.2 provide a system and thermodynamic overview for a typical R-744 heat pump water heater.

Heat rejection takes place in the supercritical, high pressure part of the cycle. Heat absorption takes place in the subcritical, low pressure part of the cycle. Heat is rejected to the water, in the gas cooler with a temperature glide, while the R-744 change in

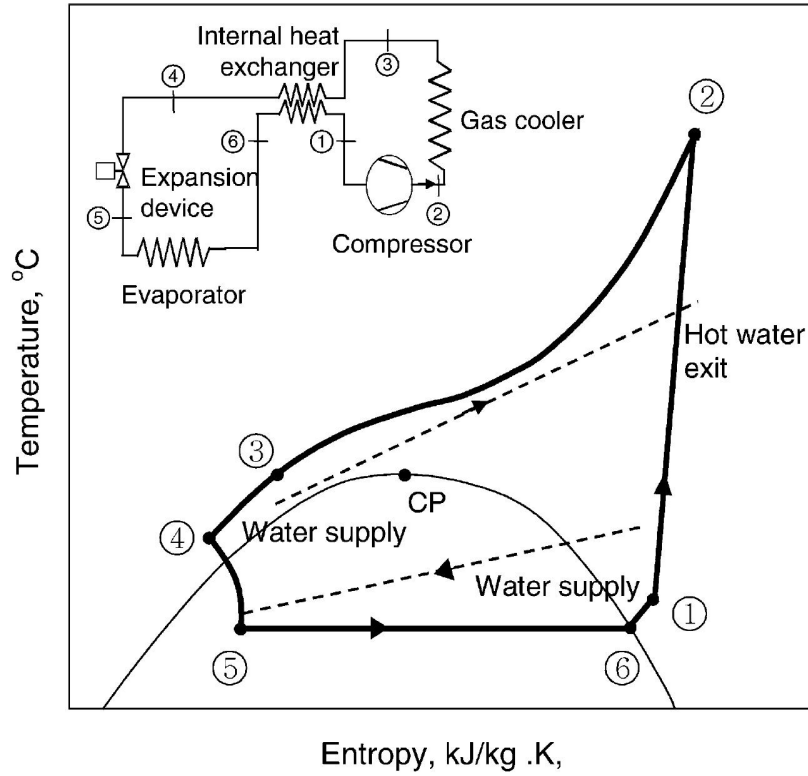


Figure 2.2: R-744 heat pump for water heating, system and temperature-entropy diagram.
(Kim et al. 2005)

density from a superheated vapour to an almost liquid like state gas (Nekså et al. 1998).

Figure 2.3 shows a traditional R-134a heat pump system with subcooler, condenser and desuperheater. Figure 2.4 shows a similar heat pump system utilizing R-744 as refrigerant. As seen from Figure 2.4, the temperature glide of R-744 can be matched very closely with the temperature glide required for water heating. In the R-134a heat pump system the maximum hot water temperature is limited by the condensing temperature, with only slight temperature increases above condensing temperature being possible, using super heating.

For R-744 the temperature of the water exiting the gas cooler is controlled by varying the water flow rate through the gas cooler. Each set of operational conditions has an optimum gas cooler operational pressure, which result in the maximum COP. As mentioned in Section 2.1, White et al. (2002) measured performance of a prototype for discharge pressures between 90 and 130 bar. The optimum discharge pressure tends to increase with an increase in required hot water temperature. Thus, in order to ensure optimum COP throughout the operation, the heat pump control system need to adjust the pressure inside the gas cooler to account for any change in the boundary conditions.

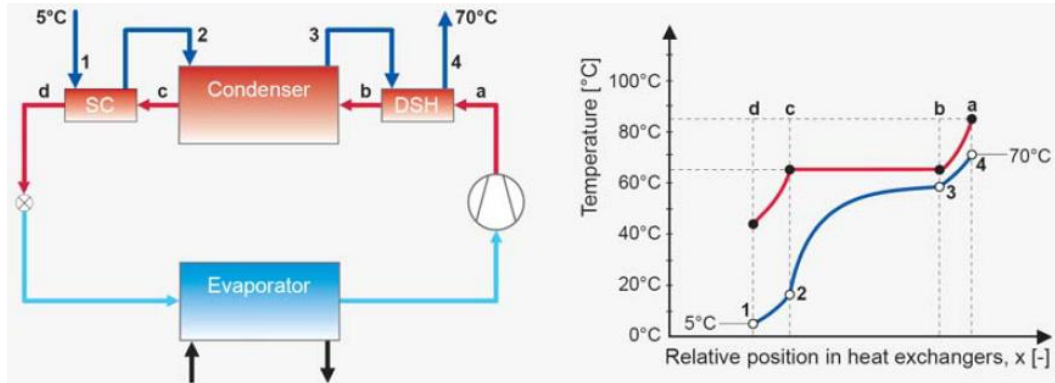


Figure 2.3: R-134a heat pump with subcooler, condenser and desuperheater (Reulens 2009).

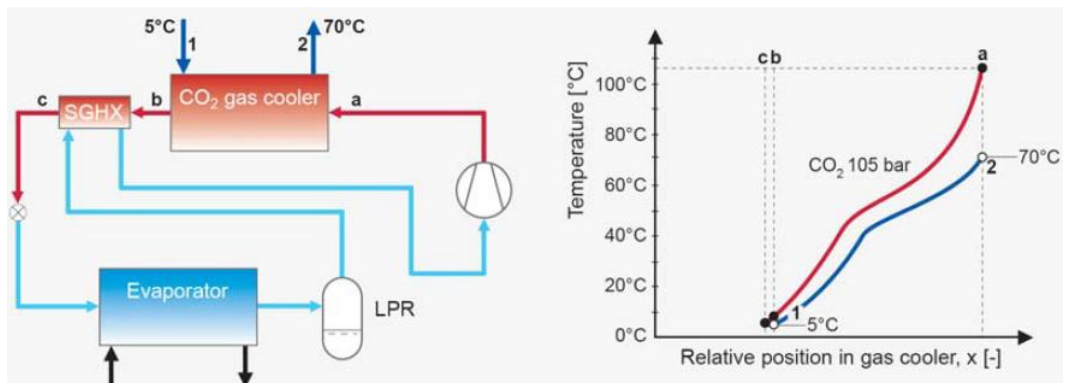


Figure 2.4: R-744 heat pump with single counter-flow gas cooler and optimised gas cooler pressure (Reulens 2009).

A liquid accumulator/receiver is used as a buffer, to supply or absorb the R-744 when the working pressure of the system is changed to a new optimum level. Neksa et al. (1998) and White et al. (2002) placed an accumulator/receiver directly after the expansion valve, allowing the two-phase mixture to collect in the accumulator and only vapour to continue to the compressor. The R-744 is circulated by pump through a separate evaporator, usually plate fin or other proven high pressure evaporator designs. In the case of a small gas leak the accumulator will also serve to prevent a reduction in the COP due to the refrigerant loss.

The expansion device for an R-744 system has several different roles. Unlike a traditional system where the expansion device only maintain the superheat, the expansion device in a R-744 system has to maintain an optimum gas cooler operational pressure as well (Neksa et al. 1998).

Although most heat pump systems implement an accumulator other configurations are also possible. A system without an accumulator and active charge management was tested by Fernandez et al. (2010).

Most systems also use an internal heat exchanger to increase the COP of the cycle. The internal heat exchanger is used to lower the temperature of the high pressure R-744 exiting the gas cooler, using low temperature R-744 vapour exiting the evaporator. Kim et al. (2005) and Fernandez et al. (2010) reported an improvement in efficiency when an internal heat exchanger is used. Both Figure 2.2 and Figure 2.4 show this configuration.

When an internal heat exchanger is used after the evaporator, superheating of the refrigerant vapour is not necessary as superheating will take place in the internal heat exchanger and the accumulator will prevent the transfer of liquid to the compressor. Where these components are not present the refrigerant vapour will need to be superheated in the evaporator to ensure that no liquid R-744 enters the compressor. Typically a value of at least 5°C to a maximum of 10°C is advised.

2.3 South African conditions

As part of this study it is necessary to take local conditions into account. This includes expected operational temperatures (air temperatures and municipal water supply temperature) as well local manufacturing capabilities.

2.3.1 Ambient temperature range

It is important to keep in mind that for an evaporator to function, a temperature difference are required between the evaporating fluid and the heat source. In this case between R-744 and air. Figure 2.2 gives an overview of the maximum and minimum temperatures for several big cities in South Africa. This data is based on the ASHRAE climatic data for South Africa (*ASHRAE Handbook, Fundamentals* 2001), and is commonly used for the design of HVAC equipment. The 0.4% value, given in the table are typically exceeded only 35 hours per year. The minimum or maximum values are exceeded, on rare occasions, by 0.5°C.

Table 2.2: Typical design temperatures and pressures for several large cities in South Africa.

City	Winter			Summer	
	Standard Air	Coldest 0.4% Days [°C]	Minimum Temp [°C]	Warmest	
	Pressure [kPa]			0.4% Days [°C]	Maximum Temp[°C]
Bloemfontein	86.15	-3.5	-7.3	34	38
Cape Town	100.82	3.6	0.5	30.3	36.1
Durban	101.23	10	6.5	30.3	35.2
Johannesburg	82.5	1	-3.3	29	32.6
Port Elizabeth	100.61	6.3	2.3	29.2	38
Pretoria	86.42	3.9	0.5	31.9	36.3

Of all the major cities in South Africa only Bloemfontein experiences temperatures below 0°C for more then 35 hours yearly. Heat pumps operational in South Africa will therefore only rarely be expected to operate in temperatures below 0°C.

For a tube-in-tube evaporator the operating temperatures are limited to a similar range. Water freezes at 0°C and the practical limit for the water supply temperature is

normally 2°C. Using brine solutions even lower temperatures can be achieved. White et al. (2002) tested a heat pump designed to provide refrigeration at -5°C while heating water from 20°C to 90°C. They tested for a minimum evaporation temperature of -6.4°C using water as the secondary fluid. Neksa et al. (1998) kept the evaporation temperature constant at 0°C while using a brine solution as the secondary fluid.

Depending on heat exchanger efficiencies a temperature difference of at least 5° to 10° is required between the refrigerant temperature in the evaporator and the ambient air or secondary fluid temperature. An evaporator will therefore evaporate at temperatures in the range of -15° minimum to 25° maximum. This temperature range will be kept under consideration for selection of the most appropriate correlations.

2.3.2 Water supply temperature

As previously seen the inlet water temperature influences the efficiency of a R-744 heat pump cycle. Neksa et al. (1998) used a constant cold water supply temperature of 8°C as an typical average cold water supply temperature for Norway. Rankin & van Eldik (2008) assumed a constant cold water temperature of 14°C, while being conservative this was an adequate assumption. Meyer & Greyvenstein (1992) in one of the early studies on using heat pumps for water heating in South Africa, used the assumption that the cold supply water temperature are equal to the ground temperature at a depth of 1.2 meter. Based on this they calculated a cold water temperature for each month of the year for the main metropolitan areas. This is represented in Figure 2.5.

For the gascooler, the inlet temperature will only be the same as the cold water supply temperature if complete stratification of the water levels occurs in the hot water storage tank. A good example is the modified Eco cute system for the Swedish conditions. Due to excessive mixing in hot water storage tank Chen et al. (n.d.) measured inlet water temperatures in the range of 30°C to 40°C. Yamaguchi et al. (2011) tested an industrial heat pump system for inlet temperatures from 10°C to 40°C.

If good stratification is achieved (as it must in order to obtain high system COP's) the expected cold water inlet temperature of the gas cooler will not typically exceed 30°C. For the purposes of this study it will be assumed that good stratification takes place inside the hot water storage tank. The gas cooler inlet water temperature will

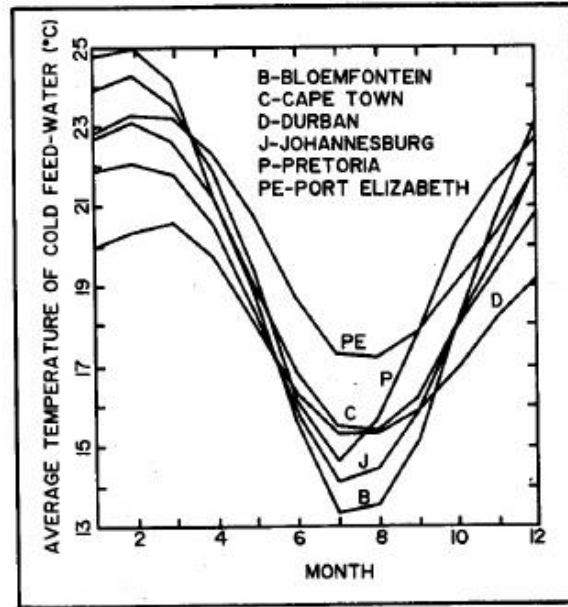


Figure 2.5: Monthly average cold water supply temperatures for the main metropolitan areas (Meyer & Greyvenstein 1992).

therefore range from a lower limit 10°C minimum to 30°C maximum with an yearly average value of 20°C.

2.3.3 Evaporation manufacturing technology

The use of finned tube evaporators is a mature technology and these are manufactured locally. The success of finned tube heat exchangers can be attributed to high reliability, good cost to performance ratio and flexibility in possible design geometries (Reulens 2009).

Finned tube coils are classified based on several criteria such as tube diameter, fin type, fin spacing and tube spacing. Figure 2.6 is a photo of a typical fin and tube evaporator.

The following is a summary of the default geometries for coils that is normally used in South Africa (HC 2009):

- Copper is the material of choice with stainless steel available where higher pressures or corrosion resistance is required.
- The tube diameters used is $\frac{3}{8}$ " (10.05 mm) or $\frac{1}{2}$ " (12.6 mm) tube where stainless steel tubes is only available in the latter.



Figure 2.6: Typical fin-and-tube evaporator with hydrophilic coated wavy fins. (Courtesy of Booyco Engineering)

- For the $\frac{3}{8}$ " the tube spacing (S_y) is 25.4 mm and the row spacing (S_x) is 22 mm. For the $\frac{1}{2}$ " tubes the tube spacing is 31.75 mm and the row spacing is 27.5 mm.
- Both smooth tubes and micro-fin tubes are available. The micro-fin tubes are only available for certain wall thicknesses. The wall thicknesses are presented in Table 2.3. Micro-fin tubes are referred to a ruffle bore tubes in general practice. The use of micro-fin tubes are advantageous since the heat transfer enhancement are 150-200% higher than the heat transfer for the same diameter smooth tube while the pressure drop penalty factor is 1.2-1.35 at the same conditions according to Cho & Kim (2007).
- Any number of rows from 1 row to 6 row coils can be manufactured, although 4 row coils are the most common.
- Different fin geometries, smooth fins, wavy fins and louvred fins, are available, with wavy fins being the most common.
- Aluminium fins are used by default. The fin thickness is 0.14 mm. Copper fins are also available for special applications.

- Fin spacing are expressed in FPI (fins per inch) and are available for a range of 6-18 FPI. For external use it is usually limited to 12 FPI to prevent excessive blocking of the fins by dust and dirt.

Table 2.3 gives an overview of the working pressure for different tube diameters, wall thicknesses, tube material and internal finishes as discussed above. The working pressure in Table 2.3 was calculated according to ASME Standard B31.9 allowable pressures. A safety factor of four is used and a 5% mill tolerance on the tube diameter was allowed for.

Table 2.3: Working pressures for different tube geometries as used in fin-and-tube heat exchangers.

Tube Material	Outer Diameter [mm]	Wall Thickness [mm]	Type	Working Pressure [bar]
Copper	10.05	0.30	Plain	29.35
		0.37	Rifle	34.24
		0.41	Plain	40.11
		0.61	Plain	59.68
	12.6	0.35	Plain	28.68
		0.35	Rifle	28.68
		0.5	Plain	40.97
		1	Plain	81.94
Stainless steel	12.6	0.7	Plain	157.38

Currently there are no local manufacturers of aluminium micro-channel heat exchangers, for use with R-744. These components have to be imported.

Most evaporators used in comfort cooling, marine environments and in industrial environments are epoxy or hydrophilic coated in order to increase the component life (Friterm 2004). With this said however, in South Africa for the most part, heat pump evaporators are provided without these protective coatings.

For construction of the concentric tube-in-tube heat exchanger, commercially of the self, schedule 40 stainless steel pipe is used. The welding process can be done by any

engineering works or technician with the required hardware. The working pressure for schedule 40 steel pipe is presented in Table 2.4.

Table 2.4: Schedule 40 steel pipe working pressure (*ASHRAE Handbook, Systems and Equipment* 2000).

Nominal Pipe Size [Inch]	Outer Diameter [mm]	Wall Thickness [mm]	Inner Diameter [mm]	Working Pressure [bar]
1/4	13.7	2.24	9.22	1296
3/8	17.1	2.31	12.48	1400
1/2	21.3	2.77	15.76	1476
3/4	26.7	2.87	20.96	1496
1	33.4	3.38	26.64	1558
1 1/4	42.2	3.56	35.08	1579
1 1/2	48.3	3.68	40.94	1593
2	60.3	3.91	52.48	1586

2.4 Evaporator modelling

The success of the implementation of R-744 as a working fluid rests on the ability to develop compact and efficient components (Pettersen et al. 1998). This is especially true for the heat exchangers including the evaporator and gas coolers. It is therefore of paramount importance to develop a suitable evaporator model that can be used to optimize the evaporator geometry and ensure a compact, low weight solution.

2.4.1 Evaporator design pressures

As discussed in Section 2.2 the operational pressure for R-744 is between 5 to 10 times higher than for traditional refrigerants. The operating pressures range for a typical R-744 heat pump are presented in Table 2.5 and Table 2.6. As shown in these tables the high side operational pressures are at least a factor of 2 to 4 times higher than the low side operational pressures. It is therefore standard practice to define different

design pressures for the low pressure evaporator side and the high pressure gas cooler side of R-744 systems.

Table 2.5 gives an overview of the saturated pressures corresponding to different evaporation temperatures. The critical pressure of R-744 is 73.77 bar with a corresponding critical temperature of 31.4°C. This data was obtained using the inbuilt fluid property functions for R-744 in EES (Klein 2011).

Table 2.5: Saturated pressure for R-744 at different temperatures during evaporation.

Saturated Pressure [bar]	Evaporation
	Temperature [°C]
26.49	-10
34.85	0
45.02	10
57.29	20
64.34	25

As mentioned in Section 2.2.2 the discharge pressure varies for different water supply temperatures. For an evaporation temperature of -6.4°C, the optimum gas pressures, as presented in Table 2.6 was predicted by White et al. (2002). This closely match the optimum pressures obtained by Neksa et al. (1998).

Table 2.6: Optimum discharge temperature for various hot water temperatures by White et al. (2002).

Optimum Pressure [bar]	Hot Water Temperature
	[°C]
100	70
110	90
120	106
130	120

For traditional systems the high side pressure is used, throughout the system, as the design pressure. At present, there is no consensus on the optimum design pres-

sures required for a R-744 system. The following are a few examples of typical design pressures, for R-744 heat pump systems, as quoted in the literature.

- Neksa et al. (1998) recommends typical maximum design pressures of 80 bar low side and 150 bar high side.
- Kim et al. (2004) refers to a draft SAE standard, J639, that requires that the low side ultimate burst pressure must be at least two times higher than the release pressure of the safety release device.
- One compressor supplier, Copeland provide a R-744 transcritical compressor that is designed for a low side pressure of 90 bar and high side pressure of 130 bar using a safety factor of three.
- *ASHRAE Handbook, Systems and Equipment* (2000) recommend a design safety factor of four, for refrigerant tubing.
- Reulens (2009) mentioned that due to the recent development and implementation of the R-744 cycle, there is a possibility that the burst pressures could be mitigated in regulating standards, especially if measures are taken to limit the increase in operational pressure during standby periods. For example, secondary cooling of the liquid receiver during the standby periods.

As seen in Section 2.3.1 ambient temperatures in South Africa can be in excess of 31°, exceeding the critical temperature of R-744 in the process. When a heat pump is in standby for a period of time, the component and refrigerant temperature will eventually achieve temperatures approaching ambient. In the case where the heatpump is in direct sunlight the ambient temperature can even be exceeded.

From this it is clear that a minimum design pressure, which exceeds the critical pressure, is required. It is therefore sensible to define a design pressure in the range of 80 to 90 bar for low pressure side and 150 bar for the high side.

For a concentric tube-in-tube evaporator, these design pressures do not present any problems. As seen in Table 2.4 the lowest pressure still comfortably exceeds the highest operational pressure.

For a fin-and-tube evaporator this is not the case. From Table 2.3 it can be seen that the only two tubes that meet the 80 bar low side design pressure are the, $\frac{1}{2}$ "

copper tube with the 1 mm wall thickness and the $\frac{1}{2}$ " stainless steel tube. Under certain conditions it might also be possible to use $\frac{3}{8}$ " copper tube with 0.61 mm wall thickness. This will require the use of a safety release valve with a release pressure of 80 bar. This seems to be a reasonable requirement as the burst pressure for this tube is 80 bar if a safety factor of three instead of four is used.

The range of micro-fin tubes available at present are however not able to handle to operational pressure required and the use of micro fin tubes will therefore not be investigated in this study.

2.4.2 Pressure drop and heat transfer considerations

The pressure drop through the heat exchanger is important, since change in pressure leads to a change in evaporating temperature, changing the temperature difference between the R-744 and secondary fluid. Pressure drops should therefore be kept as low as possible to ensure the biggest possible temperature difference. In this regard an important property of R-744 is its low viscosity. For the same mass flow rate R-744 have a much smaller pressure drop than other refrigerants. Not only is the pressure drop lower but the temperature pressure dependency are also much lower. Pettersen et al. (1998) did a comparison between equal length, equal capacity heat exchanger tubes for R-134a and R-744. It was found that for a pressure drop corresponding to a temperature drop of 1 Kelvin, R-744 had a 9 times higher pressure drop than R-134a.

R-744 also has a very high volumetric refrigerating capacity. This, combined with the high heat transfer and low pressure drop, require the use of much smaller tube diameters and high mass fluxes to ensure optimum and compact evaporator design (Kasap et al. 2011, Pettersen et al. 1998). A further advantage of smaller tubes is that they can withstand the higher pressures required.

Due to the use of smaller tubes and a tolerance to high pressure drop characteristics as mentioned above, R-744 heat exchangers tend to have very high mass fluxes. Pettersen et al. (1998) found that a R-744 heat exchangers had a mass flux of $498.5 \text{ kg.m}^{-2}.\text{s}^{-1}$, which is 5 times higher than the mass flux for a comparable R-134a heat exchanger. Another example is the industrial R-744 heat pump Yamaguchi et al. (2011) investigated (refer to Section 2.1). In this commercially available heat pump, the mass flux varied between 470 to $795 \text{ kg.m}^{-2}.\text{s}^{-1}$ depending on the different operating condi-

tions.

Due to the high heat transfer coefficient for R-744 and the compact geometry R-744 evaporators also tend to have relative high heat fluxes. For the same evaporators mentioned in the previous paragraph, Pettersen et al. (1998) obtained a heat flux of 4.57 kW.m^{-2} and Yamaguchi et al. (2011) obtained heat fluxes in the range of 4.78 to 7.84 kW.m^{-2}

Given that the data from Pettersen et al. (1998) and Yamaguchi et al. (2011) are very similar and given the fact that the industrial heat pump investigated by Yamaguchi et al. (2011) are commercially available, evaporators for heat pump systems should have similar values for heat and mass fluxes. The refrigerant correlation used in this study will therefore need to apply for refrigerant mass fluxes in the region of 450 to $800 \text{ kg.m}^{-2}.\text{s}^{-1}$ and heat fluxes in the region of 4.5 to 8 kW.m^{-2} .

For a fin-and-tube heat exchanger the fin side correlations is even more important than the tube side correlations. The fin side, air heat transfer coefficient, for wavy fin-and-tube heat exchangers, is typically 100 to $150 \text{ W/m}^2\text{-K}$ while the heat transfer coefficient for R-744 is very high with values high exceeding $3000 \text{ W/m}^2\text{-K}$. A change in the tube side heat transfer coefficient, from $2000 \text{ W/m}^2\text{-K}$ to $3000 \text{ W/m}^2\text{-K}$, leads to an increase of about 6% in the overall heat transfer coefficient for a typical industrial style air to refrigerant heat exchanger (Handschuh 2008). Furthermore, an increase in tubes side heat transfer will increase the overall heat transfer coefficient, resulting in increased cooling in the first coil rows (Iu 2007). The increased cooling lead to lower air temperatures and reduced temperatures in the proceeding tube rows. This effect contributes to limit the effect of changes in tube side head transfer on the overall heat transfer of the evaporator.

Heat transfer coefficients for R-744 are typically very high. From the test data of Cho & Kim (2007) it can be seen that the heat transfer coefficients is in excess of $5000 \text{ W/m}^2\text{-K}$. Uncertainties in predicting these values will therefore only marginally impact the overall heat transfer of the evaporator.

2.4.3 Heat exchanger models

Heat exchanger models can generally be classified by their calculation domains (Iu 2007) (Iu et al. 2007). Based on the type of discretization used they can be divided in

the following criteria:

- Zone-by-zone
- Tube-by-tube
- Segment-by-segment (elemental method)

These are discussed in detail below and are shown in Figure 2.7.

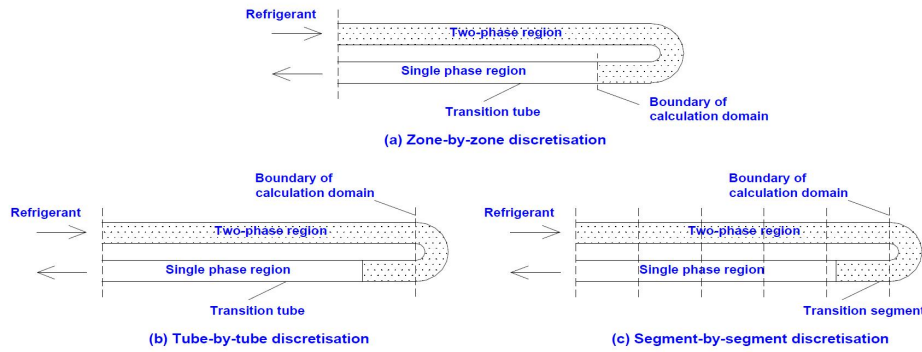


Figure 2.7: Discretization of heat exchanger tubes (Iu 2007).

Zone-by-zone model

In this model, the heat exchanger is divided into different zones, depending on the state of the refrigerant, with each zone representing a different state. Heat exchangers will typically be divided into three zones. The first zone will be subcooled liquid, the second zone will be two-phase flow and the third zone will be superheated vapour.

Although the zone-by-zone method is computationally efficient it is unable to handle fin-and-tube coil geometry adequately, mainly due to the inability to handle the coil circuitry (Iu 2007). The zone-by-zone model assumes constant air properties and heat transfer for each row, and identical circuitry for each circuit. Due to these assumptions this model tends to over predict performance, especially for heat exchangers where the air flow onto the coil is non-uniform or the circuitry are complex. If a flow pattern map, such as the one presented by Cheng et al. (2008b) is used the zone-by-zone model can be expanded, increasing the number of zones to match the number of flow patterns.

The zone-by-zone model can still be useful for conceptual design where it can be used to predict heat exchanger performance with reasonable accuracy, even without exact circuiting. Nellis & Klein (2009) showed that the results for, a zone by zone

model for a condenser, predicted the results within 4% of the results calculated using the EVAP-COND software (NIST 2010). Given the above mentioned limitations, the zone-by-zone model will not be used in this study.

Tube-by-tube model

In the tube-by-tube model each tube of the heat exchanger is treated as a separate heat exchanger and it is assumed that the fluid properties for each tube are constant. Using this method, it is possible to take into account the effect of the circuiting as well as the effect of non-uniform air distribution. The main downside of the tube-by-tube model is that it is more computational intensive (Iu et al. 2007).

Domanski, Yashar & Lee (n.d.) found that for the specific case they investigated, a 5.6 % improvement in capacity was possible, if the evaporator circuitry is optimised to account for airflow distribution over the coil. This type of optimization study, is not possible if a zone-by-zone model is used.

The constant fluid property assumption does introduce some inaccuracy when the transition zone between two phases, fall within a tube. Depending on where the transition takes place inside the tube, the performance for the tube will either be over predicted or under predicted. Iu et al. (2007) implemented a novel algorithm to simulate transition elements, splitting the element into a single phase and two phase zones and calculating the heat transfer in each zone. This algorithm increased accuracy but will be difficult to implement in EES as part of a heat exchanger model. Implementing a flow pattern map, such as the one presented by Cheng et al. (2008b), will require that the above mentioned algorithm be modified to enable the tubes to take the different flow patterns into account.

Elemental model

In the elemental model, each heat exchanger tube is further divided into smaller segments and the same constant properties assumptions, as used in a tube-by-tube model, are used here. In effect, the tube-by-tube model is an extension of the elemental model, where the element is chosen as the complete tube.

The elemental model has the same advantages as the tube-by-tube model (Iu 2007). Reducing the element size also has the added benefit of reducing the transitional el-

ement error. This type of model is also better suited for implementation of a flow pattern based heat transfer and pressure drop models.

Although the elemental model is computationally the most intensive, it is possible to take detail geometry, variations in airflow, variations in local heat transfer and pressure drop into account, while minimising the errors due to the constant property assumption.

Effectiveness-NTU and numerical methods

One method that is universally applied for solving heat exchanger models is the effectiveness-NTU method. The effectiveness-NTU method is a general analytical solution, such as the LMTD-method, applicable to same heat exchanger geometries, with the upside of flexibility and ease of use. With the effectiveness-NTU method the outlet temperatures for a heat exchanger, can directly be calculated, when the heat exchanger conductance is known (Nellis & Klein 2009). The effectiveness-NTU is mostly used for solving the performance of the complete heat exchanger in a single calculation (Nellis & Klein 2009, Incropera et al. 2007).

Implementing the effectiveness-NTU method for a complete heat exchanger, although computationally very efficient, can be very inaccurate (Reulens 2009). This is due to the assumption that fluid properties stay constant for the complete heat exchanger. This is especially problematic for an evaporator where large discrepancies in the fluid (and therefore heat transfer properties) exist, in the different zones. The configuration of a real heat exchanger is almost never pure counterflow or parallel flow as assumed in the use of the E-NTU method, but inadvertently some combination. Thus the effect of different circuitry arrangements is difficult to take into account.

Nellis & Klein (2009) discusses the use of numerical methods as an alternative to the effectiveness-NTU method to solve heat exchangers, especially for cases where the constant fluid property assumption is not valid. The most straight forward method is where the state equations is solved using numerical integration techniques.

Another more general numerical method is subdividing the heat exchanger into many elements and solving the governing equation in each element (Elemental model) (García-Valladares et al. 2004, Morales-Ruiz et al. 2009). As discussed previously, by using an elemental approach, any geometry and flow variations can be taken into

account.

One option is to combine the analytical method and numerical method, as discussed by Nellis & Klein (2009) and implemented by Iu (2007) and Iu et al. (2007). They followed the elemental approach by subdividing the heat exchanger into sub-heat exchangers, and then apply the analytical effectiveness-NTU method for each element. This not only ensures that any configuration can be solved, but also by subdividing the heat exchanger into sub-elements ensure the validity of the underlying assumption of constant fluid properties in each element (Nellis & Klein 2009), with the exception of the transition element (Iu et al. 2007).

In the present study the elemental model will be used with each tube subdivided into a minimum of one or several elements. Although there is merit in combining the effectiveness-NTU method with the elemental approach, the advantage is lost when smaller elements is used, since the elemental method approach the analytical solution under these conditions. The effectiveness-NTU method will therefore not be employed.

2.4.4 Fouling, contact and tube wall resistances

Fouling and contact resistances can be a significant contributor to overall heat transfer resistance. (Thome 2010). Contact resistance depends strongly on the manufacturing method and fouling resistance depends strongly on the operational environment. For collared fins the contact resistance can vary from negligible to very large. Generally the refrigerant side fouling factor is negligible and the air and water side can vary from negligible to very high depending on the quality of the fluid and heat exchanger maintenance.

Due to the lack of general data and the strong dependency on outside influences it is general practice to ignore the effect of contact resistance and fouling factor for design purposes. For example, Lee & Domanski (1997), Iu (2007), Bendaoud et al. (2010), Thome (2010) and Yamaguchi et al. (2011) did not take it into account.

As a norm the tube wall thermal resistance for thin-wall copper tubes are negligible with regards to the overall heat transfer coefficient. Lee & Domanski (1997) and Yamaguchi et al. (2011) takes the tube wall resistance into account, but this is more the exception than the rule. If thick stainless steel tubing is used in the evaporator this assumption is not valid anymore and the tube wall thermal resistance has to be

taken into account.

In the present study the effect of fouling and contact thermal resistances will be ignored. The tube wall resistance will however be taken into account.

2.4.5 Water side correlations

The correlations for turbulent water flow in tubes is well developed and correlations are chosen on basis of accuracy as well as ease of use. All tube is horizontal and only correlations for horizontal flow need to be considered.

Pressure drop correlations

The implicit Colebrook equation is widely seen as the standard for calculation of the turbulent friction factor (Genić et al. 2011). The well-known Moody chart is an explicit graphical presentation of this equation. Several explicit correlations are available in the literature. For example, Morales-Ruiz et al. (2009) used the Churchill correlation for water flow in an annulus and Nellis & Klein (2009) presented the correlation of Wigrang and Sylvester as the explicit correlation of choice.

A recent study by Genić et al. (2011), evaluated different friction factor correlations and found that the correlation of Wigrang and Sylvester provided the most accurate results, followed closely by the Haaland correlation. They found the Haaland correlation less computational expensive since it required only eight calculation steps where the Wigrang and Sylvester correlation required 16 calculation steps.

Another recent study by Ghanbari et al. (2011) proposed a new correlation, the Ghanbari-Farshad-Rieke equation for the friction factor. This new explicit correlation was found to be more accurate than any of the explicit correlations above-mentioned with the upside of requiring only 8 calculations steps.

The Ghanbari-Farshad-Rieke friction factor correlation will be used in this study.

Heat transfer correlations

Both the Nusselt number correlations of Dittus-Boelter (Venter 2010, Jung & Radermacher 1989) and Gnielinski (Morales-Ruiz et al. 2009, Kim et al. 2005) are often cited in literature. According to Incropera et al. (2007) the Gnielinski correlation is the more accurate of the two. The downside of the Gnielinski correlation however is

that it is slightly more complex and also requires the friction factor as an input. Since the friction factor is already calculated in order to predict the pressure drop, this does not overly increase the complexity of the simulation and the Gnielinski correlation will therefore be used in this study.

Annulus flow

According to Incropera et al. (2007), the aforementioned heat transfer and pressure drop correlations apply for flow in an annulus, if the correct hydraulic diameter is used and the flow is turbulent. The reason is that turbulent heat transfer and pressure drop is normally affected by the duct shape (Nellis & Klein 2009). However, Dirker (2002) and Lu & Wang (2008) found that the heat transfer coefficient for an annulus is depended on the geometry of the annulus, with the heat transfer coefficient increasing for a narrow annulus.

It was decided to use the Gnielinski correlation for the annulus since it is used throughout for single phase flow. The flow regime will be evaluated throughout this study and if the water flow is laminar, the Nusselt number correction factors, as presented by Incropera et al. (2007) will be applied.

2.4.6 Refrigerant side correlations

The two phase flow characteristics of R-744 is completely different from those of conventional refrigerants (Kim et al. 2004, Cheng et al. 2008a). Several studies have been done in recent years in order to obtain more accurate data for R-744 and to evaluate the available correlations for use with R-744.

As identified in the previous sections of this study the refrigerant side correlations will have to apply for the following range of conditions:

- Evaporation temperature of -15 to 25° (Section 2.3.1).
- Mass flux of 470 to 795 kg.m⁻².s⁻¹ (Section 2.4.2).
- Heat flux of 4.5 to 8 kW.m⁻² (Section 2.4.2).
- Internal tube diameters of 8.8 mm and greater (Section 2.3.3). This literature survey will therefore focus on heat transfer and pressure drops for R-744 in macro

channels ($D_i > 3\text{mm}$). The definition of a macro channel as $D_i > 3\text{mm}$ is adopted by several other authors including Yun et al. (2003), Thome & Ribatski (2005) and Cheng et al. (2008b).

- Smooth tubes only (Section 2.4.1).
- All tubes are horizontal

Pressure drop correlations

In general it is found that pressure drop for R-744 increase with an increase in mass flux and decrease with an increase in saturation temperature. Almost all existing correlations also tend to over predict the pressure drop for R-744 (Thome & Ribatski 2005, Cheng et al. 2008a, Oh et al. 2008, Oh & Son 2011).

Pressure drop for two-phase flow is typically calculated utilizing a two-phase multiplier (Reulens 2009, Rousseau & van Eldik 2011). The single phase pressure drop for one of the phases is calculated as if all flow are at the condition of that specific phase. The two-phase pressure drop is then obtained by multiplying the pressure drop with the two-phase multiplier. For traditional refrigerants, various such methods exists.

Didi et al. (2002) did a comprehensive evaluation of five of the most quoted two phase pressure drop correlations in literature, for horizontal tubes with diameters of 10.92 and 12.00 mm and mass velocities from 100 to 500 $\text{kg.m}^{-2}.\text{s}^{-1}$. An extensive two-phase database for five different refrigerants (R-134a, R-123, R-404A and R-502) was investigated. It was found that the method of Müller-Steinhagen and Heck is the best for conventional refrigerants, followed by the method of Grönnerud and the method of Friedel. Thome & Ribatski (2005) compared these three correlations against their available R-744 pressure drop data. It was found that the method of Friedel gave the best overall results, followed by the method of Müller-Steinhagen and Heck. These methods were found to outperform various other correlations, including the correlation of Yoon et al. (2004) specifically developed for R-744.

Although the Yoon et al. (2004) correlation was developed for conditions which closely matched the required conditions identified (Tube diameter 7.53mm, Saturation temperature of -4 to 20°C, mass flux of 200 to 500 $\text{kg.m}^{-2}.\text{s}^{-1}$), the findings by both Thome & Ribatski (2005) and Cheng et al. (2008a) indicate that the correlation is

a poor choice. This is probably due to the fact that the correlations was developed, based on a limited range of test conditions and only one tube diameter.

To overcome the limitation of current pressure drop correlations, Cheng et al. (2008a) develop a two-phase flow pattern map for R-744. For each flow pattern they applied a different correlation to predict the pressure drop for R-744 for the specific flow pattern. The updated flow pattern map is applicable for the identified range (Tube diameters from 0.6 to 10mm, mass velocities from 50 to 1500 kg.m⁻².s⁻¹, heat fluxes from 1.8 to 46 kW.m⁻² and saturation temperatures of -28 to 25°C). As part of their study they evaluated all of the above mentioned correlations against their database. They found that their new flow-pattern phenomenological pressure drop model predicted the pressure drop data the best, with the Friedel method giving the second best results. The downside of this model is that it is difficult to implement. The flow patterns will need to be identified throughout and the correct correlation applied to each zone. This will also lead to transition element problems for each element with a flow pattern change.

Choi et al. (1999) evaluated different pressure drop correlations for a large database of pure and mixed refrigerants in evaporation and condensation. They found the homogeneous flow correlation of Bo Pierre to be the most accurate. Bo Pierre published his model in 1964 for the evaporation pressure drop for R-12. As stated by Jung & Radermacher (1989) and Choi et al. (1999) the model is still appealing today because of its simplicity and accuracy. Choi et al. (1999) modified the Bo Pierre model to better predict their database. The model was also extended to allow for the prediction of pressure drop in micro-fin tubes and refrigerant oil mixtures.

Until recently this model was not evaluated for use with R-744. Oh et al. (2008) evaluated this model against experimental data. The experimental data was acquired for a tube diameter of 7.75mm, mass flux of 200 to 500kg.m⁻².s⁻¹, heat flux from 10 to 40 kW.m⁻² and saturation temperature of -5 to 5°C. Except for the higher heat fluxes this closely matched the identified range. Oh et al. (2008) found that the method of Choi et al. (1999) predicted the experimental data the best, outperforming the Friedel correlation significantly. This result was replicated by Oh & Son (2011), with the method of Choi et al. (1999) predicting the experimental data the best. Except for the smaller pipe diameter of 4.57mm the rest of the parameters of the experimental data

closely matched the identified range (Mass fluxes ranged from 200 to 1000 kg.m⁻².s⁻¹, heat fluxes ranged from 10 to 40 kW.m⁻² and saturation temperature ranged from 0 to 40°C).

The Choi et al. (1999) correlation will therefore be used in this study, for two phase pressure drop, due to the combination of simplicity and accuracy.

For single phase R-744 flow (vapour or liquid) pressure drop, the Ghanbari-Farshad-Rieke friction factor correlation will be used. Refer to Section 2.4.5 for more detail.

The pressure drop in bends are of the same order of magnitude as the pressure drops in straight pipes and need to be taken into account. Domanski & Hermes (2008) develop an improved correlation for two-phase pressure drop in 180° return bends for R-22 and R-410A and smooth tubes with diameters of 3.3 to 11.6mm. Their new correlation outperformed several other correlations tested. The Domanski & Hermes (2008) correlation use the Müller-Steinhagen and Heck correlation to predict the flow in the straight tube and a multiplier to take into account the effect of the curvature. Recently Padilla et al. (2011) compared four correlations from literature (including the correlation of Domanski & Hermes (2008)) against a database of 238 experimental data points. They found that none of the methods predicted the experimental results satisfactorily.

In this study the Choi et al. (1999) correlation will be used to calculate the pressure drop in the straight tube while the Domanski & Hermes (2008) multiplier will be applied to take the curvature into account.

Heat transfer correlations

Several studies, focussing on obtaining heat transfer data and determining suitable heat transfer correlations for R-744, have been completed in recent years.

Similar to their pressure drop model, Cheng et al. (2008b) developed a prediction method for heat transfer based on flow patterns. This heat transfer model is applicable for all of the identified ranges, with horizontal tube diameters of 0.6 to 10mm, mass fluxes from 50 to 1500 kg.m⁻².s⁻¹, heat fluxes from 1.8 to 46 kW.m⁻² and saturation temperature ranged from -28 to 25°C. They found good agreement between the predicted values using their flow pattern based heat transfer model and the experimental values in their extensive database. In this study they did not evaluate other leading

correlations against their database. Although accurate and applicable to the identified range of conditions, this correlation will not be used as it is highly complex. Cheng et al. (2008b) did, however complete an extensive literature survey and careful analysis of the available experimental data, from the literature, before including it in their database. Of the data included in the Cheng et al. (2008b) database, those of Yun et al. (2003) and Yoon et al. (2004) are the most applicable to this study.

Yun et al. (2003) investigated the evaporation heat transfer characteristics for R-744 in a horizontal tube with inner diameter of 6.0 mm. Saturation temperatures of 5°C and 10°C were investigated with the mass flux ranging from 170 to 320 kg.m⁻².s⁻¹ and heat fluxes ranging from 10 to 20 kW.m⁻². It was found that the heat transfer for R-744 is on average 47 percent higher than R134a for the same operational conditions. They evaluated the correlation of Gungor and Winterton with their data and it was found that this correlation showed large deviations at low mass flux but good agreement at high mass flux. The highest mass flux, 320kg.m⁻².s⁻¹, investigated by Yun et al. (2003) is still lower than the lowest expected mass flux typically found in a R-744 evaporator, as previously identified. Although it is expected that the trend will hold for the even higher mass fluxes, this might not be the case.

The study of Yoon et al. (2004) was done for a tube diameter of 7.53 mm, mass fluxes from 200 to 530 kg.m⁻².s⁻¹, heat fluxes from 12 to 20 kW.m⁻² and saturation temperatures of -4 to 20°C. The tube diameter, mass fluxes and saturation temperature ranges more closely match the identified ranges than the data of Yun et al. (2003). Yoon et al. (2004) did a comparison of various heat transfer correlations as well as developing a new correlation. It was found that all the correlations under predicted the experimental R-744 heat transfer coefficients. Of the correlations they evaluated their newly developed correlation was the best, with the correlation of Gungor and Winterton second best, but on par with the other correlations. The correlation of Jung et al. (1989) showed a decrease in heat transfer with quality matching the trend of the experimental results.

Cho & Kim (2007) did an experimental study on heat transfer in smooth and micro-fin tubes. The diameters under investigation were 5 and 9.52 mm. For their tests, mass fluxes ranged from 212 to 656 kg.m⁻².s⁻¹, heat fluxes from 6 to 20 kW.m⁻² and saturation temperatures from 0 to 8°C were used. For the most part, this data

coincides with the required ranges of this study. They found that the heat transfer enhancement from the micro-fins, for the 9.52 mm tube, was in the range of 150 to 200% while the pressure drop penalty factor was in the range of 1.2 to 1.35. For the smooth tube they found the correlation of Yoon et al. (2004) and Kandlikar to give good results. The correlations of Jung et al. (1989) and Gungor and Winterton showed a decrease in heat transfer with an increase in quality. The experimental data also indicated a decrease in the heat transfer coefficient with an increase in quality.

Recently Mastrullo et al. (2010) did an assessment of existing heat transfer correlations in order to find the best predictive methods for R-744 in macro channels. An extensive literature survey was conducted and experimental results were obtained for a horizontal circular tube of 6 mm, mass fluxes ranged from 200 to 349 $\text{kg.m}^{-2}.\text{s}^{-1}$, heat fluxes from 10.1 to 20.6 kW.m^{-2} and saturation temperatures of -7.8 to 5.8°C. It was found that almost all correlations under predicted the experimental data and that the correlation of Jung et al. (1989) predicted their experimental data with the lowest absolute mean error. The correlation of Yoon et al. (2004) did not predict the data satisfactorily. This complements the finding of Cheng et al. (2008b) which found that the correlation of Yoon et al. (2004), although developed for R-744, is only applicable to the specific conditions of that study and caution must be exercised in the use of this correlation for any other conditions. They found the correlation of Cheng et al. (2008b) was the only one able to follow the experimental data trends with precision.

The study of Oh & Son (2011) found that the method of Cheng et al. (2008b) predicted their experimental results the best and was the only method able to follow the experimental trends. The method of Jung et al. (1989) was less accurate than the Cheng et al. (2008b) method but out performed the rest of the non flow pattern based correlations.

As discussed in Section 2.4.2, inaccuracies in the heat transfer coefficient of evaporating R-744 are not critical to the accuracy for the overall heat exchanger model. The correlation of Jung et al. (1989) will therefore be used as it provides the best balance between accuracy and ease of use.

For vapour or liquid R-744 heat transfer the Gnielinski correlation will be used. This choice is based on the considerations in Section 2.4.5 and augmented by the recommendations of Reulens (2009) and Yin et al. (2001), recommending the use of

the Gnielinski correlation as the best option for subcritical R-744.

2.4.7 Simultaneous heat and mass transfer

When the air side surface temperature is lower than the dew-point of the air stream, both heat and mass transfer takes place. Two main methods, namely the enthalpy difference method and the equivalent temperature difference method (ETD) are commonly used in the literature to take into account simultaneous heat and mass transfer. Both these methods were reviewed in detail by Pirompugd, Wang & Wongwises (2009).

In the aforementioned study these methods were reviewed based on the use as a reduction method to obtain heat transfer correlations from empirical data. It was found that the enthalpy difference method was the preferred choice for the simulation models of both Ding et al. (2011) and Iu (2007). The accuracy of both methods are similar and Huzayyin et al. (2007) found the ETD method slightly better at predicting their experimental data. Pirompugd, Wang & Wongwises (2009) reported that for partially wet conditions a combination of ETD and enthalpy difference method produces the most accurate results. The ETD method is suitable to take into account local variations of parameters and complex geometries (Wang & Hihara 2003).

Although the enthalpy difference method is the most commonly used method, the author of this study, found that the ETD method was easier to implement. Since both methods is similar in accuracy, the ETD method will be used in this study.

In the enthalpy difference method, the analogy between heat and mass transfer is used to obtain the mass transfer coefficient if the heat transfer coefficients are known. As discussed by Pirompugd, Wang & Wongwises (2009) and Rousseau & van Eldik (2011), it is often assumed, for simplicity, that the ratio of heat transfer to mass transfer is equal to one. This assumption is also used by Lee & Domanski (1997) in their simulation model. For the equivalent temperature difference method, the outlet enthalpy and drybulb temperature of the air stream are determined directly. The mass transfer can then be calculated since the outlet condition is known (Wang & Hihara 2003).

Another aspect to take into account is the liquid film which forms on the fin surface during condensation. The thermal resistance due to this water film under dehumidifying conditions is typically in the order of 0.5 to 5% of the air side convective heat

transfer coefficient, and is typically ignored (Oliet et al. 2007, Kuvannarat et al. 2006, Pirompugd, Wongwises & Wang 2009). Due to the fact that this is ignored in the reduction methods used, when acquiring the heat transfer correlations, it will also be ignored in the simulation develop in this study.

2.4.8 Fin side correlations

Heat transfer on the fin side is the dominant factor in determining the total heat transfer coefficient, where it can account for 85% of the total resistance (Wang et al. 1997). Any inaccuracies will affect the accuracy of the heat exchanger far more than is the case for the refrigerant side. As discussed in Section 2.3.3 different coil options and geometries are available. The default coil geometry for this study is staggered, wavy fin coils. Only correlations relevant to this geometry will be investigated.

For increased accuracy, it would be best to use a correlation that was obtained using a reduction method, similar to the method used in simulation (ETD method, elemental approach) and operational conditions similar to those present in the evaporator. Thus correlations obtained using the ETD method and elemental approach and with refrigerant as the operational fluid have preference. However, most correlations have been obtained using the enthalpy potential method and using water as the operational fluid inside the coil (Kuvannarat et al. 2006). Previously heat exchanger correlations were also exclusively obtained using a lumped heat exchanger approach instead of a tube-by-tube or segment approach implemented more recently.

Furthermore, Iu (2007) noted that the correlations is not only dependent on the number of rows (N) in a coil but also differ from row-to-row. For an elemental simulation method, a row-by-row correlation will be the most accurate as it will allow you to take the different heat transfer coefficients in each tube row into account. Unfortunately, no such correlations is currently available for wavy fin heat exchangers and it will be assumed that the heat transfer and pressure drop is not dependent on the specific row number, but only on the total number of rows.

The following table, Table 2.7 provides an overview of the fin side correlations from the literature as well as the required parameters of this study.

Table 2.7: Fin side correlation overview.

Author (year)	Correlations	Uncertainties	Surface	FPI	Dc [mm]	Sy [mm]	S-x [mm]	N	t [mm]
Required for this study	j - factor		dry / wet	8-12	10.33 and	25.4 and	22 and	4	0.14
	f - factor				12.88	31.75	27.5		
Wang et al. (1997)	j - factor	94% within 10%	dry	7-15	10.3	25.4	19.05	1-4	0.12
	f - factor	95% within 15%							
From Thome (2010)	j - factor	93.8% within 15%	wet	8-15	8.62 and	25.4	19 and	1-6	0.12
	f - factor	84.1% within 15%			10.38		22		
Pirompugd et al. (2009)	j - factor	94.19% within 15%	wet	7-17	8.62 and	25.4	19.63	1-6	0.12
	f - factor				10.38		26.27		
Kuvannarat et al. (2006)	j - factor	91.3% within 15%	wet	10-18	9.53	25.4	19.05	2-6	0.115 and
	f - factor	87.5% within 15%							0.25

Dry surface

Wang et al. (1997) found that the work of previous researchers on wavy fin correlations were limited as the fin-and-tube geometries investigated were different from those typically found in practice. Wang et al. (1997) developed a new correlation taking into account the effect of number of tube rows. For a dry surface Thome (2010) recommends the use of the Wang et al. (1997) correlation. This correlation will be used for pressure drop and heat transfer under dry surface conditions for the present study.

Wet surface

The effect of fin thickness under dehumidifying conditions was investigated by Kuvannarat et al. (2006). It was found that the influence of fin thickness on the heat transfer coefficient was higher for high fin spacing ($FPI \approx 18$) and heat exchangers with two rows. For a 4 row heat exchanger with lower fin spacing the effect of fin thickness decreases considerably. The two fin thicknesses evaluated in this study were 0.115 mm and 0.25 mm (more than double the first value). From Table 2.7 it can be seen that the default fin thickness for the simulation is 0.14mm where most correlations are for a fin thickness of 0.12 mm. These are roughly the same order of magnitude and the effect of the difference in fin thickness is assumed negligible for the purposes of this study.

Thome (2010) presented a method of Wang and co-workers from 1999 as the preferred correlation for wet surface conditions. As seen in Table 2.7 this correlation is applicable for a wide range of conditions. Pirompugd, Wongwises & Wang (2009)

derived heat and mass transfer correlation covering almost the exact same geometric range with the latter being slightly more accurate.

More recently, Huzayyin et al. (2007) presented a correlation, derived using the ETD method and using R134a as the operational fluid for the coil. This correlation, however is only valid for one coil geometry and a small range of operational conditions. It can therefore not be used.

Thus in summary, the air side heat transfer correlations proposed by Pirompuang, Wongwises & Wang (2009) and the pressure drop correlation of Kuvannarat et al. (2006) will be used in the present study.

Extrapolation of the correlations

These correlations are mainly based for tube diameters in the range of 10 mm, tube spacing (S_y) of 25.4 mm and row spacing (S_x) of 19.05 mm. Generally it is not advised to apply these correlations outside the geometric parameters. Data for other tube diameters and spacings are scarce. Although Wang et al. (2011) did a study examining the airside performance of larger tube diameters they did not propose any new correlations.

Due to the lack of more suitable correlations the chosen correlations will be applied over the complete range of geometries.

2.4.9 Heat conduction between tubes

For the two phase region, the refrigerant temperature, and therefore tube temperature is almost constant and negligible heat transfer due to conduction takes place between the tubes. Normally this is not taken into account, but in the superheated region, the refrigerant and resultant tube temperature is much higher and conduction does take place. The finned-tube evaporator model of Lee & Domanski (1997) takes it into account in.

Domanski, Choi & Payne (n.d.) investigated the effect of longitudinal heat conduction on the capacity of finned tube evaporators. They evaluated the performance of evaporators at different exit superheats. They found that for superheats of about 5.6°C the effect is negligible but for higher superheats of about 16.7°C the effect was significant. They found capacity over predictions as high as 23% if this effect is ig-

nored. Domanski, Choi & Payne (n.d.) also established that the assumption that conduction in the axial direction of the tube is negligible is applicable for finned tube heat exchangers.

Heat conduction between tubes are only significant at high superheat. For the purposes of this study the superheat will be limited to a maximum of 10°C (Refer to Section and the effect of heat conduction between tubes will therefore not be taken into account.

2.5 Summary

In this chapter the available literature on the design of a R-744 evaporator were evaluated. This was not limited to the heat transfer and pressure drop correlations of R-744, but also extended to an overview of the complete heat pump system, its operational conditions as well as the local conditions.

For two phase R-744 pressure drop the Choi et al. (1999) correlation will therefore be used in this study due to the combination of simplicity and accuracy. For two phase R-744 heat transfer coefficient the Jung et al. (1989) will be used as it provides the best balance between accuracy and ease of use. For the air side, the correlations proposed by Pirompugd, Wongwises & Wang (2009) for pressure drop and Kuvannarat et al. (2006) for heat transfer will be used. The details about these correlations are presented in the next chapter.

Based on the results of this literature survey it was decided to use the elemental method to model the evaporator. The implementation of the elemental method is discussed in detail in Chapter 4.

Chapter 3

Theoretical background

This chapter presents the theoretical background for the detailed simulation of a concentric tube-in-tube and fin-and-tube heat exchanger. In this study, the heat exchangers are divided into smaller elements which can be seen as sub heat exchangers.

In order to simulate each element, the following is required (Rousseau & van Eldik 2011):

- Conservation laws.
- Component characteristics.
- Fluid properties.
- Boundary conditions.

In the first part of this chapter, the conservation laws will be discussed. The conservation equations need to be derived for the refrigerant side as well as for the air side. The second part will look at the details of the physical and empirical models to fully describe the component characteristics. The built-in fluid properties of EES will be utilised and the detailed fluid properties will not be discussed in this study. The relevant boundary conditions are discussed in Chapter 4.

3.1 Conservation equations - Tube side

The conservation equations for the refrigerant needs to apply for all zones i.e: liquid flow, two-phase flow as well as all vapour flow. The conservation equations for water

flow in the annulus will be the same as for refrigerant liquid flow. The incremental control volume shown in Figure 3.1 is used for analysis. The integral form of the conservation equations are from Rousseau & van Eldik (2011)

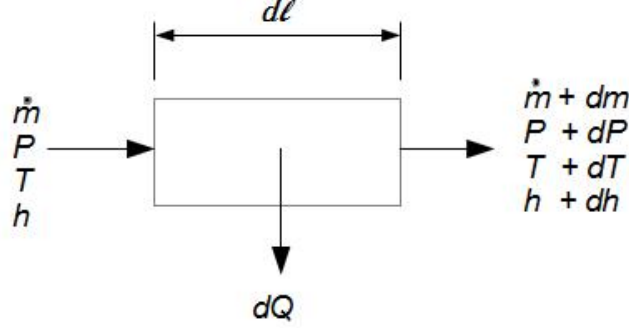


Figure 3.1: Control volume - tube side

3.1.1 Conservation of mass

For the finite control volume in Figure 3.1, the integral form of the mass conservation equation is given by

$$\frac{\partial}{\partial t} \left(\iiint \rho dV \right) + \oint \rho \vec{V} \cdot \vec{dA} = 0 \quad (3.1)$$

where V is the volume and V the velocity relative to the control volume.

For steady state conditions, the following generic equation can be derived for the conservation of mass,

$$\dot{m}_e - \dot{m}_i = 0 \quad (3.2)$$

where the subscripts $_e$ and $_i$ indicating the outlet and inlet respectively.

3.1.2 Conservation of momentum

For the finite control volume in Figure 3.1, the integral form of the momentum conservation equation is given by

$$\oint \tau \vec{dA} + \iiint \vec{B} \rho dV = \frac{\partial}{\partial t} \left(\iiint \rho \vec{V} dV \right) + \oint \rho \vec{V} (\vec{V} \cdot \vec{dA}) \quad (3.3)$$

For steady state conditions, the momentum equation accounts for total pressure loss and for flow inside tubes can be written as (Didi et al. 2002),

$$\Delta p_{total} = \Delta p_{static} + \Delta p_{mom} + \Delta p_{frict} \quad (3.4)$$

where the total pressure drop Δp_{total} is the sum of the static pressure drop Δp_{static} , the momentum pressure drop Δp_{mom} and the friction pressure drop Δp_{frict} .

For horizontal tubes there are no difference in elevation. It can therefore be assumed that the effect of the overall height difference in the heat exchanger is negligible, thus $\Delta p_{static} = 0$.

3.1.3 Conservation of energy

For the finite control volume in Figure 3.1, the integral form of the energy conservation equation is given by

$$\dot{Q} + \dot{W} = \frac{\partial}{\partial t} \left(\iiint (u + \frac{1}{2}V^2 + gz) \rho dV \right) + \oint (h + \frac{1}{2}V^2 + gz) \rho \bar{V} \cdot d\bar{A} \quad (3.5)$$

where $\dot{Q}$ is the total heat transfer rate to the fluid and $\dot{W}$ is the total rate of work done on the fluid.

For steady state conditions, the following generic equation can be derived for the conservation of energy:

$$\dot{Q} + \dot{W} = \dot{m}_e h_e - \dot{m}_i h_i + \dot{m}_e g z_e - \dot{m}_i g z_i \quad (3.6)$$

Taking into account that no work is done on the fluid and that the change in elevation for horizontal tubes is negligible, the equation for conservation of energy inside a tube is:

$$\dot{Q} = \dot{m}_e h_e - \dot{m}_i h_i \quad (3.7)$$

3.2 Conservation equations - fin side

On the fin side there is water vapour entrained in the air stream. Under certain conditions the water vapour can condense and form a liquid film on the surface of the heat exchanger. The incremental control volume shown in Figure 3.2 is used for analysis of this condition.

The integral form of the conservation equations used in the subsequent sections are from Rousseau & van Eldik (2011).

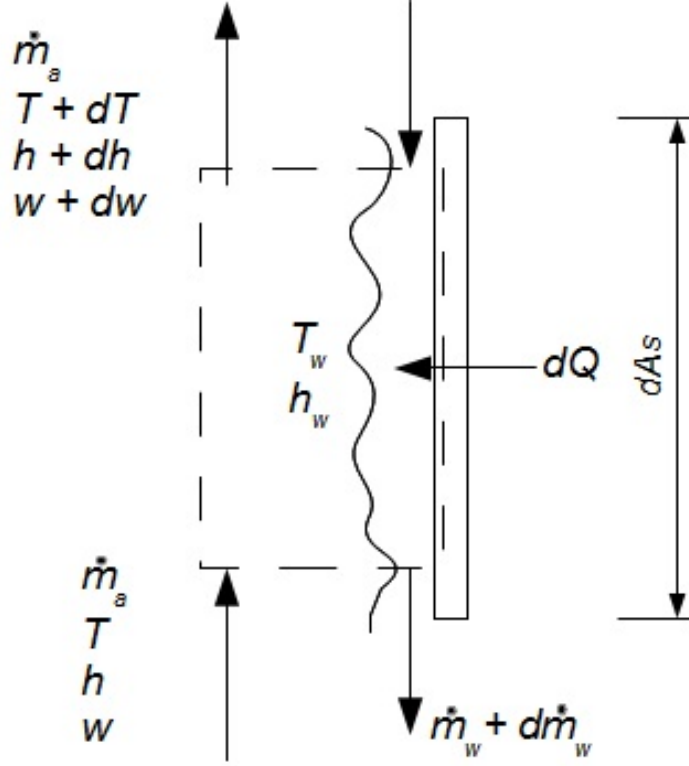


Figure 3.2: control volume, fin side

3.2.1 Conservation of mass

For dry air, the conservation of mass is derived in a similar manner as in Section 3.1.1 and can be written as:

$$\dot{m}_{a,e} - \dot{m}_{a,i} = 0 \quad (3.8)$$

The subscript $_a$ is used to indicate dry air.

For the finite control volume in Figure 3.2, the integral form of the mass conservation for the water on the fin side is given by

$$\frac{\partial}{\partial t} \left(\int \rho_w dV_w \right) + \oint \rho_w \bar{V} \cdot \overline{dA}_w + \oint \rho_w \bar{V} \cdot \overline{dA} = 0 \quad (3.9)$$

The following generic, steady state equation, can then be derived for the conservation of mass for water

$$\dot{m}_{w,e} - \dot{m}_{w,i} + \dot{m}_{a,e}w_e - \dot{m}_{a,i}w_i = 0 \quad (3.10)$$

where w is the specific humidity ratio and the subscript $_w$ denotes water.

3.2.2 Conservation of momentum

The condensate flow rate is extremely low and the contribution of water film to the momentum change of the air stream can be assumed negligible. Therefore the momentum change of moist air can be calculated on the same basis as in equation (3.4).

In general, the air side pressure drop through the coil is presented as a function of the fanno friction factor and can be written as (Iu 2007),

$$\Delta P_a = \frac{G_a^2}{2\rho_i} \left[(K_i + 1 - \sigma^2) + 2 \left(\frac{\rho_i}{\rho_e} - 1 \right) + f \left(\frac{A_{tot}}{A_c} \frac{\rho_i}{\rho_e} \right) - (1 - \sigma^2 - K_e) \frac{\rho_i}{\rho_e} \right] \quad (3.11)$$

where σ is the ratio of the minimum free-flow (cross sectional) area to the frontal area

$$\sigma = \frac{A_c}{A_{fr}} \quad (3.12)$$

and K_i and K_e are the entrance and exit loss coefficients. These coefficients vary depending on the inlet and exit geometry of the evaporator. For elements inside the evaporator these coefficients do not apply and for purposes of this study it will be assumed that $K_i = K_e = 0$

G is the mass flux through the finned passage and is given by:

$$G_a = \rho V = \frac{\dot{m}}{\sigma A_{fr}} \quad (3.13)$$

3.2.3 Conservation of energy

Both the air stream and water film have energy transport associated with them. The integral form of the control volume, as shown in Figure 3.2, is given by:

$$\begin{aligned} \dot{Q} + \dot{W} = & \frac{\partial}{\partial t} \left(\iiint (u_w + \frac{1}{2}V_w^2 + gz_w) \rho_w dV_w \right) + \oiint (h_w + \frac{1}{2}V_w^2 + gz_w) \rho_w \bar{V}_w \cdot \bar{dA}_w \\ & + \frac{\partial}{\partial t} \left(\iiint (u + \frac{1}{2}V^2 + gz) \rho dV \right) + \oiint (h + \frac{1}{2}V^2 + gz) \rho \bar{V} \cdot \bar{dA} \end{aligned} \quad (3.14)$$

For steady state conditions, with no work being done on the air and with no change in elevation the following generic equation can be derived (Oliet et al. 2007),

$$\dot{Q} = \dot{m}_{a,e} h_e - \dot{m}_{a,i} h_i + \dot{m}_{w,e} h_{w,e} - \dot{m}_{w,i} h_{w,i} \quad (3.15)$$

where h denotes enthalpy of moist air. Since the liquid water on the fin surface will be at the same temperature as the fin surface. For an evaporator the saturated temperature and resultant temperature is almost constant throughout. The third and fourth term dealing with the change in enthalpy of the liquid water is therefore negligible and the equation for conservation of energy can be simplified to:

$$\dot{Q} = \dot{m}_{a,e}h_e - \dot{m}_{a,i}h_i \quad (3.16)$$

3.3 Element pressure drop

As shown in equation (3.4) the total pressure drop consists of three distinct terms. As previously indicated, the $\Delta p_{static} = 0$. In the following sections Δp_{mom} and Δp_{fric} will be determined.

3.3.1 Single phase pressure drop

As determined in the literature review, the same correlations will be used to determine the pressure drop for all instances of single phase flow. Thus for water flow, refrigerant liquid flow and refrigerant vapour flow the following apply.

Momentum pressure drop

The momentum pressure drop reflects the change in kinetic energy of the fluid (Didi et al. 2002). For single phase flow it follows that:

$$\Delta p_{mom} = G^2 \left(\frac{1}{\rho_e} - \frac{1}{\rho_i} \right) \quad (3.17)$$

For water and liquid refrigerant, the change in density is negligible and the momentum pressure drop term can be ignored for most applications. Similarly the small density changes taking place for superheated R-744 will also be ignored.

G is the mass flux and is defined as

$$G = \frac{\dot{m}}{A_c} \quad (3.18)$$

where $\dot{m}$ is the mass flow rate and A_c is the cross sectional flow area perpendicular to the direction of the flow.

Friction pressure drop

Δp_{fric} is the dominant factor in the total pressure drop calculation and is determined in terms of a friction factor

$$\Delta p_{fric} = 2f \frac{L}{D_h} \frac{G^2}{\rho} \quad (3.19)$$

where f is the friction factor, D_h the hydraulic diameter and L the length of the element.

The hydraulic diameter is defined as

$$D_h = \frac{4A_c}{P_w} \quad (3.20)$$

where P_w is the wetted perimeter. Note that for the special case of flow inside a circular pipe $D_h = D$.

The friction factor is calculated using the recently developed Ghanbari, Farshad and Rieke correlation (Ghanbari et al. 2011)

$$f = \left(-1.52 \log \left(\left(\frac{e/D_h}{7.21} \right)^{1.042} + \left(\frac{2.731}{Re} \right)^{0.9152} \right) \right)^{-2.169} \quad (3.21)$$

e is the relative roughness of the tube wall and Re is the Reynolds number. This equation is applicable for the range $2100 < Re < 10^8$ and $0 < e < 0.05$.

The Reynolds number provides an indication of the ratio of inertia to viscous forces and is defined as:

$$Re = \frac{\rho V D_h}{\mu} = \frac{G D_h}{\mu} \quad (3.22)$$

Return bend pressure drop

For single phase refrigerant flow in a return bend, the correlation of Paliwoda (1992), as presented by Iu (2007), is used:

$$\Delta p_{rb} = \frac{1}{2} K_{dp} \frac{G^2}{\rho} \quad (3.23)$$

where K_{dp} is the curve fit correction factor

$$K_{dp} = \frac{1}{3.426 \ln \left(\frac{S_x}{2D} \right) + 3.8289} \quad (3.24)$$

S_x is the tube pitch of the heat exchanger and D is the internal diameter of the tubes.

3.3.2 Two phase pressure drop

For the two phase pressure drop, the homogeneous Bo Pierre method, as modified and presented by Choi et al. (1999) is used. This method calculate the total pressure drop directly:

$$\Delta p_{tot} = \Delta p_{mom} + \Delta p_{fric} = \left(\frac{f_N L (v_e + v_i)}{D_h} + (v_e - v_i) \right) G^2 \quad (3.25)$$

The specific volume v of the two phase fluid is obtained by a quality weighted sum of the vapour and liquid specific volumes at either the inlet i or outlet o of an element with a length L ;

$$v = \frac{x}{\rho_v} + \frac{1-x}{\rho_l} \quad (3.26)$$

The mass flux G and the two phase friction factor F_N are evaluated at the average refrigerant temperature. The two phase friction factor is

$$f_N = 0.00506 \text{Re}_l^{-0.0951} K_f^{0.1554} \quad (3.27)$$

In the above equation, the Reynolds number is used as an all liquid Reynolds number:

$$\text{Re}_l = \frac{G D_h}{\mu_l} \quad (3.28)$$

The two phase number K_f is

$$K_f = \frac{(x_e - x_i) h_{fg}}{Lg} \quad (3.29)$$

where x is quality of the two phase fluid, g is the gravitational constant and h_{fg} is the latent heat of evaporation.

Choi et al. (1999) do not provide any limits for their correlation. The original Bo Pierre correlation was only applicable for ratio of $Re/K_f > 1$ (Jung & Radermacher 1989) and it is assumed that these limits are still applicable for the new correlation. This usually only happens where L is large and the use of the elemental approach circumvents this restriction.

Return bend pressure drop

Pressure drop in bends is of the same order of magnitude as the pressure drop in straight tubes and needs to be taken into account. The method of Domanski & Hermes (2006) calculated the pressure drop gradient through the bend $\left. \frac{dp}{dl} \right|_{rb}$ as if the bend is a straight

tube and then use a curvature multiplier (Λ) to account for the bend curvature. This correlation is applicable for qualities in the range $0.2 < x < 0.8$. The pressure drop gradient is defined as follows:

$$\left. \frac{dp}{dl} \right|_{rb} = \Lambda \left. \frac{dp}{dl} \right|_{st} \quad (3.30)$$

For the straight tube pressure drop gradient $\left. \frac{dp}{dl} \right|_{st}$, they used the method of Muller-Steinhagen and Heck. Multiplying the above result with the length of the return bend $L_{rb} = \pi R$ the pressure drop is obtained.

$$\Delta p_{rb} = \Lambda L_{rb} \left. \frac{dp}{dl} \right|_{st} \quad (3.31)$$

In this study the straight tube pressure drop for two phase flow is calculated using equation (3.25) replacing L in equation (3.29) with L_{rb}

The curvature multiplier is

$$\Lambda = a_0 \left(\frac{GxD}{\mu_v} \right)^{a_1} \left(\frac{1}{x} - 1 \right)^{a_2} \left(\frac{\rho_l}{\rho_v} \right)^{a_3} \left(\frac{2R}{D} \right)^{a_4} \quad (3.32)$$

where $a_0 = 5.2 \cdot 10^{-3}$, $a_1 = 0.54$, $a_2 = 0.21$, $a_3 = 0.34$, $a_4 = -0.67$ and R is the radius of the bend.

For qualities outside the applicable range of $0.2 < x < 0.8$ the average between the bend pressure drop calculated by equation (3.23) and equation (3.31) will be used. For cases where the two phase bend pressure drop predicted is lower than the single phase bend pressure drop, the single phase bend pressure drop will be used.

3.4 Element heat transfer

From Fourier's law the following generic equation, for heat transfer due to a temperature difference, can be derived

$$\dot{Q} = R_t \Delta T \quad (3.33)$$

where R_t is the thermal resistance and ΔT is the temperature difference. Different R_t values can be formulated to account for conduction, convection and radiation heat transfer.

Using equation (3.33) in conjunction with equations (3.7) and (3.16) the outlet conditions for each fluid stream and the total heat transfer between the fluid streams can be determined.

Heat transfer in a heat exchanger is driven by temperature difference. For an annular tube-in-tube heat exchanger this is solely due to the temperature difference between the tubes. In a fin-and-tube heat exchanger, however, there is also conduction taking place, through the fins, between heat exchanger tubes at different temperatures. The total heat transfer is give by

$$\dot{Q}_{tot} = \dot{Q}_f + \dot{Q}_c \quad (3.34)$$

where $\dot{Q}_f$ represents the heat transfer due to a temperature difference between the fluid streams and $\dot{Q}_c$ represent the conducted heat transfer through the fins due to a temperature difference between the tubes. For the purposes of this study $\dot{Q}_c$ is assumed to be negligible.

3.4.1 Overall heat transfer coefficient

For composite systems such as heat exchangers, it is more convenient to work with an overall heat transfer coefficient:

$$\dot{Q}_f = UA\Delta T \quad (3.35)$$

where U is the overall heat transfer coefficient, A is the heat transfer area and ΔT is the temperature difference between the fluid streams.

Although fouling factors and contact resistances can be significant, as discussed in Chapter 2, it will not be taken into account for purposes of this study. The tube wall resistance will, however, be taken into account.

For an annular tube-and-tube, water-to-refrigerant heat exchanger, without any heat transfer enhancement on the tube surfaces, the overall heat transfer coefficient can be given as follows

$$\frac{1}{UA} = \frac{1}{h_{c,r}A_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL} + \frac{1}{h_{c,w}A_o} \quad (3.36)$$

where $h_{c,r}$ and $h_{c,w}$ are the refrigerant and water convection heat transfer coefficients respectively. A_o and A_i is the heat transfer area corresponding to D_o and D_i respectively ($A = \pi DL$), k the conductivity of the tube wall and L the length of the element.

Similarly, for a finned-tube, air-to-refrigerant heat exchanger without heat transfer enhancement on the refrigerant side, the overall heat transfer coefficient is

$$\frac{1}{UA} = \frac{1}{h_{c,r}A_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL} + \frac{1}{\eta_o h_{c,a}A_o} \quad (3.37)$$

with $h_{c,a}$ being the air side convection heat transfer coefficient, η_o the overall surface efficiency and A_o the total outside heat transfer area. Refer to Sections 3.4.4 and 3.4.5 for details on the determination of the convection heat transfer coefficients. The calculation of η_o for a finned surface is discussed in Section 3.4.3.

For a totally dry process, the process takes place at constant specific humidity, thus the inlet and outlet specific humidity stays constant.

3.4.2 Equivalent dry-bulb temperature method

For totally wet conditions obtained during cooling under dehumidifying conditions, the equivalent dry bulb temperature method is used to calculate the heat transfer.

For any cooling process involving moist air with specific inlet and outlet states, an equivalent dry process with an identical cooling capacity can be defined (Wang & Hihara 2003),(Huzayyin et al. 2007). This is illustrated in Figure 3.3. The inlet and outlet air states are represented by point 1 and 2 respectively. For the cooling process $1 - 2 - 3$, the equivalent dry process is $1^e - 2^e - 3$. The lines $1 - 1^e$ and $2 - 2^e$ are constant enthalpy lines.

As shown by Wang & Hihara (2003) using the equivalent dry bulb temperature method, the heat transfer is given by

$$\dot{Q} = UA_w (T_{ad}^e - T_r) \quad (3.38)$$

where T_{ad}^e is the equivalent dry bulb temperature and the overall heat transfer coefficient for the totally wet condition, UA_w is given by:

$$\frac{1}{UA_w} = \frac{1}{h_{c,r}A_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2\pi kL} + \frac{1}{\eta_{o,w}h_{c,w}A_o} \quad (3.39)$$

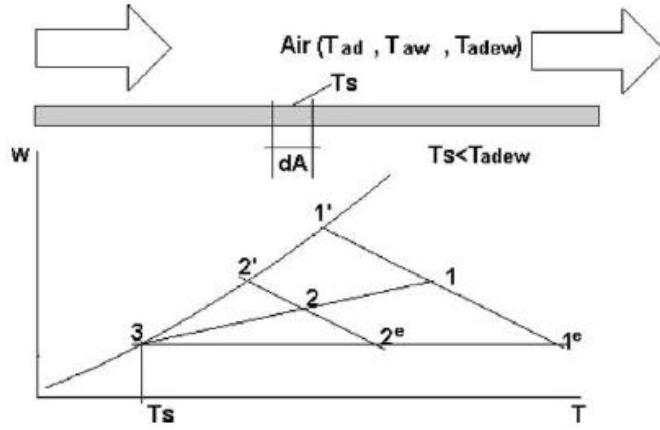


Figure 3.3: Cooling and dehumidifying of moist air over a flat plate (Wang & Hihara 2003)

The only change between equation (3.37) and (3.39) is the use of a different fin efficiency and convection heat transfer coefficient to account for the wet condition.

It must be noted that during de-humidification there forms a thin water layer on the fin-side. The thermal resistance to the water layer has been ignored.

Combining equation (3.38) with equation (3.16), it is possible to determine the outlet enthalpy of the air stream directly. It is, however, still necessary to obtain an additional parameter to fix the outlet air condition. For the purposes of this study it will be assumed that the fin temperature and water film temperature is equal to the tube wall temperature.

The outlet drybulb temperature can then be approximated by linear interpolation from the psychometric chart, shown in Figure 3.3.

$$\frac{T_{a1} - T_{a2}}{T_{a1}^e - T_{a2}^e} = \frac{T_{a1} - T_s}{T_{a1}^e - T_s} \quad (3.40)$$

where T_a is the dry bulb air temperature for point 1 and point 2, T_a^e the equivalent dry bulb temperatures and T_s the surface temperature.

With the outlet dry bulb temperature and enthalpy known, the outlet conditions of the air stream can be fully defined. If required, the mass transfer can be calculated based on the difference of the specific humidity ratio between the inlet and outlet conditions.

3.4.3 Surface efficiency

The surface efficiency for a finned surface is defined as:

$$\eta_o = 1 - \frac{A_f}{A_o} (1 - \eta_f) \quad (3.41)$$

The fin efficiency is calculated based on the method recommended by Huzayyin et al. (2007) and Oliet et al. (2007) with

$$\eta_f = \frac{\tanh(Mr_i\phi) \cos(0.1Mr_i\phi)}{Mr_i\phi} \quad (3.42)$$

where

$$M = \sqrt{\left(\frac{2h_c}{k_f t_f}\right)} \quad (3.43)$$

for dry conditions $C_w = 1$. For fully wet conditions C_w , is the slope of the saturated enthalpy curve at the fin temperature and is given by:

$$C_w = \frac{dh/dT}{C_p} \quad (3.44)$$

Determining C_w adds complexity as this requires the fin temperature to be calculated. Schmidt's approximation also did not take factor into account. For the purposes of this study $C_w = 1$ will be used under dry and wet conditions.

ϕ is given by

$$\phi = \left(\frac{r_o}{r_i} - 1\right) \left(1 + 0.35 \ln\left(\frac{r_o}{r_i}\right)\right) \quad (3.45)$$

For a staggered tube layout the octagonal fin can be represented by an equivalent circular fin. The approximation of Schmidt as presented by Oliet et al. (2007) is used:

$$r_{o,eq} = 1.27X_M \sqrt{\frac{X_L}{X_M} - 0.3} \quad (3.46)$$

where X_M is

$$X_M = 0.5S_y \quad (3.47)$$

and X_L is

$$X_L = 0.5\sqrt{(0.5S_y)^2 + S_x^2} \quad (3.48)$$

3.4.4 Single phase heat transfer

The heat transfer coefficient is calculated from the Nusselt number. The Nusselt number (Nu) is a non-dimensional number that provides a measure of the ratio of convection to conduction heat transfer and is defined as

$$Nu = \frac{h_c D_h}{k} = f(Re, Pr) \quad (3.49)$$

where the Nusselt number is a function of the non-dimensional Reynolds (Re) and Prandtl (Pr) numbers. The Re number is defined by equation (3.22) and h_c is the heat transfer coefficient.

The Pr number provides a measure of the ratio of momentum and thermal diffusivities and is defined as

$$Pr = \frac{c_p \mu}{k} \quad (3.50)$$

with c_p the heat capacity, μ the viscosity and k the conductivity of the fluid.

Except for the case of laminar flow where it is possible to determine the Nusselt number analytically the Nusselt number is provide by empirical correlations. For all single phase flows (water, refrigerant liquid and vapour) inside a tube, with diameter D the Nusselt number correlation used, is the correlation of Gnielinski as presented by Incropera et al. (2007)

$$Nu_D = \frac{\left(\frac{f}{8}\right) (Re_D - 1000) Pr}{1 + 12.7 \left(\frac{f}{8}\right)^{\frac{1}{2}} \left(Pr^{\frac{2}{3}} - 1\right)} \quad (3.51)$$

where f is the friction factor as defined by equation 3.21. The friction factor, Reynolds number and Prandtl number are evaluated at the average fluid temperature.

3.4.5 Two phase heat transfer

For two phase flow the heat transfer equation of Jung et al. (1989) was chosen. They define the heat transfer correlation for pure refrigerant as follows

$$h_{tp} = h_{nbc} + h_{cec} = N h_{SA} + F_p h_{lo} \quad (3.52)$$

where N , h_{SA} , F_p and h_{lo} are defined in equations (3.53), (3.56), (3.58) and (3.60).

N is a factor due to the nucleate boiling effect and is represented using the dimensionless Martinelli parameter (X_{tt}) and the boiling number (B_o).

$$N = f(X_{tt}, B_o) = 4048 X_{tt}^{1.22} B_o^{1.13} \quad (3.53)$$

The Martinelli parameter is obtained from the following equation, as used by Jung et al. (1989)

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_v}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_v} \right)^{0.1} \quad (3.54)$$

and the Boiling number is given by:

$$B_o = \frac{\dot{Q}}{G h_{lv}} \quad (3.55)$$

where h_{lv} is the latent heat of vaporization of the refrigerant.

h_{SA} is the pool boiling heat transfer and the correlation of Stephan and Abdelsalam:

$$h_{SA} = 207 \frac{k_l}{bd} \left(\frac{\dot{Q} bd}{k_l T_s} \right)^{0.745} \left(\frac{\rho_v}{\rho_l} \right)^{0.581} \text{Pr}_l^{0.533} \quad (3.56)$$

where bd is

$$bd = 0.0146 \beta \left(\frac{2\sigma}{g(\rho_l - \rho_v)} \right)^{0.5} \quad (3.57)$$

and the contact angle $\beta = 35^\circ$

The F factor, as used in equation (3.52), has the following correlation for $x_{tt} < 1$

$$F_p = 2.37 \left(0.29 + \frac{1}{X_{tt}} \right)^{0.85} \quad (3.58)$$

and for $1 < x_{tt} < 5$ (Oh & Son 2011)

$$F_p = 2 - 0.1 (X_{tt})^{-0.28} (B_o)^{-0.33} \quad (3.59)$$

and lastly h_l is determined using the Dittus-Boelter equation

$$h_l = 0.023 \frac{k_l}{D_h} \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \quad (3.60)$$

where Re_l is determined by substituting G in equation (3.22) with $G(1-x)$.

3.4.6 Air side heat transfer correlations

The fin-side heat transfer is typically calculated using Colburn j-factor correlations. The Nusselt number is a function of the Colburn j-factor, the Reynolds number and the Prandtl number as shown by Rousseau & van Eldik (2011),

$$Nu = jRePr^{\frac{1}{3}} \quad (3.61)$$

The heat transfer coefficient can then be calculated using equation (3.49).

For the fin side under dry surface conditions the Colburn j-factor correlation of Wang et al. (1997) is used.

$$j = \frac{1.201}{[\ln(Re_{Dc}^{\sigma})]^{2.921}} \quad (3.62)$$

where N is the number of tube rows, Re_{Dc} is the Reynolds number for the collar diameter with $D_c = D + 2\delta_f$ and σ is defined as:

$$\sigma = \frac{A_{ff}}{A_{fr}} \quad (3.63)$$

For a totally wet surface condition the Colburn j-factor correlation of Pirompugd, Wongwises & Wang (2009) is used:

$$j = 0.171 \left(\frac{A_{tot}}{A_{tube}} \right)^{0.377N} Re_{Dc}^{(-0.0142N-0.478)} \left(\frac{S_p}{D_c} \right)^{(0.00412N-0.0217)} \left(\frac{A_{tot}}{A_{tube}} \right)^{(-0.114N+0.440)} \quad (3.64)$$

3.4.7 Air side pressure drop

The fin side pressure drop under dry surface conditions is calculated with the correlation of Wang et al. (1997):

$$f = \frac{16.67}{[\ln(Re_{Dc})]^{2.64}} \left(\frac{A_{tot}}{A_{tube}} \right)^{-0.096} N^{0.098} \quad (3.65)$$

Under fully wet surface conditions the pressure drop is calculated by the equation of Kuvannarat et al. (2006):

$$f = 64.0542 Re_{Dc}^{-0.69284} N^{-0.5237} \left(\frac{A_{tot}}{A_{tube}} \right)^{-0.54736} \left(\frac{\delta_f}{S_x} \right)^{-0.098371} \quad (3.66)$$

3.5 Summary

The theoretical background for the detailed simulation of the two different evaporators was presented in this chapter. In the first part of this chapter the conservation laws were discussed, while in the second part the details of the physical and empirical models necessary to fully describe the component characteristics were presented.

The next chapter will provide the detail about how the conservation laws and component models are integrated into the detailed evaporator simulation models.

Chapter 4

Simulation overview

The following section will provide an overview of the implementation of the simulation models. The first simulation model discussed is for the tube-in-tube evaporator and the second model for the fin-and-tube evaporator model. These models will be implemented using EES (Klein 2011).

4.1 Tube-in-tube evaporator

Figure 4.1 provides an overview of a tube-in-tube evaporator with the refrigerant inside the inner tube and the water flowing within the annulus between the tubes.

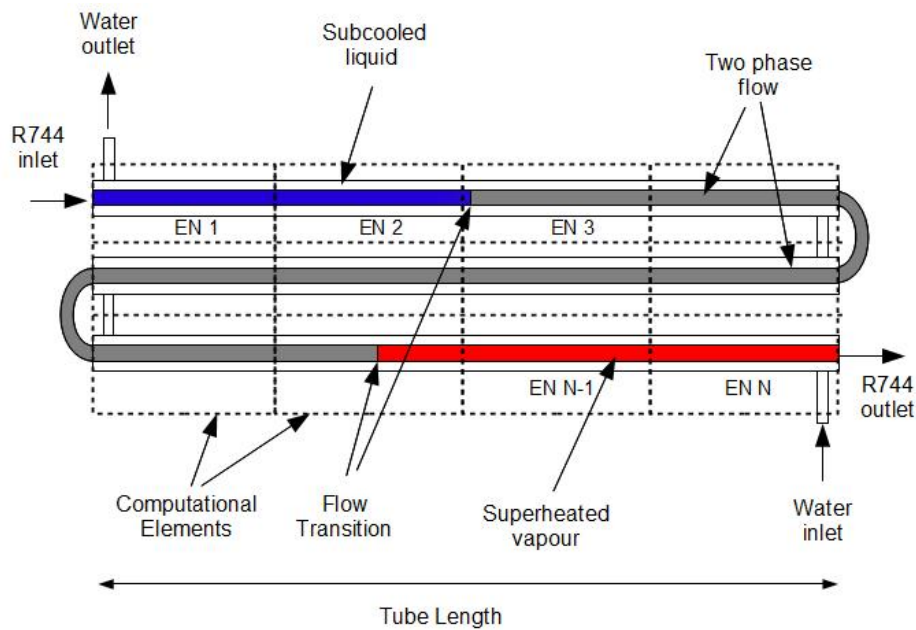


Figure 4.1: Tube-in-tube evaporator overview.

As seen from Figure 4.1 the evaporator consists of several tube sections connected with return bends. In order to take the return bends into account the largest computational element possible is one tube. For improved accuracy each of these tube sections can be further subdivided into an equal number of equal length elements. Each element are then regarded as a sub heat exchanger. The elements are numbered in the direction of the R-744 flow. Thus the inlet element is element one and the outlet element is the product of the number of tubes and number of elements per tube. Although the heat exchanger elements are defined in a counterflow fashion the simulation model can handle a parallel flow simulation as well.

The main assumptions for the model are as follows:

- Difference in elevation is negligible.
- No heat transfer to the environment.
- Return bends are adiabatic.
- Steady state heat transfer process.
- Axial conduction is negligible.
- Constant thermal conductivity for the tube material.
- Fluid properties are constant for each element.

For the most part these assumptions are a good approximation of reality. In most cases heat exchangers are installed in a horizontal position where only the distance between the tubes effect the elevation. Even then the effect is still negligible with regards to the overall pressure drop. The water side outside tube will also be thoroughly insulated ensuring limited heat transfer to the environment. The evaporation temperature between segments is almost constant thus negligible heat transfer takes place in the direction of the tube. Similarly the tube wall conductivity can be assumed constant.

The constant fluid property assumption breaks down for transition elements. As seen from Figure 4.1 the transition can take place anywhere inside the element. As discussed in section 2.4.3, this simulation model will not treat these transition elements separately.

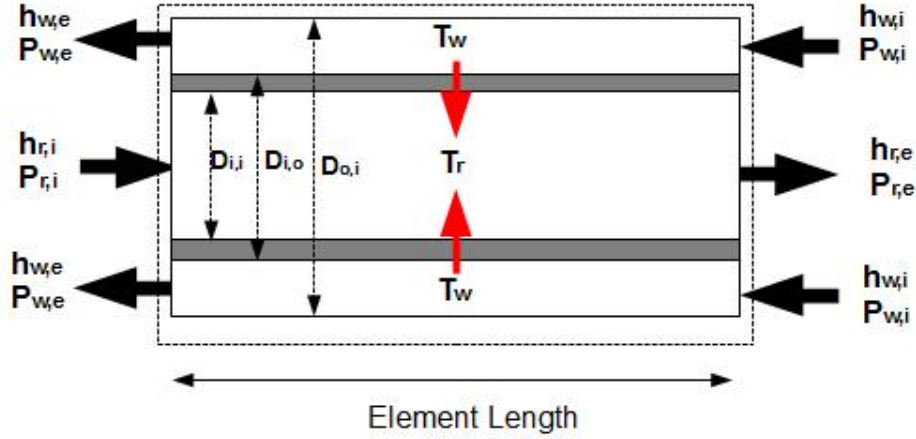


Figure 4.2: Tube-in-tube element detail.

The detail for a computation element is given in Figure 4.2. All fluid properties are calculated at the average enthalpy and pressure for the element. The temperature differences, heat transfer coefficients, total heat transfer and pressure drop for each segment is calculated using these average properties. It must be noted that as discussed in Section 2.4.3 this simulation will not make use of the effectiveness-NTU method but will solve directly for the heat transfer using the equations in Chapter 3.

The refrigerant outlet conditions for element one is the inlet conditions for element two. As the evaporator is counterflow the water outlet conditions for element two is the inlet conditions for element one.

Where a bend is encountered as between element four and element five the refrigerant inlet pressure of element five is the sum of the outlet pressure of four and the pressure drop through the bend. All bends are assumed adiabatic and no heat transfer takes place between the bends and the environment.

Thus in order to simulate a tube-in-tube, air-to-refrigerant evaporator, using the elemental approach at least the following inputs is required:

Water inputs

- Inlet enthalpy.
- Inlet pressure.
- Mass flow rate of water.

Refrigerant inputs

- Inlet enthalpy.
- Inlet pressure.
- Mass flow rate of R-744.

Geometry inputs

- Outer tube inside diameter.
- Inner tube inside diameter.
- Inner tube wall thickness.
- Tube material.
- Tube length.
- Number of tubes.
- Number of elements per tube section.

Since the inlet conditions of both the water and refrigerant are known it is possible to calculate the inlet and outlet conditions for each sub element and using the average conditions for the element, calculate the heat transfer and pressure drop for the element. The build-in solver of EES is able to handle this iterative process.

4.2 Finned tube evaporator

Figure 4.3 provide an overview of a finned tube evaporator, with the refrigerant inside the tubes and air flowing over the finned tubes.

As seen from Figure 4.3 the evaporator consists of several finned tube sections connected with return bends. In order to take into account the return bends the largest computational element possible is one tube. For improved accuracy each of these tube sections can be further subdivided into an equal number of equal length elements. Each element is then regarded as a sub heat exchanger. The elements are numbered in the direction of the R-744 flow. Thus the inlet element is element one and

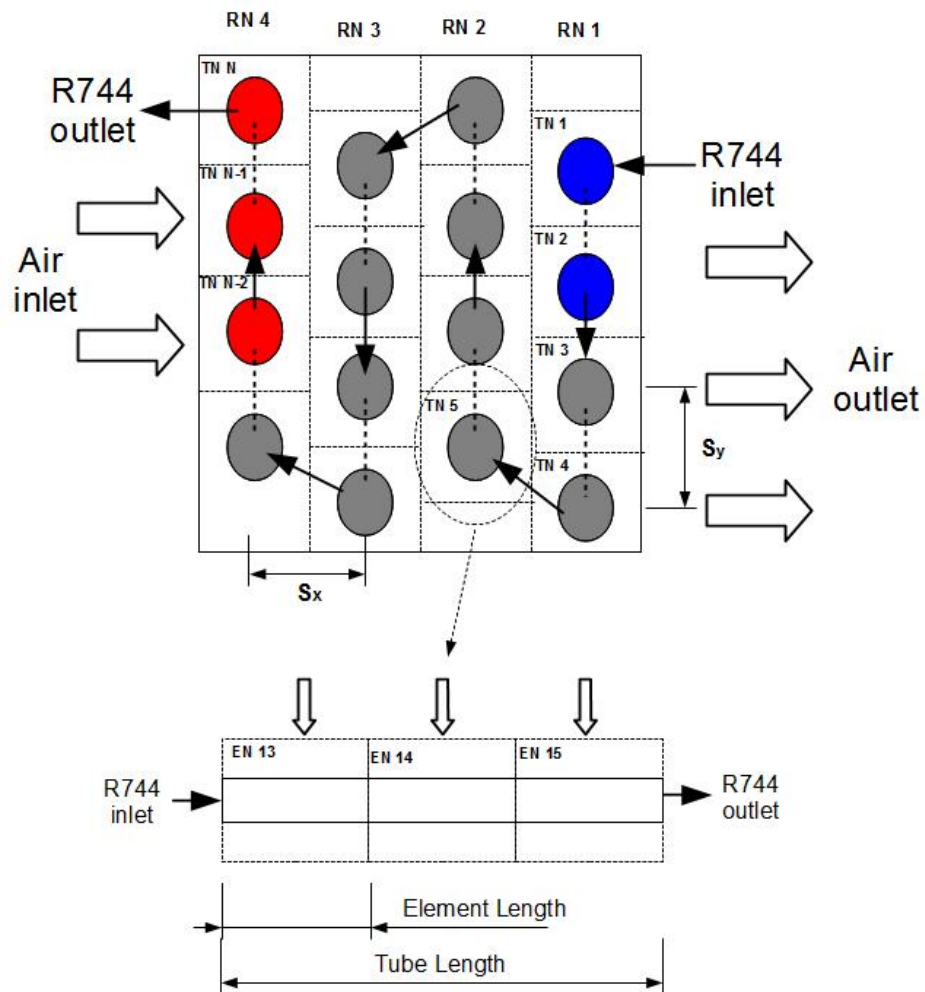


Figure 4.3: Finned tube evaporator overview.

the outlet element is the product of the total number of tubes and number of elements per tube. The total number of tubes is the product of the number of rows and the number of tubes per row.

For ease of simulation all finned tube evaporators are modelled as a single counterflow circuit. This eliminates the need of special algorithms to handle refrigeration circuitry.

The same assumptions as in Section 4.1 apply for the finned tube heat exchanger with a few additions:

- Difference in elevation is negligible.
- No heat transfer to the environment.
- Return bends are adiabatic.
- Steady state heat transfer process.
- Axial conduction is negligible.
- Constant thermal conductivity for the tube material.
- No conduction between tube rows.
- Uniform flow distribution over the coil.
- Air side ambient pressure is constant through the coil.
- The fin surface of an element is either completely dry or completely wet.

For details about these assumptions refer to Section 4.1. Some of the assumptions are explained in more detail below:

- For a typical evaporator coil the heat transfer area is mostly in contact with the air stream, with only a small fraction in contact with the environment, ensuring limited heat transfer to the environment.
- Although the airflow distribution over coils are rarely uniform in practice, this assumption simplifies the model and will be applied in this study.

- The air side ambient pressure are orders of magnitude higher than the pressure drop through the coil. To simplify the model the effect of air side pressure drop is not taken into account for the capacity calculation.
- The fins surface for any elements can be dry, partially wet or completely wet. Due to a lack of correlations for the partially wet region, elements will be assumed either completely dry or completely wet. Whenever the outside surface temperature of the tube is lower than the dew-point temperature of the air stream the element will be assumed to be completely wet.

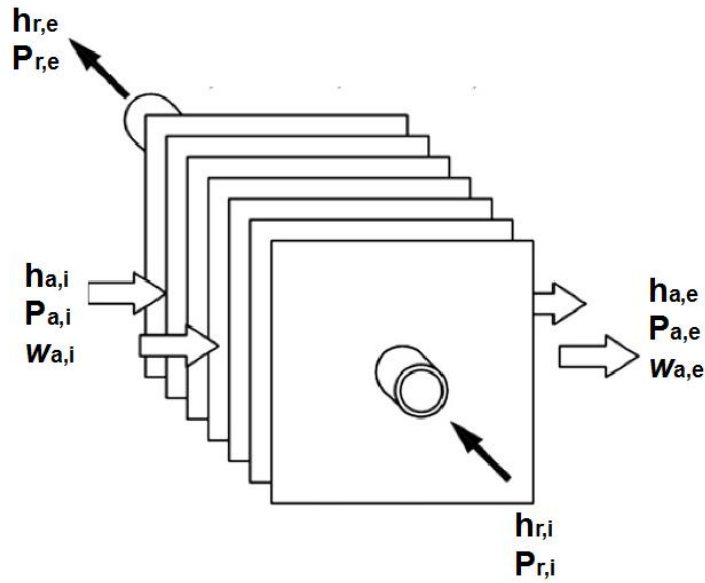


Figure 4.4: Finned tube element detail (Ding et al. 2011).

The detail for a computation element is given in Figure 4.4. The fluid properties for both air and refrigerant are calculated at the average enthalpy and pressure for the element. The temperature difference, heat transfer coefficients, total heat transfer and pressure drop for each segment are calculated using these average properties. As with the tube-in-tube model (Section 4.1) this simulation will not make use of the effectiveness-NTU method but will solve directly for the heat transfer using the equations in Chapter 3.

The refrigerant outlet conditions for element one is the inlet conditions for element two. Due to the staggered arrangement of the tubes considered in this study, the air side inlet conditions is an average of the outlet conditions of the two proceeding

elements. This is shown in Figure 4.5. This applies for both the air inlet enthalpy and inlet relative humidity.

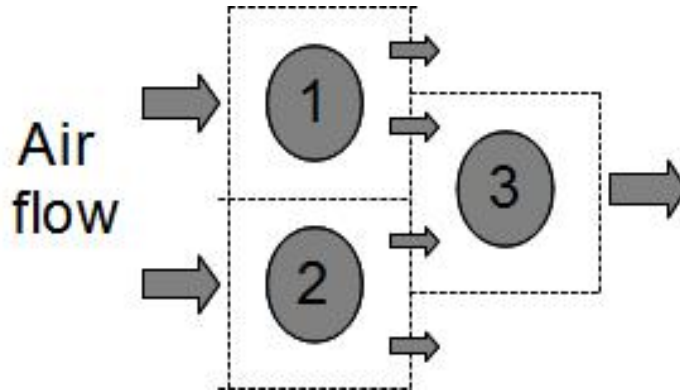


Figure 4.5: Element airflow handling.

Where a bend is encountered, as between all tubes, the refrigerant inlet pressure of the next tube is the sum of the outlet pressure of the previous tube and the pressure drop through the bend. All bends are assumed adiabatic and no heat transfer takes place between the bend and the environment.

In summary, at least the following is required to simulate a fin-and-tube, air-to-refrigerant evaporator:

Air side

- Inlet enthalpy.
- Inlet relative humidity.
- Air pressure.
- mass flow of air.

Refrigerant inputs

- Inlet enthalpy.
- Inlet pressure.
- Mass flow of the refrigerant.

Coil geometry

- *FPI* or fin spacing.
- Fin material.
- Fin thickness.
- Fin characteristics - Wavy for this study.
- Tube diameter.
- Tube wall thickness.
- Tube material.
- Number of rows.
- Number of tubes heigh.
- Tube spacing - distance between tubes.
- Tube lacing - Counterflow, single circuit for this study.

4.3 Summary

The elemental approach was successfully applied to both detailed evaporator simulation models. The geometric and boundary conditions and all inputs required for each simulation model was identified. In both cases the details for each computation elements was provide as well as how each element are linked to the adjacent elements.

The simulation inputs and results are discussed on a case-by-case basis in the next chapter.

Chapter 5

Evaporator model verification

In the first part of this chapter the tube-in-tube evaporator model is verified using experimental data. This verification will serve as validation of the chosen heat transfer and pressure drop correlations for R-744. The finned-tube heat exchanger, using these validated R-744 correlations will then be verified against the program EVAP-COND (NIST 2010). In the last part of this chapter the verified fin-and-tube evaporator model are used to evaluate the different pipe diameters locally available for evaporator coils, to determine the most cost effective solution.

5.1 Tube-in-tube heat exchanger

The R-744 heat pump test bench at the North West University uses a tube-in-tube heat exchanger for an evaporator. Data obtained from the test set-up is used to verify the results for the tube-in-tube heat exchanger model as discussed in the previous sections.

1

5.1.1 Evaporator description

In this section only a brief description on the test bench will be provided. The evaporator consists of a tube-in-tube heat exchanger with evaporating R-744 flowing inside the inner tube and water flowing in the annulus between the tubes. The inner tube is a $\frac{1}{2}$ " schedule 40 stainless steel pipe. For details about the pipe dimensions and working pressures refer to Table 2.4. The outer tube is an 1" copper water pipe with an inner

¹The author was not involved with the testing and cannot vouch for the accuracy of the data.

diameter of 25 mm. The outside of the copper tube is insulated to minimise heat transfer to the environment. The evaporator consists of eight equal length sections of 2 meters in length. Figure 5.1 provide an overview of the evaporator.

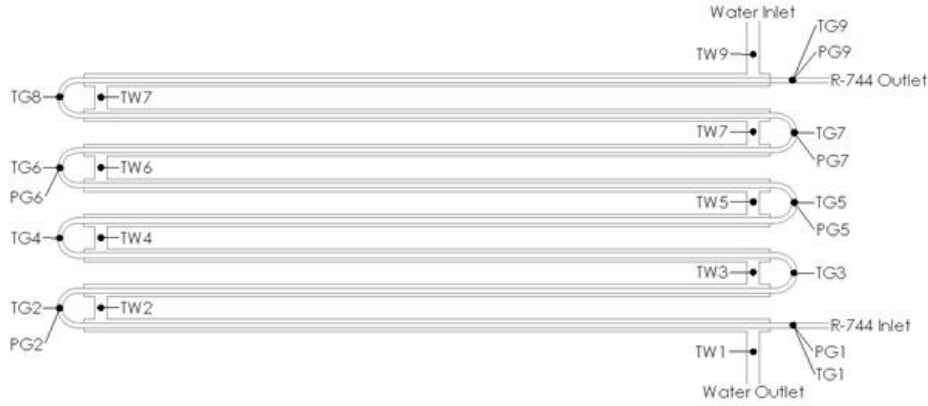


Figure 5.1: Temperature and pressure measurement points for the evaporator test bench at the North West University.

From Figure 5.1 it can be seen that the evaporator is a counter flow configuration with water and R-744 entering at opposite ends. The water temperatures (TW) are measured between each section as well as at the inlet and the outlet. The R-744 temperatures (TG) and pressures (PG) is measured at both the inlet and outlet as well as on some of the return bends between the sections. The mass flow rates of refrigerant and water are measured elsewhere in the system.

Although not discussed here the gas cooler temperatures and pressures were measured in similar detail. The only data that will be used from the gas cooler are the outlet temperature and pressure. Assuming a constant enthalpy expansion process over the expansion device, it is possible to calculate the inlet enthalpy and quality for the evaporator using the outlet conditions of the gas cooler.

5.1.2 Data reduction

The dataset received from the NWU contained data for a total of 99 separate test runs. This dataset cover a large set of operational conditions, with typical variables being mass flux, heat flux and saturation temperature.

In some of the tests at lower saturated pressures, freezing of water in the evaporator coil took place. As freezing in the evaporator will introduce additional heat transfer resistances, these data sets were discarded.

Due to the wide range of test conditions the evaporator was completely oversized for some of the operational conditions and the outlet gas temperature approached the inlet water temperature. Under these conditions, where the heat exchanger effectiveness is very high, the calculated heat exchanger capacity will approach the measured heat exchanger capacity regardless of the accuracy of the heat transfer correlations used. It was therefore decided to disregard any datasets where the outlet gas temperature is within 1°C of the inlet water temperature.

The energy balance between the water side and the R-744 side was also evaluated against each other, for the complete heat exchanger. In some of the datasets discrepancies larger than 15% were found. All datasets where the energy balances were not within 2% of each other were disregarded. Lastly, as the aim is to evaluate two phase flow, all datasets where the inlet quality was higher than 0.85 was disregarded.

Although these changes eliminated many datasets it will ensure that the results obtained using the remaining data sets are more accurate. The size of the database was reduced from 99 sets to 10 sets.

The details of these 10 datasets are provided in Table 5.1.

Table 5.1: Datasets used from the tube-in-tube evaporator test bench of the North West University.

Run	Gas Cooler		Refrigerant Side					Water side	
	Outlet Temp	Outlet Pressure	Inlet Temp	Mass Flow	Inlet Quality	Mass Flux	Outlet Temp	Inlet Temp	Mass Flow
	[°C]	[Bar]	[°C]	[kg/s]	[-]	[kg/m ²]	[°C]	[°C]	[kg/s]
1	44.4	88.3	1.8	0.139	0.84	713	14.3	15.7	0.316
2	43.3	93.8	-8.9	0.109	0.71	557	10.5	12.0	0.263
3	52.4	104.9	-3.3	0.135	0.79	692	11.2	12.5	0.367
4	54.3	107.1	-3.6	0.134	0.81	685	9.7	11.1	0.317
5	46.3	98.2	-8.8	0.128	0.73	655	11.1	12.3	0.363
6	38.1	94.8	-8.2	0.118	0.51	606	29.2	31.5	0.315
7	33.6	86.8	8.2	0.168	0.36	859	26.8	44.6	0.264
8	31.9	89.7	7.9	0.158	0.31	810	32.9	44.9	0.314
9	37.3	101.3	8.0	0.150	0.38	768	33.2	41.3	0.364
10	42.1	111.4	8.0	0.192	0.44	986	27.7	41.6	0.366

Both the pressure and the temperature of the R-744 at the inlet of the evaporator were measured. Due to some differences between the measured pressure and the saturated pressure corresponding to the measured temperatures, the temperature was used

and the pressure calculated.

5.1.3 Simulation results

The inlet enthalpy is calculated using the condenser outlet temperature and pressure. The inlet quality is then calculated from the inlet enthalpy and temperature for the evaporator.

The simulation was completed for each of the 10 datasets and in each instance the total heat transfer and pressure drop were calculated. The complete program can be found in Appendix A.

As seen from Figure 5.2 the simulated capacity closely match the experimental capacity for run 1 to run 6 and over predict the evaporator capacity for run 7 to run 10. This over prediction is much more severe than indicated by the percentage difference between the simulated and experimental capacities. For run 7 to run 10 the simulated capacity approach the maximum capacity possible for the heat exchanger, with the superheated R-744 approaching the water inlet temperature.

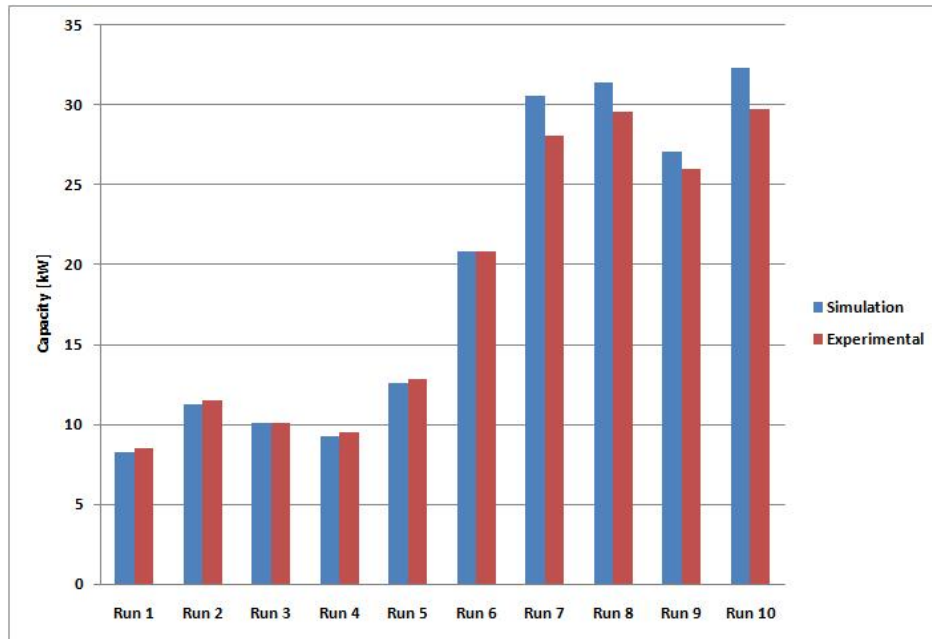


Figure 5.2: Results of simulated and experimental R-744 evaporator capacities.

The heat transfer results are discussed in detail in Section 5.1.5.

From Figure 5.3 it is clear that the simulated R-744 pressure drop and experimental pressure drop differs by an order of magnitude. This implies that either an error

was introduced during measurement of the experimental values or the pressure drop correlation completely fails to predict the pressure drop. The pressure drop results are discussed in more detail in Section 5.1.6

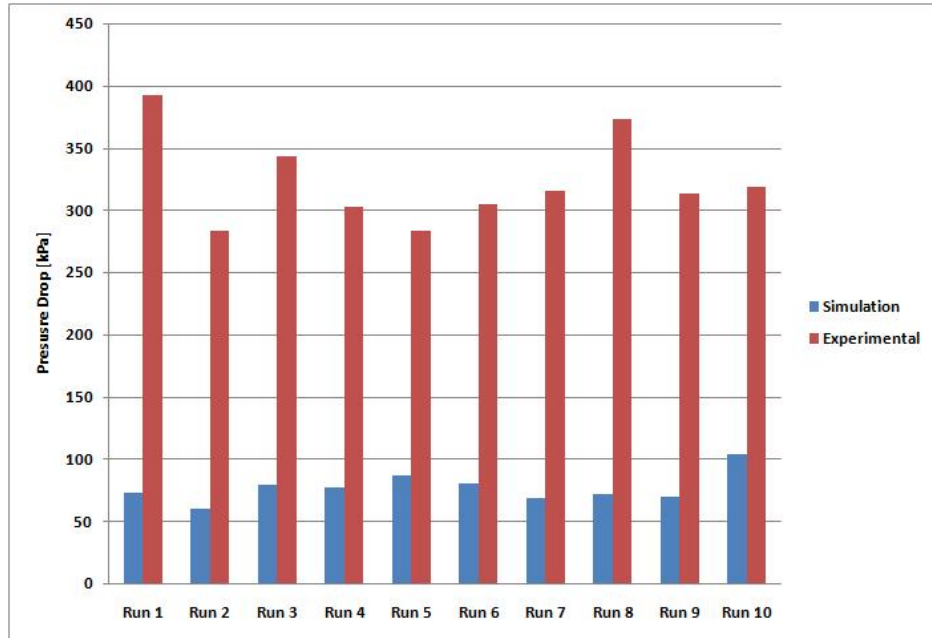


Figure 5.3: Results of simulated and experimental R-744 pressure drop.

5.1.4 Effect of number of elements

An increase in elements will typically improve accuracy but with the downside of increased solving time. The number of elements also increase the number of variables used in EES. This requires additional guess values for each extra variable and there is also an upper limit on the total number of variables in EES. As the heat exchanger exists out of several tube sections the smallest amount of elements possible is one element per tube section and thereafter elements are added to all tube sections.

Before continuing with the detail calculations it is necessary to determine what the minimum number of elements are that need to be used in simulating the heat exchanger accurately. This was done by solving run 7 for different numbers of elements.

Figure 5.4 give the results for capacity and Figure 5.5 the results for pressure drop with regards to the number of elements.

The overall capacity decrease slightly with an increase in the number of elements. By increasing from one element per tube to 3 elements per tube the capacity is changed

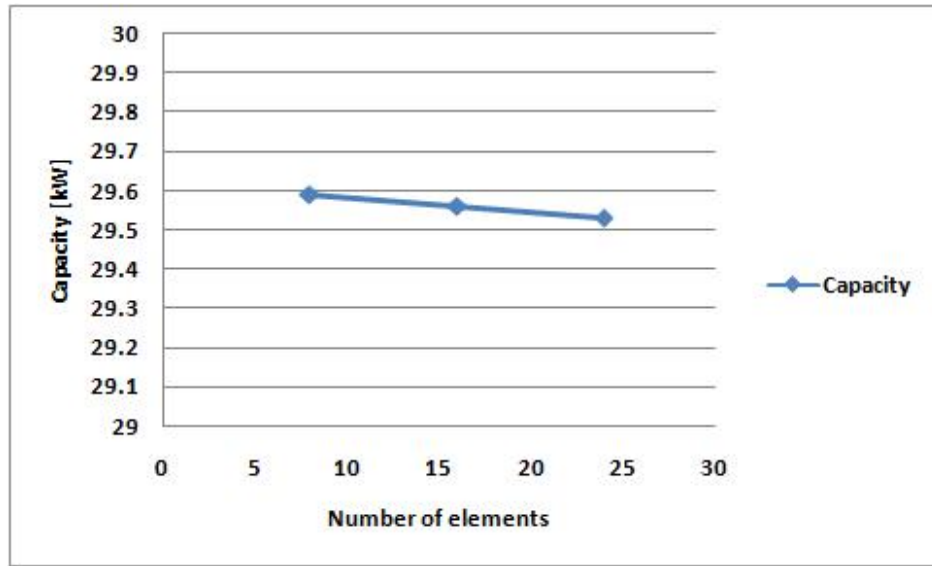


Figure 5.4: Capacity vs number of elements.

by only 0.2% in total. Further increasing the number of elements will have an even smaller effect on the capacity.

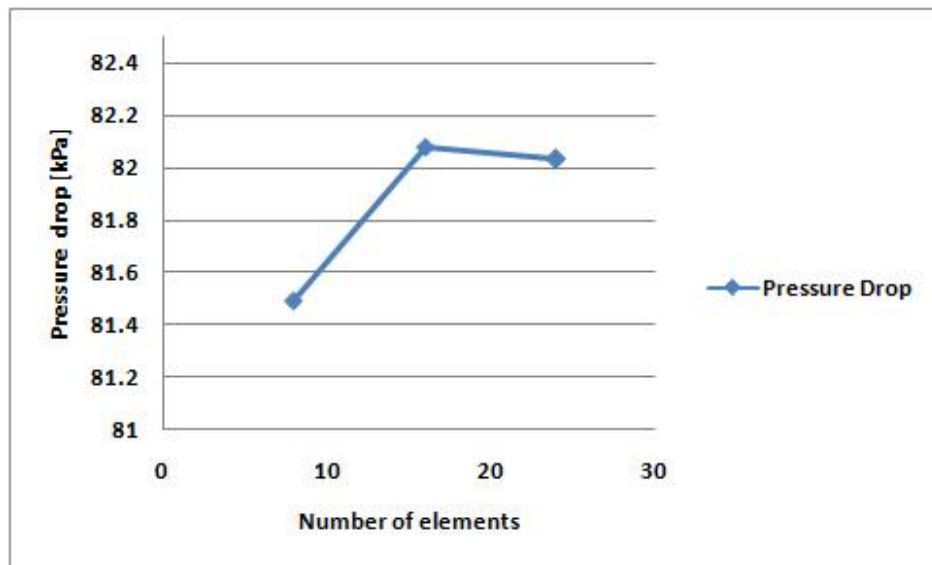


Figure 5.5: Pressure drop vs number of elements.

The pressure drop first increased and then slightly decreased with an increase in the number of elements. The first change is 0.66% with the second change being smaller than 0.1%.

As seen the change in overall capacity and pressure drop is negligible for an increase in the number of elements. The only distinct change was the pressure drop change for two elements per tube. The rest of the simulations were all done with two elements per

tube as this gave a good trade-off between smallest number of elements and improved accuracy.

5.1.5 Heat transfer discussion

As mentioned previously the simulation results for run 7-10 are completely over predicted. There are several factors that need to be taken into account that can explain this over prediction:

- The R-744 heat transfer equations do not apply for the operational conditions of the heat exchanger. This includes the effect of mass flux, heat flux and tube diameter.
- Fouling factors was not taken into account. Fouling can take place both on the inner and outer surfaces of the tube. As the evaporator was only operational for the duration of the testing and the water loop is sealed, fouling is not expected to have played a role.
- Oil (lubricant) is present in the evaporator and suppresses R-744 heat transfer.

Operational conditions In Section 2.4.2 the heat transfer and pressure drop equations were selected to apply for refrigerant mass fluxes in the region of 450 to 800 $\text{kg.m}^{-2}.\text{s}^{-1}$ and heat fluxes in the region of 4.5 to 8 kW.m^{-2} .

The chosen heat transfer correlation of Jung et al. (1989) was evaluated by Cho & Kim (2007) for mass fluxes ranging from 212 to 656 $\text{kg.m}^{-2}.\text{s}^{-1}$ and heat fluxes from 6 to 20 kW.m^{-2} and by Mastrullo et al. (2010) for mass fluxes ranging from 200 to 349 $\text{kg.m}^{-2}.\text{s}^{-1}$ and heat fluxes from 10.1 to 20.6 kW.m^{-2} . The tube diameters varied from 5mm to 9.52mm.

From Table 5.1 it can be seen that the mass fluxes for the experimental runs varied from 557 to 986 $\text{kg/m}^2\text{s}$. Typically this is at the upper range or slightly higher the evaluated range of the heat transfer correlation. Oh et al. (2008) found that variations in mass flux have a small impact on the overall heat transfer coefficient. The slightly higher mass flux is therefore not expected to contribute excessively to inaccuracies in the heat transfer.

Table 5.2 present the maximum simulated heat flux in a single tube section, for each run. These maximum values are 3 to 4 times higher than the evaluated range of the correlation. The reasons for the high heat fluxes are twofold. It can be contributed to the high temperature differences between the water and evaporating R-744 as well as the high heat transfer coefficients typically found with evaporating R-744.

Table 5.2: Maximum simulated heat flux for each run.

Run	Heat Flux [kw/m ²]
1	26.31
2	36.11
3	32.43
4	25.88
5	44.52
6	92.03
7	73.99
8	86.8
9	79.41
10	81.77

Oh et al. (2008) observed that for all qualities, the heat transfer coefficient increase with an increase in heat flux and decrease for an increase in quality. The increase in heat transfer coefficient with the increase of heat flux is therefore expected. Keeping all the parameters constant and changing only the heat flux it can be observed from Figure 5.6 that for the heat transfer correlation of Jung et al. (1989) as implemented in this study, follow this trend in general. The inlet quality for the evaporator was always greater than 0.3 excluding the extremely high heat fluxes in the low quality region.

Since the accuracy of the heat transfer correlation is not known for heat fluxes between 20-80 kW.m⁻², inaccuracies in the heat transfer coefficient prediction is expected.

The inner diameter for the evaporator test bench tube is 15.7 mm. This is 1.5 to 3 times larger than the evaluated tube diameters. The larger tube diameters are however not expected to have a large impact on the accuracy of the correlation. Keeping the mass flux and heat flux constant and changing the tube diameter, it can be seen from

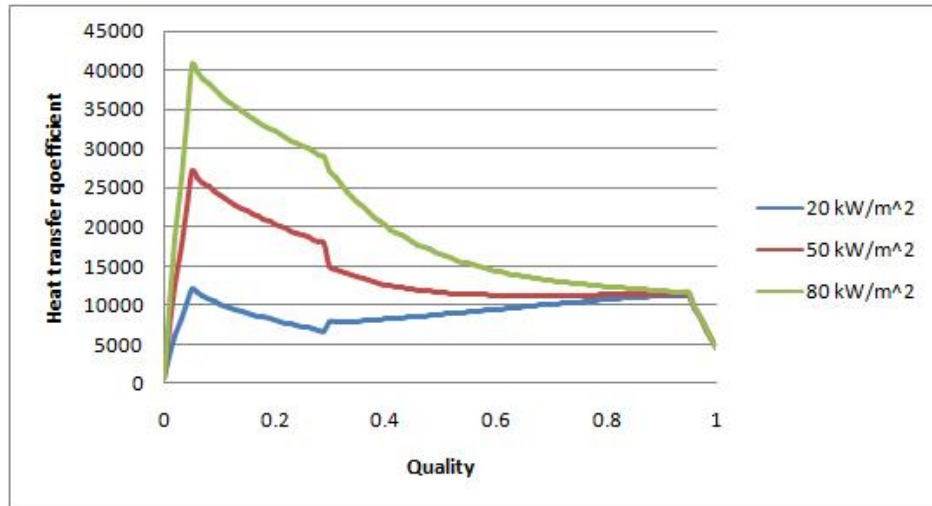


Figure 5.6: Heat transfer coefficients for different heat fluxes with $G = 800 \text{ kg/m}^2\text{s}$, $T_{sat} = 8$, $D_{eq} = 15.7\text{mm}$.

Figure 5.7 that the heat transfer does not really change. The effect of the tube diameter on accuracy is therefore minimal.

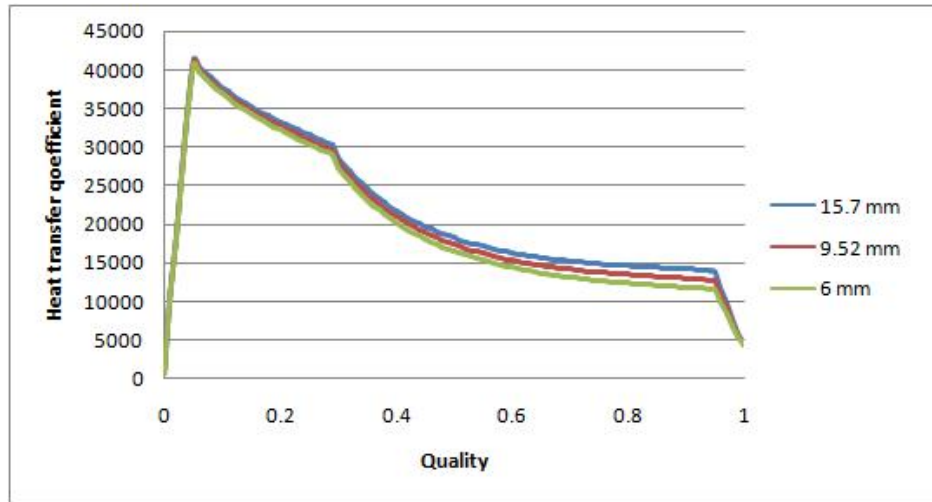


Figure 5.7: Heat transfer coefficients for different tube diameters with $G = 800 \text{ kg/m}^2\text{s}$, $T_{sat} = 8$ and $q_{flux} = 80\text{kW/m}^2$.

Lubricant in the refrigerant Several authors including Wang et al. (2011), Zhao & Bansal (2009) and Wang et al. (2012) note that the presence of lubricant in an R-744 evaporator suppresses the heat transfer coefficient. Reduction of the heat transfer coefficient by a factor of 3 to 4 is very likely. Based on the high mass flux through the system it is very likely that a substantial amount of oil pick up takes place in the compressor. The heat pump test system do not employ any method of preventing

compressor lubricant from circulation through the system. Fernandez et al. (2010) for example used two oil filters in series to prevent lubricant from entering the system.

From the data of Wang et al. (2011), as shown in Figure 5.8 it can be observed that the heat transfer coefficient is partially suppressed with the onset of the presence of oil in the system. For all fractions of oil of 0.1% and greater the heat transfer coefficient is completely suppressed. The heat transfer coefficient is also suppressed to similar levels regardless of the heat flux.

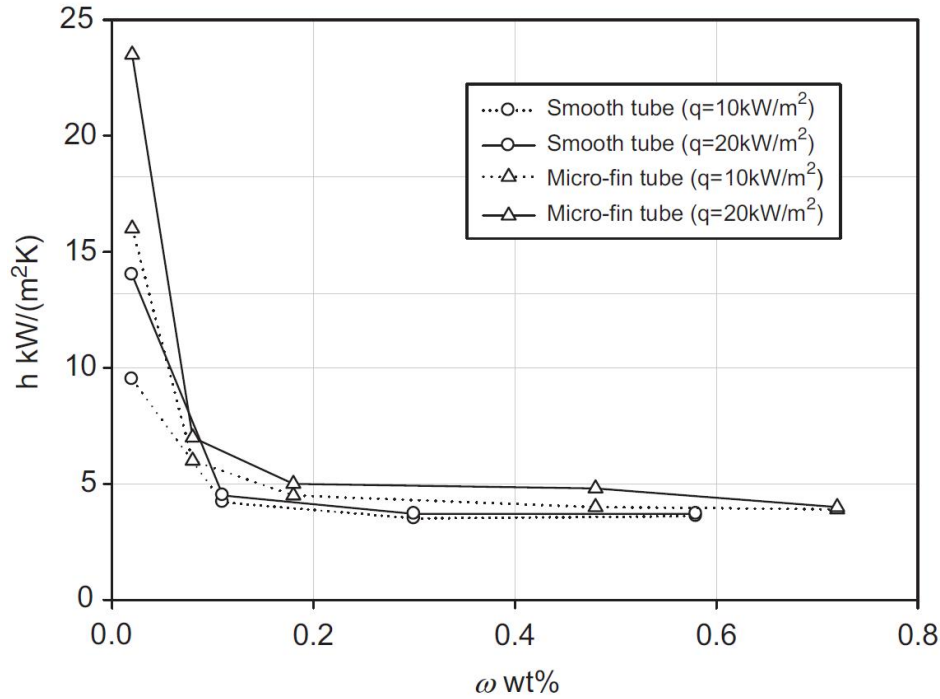


Figure 5.8: The effect of lubricant and heat flux on the heat transfer coefficient of R-744 (Wang et al. 2011).

This is an important observation. As discussed previously in this section, the heat transfer increase with increase in heat flux. The high heat fluxes in the evaporator therefore lead to high heat transfer coefficients. These will then be suppressed to low levels by the presence of oil in small quantities regardless of the high heat fluxes present.

Since it is known that there will be lubricant present in the system and that this will lead to suppression of the heat transfer coefficient it was decided to adjust the heat transfer correlation with a correction factor to simulate the presence of oil. The correction factor for the two-phase region was taken as 3.33 and a correction factor of 1.33 was applied to the single phase region. Runs 7 to 10 showed the most deviation from the measured results and were recalculated. The results are provided in Figure

5.9. Overall the results show improvement with the simulated values of run 7 and run 8 closely matching the experimental results. Run 9 and run 10 is also predicted with increased accuracy. It is therefore clear that heat transfer suppression do take place in the evaporator.

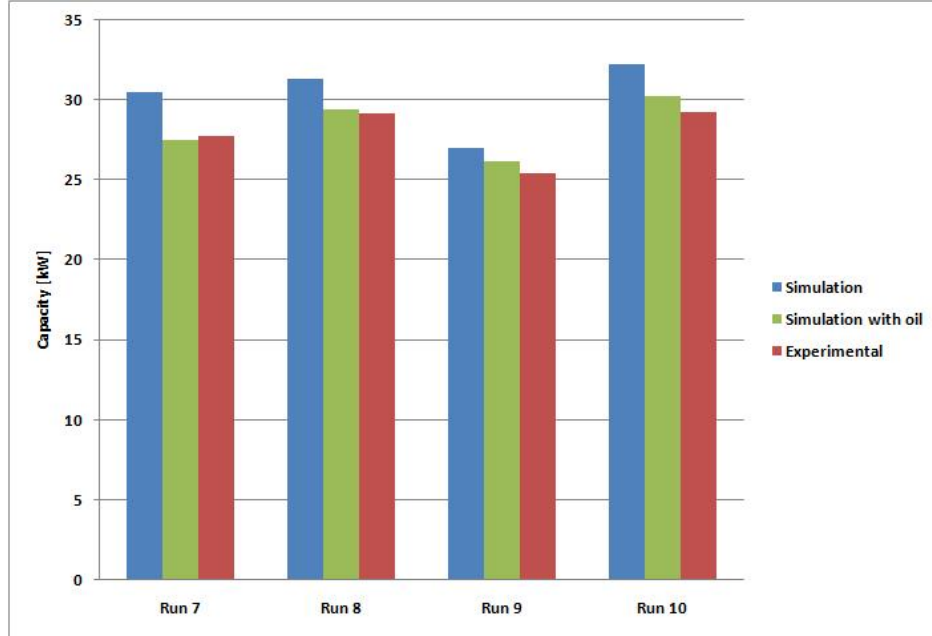


Figure 5.9: Simulated capacity with heat transfer adjusted for the presence of lubricant.

Temperature distribution through the evaporator A good simulation model must not only be able to predict the overall capacity of the heat exchanger but also the temperature distribution with accuracy. As the temperatures of the evaporator were measured in each return bend and the outlet the temperature distributions can be evaluated on a tube-by-tube basis. The simulated temperature distributions of both run 7 (Figure 5.10) and run 10 (Figure 5.11) was evaluated against the experimental results. This was done for both cases, with and without lubricant.

From Figure 5.10 it can be seen that the temperature distribution for the simulation taking the presence of lubricant into account closely match the experimental results with transition to single phase flow taking place somewhere in tube 6 or 7. For the original simulation the heat transfer is over predicted and this lead to transition from two phase to single phase as early as tube 4 or 5 and the outlet temperature are over predicted, directly leading to the capacity over prediction.

From Figure 5.11 it is clear that run 10 and run 7 shows the same trends. The

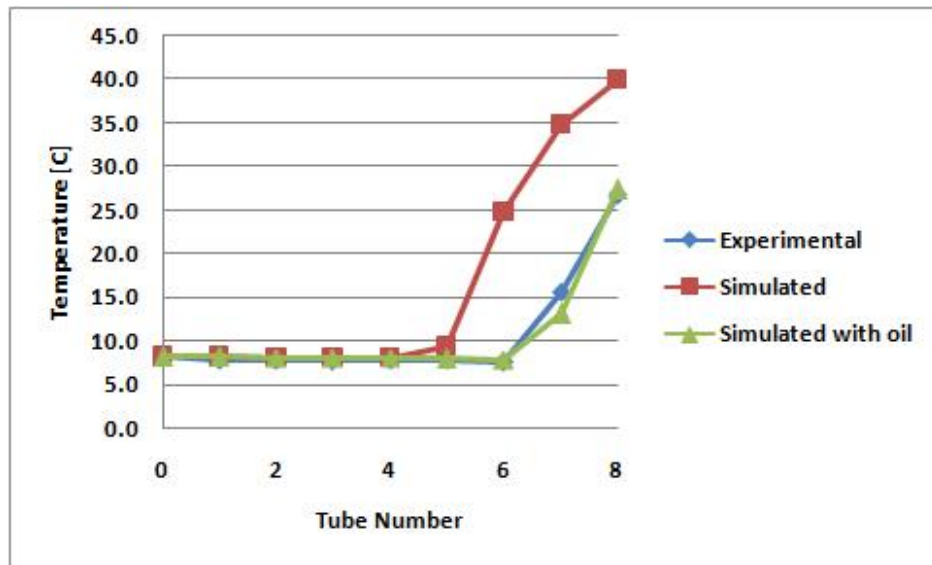


Figure 5.10: Detail temperature distribution for the refrigerant through the evaporator - Run 7.

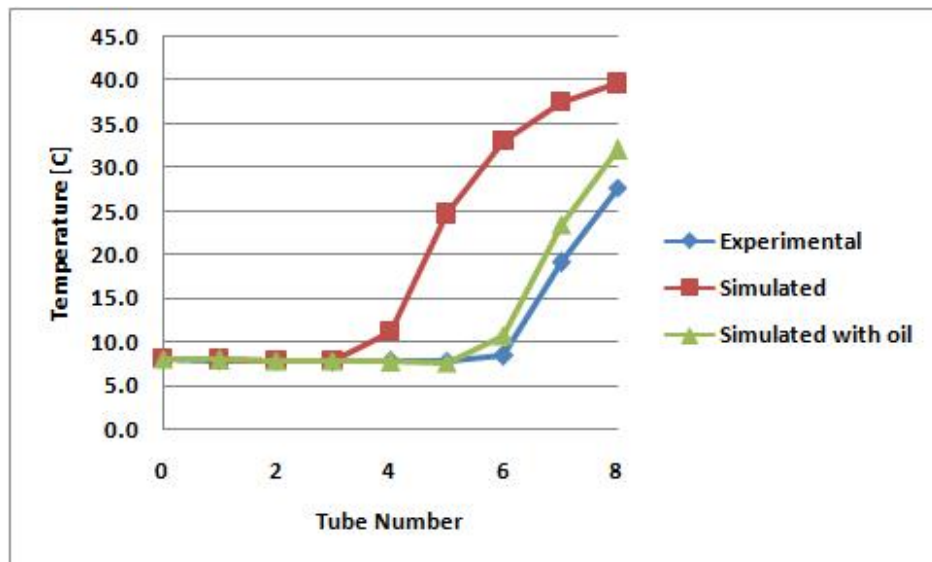


Figure 5.11: Detail temperature distribution for the refrigerant through the evaporator - Run 10.

original simulation completely over predicted the heat transfer in the two phase zone leading to early transition into single phase and the resultant capacity over prediction. The simulations with oil taken into account are able to follow the trend carefully with the exception that transition taking place in tube 5 or 6 instead of tube 6 or 7. This lead to a slight over prediction of the capacity indicating that the heat transfer suppression factor for run 10 is actually greater than 3.33.

5.1.6 Pressure drop discussion

The simulated pressure drops do not match the experimental values. As these values differ order of magnitude this indicate that either the chosen correlation is not suitable for the operational conditions of the evaporator or that the errors were introduced during the experimental measurements.

Since the saturated pressure for evaporating R-744 is a function of the saturated temperature it is also possible to calculate the pressure drop for the two phase region using the measured temperatures. The change in temperature per tube however is very small and it is difficult to obtain an accurate indication of the pressure drop per tube. By using the complete two phase region in the evaporator it is possible to get a good indication of the pressure drop in the two phase region. This was done for several runs and these results are given in Figure 5.12.

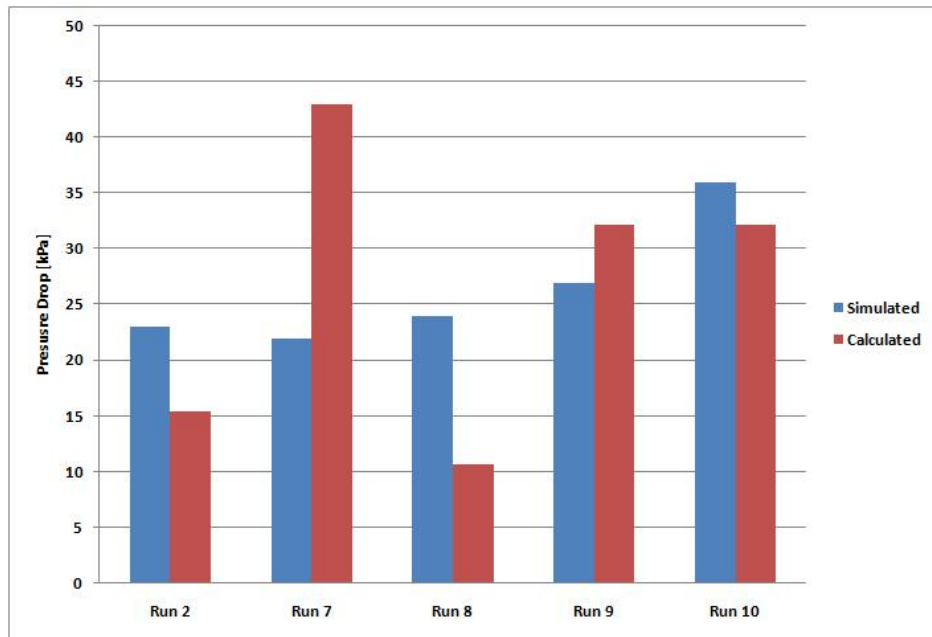


Figure 5.12: Simulated pressure drop vs calculated pressure drop.

Although the predicted and calculated values differ they are in the same order of magnitude and it is clear that the experimental values were inaccurate rather than the pressure drop correlation. Since very small change in saturated temperature result in large change of saturated pressure for R-744, the differences can be attributed to errors in measuring the temperature correctly.

The experimental pressure drop data is measured inside the return bends as indicated in Figure 5.1. It is suspected that due to the high mass fluxes the pressure drop data in the bend is inaccurate. There are several other reasons that can contribute to inaccurate measurement results but it will not be discussed here. It suffice to show that the errors are mostly due to measurement and that the correlation give adequate results.

As it is impossible to verify the pressure drop correlation against the measured results it is necessary to verify it using data from the literature. The experimental data of Bredensen et al. was obtained from the study by Cheng et al. (2008a). The pressure drop gradient increase with an increase in quality and than drop off suddenly as the quality approached 0.7. As seen from Figure 5.12 the simulated pressure drop gradient are able to follow the the experimental data closely for the lower qualities with some deviation at the higher qualities. The simulated values also do not show the sudden drop off in pressure drop gradient that is observed for high pressure drop gradients. This said, the deviation for the most part are small and the simulated and experimental values match closely.

5.1.7 Conclusion

The tube-in-tube heat exchanger model were unable to accurately predict the capacity of the experimental set-up. This can mainly be attributed to presence of oil and the possible inaccuracy of the heat transfer correlation at very high heat fluxes present in the evaporator. Accounting for this it is possible to predict evaporator capacity more accurately, with the simulated temperature distribution closely matching the experimental temperature distribution.

For the pressure drop the experimental data was discarded completely and the relevance of the correlation was tested against the temperature change due to pressure and against data from the literature. The pressure drop correlation is able to closely

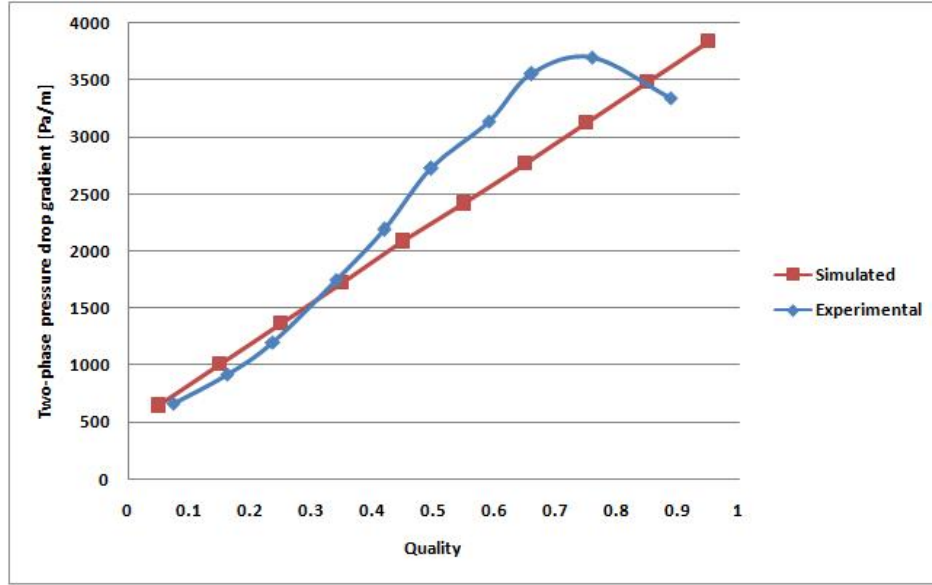


Figure 5.13: Simulated pressure drop vs the experimental data of Bredensen et al. as presented by Cheng et al. (2008a) with $G = 400 \text{ kg/m}^2\text{s}$, $T_{sat} = -10$, $D_{eq} = 7\text{mm}$ and $q_{flux} = 9\text{kW/m}^2$.

predict the experimental values.

It can be concluded that both the heat transfer correlation of Jung et al. (1989) and the pressure drop correlation of Choi et al. (1999) are able to predict the experimental values accurately and are valid for use in the fin-and-tube evaporator.

5.2 Fin-and-tube heat exchanger

Limited data on the performance of fin-tube heat exchangers are available in literature. Due to the lack of data it was decided to verify that the detail fin-and-tube model developed in this study, against the simulation program EVAP-COND. This software package contains the simulation models for a finned-tube evaporator and condenser developed by NIST (NIST 2010). EVAP-COND is ideally suited for the comparison as it also uses the elemental method and contains R-744 as an in-built refrigerant option. The main differences between this simulation model and EVAP-COND are that the correlations use in this study was specifically chosen to be applicable for R-744.

5.2.1 Evaporator geometry

The fin-and-tube heat exchanger was evaluated against the exact same evaporator in EVAP-COND. Since the fin-side heat transfer and pressure drop correlations are applicable for the 10mm tube only, the two programs are only evaluated against each other for this geometry.

In Section 2.4.8 the applicable range and selection criteria for the fin-side correlations were discussed in detail. Similarly the local manufacturing capabilities is discussed in Section 2.3.3. The following geometric parameters will be used in the comparison study as the most applicable for the range of the correlations.

- Fin spacing of 12 FPI.
- Fin thickness of 0.12mm
- Tube collar diameter of 10.38 mm and therefore outer tube diameter of 10.14mm
- Tube thickness of 0.61 mm
- Tube pitch (Sy) of 25mm
- Depth row pitch (Sx) of 19.05 mm
- Coil depth of four rows as the default coil thickness
- Tube length is fixed at 1600 mm
- Number of rows high per circuit is 4 rows high

The higher fin spacing was chosen to ensure higher heat transfer values while the long tube length was chosen to ensure high mass fluxes for the circuit. Each simulation is for a single coil with one circuit in a counterflow arrangement.

5.2.2 Evaporator operating conditions

In order to cover a range of operating conditions with the least amount of simulations it was decided to do a run for the minimum and maximum operating conditions. As discussed in Section 2.3.1 the ambient temperatures typically range from 0 to 30°C. The saturated temperature at the outlet will be fixed to be 15°C below the ambient

temperature. For the high temperature condition the air humidity will be low and for the low temperature it will tend to saturation. A few simulations will also be done for an average temperature of 20°C. Again the saturated evaporating temperature will be fixed at 15°C below ambient. In all cases the mass flow is then determined to deliver a superheat of 10°C.

The inlet quality into the evaporator is determined by the operating conditions in the gas cooler as well as the pressure inside the evaporator. For a very efficient gas cooler the gas temperature will approach the inlet water temperature. As discussed in Section 2.3.2, assuming good stratification, the inlet water temperature will range from a lower limit 10°C minimum to 30°C maximum with an yearly average value of 20°C. The following gas temperature is therefore used: 10°C for the low temperature run, 30°C for the high temperature run and 20°C for the rest.

For simplicity it is assumed that hot water is required at 90°C and based on the results of White et al. (2002) the optimum pressure in the gas cooler is fixed at 110 bar. This value is used throughout. Using the optimum gas cooler pressure and the values for the gas cooler outlet as discussed above, it is found that the inlet quality is very low for all operating conditions. In practice the inlet quality is higher due to non-ideal stratification and working pressures. It was decided to add an additional run to test for the higher inlet qualities as found in practice. An inlet quality of 0.4 was chosen for run 5 to take this effect into account.

The air velocity onto the coil was fixed at 2.5 m/s and the air pressure at 83.5 kPa. The superheat for each circuit was fixed at 10°C.

The test conditions identified is summarised in Table 5.3:

Table 5.3: Fin-and-tube evaporator simulation operating conditions.

	Gas cooler		Evaporator		Airside		
	Temperature [°C]	Pressure [bar]	Saturate Temperature [°C]	Inlet quality [-]	Temperature [°C]	Relative Humidity [%]	Pressure [kPa]
Run 1	10	110	-15	0.19	0	90	83.5
Run 2	30	110	20	0.16	35	30	83.5
Run 3	20	110	5	0.13	20	20	83.5
Run 4	20	110	5	0.13	20	50	83.5
Run 5	20	110	5	0.13	20	80	83.5
Run 6	20	110	5	0.4	20	20	83.5

5.2.3 Simulation results

From Figure 5.14 it can be seen that although the coil capacities do not match exactly they do follow similar trends. The developed simulation predicted higher capacities throughout. The dry coil conditions are predicted to within 6% to 10%. For the completely wet coil conditions the results diverged more widely with changes of up to 14%. This higher deviation for wet coils are probably due to the different methods implemented in the programs to calculate the air side heat transfer coefficient under completely and partially wet conditions.

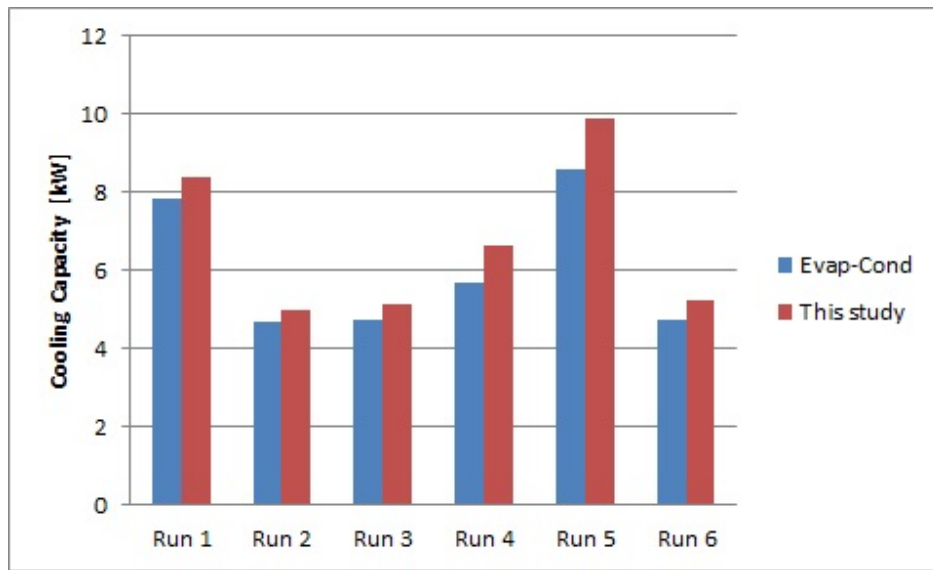


Figure 5.14: Cooling capacity comparison between the EES fin-and-tube simulation model and EVAP-COND.

The pressure drop results is shown in Figure 5.14. EVAP-COND consistently predict higher pressure drops in the range of 30% to 50%. This is to be expected. As discussed in Section 2.4.6 almost all existing correlations tend to over predict the pressure drop for R-744. The pressure drop differences are therefore adequately explained.

For the above simulations the mass fluxes ranged from 402.3 to 765 $\text{kg.m}^{-2}.\text{s}^{-1}$ and with an average heat flux for the coils ranging from 7 to 13 kW.m^{-2} . In Section 2.4.2 the heat transfer and pressure drop equations were selected to apply for refrigerant mass fluxes in the region of 450 to 800 $\text{kg.m}^{-2}.\text{s}^{-1}$ and heat fluxes in the region of 4.5 to 8 kW.m^{-2} . The heat transfer and pressure drop correlations are therefore not applied outside the applicable range.

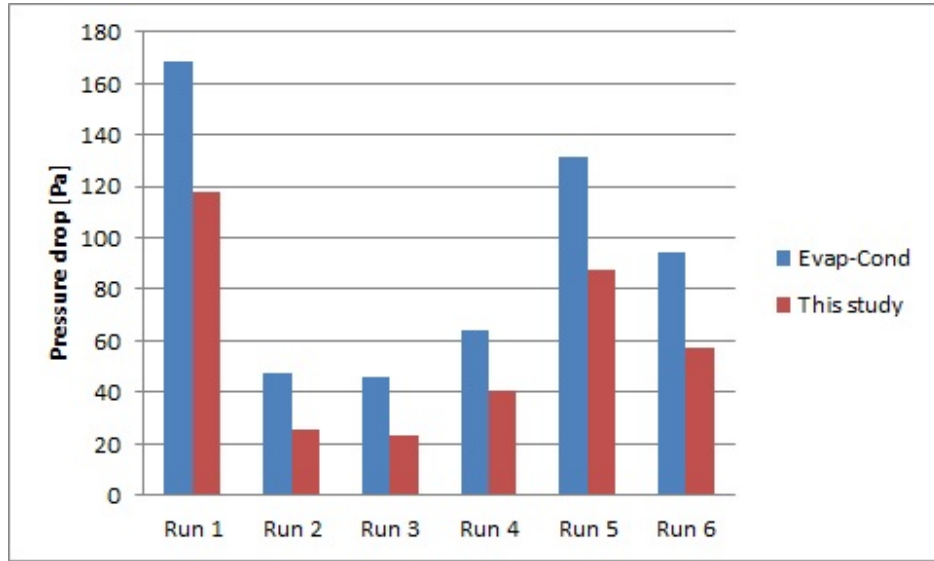


Figure 5.15: Pressure drop comparison for R-744 between the EES fin-and-tube simulation model and EVAP-COND.

5.2.4 Effect of number of elements

An increase in elements will typically improve accuracy but with the downside of increased solving time. The number of elements also increase the number of variables used in EES. This requires additional guess values for each extra variable and there is also an upper limit on the total number of variables in EES.

The default evaporator model evaluated contains 16 tube sections. The minimum number of elements that can therefore be used for this simulation is 16 elements. As seen in Section 5.1.4 the change in overall capacity and pressure drop for the tube-in-tube evaporator is negligible for an increase in the number of elements higher than 16 elements. Based on this results it is expected that increasing the number of elements per tube will not significantly affect the accuracy of this simulation. The fin-and-tube simulation was therefore done by defining each element as a single tube for the sake of simplicity.

5.2.5 Conclusion

The fin-tube simulation program was developed and there are good agreement between the capacity results of the developed simulation and the program EVAP-COND.

The pressure drop differences are expected since the pressure drop correlations in EVAP-COND is not specifically selected for use by R-744.

It can be concluded that the fin-and-tube detailed simulation model was successfully developed and can be used for detail evaporator development.

5.3 Coil geometry evaluation

To demonstrate the use of the detailed fin-and-tube model in optimisation of an evaporator coil it was decided to evaluate the local coil geometries available for the most cost effective options.

The different fin-and-tube evaporator options available is discussed in detail in Section 2.3.3 and can be found in Tabel 2.3. In Section 2.4.1 these options are evaluated on hand of the design pressure for R-744 evaporators.

5.3.1 Simulation inputs

The only two tubes that meet the 80 bar low side design pressure are the, $\frac{1}{2}$ " copper tube with the 1 mm wall thickness and the $\frac{1}{2}$ " stainless steel tube. If the 80 bar requirement is mitigated by the use of safety equipment or design it might also be possible to use $\frac{3}{8}$ " copper tube with 0.61 mm wall thickness. These three tube diameters will therefore be evaluated to determine the optimum tube diameter for a R-744 evaporator.

In order to differentiate between different heat exchanger designs, some form of performance criteria must be defined. The most logical criteria is the one of cost. The costs involved with a heat exchanger are the direct cost of acquiring the hardware and the indirect cost of operating the heat pump with the chosen heat exchanger.

The hardware cost can be determined as follows:

- The size of the coil is determined by the face area, number of rows deep and number of rows high.
- This will determine the total number of tubes required as well as the length of the different tubes. From this it is possible to calculate the total amount of material used for the tubes.
- The fin spacing will determine the number of fins used. This combined with the area per fin will provide the total amount of aluminium used.

- Due to the simplicity of the manufacturing process it can be assumed that the price of an evaporator coil are a function of the cost of the raw material used. Thus the weight of the coil can give a good indication of the total weight.

The indirect running cost can be determined as follows:

- The first criteria that has to be considered is the effect of the evaporator on the COP of the system. The lower the outlet pressure for the same evaporation pressure, the lower the COP of the cycle and the higher the running cost. This criteria is best evaluated at the normal operational conditions of the heat pump. For this excise the saturated pressures at the outlet is kept the same for all coils and it will not be taken into account.
- Fan power to move the secondary fluid is also of importance. The higher the volume flow or pressure drop required, the more power is required by the fan.

5.3.2 Simulation Results

Different heat exchangers geometries can therefore be evaluated against each other on a basis of material weight and fan power required. In order to account for different parameters such as difference in geometry and air volume flow rates, the comparison is done on a basis of weight or fan power vs unit of cooling capacity.

As seen from Figure 5.16 and 5.17 the best option is to use $\frac{3}{8}$ " (10.05mm) copper tube. It offers both the lowest hardware cost as well as the lowest operational cost. The least cost effective option is $\frac{1}{2}$ " (12.6 mm) stainless steel tubes.

5.4 Summary

The tube-in-tube heat exchanger model were unable to accurately predict the capacity of the experimental set-up. Accounting for oil it was possible to predict evaporator capacity more accurately, with the simulated temperature distribution closely matching the experimental temperature distribution. The pressure drop the experimental data was discarded completely and the relevance of the correlation was tested against the temperature change due to pressure and against data from the literature. The pressure drop correlation is able to closely predict the experimental values. It can be concluded

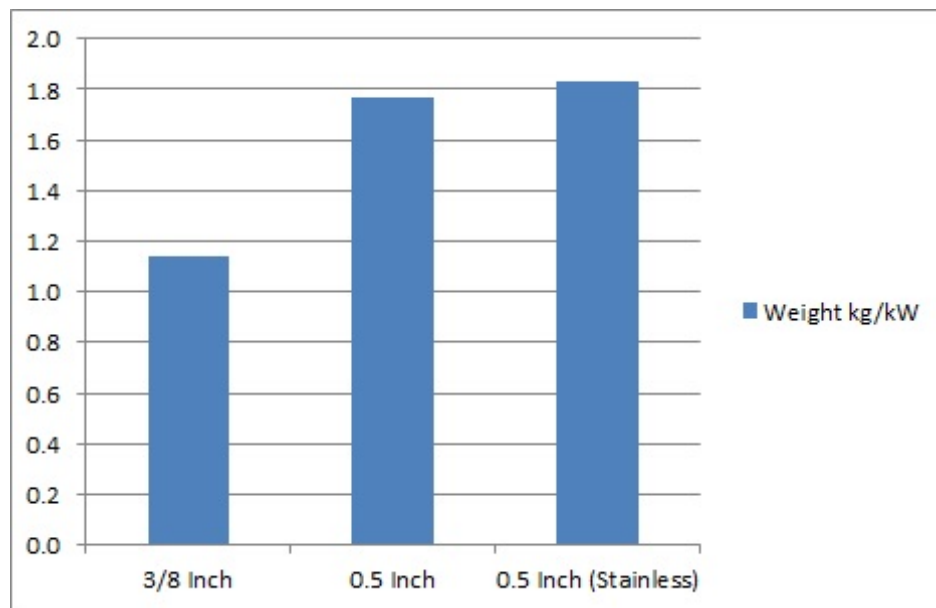


Figure 5.16: Coil weight vs Cooling capacity for different tube options.

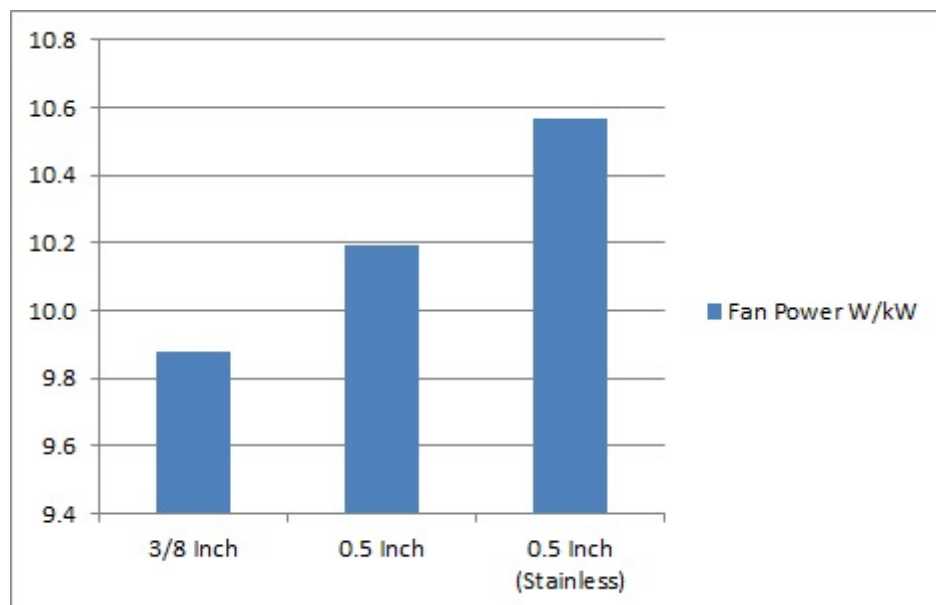


Figure 5.17: Fan power vs Cooling capacity for different tube options.

that both the heat transfer correlation of Jung et al. (1989) and the pressure drop correlation of Choi et al. (1999) are able to predict the experimental values accurately and are valid for use in the fin-and-tube evaporator.

The fin-tube simulation program was developed and there are good agreement between the capacity results of the developed simulation and the program EVAP-COND. The pressure drop difference are expected since the pressure drop correlations in EVAP-COND is not specifically selected for use by R-744. It can be concluded that the fin-and-tube detailed simulation model was successfully developed and can be used for detail evaporator development.

The purpose of this study therefore was to develop a detail simulation model of a concentric tube-in-tube water-to-refrigerant evaporator, as well as a fin-and-tube air-to-refrigerant evaporator model. Data from the North-West University R-744 heat pump test bench were used to verify the tube-in-tube evaporator simulation model. The discrepancies in the cooling capacity between the simulation and test bench can be attributed to the presence of lubricant in the system. The fin-and-tube model was verified by testing it against the NIST program EVAP-COND (NIST 2010). Overall there was good agreement between the results of the two programs and it can therefore be concluded that the fin-and-tube model is adequate. It was found that both the heat transfer correlation of Jung et al. (1989) and the pressure drop correlation of Choi et al. (1999) are able to predict the experimental values accurately and are valid for use in both the evaporator models developed.

To demonstrate the use of the detail evaporator fin-and-tube model, an evaluation of the different tube geometries, commercially available in South Africa, for use with R-744 fin-and-tube evaporators was done. For a fin-and-tube evaporator it was found that the most cost effective option is to use $\frac{3}{8}$ " (10.05 mm) copper tubes and the least effective is $\frac{1}{2}$ " (12.6 mm) stainless steel tubes.

Chapter 6

Conclusion

6.1 Tube-in-tube evaporator

In the literature study the R-744 two-phase pressure drop correlation of Choi et al. (1999) and the heat transfer coefficient correlation of Jung et al. (1989) was identified as the most suitable as it provides the best balance between accuracy and ease of use.

The tube-in-tube heat exchanger model was unable to accurately simulate the capacity of the experimental set-up using the above-mentioned correlations. If the effect of oil in the R-744 is taken into account, it is possible to predict evaporator capacity more accurately, with the simulated temperature distribution closely matching the experimental temperature distribution.

The pressure drop correlation was tested against data from the literature. The selected pressure drop correlation is able to closely predict the data from the literature.

It can be concluded that both the heat transfer correlation of Jung et al. (1989) and the pressure drop correlation of Choi et al. (1999) are able to predict the experimental values accurately and are valid for use in the fin-and-tube evaporator.

6.2 Fin-and-tube evaporator

A model was developed for a fin-and-tube evaporator using the above correlations. The overall model was verified by testing it against the EVAP-COND program. Overall there was good agreement between the fin-and-tube simulation and results from EVAP-COND. Both programs followed the same trend, but with EVAP-COND predicting a

lower cooling capacity (6% to 14% lower) and a higher pressure refrigerant pressure drop (30% to 50%) for a R-744 evaporator.

The pressure drop difference is expected since the pressure drop correlations in EVAP-COND is not specifically selected for use by R-744. It can be concluded that the fin-and-tube detailed simulation model was successfully developed and can be used for detail evaporator development.

6.3 Evaluate coil geometries

A detail evaluation study was completed on the different geometries commercially available in South Africa, for use with R-744 fin-and-tube evaporators. For a fin-and-tube evaporator it was found that the most cost effective option is to use $\frac{3}{8}$ " (10.05 mm) copper tubes and the least effective is $\frac{1}{2}$ " (12.6 mm) stainless steel tubes.

It is therefore recommended that $\frac{3}{8}$ " tubes is used for R-744 heat exchangers and that the use of even smaller tubes need to be investigated for the future.

6.4 Future studies

Although R-744 heat pump systems are currently still expensive, opportunities for cost reduction do exist. The properties of R-744 lead to compact and therefore cheaper components. In order to do a better design it is necessary to improve the current simulation models further.

Areas for improvement and future studies are:

- More detailed heat transfer correlations are still required. Current heat transfer correlations need to be adjusted to predict R-744 heat transfer more accurately.
- Expand the pressure drop and heat transfer models to take oil into account. Designs either need to be such as to prevent oil in the evaporator or this need to be taken into account for the heat transfer model.
- The reduction method for the fin-side heat transfer need to use refrigerant in the tubes instead of water. Currently most studies were done using water coils.

- The fin side heat transfer correlations need to be done on a row by row basis. No row-by-row correlations are currently available in the literature for the wavy fin geometry.
- There are currently very limited information available for heat transfer via conduction between tube rows.

Bibliography

ASHRAE Handbook, Fundamentals (2001).

ASHRAE Handbook, Systems and Equipment (2000).

Bendaoud, A., Ouzzane, M., Aidoun, Z. & Galanis, N. (2010), ‘A new modelling approach for the study of finned coils with CO₂’, *International Journal of Thermal Sciences* pp. 1–10.

Brown, J. S., Yana-Motta, S. F. & Domanski, P. A. (2002), ‘Comparitive analysis of an automotive air conditioning systems operating with CO₂ and R134a’, *International Journal of Refrigeration* **25**(19-32).

Cecchinato, L., Corradi, M., Fornasieri, E. & Zamboni, L. (2005), ‘Carbon dioxide as refrigerant for tap water heat pumps: A comparison with the traditional solution’, *International Journal of Refrigeration* **28**, 1250–1258.

Chen, Y., Lundqvist, P., Alves, A. & Brachert, L. (n.d.), Test and analysis of the sanyo eco-cute heat pump modified for swedish conditions, Technical report. Effys2 - Project report.

Cheng, L., Ribatski, G., Quibén, J. M. & Thome, J. R. (2008a), ‘New prediction methods for CO₂ evaporation inside tubes: Part I - a two-phase flow pattern map and a flow pattern based phenomenological model for two-phase flow frictional presure drops’, *International Journal of Heat and Mass Transfer* **51**, 111–124.

Cheng, L., Ribatski, G. & Thome, J. R. (2008b), ‘New prediction methods for CO₂ evaporation inside tubes: Part II - an updated general flow boiling heat transfer model based on flow patterns’, *International Journal of Heat and Mass Transfer* **51**, 125–135.

- Cho, J. M. & Kim, M. S. (2007), 'Experimental studies on the evaporative heat transfer and pressure drop of CO₂ in smooth and micro-fin tubes of diameters 5 and 9.52mm', *International Journal of Refrigeration* **30**, 986–994.
- Choi, J., Kedzierski, M. A. & Domanski, P. A. (1999), A generalized pressure drop correlation for evaporation and condensation of alternative refrigerants in smooth and micro-fin tubes, Technical report, NIST, Building and Fire Research Laboratory, National Institute of Standards and Technology, Gaithersburg, MD 20899.
- Didi, M. B. O., Kattan, N. & Thome, J. R. (2002), 'Prediction of two-phase pressure gradients of refrigerants in horizontal tubes', *International Journal of Refrigeration* **25**, 935–947.
- Ding, W. K., Fan, J. F., He, Y. L., Tao, W. Q., Zheng, Y. X., Gao, Y. F. & Song, J. (2011), 'A general simulation model for performance prediction of plate fin-and-tube heat exchanger with complex circuitry configuration', *Applied Thermal Engineering* **31**, 3106–3116.
- Dirker, J. (2002), Heat transfer coefficients in concentric annuli, Master's thesis, Rand Afrikaans University.
- Domanski, P. A., Choi, J. M. & Payne, W. V. (n.d.), Longitudinal heat conduction in finned-tube evaporators, *in* 'International Congress of Refrigeration 2007, Beijing'.
- Domanski, P. A. & Hermes, C. J. L. (2006), An improved correlation for two-phase pressure drop of R-22 and R-410A in 180° return bends, *in* 'Proceedings of the 11th Brazilian Congress of Thermal Sciences and Engineering - ENCIT 2006', number Paper CIT06-1051, Braz. Soc. of Mechanical Sciences and Engineering - ABCM.
- Domanski, P. A. & Hermes, C. J. L. (2008), 'An improved correlation for two-phase pressure drop of R-22 and R-410A in 180° return bends', *Applied Thermal Engineering*.
- Domanski, P. A., Yashar, D. A. & Lee, S. (n.d.), Evaporator optimization for non-uniform air distribution, Technical report, HVACR Equipment Performance Group, National Institute of Standards and Technology, Gaithersburg, MD 20899-8631, USA.

Eco Cute sales on the rise (2011). Date of access: 10 Aug. 2011.

URL: www.r744.com/news/view/1184

Fernandez, N., Hwang, Y. & Radermacher, R. (2010), 'Comparison of CO₂ heat pump water heater performance with baselin cycle and two high COP cylces', *International Journal of Refrigeration* **33**, 635–644.

Friterm (2004), *Epoxy and hydrophilic coatings used in aluminium finned heat exchangers*. Rev 0.0.

García-Valladares, O., Pérez-Segarra, C. D. & Rigola, J. (2004), 'Numerical simulation of double-pipe condensers and evaporators', *International Journal of Refrigeration* **27**, 656–670.

Genić, S., Arandjelović, I., Kolendić, P., Jarić, M., Budimir, N. & Genić, V. (2011), 'A review of explicit approximations of colebrook's equation', *FME Transactions* **39**, 67–71.

Ghanbari, A., Farshad, F. F. & Rieke, H. H. (2011), 'Newly developed friction factor correlation for pipe flow and flow assurance', *Journal of Chemical Engineering and Materials Science* **2**(6), 83–86.

Handschuh, R. (2008), 'Design criteria for co₂ evaporators', *Natural Refrigerants - sustainable ozone- and climate- friendly alternatives to HCFCs* .

HC (2009), *New coil designations - Sept 2009*. APGT 16-10-09.

Hepbasli, A. & Kalinci, Y. (2009), 'A review of heat pump water heating systems', *Renewable and Sustainable Energy Reviews* .

Huzayyin, A. S., Nada, S. A. & Elattar, H. F. (2007), 'Air-side performance of wavy-finned-tube direct expansion cooling and dehumidifying air coil', *International Journal of Refrigeration* **30**, 230–244.

Incropera, F. P., Dewitt, D. P., Bergman, T. L. & Lavine, A. S. (2007), *Fundamentals of Heat and Mass Transfer*, edition 6 edn, John Wiley and Sons.

Integrated Demand Management (2012). Date of access: 23 Sep. 2012.

URL: <http://www.eskomidm.co.za/>

- Iu, I. S. (2007), Development of air-to-air heat pump simulation program with advanced heat exchanger circuitry algorithm, PhD thesis, Oklahoma State University.
- Iu, I., Weber, N. A., Bansal, P. & Fisher, D. E. (2007), ‘Applying the effectiveness-ntu method to elemental heat exchanger models’, *ASHREA Transactions* **113**.
- Jung, D. S., McLinden, M., Radermacher, R. & Didion, D. (1989), ‘Prediction of flow boiling heat transfer with refrigerant mixtures’, *International Journal of Heat Mass Transfer* **32**(9), 1751–1764.
- Jung, D. S. & Radermacher, R. (1989), ‘Prediction of pressure drop during horizontal annular flow boiling of pure and mixed refrigerants’, *International Journal of Heat Mass Transfer* **32**(12), 2435–2446.
- Kasap, F., Acül, H., Canbaz, H. & Erbil, S. (2011), R744 (CO₂) refrigerant cooling systems, the finned tube R744 (CO₂) evaporator and gas cooler design principles, Technical report, FRITERM.
- Kim, M., Pettersen, J. & Bullard, C. W. (2004), ‘Fundamental process and system design issues in CO₂ vapor compression systems’, *Progress in Energy and Combustion Science* .
- Kim, S. G., Kim, Y. J., Lee, G. & Kim, M. S. (2005), ‘The performance of a transcritical CO₂ cycle with an internal heat exchanger for hot water heating’, *International Journal of Refrigeration* **28**, 1064–1075.
- Klein, S. A. (2011), ‘Ees version 9.389’. Engineering Equation Solver.
URL: <http://www.fChart.com>
- Kuvannarat, T., Wang, C. & Wongwises, S. (2006), ‘Effect of fin thickness on the air-side performance of wavy fin-and-tube heat exchangers under dehumidifying conditions’, *International Journal of Heat and Mass Transfer* **49**, 2587–2596.
- Kyo (1998), *The Kyoto Protocol to the United Nations Framework Convention on Climate Change*.
- Lee, J. & Domanski, P. A. (1997), Impact of air and refrigerant maldistributions on the performance of finned-tube evaporators with R-22 and R-407C, Technical report, Building Environment Division National Institute of Standards and Technology.

- Lorentzen, G. (1995), ‘The use of natural refrigerants: a complete solution to the CFC/HCFC predicament’, *International Journal of Refrigeration* .
- Lu, G. & Wang, J. (2008), ‘Experimental investigation on heat transfer characteristics of water flow in a narrow annulus’, *Applied Thermal Engineering* **28**, 8 – 13.
- Mastrullo, R., Mauro, A. W., Rosato, A. & Vanoli, G. P. (2010), ‘Carbon dioxide heat transfer coefficients and pressure drops during flow boiling: Assessment of predictive methods’, *International Journal of Refrigeration* pp. 1–18.
- Meyer, J. P. & Greyvenstein, G. P. (1992), ‘Hot water for large residential units, hospitals and laundries with heat pumps in south africa: A techno-economic analysis’, *Energy Convers. Mgmt* **33**(2), 135–143.
- Mon (2000), *The Montreal Protocol on Substances that Deplete the Ozone Layer*.
- Morales-Ruiz, S., Rigola, J., Pérez-Segarra, C. D. & García-Valladares, O. (2009), ‘Numerical analysis of two-phase flow in condensers and evaporators with special emphasis on single-phase/two phase transition zones’, *Applied Thermal Engineering* **29**, 1032–1042.
- Nekså, P., Rekstad, H., Zakeri, G. R. & Schiefloe, P. (1998), ‘CO₂-heat pump water heater: characteristics, system design and experimental results’, *International Journal of Refrigeration* **21**(3), 172–179.
- Nellis, G. & Klein, S. (2009), *Heat Transfer*, Cambridge University Press.
- NIST (2010), ‘Evap-cond version 3.0’. Simulation Models for Finned-Tube Heat Exchangers with Circuitry Optimization.
URL: http://www.nist.gov/el/building_environment/evapcond_software.cfm
- Oh, H., Ku, H., Roh, G., Son, C. & Park, S. (2008), ‘Flow boiling heat transfer characteristics of carbon dioxide in a horizontal tube’, *Applied Thermal Engineering* **28**, 1022–1030.
- Oh, H. & Son, C. (2011), ‘Flow boiling heat transfer and pressure drop characteristics of CO₂ in horizontal tube of 4.57-mm inner diameter’, *Applied Thermal Engineering* **31**, 163–172.

- Oliet, C., Pérez-Segarra, C. D., Danov, S. & Oliva, A. (2007), ‘Numerical simulation of dehumidifying fin-and-tube heat exchangers: Semi-analytical modelling and experimental comparison’, *International Journal of Refrigeration* **30**, 1266–1277.
- Padilla, M., Revellin, R., Haberschill, P. & Bonjour, J. (2011), ‘Two-phase pressure drop in return bends: Experimental results for R-410A’, *International Journal of Refrigeration* **34**, 1854–1865.
- Pettersen, J., Hafner, A., Skaugen, G. & Rekstad, H. (1998), ‘Development of compact heat exchangers for CO₂ air conditioning systems’, *International Journal of Refrigeration* **21**(3), 180–193.
- Pirompugd, W., Wang, C. & Wongwises, S. (2009), ‘A review on reduction method for heat and mass transfer characteristics of fin-and-tube heat exchangers under dehumidifying conditions’, *International Journal of Heat and Mass Transfer* **52**, 2370–2378.
- Pirompugd, W., Wongwises, S. & Wang, C. (2009), ‘Simultaneous heat and mass transfer characteristics for wavy fin-and-tube heat exchangers under dehumidifying conditions’, *International Journal of Heat and Mass Transfer* **52**, 2370–2378.
- Rankin, R., Rousseau, P. G. & van Eldik, M. (2004), ‘Demand side management for commercial buildings using an inline heat pump water heating methodology’, *Energy Conversion and Management* **45**, 1553–1563.
- Rankin, R. & van Eldik, M. (2008), ‘An investigation into the energy savings and economic viability of heat pump water heaters applied in the residential and commercial sectors - a comparison with solar water heating systems’.
- Reulens, W., ed. (2009), *Natural Refrigerant CO₂*, Leonardo Project. NARECO₂.
- Rousseau, P. P. & van Eldik, D. M. (2011), *Thermal-Fluid systems modelling II, Study guide for MGII 886 E*, North West University, Potchefstroom Campus.
- Thome, J. R. (2010), *Wolverine Engineering Data Book III*, Wolverine Tube, Inc.
- Thome, J. R. & Ribatski, G. (2005), ‘State-of-the-art of two-phase flow and flow boiling heat transfer and pressure drop of CO₂ in macro- and micro-channels’, *International Journal of Refrigeration* **28**, 1149–1168. Review.

- van Eldik, M. (2012), 'Heat pump photos'. Photos of heat pumps provided via E-mail to the Author by M. van Eldik.
- Venter, P. (2010), A supercritical R-744 heat transfer simulation implementing various nusselt number correlations, Master's thesis, School of Mechanical Engineering, Faculty of engineering at the North-West University.
- Wang, C. C., Fu, W. L. & Chang, C. T. (1997), 'Heat transfer and friction characteristics of typical wavy fin-and-tube heat exchangers', *Experimental Thermal and Fluid Science* **14**, 174–186.
- Wang, C., Hafner, A., Kuo, C. & Hsieh, W. (2012), 'An overview of the effect of lubricant on the heat transfer performance on conventional refrigerants and natural refrigerant R-744', *Renewable and Sustainable Energy Review* **16**, 5071–5086.
- Wang, C., Liaw, J. & Yang, B. (2011), 'Airside performance of herringbone wavy fin-and-tube heat exchangers - data with larger tube diameters', *International Journal of Heat and Mass Transfer* **54**, 1024–1029.
- Wang, J. & Hihara, E. (2003), 'Prediction of air coil performance under partially wet and totally wet cooling conditions using equivalent dry-bulb temperature method', *International Journal of Refrigeration* **26**, 293–301.
- White, S. D., Yarrall, M. G., Cleland, D. J. & Hedley, R. A. (2002), 'Modelling the performance of a transcritical CO₂ heat pump for high temperature heating', *International Journal of Refrigeration* **25**, 479–486.
- Yamaguchi, S., Kato, D., Saito, K. & Kawai, S. (2011), 'Development and validation of static simulation model for CO₂ heat pump', *International Journal of Heat and Mass Transfer* **54**, 1896–1906.
- Yin, J. M., Bullard, C. W. & Hrnjak, P. S. (2001), 'R-744 gas cooler model development and validation', *International Journal of Refrigeration* **24**, 692–701.
- Yoon, S. H., Cho, E. S., Hwang, Y. W., Kim, M. S., Min, K. & Kim, Y. (2004), 'Characteristics of evaporative heat transfer and pressure drop of carbon dioxide and correlation development', *International Journal of refrigeration* **27**, 111–119.

- Yun, R., Kim, Y., Kim, M. S. & Choi, Y. (2003), ‘Boiling heat transfer and dryout phenomenon of CO₂ in a horizontal smooth tube’, *International Journal of Heat and Mass Transfer* (46), 2353–2361.
- Zhao, X. & Bansal, P. (2009), ‘Critical review of flow boiling heat transfer of CO₂-lubricant mixtures’, *International Journal of Heat and Mass Transfer* **52**, 870–879.

Chapter 7

Tube-in-Tube Simulation Program

File:case 10.EES

2013/03/04 03:34:18 AM Page 1

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

Function ReNumber(G,Dh,x,T,P,h)
{function to calculate reynolds number - refrigerant}
if x <= 0 then
mu = viscosity(R744,x = 0, T = T)           {viscosity of liquid refrigerant}
else
if x > 1 then
mu = viscosity(R744,P = P, T = T+0.1)       {viscosity of vapor refrigerant}
else
If x = 1 then
mu = viscosity(R744,x = 1, T = T)           {viscosity of vapor refrigerant}
else
mu_l = viscosity(R744,x = 0, T = T)
mu_v = viscosity(R744,x = 1, T = T)
mu = mu_l*(1-x) + mu_v*x                   {quality averaged viscosity}
endif
endif
endif
ReNumber = (G*Dh)/mu
end

Function ReNumberw(G,Dh,T)
{function to calculate reynolds number - water}
mu = viscosity(water,x = 0, T = T)
ReNumberw = (G*Dh)/mu
end

Function friction(RR,Re)
{Calculate the friction factor for single phase flows}
{NOTE: all flows are assumed to be turbulent}
{Ghanbari-Farshad-Rieke correlation is used}
RR = abs(RR)
If Re < 3000 then
Re = 3000
endif

friction = (-1.52*log10((RR/7.21)^1.042+(2.731/Re)^0.9152))^(-2.169)

end

Function dpwater(G,L,Dh,rho,f)
{calculate single phase pressure drop, liquid or vapour}
{dp_mom = G^2*(1/rho_e - 1/rho_i)}           {density change neglible, ignore momentum for single phase}
dp_fric = 0.5*(L/Dh)*(G^2/rho)               {friction pressure drop}
dpwater={dp_mom +} dp_fric                  {total pressure drop}
end

Function dp1phase(G,L,Dh,rho,rho_e,rho_i,f)
{calculate single phase pressure drop, liquid or vapour}
{dp_mom = G^2*(1/rho_e - 1/rho_i)}           {momentum pressure drop}
dp_fric = 0.5*(L/Dh)*(G^2/rho)               {friction pressure drop}
dp1phase={dp_mom +} dp_fric                 {total pressure drop}
end

Function dp2phase(G,L,Dh,x_e,x_i,T)
{2 phase pressure drop - modified Bo Pierre method by Choi et al (1999)}

If x_e <= 0 then x_e = 0.001
if x_i <= 0 then x_i = 0.001
If x_e >= 1 then x_e = 1
if x_i >= 1 then x_i = 1

```

File:case 10.EES

2013/03/04 03:34:18 AM Page 2

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

dx = abs(x_e-x_i)
h_lv = abs(enthalpy(R744, T = T, x = 1) - enthalpy(R744, T = T, x = 0))
nu_i = volume(R744, T = T, x = x_i) {inlet specific volume}
nu_e = volume(R744, T = T, x = x_e) {outlet specific volume}
mu_l = viscosity(R744, x = 0, T = T) {viscosity of liquid refrigerant}
Re_l = (G*Dh)/mu_l {Liquid Reynolds number}
K_f = (dx)*h_lv*1000/(L*g#) {two phase number}
f_N = 0.00506*Re_l^(-0.0951)*K_f^(0.1554)
dp2phase=((f_N*L*(nu_e+nu_i))/Dh+(nu_e-nu_i))*G^2
end

Function dp(G,L,Dh,rho,rho_e,rho_i,f,x,x_e,x_i,T)
{pressure drop function - determine if single phase or two phase based on average quality of the increment}
if x <= 0.02 then {all liquid flow}
dp = dp1phase(G,L,Dh,rho,rho_e,rho_i,f)
else
if x >= 0.98 then {all vapour flow}
dp = dp1phase(G,L,Dh,rho,rho_e,rho_i,f)
else
dp = dp2phase(G,L,Dh,x_e,x_i,T) {two phase flow}
endif
endif
end

{Function dpbend1phase(G,rho,R,Dh)}
{Calculate bend pressure drop for single phase flows - Paliwoda1992}
Sx = 2*R {Tupe pitch = 2 x bend diameter}
Kp = 1/(3.426*ln(Sx/(2*Dh))+3.8289) {correction factor}
dpbend1phase = 0.5*Kp*G^2/rho
end

Function dpbend1phase2(G,rho,R,Dh,RR,T,L)
{Calculate bend pressure drop for single phase flows - Idelshik 1986}
mu_l = viscosity(R744, x = 0, T = T) {viscosity of liquid refrigerant}
Re_l = (G*Dh)/mu_l {Liquid Reynolds number}
f = friction(RR, Re_l)
K_sp = f*(L/Dh)+0.294*(R/Dh)^0.5 {correction factor}
dpbend1phase2 = 0.5*K_sp*G^2/rho
end

Function dpstraight(G,L,Dh,x_e,x_i,T)
{2 phase pressure drop - modified Bo Pierre method by Choi et al (1999)}
{modified to be used for bend pressure drop calculation}
{no heat transfer for a bend - quality change assumed negligible}
If x_e <= 0 then x_e = 0.001
if x_i <= 0 then x_i = 0.001
If x_e >= 1 then x_e = 1
if x_i >= 1 then x_i = 1

{dx = abs(x_e-x_i)}
dx = 0 {no change in quality}
h_lv = abs(enthalpy(R744, T = T, x = 1) - enthalpy(R744, T = T, x = 0))
nu_i = volume(R744, T = T, x = x_i) {inlet specific volume}
nu_e = volume(R744, T = T, x = x_e) {outlet specific volume}
mu_l = viscosity(R744, x = 0, T = T) {viscosity of liquid refrigerant}
Re_l = abs((G*Dh)/mu_l) {Liquid Reynolds number}
{K_f = (dx)*h_lv*1000/(L*g#)}
K_f = 0 {two phase number = 0 - no since dx = 0}
f_N = 0.00506*Re_l^(-0.0951)
dpstraight=((f_N*L*(nu_e+nu_i))/Dh+(nu_e-nu_i))*G^2 {no change in inlet and outlet specific volumes, thus blank out}

```

File:case 10.EES

2013/03/04 03:34:18 AM Page 3

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

end

Function dpbend1phase3(G,R,Dh,RR,T,L,x)

{Calculate bend pressure drop for single phase flows - from chen 2003}

If T > 30 then

T = 30

endif

If x < 0.5 then

rho = density(R744,x = 0, T = T)

{density of liquid refrigerant}

mu = viscosity(R744,x = 0, T = T)

{viscosity of liquid refrigerant}

else

rho = density(R744,x = 1, T = T)

{density of vapor refrigerant}

mu = viscosity(R744,x = 1, T = T)

{viscosity of vapor refrigerant}

endif

Re= (G*Dh)/mu

{Reynolds number}

Dn = Re/(2*R/Dh)

{Dean number}

a = 0.021796

b = 0.0413356

f = (2.71828^(a+b*(ln(Dn))^2))*(64/Re)

{friction factor for bend flow}

K = f*(L/Dh)

{correction factor}

dpbend1phase3 = 0.5*K*G^2/rho

end

Function dpbend2phase(G,L_st,Dh,R,x_e,x_i,x,T)

{calculate bend pressure drop for 0.2<x<0.8 - Domanski and Hermes}

dp_straight = dpstraight(G,L_st,Dh,x_e,x_i,T)

a0 = 6.5e-3

a1 = 0.54

a2 = 0.21

a3 = 0.34

a4 = -0.67

rho_l = density(R744,x = 0, T = T)

{density of liquid refrigerant}

rho_v = density(R744,x = 1, T = T)

{density of vapour refrigerant}

mu_v = viscosity(R744,x = 1, T = T)

CM = a0*((G*x*Dh)/mu_v)^a1*(1/x-1)^a2*(rho_l/rho_v)^a3*((2*R)/Dh)^a4 *{curvature multiplier}*

dpbend2phase = dp_straight * CM

end

function dpbend(G,L_st,Dh,R,x,T,RR)

{calculate bend pressure drop}

{for a bend the quality is assumed constant through the bend}

{For x < 0 and x > 0 - use single phase correlation}

{for 0.2 < x, 0.8 use two phase correlation}

{for other conditions, use the singlephase if greater than two phase correlation, otherwise use average}

If (x <= 0) or (x >= 1) then

dpbend = dpbend1phase3(G,R,Dh,RR,T,L_st,x)

{single phase}

else

if (0.2<x) and (x<0.8) then

dpbend = dpbend2phase(G,L_st,Dh,R,x,x,x,T)

{two phase - apply for 0.2<x<0.8}

else

dpb1 = dpbend1phase3(G,R,Dh,RR,T,L_st,x)

dpb2 = dpbend2phase(G,L_st,Dh,R,x,x,x,T)

if dpb1>dpb2 then

{use single phase if greater than 2 phase}

dpbend = dpb1

else

dpbend = (dpb1 + dpb2)/2

{else use average}

File:case 10.EES

2013/03/04 03:34:18 AM Page 4

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```
endif
endif
endif
end
```

```
function hwater(Dh, f, Re, T, P)
{calculate the heat transfer coefficient for liquid water}
{Gnielinski correlation is used}
Pr = Prandtl(water, T = T, P = P)
k = Conductivity(water, T = T, P = P)
Pr = abs(Pr)
Nu = ((f/8)*(Re - 1000)*Pr)/(1 + 12.7*(f/8)^0.5*(Pr^(2/3)-1))
hwater = Nu*k/Dh
end
```

```
function h1phase(Dh, f, Re, T, P, h)
{calculate the heat transfer coefficient for liquid and vapor}
{Gnielinski correlation is used}
x = quality(R744, h = h, P = P)
if x < 100 then
Pr = Prandtl(R744, P = P, x = 1)
k = Conductivity(R744, P = P, x = 1)
else
Pr = Prandtl(R744, h = h, P = P)
k = Conductivity(R744, h = h, P = P)
endif
Pr = abs(pr)
Nu = ((f/8)*(Re - 1000)*Pr)/(1 + 12.7*(f/8)^0.5*(Pr^(2/3)-1))
h1phase = Nu*k/Dh
end
```

```
function h2phase(G, Q_flux, Dh, x, Re, T)
Q_flux = abs(Q_flux)
{calculate the heat transfer coefficient for liquid and vapor}
{Jung et al (1989) correlation is used}
mu_l = viscosity(R744, T = T, x = 0) {viscosity liquid}
mu_v = viscosity(R744, T = T, x = 1) {viscosity vapour}
rho_l = density(R744, T = T, x = 0) {density liquid}
rho_v = density(R744, T = T, x = 1) {density vapour}
h_lv = enthalpy(R744, T = T, x = 1) - enthalpy(R744, T = T, x = 0) {latent heat of vaporization}
Pr_l = prandtl(R744, T = T, x = 0) {liquid pr number}
k_l = conductivity(R744, T = T, x = 0) {liquid conductivity}
T_s = T + 273.15 {Saturated temperature in kelvin}
```

```
sigma = SurfaceTension(R744, T = T)
```

```
X_tt = ((1-x)/x)^0.9*(rho_v/rho_l)^0.5*(mu_l/mu_v)^0.1 {Martinelli parameter}
Bo = Q_flux/(G*h_lv) {boiling number}
if X_tt < 1 then {nucleate boiling factor}
N = 4048*X_tt^1.22*Bo^1.13
else
N = 2-0.1*(X_tt)^(-0.28)*(Bo)^(-0.33)
endif
bd = 0.0146*35*((2*sigma)/(g*(rho_l-rho_v)))^0.5
h_SA = 207*(k_l/bd)*((Q_flux*1000*bd)/(k_l*T_s))^0.745*(rho_v/rho_l)^0.581*Pr_l^0.533 {bool boiling heat transfer}
```

```
Fp = 2.37*(0.29+1/X_tt)^0.85 {F factor}
Re_l = ReNumber(G*(1-x), Dh, 0, T, 1, 0) {Liquid reynolds number wiht G = G(1-x)}
h_lo = 0.023*(k_l/Dh)*Re_l^0.8*Pr_l^0.4 {Ditus boelter equation - liquid only}
```

File:case 10.EES

2013/03/04 03:34:18 AM Page 5

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

h_nbc = N*h_SA
h_cec = Fp*h_lo
h2phase = h_nbc + h_cec                                {two phase heat transfer}
end

function hc(G,Q,Dh,x, f,Re,T,P,h,HF,HFs)
{check for phase and use appropriate heat transfer correlation}
if (x <= 0) then                                       {all liquid flow}
hc = HFS*h1phase(Dh, f,Re,T,P,h)
else
if x >= 1 then                                         {all vapour flow}
hc = HFS*h1phase(Dh, f,Re,T,P,h)
else
{linear changeover between phases to prevent discontinuity}
if (x > 0.95) then
hv = enthalpy(R744, x = 1, T=T)
tv = temperature (R744, x = 1, P = P)
h1 = HFS*h1phase(Dh, f,Re,Tv,P,hv)
h2 = hf*h2phase(G,Q,Dh,x,Re,T)
hc = (h1-h2)/0.05*(x-0.95)+h2                         {linear change over for two phase to vapour flow}
else
if (x < 0.05) then
hv = enthalpy(R744, x = 0, T=T)
tv = temperature (R744, x = 0, P = P)
h1 = HFS*h1phase(Dh, f,Re,Tv,P,hv)
h2 = hf*h2phase(G,Q,Dh,x,Re,T)
hc = (h2-h1)/0.05*(x)+h1                               {linear change over for liquid to two phase flow}
else
hc = hf*h2phase(G,Q,Dh,x,Re,T)                         {two phase flow}
endif
endif
endif
endif
end

{Main Program}
{Geometry Inputs}
L_t = 2                                                  {Lenght of a tube section}
N_t = 8                                                  {Number of tube sections}
Inc = 1                                                  {Number of increments per tube section}
N = N_t*Inc                                             {total number of increments}
L_inc = L_t/Inc                                         {Lenght per increment}

D_i = 15.76/1000                                         {Diameter of the inner tube}
W_t = 2.77/1000                                         {Wall thickness of the inner wall}
D_io = D_i + W_t*2                                       {Outer diameter of the inner tube}
D_o = 28/1000                                           {Diameter of the outer tube}

D_hi = D_i                                               {Hydraulic diameter of the inner tube}
D_ha = D_o - D_io                                       {Hydraulic diameter of the annulus}
A_ci = pi*(D_i/2)^2                                     {Cross sectional area of the inner tube}
A_an = pi*(D_o/2)^2 - pi*(D_io/2)^2                   {Cross sectional area of the annulus}
A_hi = pi*D_i*L_inc                                     {Inner tube heat transfer area per increment}
A_ho = pi*D_io*L_inc                                    {Outer tube heat transfer area}

{materials}
k_ss= k_('Stainless_AISI304', Temperature)             {conductivity of stainless steel - tube wall}
k_cu = k_('Copper', Temperature)                      {conductivity of copper - tube wall}
k_a= k_('Aluminum', Temperature)                      {conductivity of aluminum - fin material}
Temperature = 15                                       {assumed constant for the applicable temperature range}

```

File:case 10.EES

2013/03/04 03:34:18 AM Page 6

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

k_t = k_ss                                {tubewall conductivity}

{Return bend}
D_rb = 100/1000                          {Diameter for the return bend - distance between the tubes}
R_rb = D_rb/2                            {RADIUS of the return bend}
L_st = pi*(D_rb/2)                        {equivalent straight length for the return bend}

{friction factor inputs}
e = 46e-6                                {surface roughness - steel (mechanics of fluids - shames)}
RR_i = e/D_hi                            {Relative roughness for the inner tube}
RR_an = e/D_ha                            {Relative roughness for the annulus}

{water side inputs}
m_dot_w = 0.366                           {water mass flow rate}
m_dot_w = G_w*A_an                        {mass flux}
T_wi = 41.6                               {Water inlet temperature}
P_wi = 250                                {Water inlet pressure - assumed}

{refrigerant inputs}
m_dot = 0.1923                            {mass flow refrigerant}
m_dot = G*A_ci                            {mass flux}
P_gas = 111.4*100                         {outlet pressure of the gascooler}
T_gas = 42.1                              {Outlet temperature from the gascooler}
h_gas = enthalpy(R744, T = T_gas, P = P_gas) {outlet enthalpy from the gascooler}
h_in = h_gas                              {inlet enthalpy into the evaporator}
T_in = 8

{Modification factors for oil}
HF = 0.3                                  {Heat transfer suppressing factor due to lubricant - only apply to two
phase region}
HFs = 0.75                               {Factor applying to single phase region}
PF = 1                                    {Pressure increase factor due to lubricant, apply throughout}

{Note, refrigerant properties without a subscript}
{All waterside properties with subscript o = outer}
{subscript i = inlet, e = exit}

{Refrigerant side - inlet conditions}
{inlet at increment 1}
h_i[1] = h_in
P_i[1] = P_sat(R744, T = T_in)

{water side - inlet conditions}
{inlet at increment N - counterflow arrangement}
h_iw[N] = enthalpy(water, T = T_wi, P = P_wi)
P_iw[N] = P_wi

Duplicate i = 1,N
Lenght[i] = L_inc*i
{refrigerant side}
{Inlet properties}
T_i[i] = Temperature (R744, h=h_i[i], P= P_i[i])
x_i[i] = quality(R744, P = P_i[i], h = h_i[i])
rho_i[i] = density(R744, P = P_i[i], h = h_i[i])

{outlet properties}
T_e[i] = Temperature (R744, h=h_e[i], P= P_e[i])
x_e[i] = quality(R744, h=h_e[i], P = P_e[i])
rho_e[i] = density(R744, h=h_e[i], P = P_e[i])

```

```

{average properties }
h[i] =(h_i[i]+h_e[i])/2                                {linearly average enthalpy}
P[i] = (P_i[i] +P_e[i])/2                                {linearly average pressure}
T[i] = temperature(R744,P = P[i], h = h[i])
x[i] = quality(R744, P = P[i], h = h[i])
rho[i] = density(R744, h= h[i],P = P[i])
m_dot = rho[i]*V[i]*A_ci                                {average velocity for homogenous fluid}

{conservation of momentum}
Re[i] = ReNumber(G,D_hi,x[i],T[i],P[i],h[i])
f[i] = friction(RR_i,Re[i]) {f[i]= 0.025}
dp[i] = PF*dp(G,L_inc,D_hi,rho[i],rho_e[i],rho_i[i],f[i],x[i],x_e[i],x_i[i],T[i])/1000
{dp[i] = 100/N}
dp_test[i] = PF*dp(G,L_inc,D_hi,rho[i],rho_e[i],rho_i[i],f[i],x[i],x_e[i],x_i[i],T[i])/1000
P_e[i] = P_i[i] - dp[i]

{conservation of energy}
q_dot[i] = m_dot*(h_e[i] - h_i[i])
q_flux[i] = q_dot[i]/A_hi                                {heat flux}

{water side}
{average properties }
h_w[i] = (h_iw[i] + h_ew[i])/2                            {linearly average enthalpy}
T_w[i] = temperature(water,P = P_iw[i], h = h_w[i])        {water temperature}
rho_w[i] = density(water,P =P_iw[i], h = h_iw[i])           {water density}
m_dot_w= rho_w[i]*V_w[i]*A_an                               {velocity}

{conservation of momentum}
Re_w[i] = ReNumberw(G_w,D_ha,T_w[i])
f_w[i] = friction(RR_an,Re_w[i])
dp_w[i] = dpwater(G_w,L_inc,D_ha,rho_w[i],f_w[i])/1000
P_ew[i] = P_iw[i] - dp_w[i]

{conservation of energy}
q_dot[i] = m_dot_w*(h_iw[i]-h_ew[i] )

{heat transfer between the fluids}
q_dot[i] = (UA[i]*(T_w[i] - T[i]))/1000                    {heat transfer due to temperature difference}
q_test[i] = (UA[i]*(T_w[i] - T[i]))/1000
{q_dot[i] =9.5/N}

h_cr[i] = hc(G,q_flux[i],D_hi,x[i], f[i],Re[i],T[i],P[i],h[i],HF,HFs){convection heat transfer coefficient for refrigerant}
h_cw[i] = hwater(D_ha, f_w[i],Re_w[i],T_w[i],P_iw[i])      {convection heat transfer coefficient for water}

1/UA[i] =1/(h_cr[i]*A_hi)+ ln(D_io/D_i)/(2*pi*k_t*L_inc)+1/(h_cw[i]*A_ho) {overall heat transfer coefficient}

T_two[i] = T_w[i] - q_dot[i]/(h_cw[i]*A_ho)                {Waterside tube surface temperature}
T_tw[i] = T[i] + q_dot[i]/(h_cr[i]*A_hi)                    {Waterside tube surface temperature}

end

{property transfer between increments}
Duplicate i = 0,(N_t-1)
{inlet enthalpy of next increment is outlet of previous increment}
{Inlet pressure of next increment is outlet of previous increment }
Duplicate j = 1,inc - 1
k[i,j] = (i*inc+j)
h_i[k[i,j]+1] = h_e[k[i,j]]

```

File:case 10.EES

2013/03/04 03:34:18 AM Page 8

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

P_i[k[i,j]+1] = P_e[k[i,j]]

h_ew[k[i,j]+1] = h_iw[k[i,j]]
P_ew[k[i,j]+1] = P_iw[k[i,j]]
end
end
{for water no heat transfer or pressure drop over bends}
{for refrigerant no heat transfer but pressure drop over bends}
Duplicate i = 1,(N_t-1)
  l[i] = i*inc
  h_i[l[i]+1] = h_e[l[i]]
  P_i[l[i]+1] = P_e[l[i]] - dpB[l[i]]
{ dpB[l[i]] = 0.6}
  dpB[l[i]] = dpbend(G,L_st,D_hi,R_rb,x_e[l[i]],T_e[l[i]],RR_i)/1000
  dpB_test[l[i]] = dpbend(G,L_st,D_hi,R_rb,x_e[l[i]],T_e[l[i]],RR_i)/1000

  h_iw[l[i]] = h_ew[l[i]+1]
  P_iw[l[i]] = P_ew[l[i]+1]
end

{summary}
Q_dot_tot = sum(q_dot[i], i = 1,N)
Q_dot_tot2 = m_dot*(h_e[N] - h_i[1])
dp_tot = P_i[1]-P_e[N]
dp_tot_w = P_iw[N] - P_ew[1]

```

Arrays Table: Main

	dp _i	dpB _i	dpB _{test,i}	dp _{test,i}	dp _{w,i}	f _i	f _{w,i}	h _i	h _{cr,i}	h _{cw,i}
	[kPa]	[kPa]						[kJ/kg]		
1	5.13	0.6123	0.6123	5.13	11.92	0.02666	0.04002	-190.4	3066	8108
2	5.859	0.6911	0.6911	5.859	11.86	0.02661	0.03981	-176.2	3288	8365
3	6.737	0.7744	0.7743	6.737	11.8	0.02655	0.03957	-159.5	3520	8666
4	7.797	0.8489	0.8489	7.796	11.73	0.02648	0.03932	-139.7	3751	9022
5	9.071	0.9443	0.9443	9.071	11.66	0.02639	0.03903	-116.3	3956	9441
6	9.746	0.8524	0.8524	9.756	11.58	0.02629	0.03873	-89.1	3902	9928
7	14.75	0.5263	0.5263	14.75	11.52	0.02626	0.03847	-63.46	3626	10384
8	16.5			16.5	11.48	0.02626	0.03831	-46.03	3271	10694

Arrays Table: Main

	h _{e,i}	h _{ew,i}	h _{i,i}	h _{iw,i}	h _{w,i}	l _i	Lenght _i	P _i	P _{e,i}	P _{ew,i}	P _{i,i}
	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]			[kPa]	[kPa]		[kPa]
1	-183.9	91.79	-197	98.66	95.22	1	2	4280	4278	156.4	4283
2	-168.5	98.66	-183.9	106.7	102.7	2	4	4274	4271	168.4	4277
3	-150.4	106.7	-168.5	116.2	111.5	3	6	4267	4264	180.2	4271
4	-129	116.2	-150.4	127.5	121.9	4	8	4259	4255	192	4263
5	-103.6	127.5	-129	140.8	134.2	5	10	4250	4246	203.8	4255
6	-74.58	140.8	-103.6	156.1	148.5	6	12	4240	4235	215.4	4245
7	-52.34	156.1	-74.58	167.8	161.9	7	14	4227	4219	227	4234
8	-39.71	167.8	-52.34	174.4	171.1		16	4210	4202	238.5	4219

Arrays Table: Main

P _{iw,i}	q _i	q _{flux,i}	q _{test,i}	Re _i	Re _{w,i}	P _i	P _{e,i}	P _{i,i}	P _{w,i}
-------------------	----------------	---------------------	---------------------	-----------------	-------------------	----------------	------------------	------------------	------------------

		[kJ/kg]	[kJ/kg]	[C]			[kg/m ³]	[kg/m ³]	[kg/m ³]	[kg/m ³]
1	168.4	2.513	25.38	2.513	281739	10061	231.4	220.2	243.8	997.5
2	180.2	2.953	29.82	2.953	310677	10486	208.3	197.6	220.1	997
3	192	3.485	35.19	3.485	353480	10996	186.3	176.2	197.6	996.4
4	203.8	4.123	41.64	4.123	422326	11611	165.6	156.2	176.2	995.6
5	215.4	4.875	49.23	4.875	548989	12354	146.3	137.6	156.2	994.6
6	227	5.584	56.39	5.584	843543	13240	128.8	118.6	137.6	993.3
7	238.5	4.277	43.2	4.277	984121	14096	109.7	102.1	118.5	992.3
8	250	2.428	24.52	2.428	964537	14690	98.17	94.5	102.1	991.7

Arrays Table: Main

	T _i	T _{e,i}	T _{i,i}	T _{tw,i}	T _{two,i}	T _{w,i}	UA _i	V _i	V _{w,i}	x _i	x _{e,i}
	[C]	[C]	[C]			[C]					
1	7.976	7.952	8	7.984	22.68	22.69	170.8	4.26	1.414	0.4695	0.5015
2	7.921	7.892	7.947	7.93	24.47	24.47	178.5	4.733	1.415	0.5392	0.5767
3	7.855	7.823	7.886	7.865	26.57	26.57	186.2	5.291	1.416	0.621	0.6652
4	7.779	7.743	7.816	7.79	29.05	29.05	193.8	5.952	1.417	0.7175	0.7697
5	7.692	7.65	7.735	7.705	31.98	31.99	200.7	6.737	1.419	0.8313	0.8928
6	7.595	10.73	7.641	7.61	35.4	35.4	200.8	7.651	1.42	0.9632	97.58
7	16.62	23.45	10.71	16.63	38.62	38.62	194.4	8.982	1.422	99.78	99.81
8	27.64	32.05	23.45	27.64	40.81	40.81	184.3	10.04	1.423	100	100

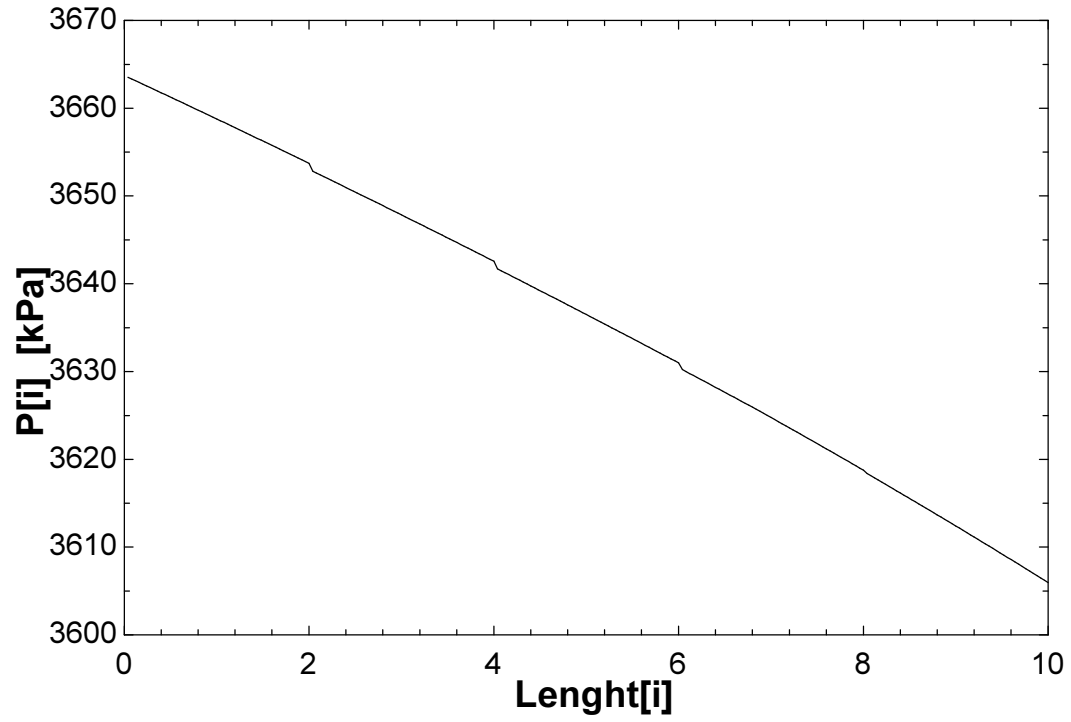
Arrays Table: Main

	x _{i,i}
1	0.4375
2	0.5016
3	0.5767
4	0.6652
5	0.7697
6	0.8928
7	97.58
8	99.81

File:case 10.EES

2013/03/04 03:34:19 AM Page 10

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa



Chapter 8

Finned Tube Simulation Program

File:Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 1

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

Function ReNumber(G,Dh,x,T,P,h)
{function to calculate reynolds number - refrigerant}
if x <= 0 then
mu = viscosity(R744,x = 0,T = T)           {viscosity of liquid refrigerant}
else
if x > 1 then
mu = viscosity(R744,P = P, T = T+0.1)      {viscosity of vapor refrigerant}
else
If x = 1 then
mu = viscosity(R744,x = 1, T = T)          {viscosity of vapor refrigerant}
else
mu_l = viscosity(R744,x = 0, T = T)
mu_v = viscosity(R744,x = 1, T = T)
mu = mu_l*(1-x) + mu_v*x                  {quality averaged viscosity}
endif
endif
endif
ReNumber = (G*Dh)/mu
end

Function ReNumberw(G,Dh,T)
{function to calculate reynolds number - water}
mu = viscosity(water,x = 0, T = T)
ReNumberw = (G*Dh)/mu
end

Function friction(RR,Re)
{Calculate the friction factor for single phase flows}
{NOTE: all flows are assumed to be turbulent. This assumption will be checked for accuracy throughout}
{Ghanbari-Farshad-Rieke correlation is used}
RR = abs(RR)
If Re < 3000 then
Re = 3000
endif

friction = (-1.52*log10((RR/7.21)^1.042+(2.731/Re)^0.9152))^(-2.169)
end

Function dpwater(G,L,Dh,rho,f)
{calculate single phase pressure drop, liquid or vapour}
{dp_mom = G^2*(1/rho_e -1/rho_i)}           {density change neglible, ignore momentum for single phase}
dp_fric = 0.5*(L/Dh)*(G^2/rho)              {friction pressure drop}
dpwater={dp_mom +} dp_fric                  {total pressure drop}
end

Function dp1phase(G,L,Dh,rho,rho_e,rho_i,f)
{calculate single phase pressure drop, liquid or vapour}
{dp_mom = G^2*(1/rho_e -1/rho_i)}           {momentum pressure drop}
dp_fric = 0.5*f*(L/Dh)*(G^2/rho)            {friction pressure drop}
dp1phase={dp_mom +} dp_fric                 {total pressure drop}
end

Function dp2phase(G,L,Dh,x_e,x_i,T)
{2 phase pressure drop - modified Bo Pierre method by Choi et al (1999)}

If x_e <= 0 then x_e = 0.001
if x_i <= 0 then x_i = 0.001
If x_e >= 1 then x_e = 1
if x_i >= 1 then x_i = 1

dx = abs(x_e-x_i)

```

```

h_lv = abs(enthalpy(R744, T = T, x = 1) - enthalpy(R744, T = T, x = 0))
nu_i = volume(R744, T = T, x = x_i)           {inlet specific volume}
nu_e = volume(R744, T = T, x = x_e)           {outlet specific volume}
mu_l = viscosity(R744, x = 0, T = T)           {viscosity of liquid refrigerant}
Re_l = (G*Dh)/mu_l                             {Liquid Reynolds number}
K_f = (dx)*h_lv*1000)/(L*g#)                  {two phase number}
f_N = 0.00506*Re_l^(-0.0951)*K_f^(0.1554)
dp2phase = ((f_N*L*(nu_e+nu_i))/Dh+(nu_e-nu_i))*G^2
end

Function dp(G,L,Dh,rho,rho_e,rho_i,f,x,x_e,x_i,T)
{pressure drop function - determine if single phase or two phase based on average quality of the increment}
if x <= 0.02 then                               {all liquid flow}
dp = dp1phase(G,L,Dh,rho,rho_e,rho_i,f)
else
if x >= 0.98 then                               {all vapour flow}
dp = dp1phase(G,L,Dh,rho,rho_e,rho_i,f)
else
dp = dp2phase(G,L,Dh,x_e,x_i,T)                 {two phase flow}
endif
endif
end

{Function dpbend1phase(G,rho,R,Dh)
{Calculate bend pressure drop for single phase flows - Paliwoda1992}
Sx = 2*R                                         {Tipe pitch = 2 x bend diameter}
Kp = 1/(3.426*ln(Sx/(2*Dh))+3.8289)            {correction factor}
dpbend1phase = 0.5*Kp*G^2/rho
end}

Function dpbend1phase2(G,rho,R,Dh,RR,T,L)
{Calculate bend pressure drop for single phase flows - Idelshik 1986}
mu_l = viscosity(R744, x = 0, T = T)           {viscosity of liquid refrigerant}
Re_l = (G*Dh)/mu_l                             {Liquid Reynolds number}
f = friction(RR,Re_l)
K_sp = f*(L/Dh)+0.294*(R/Dh)^0.5              {correction factor}
dpbend1phase2 = 0.5*K_sp*G^2/rho
end}

Function dpstraight(G,L,Dh,x_e,x_i,T)
{2 phase pressure drop - modified Bo Pierre method by Choi et al (1999)}
{modified to be used for bend pressure drop calculation}
{no heat transfer for a bend - quality change assumed negligible}
If x_e <= 0 then x_e = 0.001
if x_i <= 0 then x_i = 0.001
If x_e >= 1 then x_e = 1
if x_i >= 1 then x_i = 1

{dx = abs(x_e-x_i)}
dx = 0                                           {no change in quality}
h_lv = abs(enthalpy(R744, T = T, x = 1) - enthalpy(R744, T = T, x = 0))
nu_i = volume(R744, T = T, x = x_i)           {inlet specific volume}
nu_e = volume(R744, T = T, x = x_e)           {outlet specific volume}
mu_l = viscosity(R744, x = 0, T = T)           {viscosity of liquid refrigerant}
Re_l = abs((G*Dh)/mu_l)                       {Liquid Reynolds number}
{K_f = (dx)*h_lv*1000)/(L*g#)}                 {two phase number = 0 - no since dx = 0}
K_f = 0
f_N = 0.00506*Re_l^(-0.0951)
dpstraight = ((f_N*L*(nu_e+nu_i))/Dh+(nu_e-nu_i))*G^2
volumes, thus blank out}                       {no change in inlet and outlet specific

```

end

Function dpbend1phase3(G,R,Dh,RR,T,L,x)

{Calculate bend pressure drop for single phase flows - from chen 2003}

If T > 30 then

T = 30

endif

If x < 0.5 then

rho = density(R744,x = 0, T = T) *{density of liquid refrigerant}*

mu = viscosity(R744,x = 0, T = T) *{viscosity of liquid refrigerant}*

else

rho = density(R744,x = 0, T = T) *{density of vapor refrigerant}*

mu = viscosity(R744,x = 1, T = T) *{viscosity of vapor refrigerant}*

endif

Re = (G*Dh)/mu *{Reynolds number}*

Dn = Re/(2*R/Dh) *{Dean number}*

a = 0.021796

b = 0.0413356

f = (2.71828^(a+b*(ln(Dn))^2))*(64/Re) *{friction factor for bend flow}*

K = f*(L/Dh) *{correction factor}*

dpbend1phase3 = 0.5*K*G^2/rho

end

Function dpbend2phase(G,L_st,Dh,R,x_e,x_i,x,T)

{calculate bend pressure drop for 0.2<x<0.8 - Domanski and Hermes}

dp_straight = dpstraight(G,L_st,Dh,x_e,x_i,T)

a0 = 6.5e-3

a1 = 0.54

a2 = 0.21

a3 = 0.34

a4 = -0.67

rho_l = density(R744,x = 0, T = T) *{density of liquid refrigerant}*

rho_v = density(R744,x = 1, T = T) *{density of vapour refrigerant}*

mu_v = viscosity(R744,x = 1, T = T)

CM = a0*((G*x*Dh)/mu_v)^a1*(1/x-1)^a2*(rho_l/rho_v)^a3*((2*R)/Dh)^a4 *{curvature multiplier}*

dpbend2phase = dp_straight * CM

end

function dpbend(G,L_st,Dh,R,x,T,RR)

{calculate bend pressure drop}

{for a bend the quality is assumed constant throughout the bend}

{For x < 0 and x > 0 - use single phase correlation}

{for 0.2 < x, 0.8 use two phase correlation}

{for other conditions, use the singlephase if greater than two phase correlation, otherwise use average}

If (x <= 0) or (x >= 1) then

dpbend = dpbend1phase3(G,R,Dh,RR,T,L_st,x) *{single phase}*

else

if (0.2 < x) and (x < 0.8) then

dpbend = dpbend2phase(G,L_st,Dh,R,x,x,x,T) *{two phase - apply for 0.2<x<0.8}*

else

dpb1 = dpbend1phase3(G,R,Dh,RR,T,L_st,x)

dpb2 = dpbend2phase(G,L_st,Dh,R,x,x,x,T)

if dpb1 > dpb2 then *{use single phase if greater than 2 phase}*

dpbend = dpb1

else

dpbend = (dpb1 + dpb2)/2 *{else use average}*

File: Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 4

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```
endif
endif
endif
end
```

```
function hwater(Dh, f, Re, T, P)
{calculate the heat transfer coefficient for liquid and vapor}
{Gnielinski correlation is used}
Pr = Prandtl(water, T = T, P = P)
k = Conductivity(water, T = T, P = P)
Nu = ((f/8)*(Re - 1000)*Pr)/(1 + 12.7*(f/8)^0.5*(Pr^(2/3) - 1))
hwater = Nu*k/Dh
end
```

```
function h1phase(Dh, f, Re, T, P, h)
{calculate the heat transfer coefficient for liquid and vapor}
{Gnielinski correlation is used}
x = quality(R744, h = h, P = P)
if x < 100 then
Pr = Prandtl(R744, P = P, x = 1)
k = Conductivity(R744, P = P, x = 1)
else
Pr = Prandtl(R744, h = h, P = P)
k = Conductivity(R744, h = h, P = P)
endif
Pr = abs(pr)
Nu = ((f/8)*(Re - 1000)*Pr)/(1 + 12.7*(f/8)^0.5*(Pr^(2/3) - 1))
h1phase = Nu*k/Dh
end
```

```
function h2phase(G, Q_flux, Dh, x, Re, T)
Q_flux = abs(Q_flux)
```

```
{calculate the heat transfer coefficient for liquid and vapor}
{Jung et al (1989) correlation is used}
mu_l = viscosity(R744, T = T, x = 0) {viscosity liquid}
mu_v = viscosity(R744, T = T, x = 1) {viscosity vapour}
rho_l = density(R744, T = T, x = 0) {density liquid}
rho_v = density(R744, T = T, x = 1) {density vapour}
h_lv = enthalpy(R744, T = T, x = 1) - enthalpy(R744, T = T, x = 0) {latent heat of vaporization}
Pr_l = prandtl(R744, T = T, x = 0) {liquid pr number}
k_l = conductivity(R744, T = T, x = 0) {liquid conductivity}
T_s = T + 273.15 {Saturated temperature in kelvin}
```

```
sigma = SurfaceTension(R744, T = T)
```

```
X_tt = ((1-x)/x)^0.9*(rho_v/rho_l)^0.5*(mu_l/mu_v)^0.1 {Martinelli parameter}
Bo = Q_flux/(G*h_lv) {boiling number}
if X_tt < 1 then {nucleate boiling factor}
N = 4048*X_tt^1.22*Bo^1.13
else
N = 2*0.1*(X_tt)^(-0.28)*(Bo)^(-0.33)
endif
bd = 0.0146*35*((2*sigma)/(g*(rho_l-rho_v)))^0.5
h_SA = 207*(k_l/bd)*((Q_flux*1000*bd)/(k_l*T_s))^0.745*(rho_v/rho_l)^0.581*Pr_l^0.533 {pool boiling heat transfer}
```

```
Fp = 2.37*(0.29 + 1/X_tt)^0.85 {F factor}
Re_l = ReNumber(G*(1-x), Dh, 0, T, 1, 0) {Liquid reynolds number wiht G = G(1-x)}
h_lo = 0.023*(k_l/Dh)*Re_l^0.8*Pr_l^0.4 {Ditus boelter equation - liquid only}
```

File: Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 5

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

h_nbc = N*h_SA
h_cec = Fp*h_lo
h2phase = h_nbc + h_cec           {two phase heat transfer}
end

function hc(G,Q,Dh,x, f,Re,T,P,h)
{check for phase and use appropriate heat transfer correlation}
if (x <= 0) then                  {all liquid flow}
    hc = h1phase(Dh, f,Re,T,P,h)
else
    if x >= 1 then                 {all vapour flow}
        hc = h1phase(Dh, f,Re,T,P,h)
    else
        {linear changeover between phases to prevent discontinuity}
        if (x > 0.95) then
            hv = enthalpy(R744, x = 1, T=T)
            tv = temperature (R744, x = 1, P = P)
            h1 = h1phase(Dh, f,Re,Tv,P,hv)
            h2 = h2phase(G,Q,Dh,x,Re,T)
            hc = (h1-h2)/0.05*(x-0.95)+h2           {linear change over for two phase to vapour flow}
        else
            if (x < 0.05) then
                hv = enthalpy(R744, x = 0, T=T)
                tv = temperature (R744, x = 0, P = P)
                h1 = h1phase(Dh, f,Re,Tv,P,hv)
                h2 = h2phase(G,Q,Dh,x,Re,T)
                hc = (h2-h1)/0.05*(x)+h1           {linear change over for liquid to two phase flow}
            else
                hc = h2phase(G,Q,Dh,x,Re,T)           {two phase flow}
            endif
        endif
    endif
endif
end

Procedure wwet(T_ai,T_s,h_a,h_ai,h_ae,P_a,T_dew, w_dry: w_wet, T_eq1)
If T_s < T_dew then
    omega_s = HumRat(AirH2O,T=T_s,r = 1,P=P_a)           {specific humidity for saturated air at the
    surface temperature}
    T_eq = Temperature(AirH2O,h=h_a,w=omega_s,P=P_a)       {average equivalent dry bulb
    temperature}
    T_eq1 = Temperature(AirH2O,h=h_ai,w=omega_s,P=P_a)     {calculate equivalent dry bulb
    temperature for inlet}
    T_eq2 = Temperature(AirH2O,h=h_ae,w=omega_s,P=P_a)     {calculate equivalent dry bulb
    temperature for outlet}
    T_ae_c = -1*(T_ai-T_s)/(T_eq1-T_s)*(T_eq1-T_eq2)+T_ai {calculate outlet dry bulb temperature
    by linear interpolation from the psychometric chart}
    {for stability}
    w_wet = HumRat(AirH2O,h=h_ae,T=T_ae_c,P=P_a)
    {calculate outlet specific humidity for enthalpy and dry bulb temperature at the outlet}
else
    w_wet=w_dry
    T_eq1 = T_ai
endif

end

Procedure wetdry(T_dew,T_s,h_dry,h_wet,eta_dry,eta_wet,T_dry,T_eq,w_i,w_wet:h,eta,T,w,count)
{Test if the surface is wet or dry and use the correct values}
If T_s > T_dew then           {if dry surface}

```

File:Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 6

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

h = h_dry
eta = eta_dry
T = T_dry
w = w_i
count = 0
else                                     {if wet surface}
    h = h_wet
    eta = eta_wet
    T = T_eq
    w = w_wet
    count = 1
endif
count = count + count
end

{Main Program}
{Geometry Inputs}
L_t = 1.6                               {Lenght of the coil and therefore a single tube}
N_t = 4                                 {Number of tubes high}
N_r = 4                                 {Number of rows}
N_tubes = N_t*N_r                       {Total number of tubes}
Inc = 1                                 {Number of increments per tube}
N_row = N_t*Inc                         {Number of increments per row}
N = N_t*N_r*Inc                         {total number of increments}
L_inc = L_t/Inc                         {Lenght per increment}

{Tubeside geometry}
D_i = 8.92/1000                         {Inner diameter of the tube}
W_t = (10.14-8.92)/2000                 {Wall thickness of the tube}
D_io = D_i + W_t*2                      {Outer diameter of the inner tube}

D_hi = D_i                              {Hydraulic diameter of the inner tube}
A_ci = pi*(D_i/2)^2                     {Cross sectional area of the inner tube}
A_hi = pi*D_i*L_inc                     {Tube inner heat transfer area per increment}
A_ho = pi*D_io*L_inc                    {Tube outer heat transfer area}

Sx = 19.05/1000                         {Row pitch - distance between tube rows}
Sy = 25.4/1000                          {Tube pitch - distance between tubes}

{Finside geometry}
FPI = 12                                {Fins per inch}
delta_f = 0.12/1000                     {Fin thickness}
D_c = D_io+delta_f*2                     {finside diameter of tube - fin collar formed by pressed fins}
fin_spac = 25.4/FPI/1000                 {Distance between fins}
NOF = L_t/fin_spac                       {total number of fins}
H = Sy*N_t                               {Total height of the coil}

A_fr = H*L_t                             {Total frontal area}
sigma = (Sy-(1/fin_spac*delta_f*(Sy-D_c)+D_c))/Sy {ratio of minimum free flow area to frontal area}
A_c = sigma*A_fr                         { free flow area}
A_tube = pi*D_c*N_tubes*(L_t-NOF*delta_f) {heat transfer area of the tubes - including collar}
A_fin = (Sy*Sx-pi*(D_c/2)^2)*N_tubes*NOF*2 {heat transfer area of the tubes}
A_tot = A_tube+A_fin                     {Total heat transfer area}
FA_rat = A_fin/A_tot                     {finned area to total area ratio}
Vol = H*Sx*N_r*L_t                       {total volume of the coil}
alfa = A_tot/Vol                         {Ratio of total heat transfer area to volume}
mass_tube = pi*((D_c/2)^2-(D_i/2)^2)*N_tubes*L_t*rho_t {Total mass for the tubes}
mass_bends = pi*((D_c/2)^2-(D_i/2)^2)*N_tubes*L_t*st*rho_t {total mass for the bends - asuming every tube have a bend}

```

```

mass_fin = (Sy*Sx-pi*(D_c/2)^2)*delta_f*N_tubes*NOF*rho_f {Total mass for the fins}
mass_tot = mass_tube+mass_bends+mass_fin {total mass exluding flanges}
mass_ratio = mass_tot/Vol {ratio of mas to the total volume}
kw_ratio = Q_tot/Vol {ratio of total capacity per volume}
Q_tot = Q_tot_ref

{materials}
k_ss= k_('Stainless_AISI304', Temperature) {conductivity of stainless steel - tube wall}
rho_ss = rho_('Stainless_AISI304', Temperature) {density of stainless steel}
k_cu = k_('Copper', Temperature) {conductivity of copper - tube wall}
rho_cu = rho_('Copper', Temperature) {density of copper}
k_a= k_('Aluminum', Temperature) {conductivity of aluminum - fin material}
rho_a= rho_('Aluminum', Temperature) {Denisty of aluminum}
Temperature = 15 {assumed constant for the applicable temperature range}
k_t = k_cu {tubewall conductivity}
rho_t = rho_cu {tubewall density}
k_f = K_a {Fin conductivity}
rho_f = rho_a {fin density}

{Return bend}
D_rb = Sy {Diameter for the return bend - distance between the tubes}
R_rb = D_rb/2 {Raduis of the return bend}
L_st = pi*(D_rb/2) {equivalent straight length for the return bend}

{friction factor inputs}
e =1.5e-6 {surface roughness- drawn tubing }
RR_i = e/D_hi {Relative roughness for the inner tube}

{Air side inputs}
{m_dot_dry =0.5997} {mass flow rate of dry air - constant throughout}
V_on = 2.5 {Velocity onto the coil} {Note - need to fix either mass flow or volume flow onto coil}

m_dot_inc = m_dot_dry/(N_row+0.5*Inc) {mas flow rate through each increment - 0.5 inc at one row at the side for bypass}
m_dot_a = m_dot_dry *( omega_i+1) {mass flow rate of air mixture}
T_ai =20 {Air inlet temperature}
P_ai = 83.5 {Air inlet pressure}
h_ai = enthalpy(AirH2O, T = T_ai, P = P_ai, r = RH_i) {Air inlet enthalpy}
RH_i = 0.5 {relatvie humidy of the air at the inlet}
omega_i = HumRat(AirH2O, T = T_ai, P = P_ai, r = RH_i) {Air inlet humidity ratio}
rho_ai = density(AirH2O, T = T_ai, P = P_ai, r = RH_i) {density of air at the inlet}
Vol_on = V_on*A_fr {volume flow of moist air into the coil}
V_on = m_dot_a/(rho_ai*A_fr) {On coil air velocity}
V_ai = m_dot_a/(rho_ai*A_fr) {Velocity of air in the coil at the inlet}

m_dot_a = G_a*A_c {mass flux of air through finned area}

{Air side outlet conditions- average of all tube outlets}
T_ae= T_air[1] {average outlet temperature}
P_ae = P_ai- {dp_tot}0.1
P_a = average(P_ae,P_ai)
rho_ae = density(AirH2O, T = T_ae, P = P_ae, r = 0.8)

{wet and dry conditions}
frac_wet = count[N_r,N_t,Inc] / (N_r*N_t*Inc)

frac_dry = 1-frac_wet

{Air side conditions are calculated for the complete coil and then applied per element}
{calculate air side pressure drop}
G_a = G_ai

```

```

mu_ai = viscosity(AirH2O, T = T_ai, P = P_ai, r = RH_i) {density of air at the inlet}
cp_ai = cp(AirH2O, T = T_ai, P = P_ai, r = RH_i)
k_ai = conductivity(AirH2O, T = T_ai, P = P_ai, r = RH_i)
Pr_ai = Cp_ai*1000*mu_ai/k_ai
Re_Dc = G_a*D_c/mu_ai

f_dry = 16.67/(ln(Re_Dc))^2.64*(A_tot/A_tube)^(-0.096)*N_r^0.098 {friction factor for dry coil}
f_wet = 64.0524*Re_Dc^(-0.69284)*N_r^(-0.5237)*(A_tot/A_tube)^(-0.54736)*(delta_f/Sx)^(-0.098371)
{friction factor for wet coil}
f = 1.1285*Re_dc^(-0.4911)*(fin_spac/D_c)^(-0.6842)*(25.4/22)^(-0.5947)*(0.5)^(-0.2887)*N_r^0.0333

K_i = 0
K_e = 0
dp_dry*1000 = G_a^2/(2*rho_ai)*((K_i + 1 - sigma^2)+2*(rho_ai/rho_ae-1)+f_dry*(A_tot/A_c)*(rho_ai/rho_ae)-(1-sigma^2-K_e)
*(rho_ai/rho_ae)) {pressure drop for a dry coil}
dp_wet*1000 = G_a^2/(2*rho_ai)*((K_i + 1 - sigma^2)+2*(rho_ai/rho_ae-1)+f_wet*(A_tot/A_c)*(rho_ai/rho_ae)-(1-sigma^2-K_e)
*(rho_ai/rho_ae)) {pressure drop for a wet coil}

dp_tot = frac_wet * dp_wet + frac_dry*dp_dry {pressure drop for the coil}
P_fan = (G_a*A_c*dp_tot)/rho_ai {Fan power required}

{Calculate air side heat transfer}

j_dry = 1.201/(ln(Re_dc*sigma)^2.921)
j_wet = 0.171*(epsilon)^(0.377*N_r)*Re_dc^(-0.0142*N_r-0.478)*(fin_spac/D_c)^(0.00412*N_r-0.0217)*(A_tot/A_tube)^(-0.114
*N_r+0.440)}
j_wet = j_toets
j_toets = 0.213262*Re_dc^(-0.51507)*N_r^(0.09891)*(A_tot/A_tube)^0.600543*(delta_f/Sx)^0.072448
j_1 = 6.6412*(Sx/D_c)^(-0.00085)*(Sy/D_c)^(-2.1461)*Re_dc^(-0.2636*fin_spac/D_c-0.00091*Sx/D_c+0.1558*Sy/D_c-0.8865)
j_N = j_1*N_r^(-0.06451)*(fin_spac/D_c)^(-0.1219*N_r+0.7381)*(epsilon)^(0.1616*N_r+0.6105)*Re_dc^(0.03475*N_r+0.1145
*fin_spac/D_c-0.00521*Sx/D_c-0.03498*Sy/D_c-0.04374)
epsilon = a_tot/a_tube

Nu_aw = j_N*Re_dc*Pr_ai^(1/3)
Nu_aw = h_c_a_wet*D_c/k_ai
Nu_ad = j_dry*Re_dc*Pr_ai^(1/3)
Nu_ad = h_c_a_dry*D_c/k_ai

{Calculate surface efficiency}
eta_o_wet = 1 - (A_fin/A_tot)*(1-eta_fin_wet)
eta_o_dry = 1 - (A_fin/A_tot)*(1-eta_fin_dry)
eta_fin_wet = (tanh(M_wet*r_i*phi)*cos(0.1*M_wet*r_i*phi))/(M_wet*r_i*phi)
eta_fin_dry = (tanh(M_dry*r_i*phi)*cos(0.1*M_dry*r_i*phi))/(M_dry*r_i*phi)
M_wet = 2*h_c_a_wet/(k_f*delta_f)
M_dry = 2*h_c_a_dry/(k_f*delta_f)
r_i = D_io/2
phi = (r_o/r_i - 1)*(1+0.35*ln(r_o/r_i))
r_o = 1.27*X_M*sqrt(X_L/X_M-0.3)
X_M = 0.5*Sy
X_L = 0.5*sqrt((0.5*Sy)^2+Sx^2)

{refrigerant inputs}
m_dot = 117/3600 {mass flow refrigerant. Note - change mass flow until required superheat is reached}
m_dot = G*A_ci {mass flux}
P_gas = 101.325*100 {outlet pressure of the evaporator}
T_gas = 37.3 {Outlet temperature from the evaporator}
h_gas = enthalpy(R744, T = T_gas, P = P_gas) {outlet enthalpy from the evaporator}
h_in = h_gas {inlet enthalpy into the evaporator}

{Note, refrigerant properties without a subscript}

```

{subscibt i = inlet, e = exit}

{Refrigerant side - inlet conditions}

{Inlet conditions - exit conditions of dummy increment 0}

h_in = enthalpy(R744, T = T_in, x = x_in)

h_e[1,1,0] = h_in

T_in = 5.4

P_e[1,1,0] = Pressure(R744, T = T_in, x = x_in)

x_in = 0.13

P_in = P_e[1,1,0]

Duplicate i = 1, N_r

Duplicate j = 1, N_t

Duplicate k = 1, Inc

{refrigerant side}

{Inlet properties}

T_i[i,j,k] = Temperature (R744, h=h_i[i,j,k], P= P_i[i,j,k])

x_i[i,j,k] = quality(R744, P = P_i[i,j,k], h = h_i[i,j,k])

rho_i[i,j,k] = density(R744, P = P_i[i,j,k], h = h_i[i,j,k])

{outlet properties}

T_e[i,j,k] = Temperature (R744, h=h_e[i,j,k], P= P_e[i,j,k])

x_e[i,j,k] = quality(R744, h=h_e[i,j,k], P = P_e[i,j,k])

rho_e[i,j,k] = density(R744, h=h_e[i,j,k], P = P_e[i,j,k])

{average properties}

h[i,j,k] = (h_i[i,j,k] + h_e[i,j,k])/2 *{linearly average enthalpy}*

P[i,j,k] = (P_i[i,j,k] + P_e[i,j,k])/2 *{linearly average pressure}*

T[i,j,k] = temperature(R744, P = P[i,j,k], h = h[i,j,k])

x[i,j,k] = quality(R744, P = P[i,j,k], h = h[i,j,k])

rho[i,j,k] = density(R744, h= h[i,j,k], P = P[i,j,k])

m_dot = rho[i,j,k]*V[i,j,k]*A_ci *{average velocity for homogenous fluid}*

{conservation of momentum}

Re[i,j,k] = ReNumber(G,D_hi,x[i,j,k],T[i,j,k],P[i,j,k],h[i,j,k])

f[i,j,k] = friction(RR_i,Re[i,j,k]) *{f[i,j,k]= 0.025}*

dp[i,j,k] = dp(G,L_inc,D_hi,rho[i,j,k],rho_e[i,j,k],rho_i[i,j,k],f[i,j,k],x[i,j,k],x_e[i,j,k],x_i[i,j,k],T[i,j,k])/1000

{dp[i,j,k] = 2}

P_e[i,j,k] = P_i[i,j,k] - dp[i,j,k]

{conservation of energy}

q_dot[i,j,k] = m_dot*(h_e[i,j,k] - h_i[i,j,k])

q_flux[i,j,k] = q_dot[i,j,k]/A_hi *{heat flux}*

{air side}

{inlet properties}

T_ai[i,j,k]=Temperature(AirH2O,h=h_ai[i,j,k],w=w_ai[i,j,k],P=P_a)

{outlet properties}

T_ae[i,j,k]=Temperature(AirH2O,h=h_ae[i,j,k],w=w_ae[i,j,k],P=P_a)

{average properties}

h_a[i,j,k] = (h_ai[i,j,k] + h_ae[i,j,k])/2 *{linearly average enthalpy}*

{average temperature calculation}

T_a_d[i,j,k]=Temperature(AirH2O,h=h_a[i,j,k],w=w_ai[i,j,k],P=P_a)

{condition}

{Average temperature for dry surface}

{equivalent temperature calculation}

File: Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 10

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

{If surface is wet use equivalent temperature, h_wet, eta_w and change the specific humidity}
T_dew[i,j,k] = DewPoint(AirH2O,T=T_ai[i,j,k],w=w_ai[i,j,k],P=P_a)
{T_ae_c[i,j,k]=Temperature(AirH2O,h=h_ae[i,j,k],w=omega_1,P=P)}

call wwet(T_ai[i,j,k],T_s[i,j,k],h_a[i,j,k],h_ai[i,j,k],h_ae[i,j,k],P_a,T_dew[i,j,k],w_ai[i,j,k]: w_wet[i,j,k], T_eq1[i,j,k])
call wetdry(T_dew[i,j,k],T_s[i,j,k],h_c_a_dry,h_c_a_wet,eta_o_dry,eta_o_wet,T_a_d[i,j,k],T_eq1[i,j,k],w_ai[i,j,k],W_wet[i,j,k]:h_ca[i,j,k],eta_o[i,j,k],T_ai[i,j,k],w_ae[i,j,k],count[i,j,k])

{h_ca[i,j,k] = h_c_a_dry                                {convection heat transfer coefficient for air}
eta_o[i,j,k] = eta_o_dry
T_a[i,j,k] = T_a_d[i,j,k]
w_ae[i,j,k] = w_ai[i,j,k]}

{conservation of mass}

{conservation of energy}
q_dot[i,j,k] = m_dot_inc*(w_ai[i,j,k]+1)*h_ai[i,j,k] - m_dot_inc*(w_ae[i,j,k]+1)*h_ae[i,j,k]

{heat transfer between the fluids}
q_dot[i,j,k] = (UA[i,j,k]*(T_ai[i,j,k] - T[i,j,k]))/1000                                {heat transfer due to temperature difference}
{q_dot[i,j,k] = 0.41}

q_test[i,j,k] = (UA[i,j,k]*(T_ai[i,j,k] - T[i,j,k]))/1000

h_cr[i,j,k] = hc(G,q_flux[i,j,k],D_hi,x[i,j,k], f[i,j,k],Re[i,j,k],T[i,j,k],P[i,j,k],h[i,j,k])                                {convection heat transfer coefficient for refrigerant}
1/UA[i,j,k] = 1/(h_cr[i,j,k]*A_hi) + ln(D_io/D_i)/(2*pi*k_t*L_inc) + 1/(h_ca[i,j,k]*A_tot*eta_o[i,j,k])
{overall heat transfer coefficient}
T_s[i,j,k] = T[i,j,k] + q_dot[i,j,k]*(1/(h_cr[i,j,k]*A_hi) + ln(D_io/D_i)/(2*pi*k_t*L_inc))                                {Tube wall temperature}
end
end
end

{{Set guess values for heat transfer}
Duplicate j = 1,N_t
Duplicate k = 1,Inc
q_dot[1,j,k] = 0.09
end
end

Duplicate j = 1,N_t
Duplicate k = 1,Inc
q_dot[2,j,k] = 0.18
end
end

Duplicate j = 1,N_t
Duplicate k = 1,Inc
q_dot[3,j,k] = 0.4
end
end

Duplicate j = 1,N_t
Duplicate k = 1,Inc
q_dot[4,j,k] = 0.55

```

```

end
end}

{Airsides properties transfer}
{element defined as x[Rownumber,Tubenum,elementnumber]}
{define air inlet properties for each element}
{pressure assumed constant throughout - P_ai}
{For the last row - equal to inlet conditions}
Duplicate j = 1, N_t
Duplicate k = 1, Inc
h_ai[N_r,j,k] = h_ai
w_ai[N_r,j,k] = omega_i
end
end

{For the inner rows equal to the average of the upstream elements}
duplicate i = 1, N_r
RN[i] = step(trunc(i/2) - i/2)           {If row number even = 1, uneven = 0}
end

Duplicate i = 1, N_r-1
Duplicate j = 1, N_t
Duplicate k = 1, Inc

{air properties average of the 2 upstream elements}
h_ai[i,j,k] = (h_ae[i+1,j-RN[i],k]+h_ae[i+1,j+1-RN[i],k])/2
w_ai[i,j,k] = (w_ae[i+1,j-RN[i],k]+w_ae[i+1,j+1-RN[i],k])/2

end
end
end

{create dummy elements}
{determine if bottom or top row is at the edge and create dummy element with properties of 2 rows previously}

Duplicate i = 1, N_r- 2
a[i] = N_t+1 - RN[i]*(N_t+1)
b[i] = a[i]-1+2*RN[i]
Duplicate k = 1, Inc
h_ae[i+1,a[i],k] = h_ae[i+2,b[i],k]
w_ae[i+1,a[i],k] = w_ae[i+2,b[i],k]
end
end

Duplicate i = N_r-1, N_r- 1
a[i] = N_t+1 - RN[i]*(N_t+1)
b[i] = a[i]-1+2*RN[i]
Duplicate k = 1, Inc
h_ae[i+1,a[i],k] = h_ai
w_ae[i+1,a[i],k] = omega_i
end
end

{Determine average properties for all the elements at inlet, outlet, and between rows.}

Duplicate i = 1, N_r
h_air[i] = (average(h_ae[i,1..N_t,1..Inc])*N_row + h_ai[i+1]*0.5*Inc)/(N_row+Inc*0.5) {estimated - bypass from dummy element
only indirectly accounted for}
w_air[i] = average(w_ae[i,1..N_t,1..Inc])
T_air[i] = Temperature(AirH2O,h=h_air[i],w=w_air[i],P=P_a)

```

File: Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 12

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

```

end
h_air[N_r+1] = h_ai
T_air[N_r+1] = T_ai
w_air[N_r+1] = omega_i

Q_tot_air = m_dot_dry*(w_air[N_r+1]+1)*h_air[N_r+1] - m_dot_dry*(w_air[1]+1)*h_air[1]

{Refrigerant side propeties transfer}
{Element (1,1,1) is always defined as the refrigerant inlet}
{row numbers is used to define flow from top or bottom - thus maximum number of rows allowed is 8}

Duplicate i = 1, N_r
Duplicate j = 1, N_t
TN[i,j] = step(trunc((i+j)/2) - (i+j)/2)
{Tube number is used to track direction of flow through the tube, either 0 or 1}
Duplicate k = 1, Inc
{Property transport between elements in a tube}
h_i[i,j,k] = h_e[i,j,k+1-2*TN[i,j]]
P_i[i,j,k] = P_e[i,j,k+1-2*TN[i,j]]
{h_e[i,j,k] = h_i[i,j,k] - 1} {for testing - exit = inlet}
{P_e[i,j,k] = P_i[i,j,k] - 1} {For testing - exit = inlet}
end
{Dummy element at both sides of each tube}
c[i,j] = (Inc+1)*TN[i,j]
d[i,j] = 1+(Inc-1)*TN[i,j]
h_i[i,j,c[i,j]] = h_e[i,j,d[i,j]]
P_i[i,j,c[i,j]] = P_e[i,j,d[i,j]] - dpb[i,j] {Detuct bend pressure drop}
{dpb[i,j] = 0.1}
dpb[i,j] = dpbend(G,L_st,D_hi,R_rb,x_e[i,j,d[i,j]],T_e[i,j,d[i,j]],RR_i)/1000
end
end

{Tube linking matrix}
{The following is used to link the inlet and outlet elements of each tube in pure crossflow manner}
{Between tubes in a row}
Duplicate i = 1, N_r
Duplicate j = 1, N_t-1
h_e[i,j+1-RN[i],c[1,j]] = h_i[i,j+RN[i],c[1,j]] {No heat transfer for bends}
P_e[i,j+1-RN[i],c[1,j]] = P_i[i,j+RN[i],c[1,j]] {Bend pressure drop is taken into account at dummy element}
end
end

{Between the rows}
mx = step(trunc(N_t/2)-N_t/2)
Duplicate i = 1, N_r-1
CN[i] = N_t-(N_t-1)*RN[i],
CN2[i] = c[i,1]*(1-mx)
h_e[i+1,CN[i],CN2[i]] = h_i[i,CN[i],CN2[i]]
P_e[i+1,CN[i],CN2[i]] = P_i[i,CN[i],CN2[i]]
end

Duplicate i = N_r, N_r
CN[i] = N_t-(N_t-1)*RN[i],
CN2[i] = c[i,1]*(1-mx)
h_ex = h_i[i,CN[i],CN2[i]]
P_ex = P_i[i,CN[i],CN2[i]]
end

{Overall Result summary}
T_ex = Temperature (R744, h=h_ex, P=P_ex)

```

File:Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 13

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

Superheat = T_ex - T_sat_ex
Q_tot_ref = m_dot*(h_ex - h_in)
dp_tot_ref = P_in - P_ex
T_sat_ex = T_sat(R744,P = P_ex)

SOLUTION

Unit Settings: SI C kPa kJ mass deg

alfa = 843.2
A_fin = 9.657
A_ho = 0.05097
cp_ai = 1.023
dp_tot = 0.06862 [kPa]
D_c = 0.01038
D_io = 0.01014
ε = 13.26
η_o,dry = 0.09035
FA_rat = 0.9246
frac_dry = 0.875
f_wet = 0.05488
G_ai = 4.385
h_c,a,dry = 88.98
h_in = -265.5
j_dry = 0.01624
j_wet = 0.01423
K_ai = 0.02518
K_f = 236.1
K_t = 402.4
L_t = 1.6
mass_ratio = 545.5
μ_ai = 0.00001823
m_a = 0.3976
M_dry = 6281
NOF = 755.9
N_r = 4
N_tubes = 16
P_rai = 0.7409
P_ai = 83.5
P_in = 4010
Q_tot_ref = 6.646 [kJ/m]
p_ae = 1.024
p_f = 2704
RH_i = 0.5
r_o = 0.01251
Superheat = 10.02
Temperature = 15
T_ex = 15.02
Vol = 0.01239
V_on = 2.5
X_L = 0.01145
A_c = 0.09068
A_fr = 0.1626
A_tot = 10.44
δ_r = 0.00012
dp_tot_ref = 40.44 [kPa]
D_hi = 0.00892
D_rb = 0.0254
η_fin,dry = 0.01617
η_o,wet = 0.1053
fin_spac = 0.002117
frac_wet = 0.125
G = 520.1
H = 0.1016
h_c,a,wet = 44.61
Inc = 1
j_n = 0.008141
kW_ratio = 536.5
K_cu = 402.4
K_i = 0
L_inc = 1.6
mass_bends = 0.1263
mass_tot = 6.757
mx = 1
m_dot_dry = 0.3942
M_wet = 3149
v_ad = 36.69
N_row = 4
ω_i = 0.008834
P_a = 83.45
P_ex = 3970
Q_tot = 6.646
Re_Dc = 2497
ρ_ai = 0.9785
ρ_ss = 7904
RR_i = 0.0001682
R_rb = 0.0127
S_x = 0.01905
T_ae = 7.752
T_in = 5.4
Vol_on = 0.4064
W_t = 0.00061
X_M = 0.0127
A_ci = 0.00006249
A_hi = 0.04484
A_tube = 0.7875
dp_dry = 0.07003
dp_wet = 0.05879
D_i = 0.00892
e = 0.0000015
η_fin,wet = 0.0324
f = 0.08441
FPI = 12
f_dry = 0.06527
G_a = 4.385
h_ai = 42.55
h_ex = -60.98
j_1 = 0.01211
j_toets = 0.01423
K_a = 236.1
K_e = 0
K_ss = 14.63
L_st = 0.0399
mass_fin = 1.567
mass_tube = 5.064
m_dot = 0.0325
m_dot_inc = 0.08759
N = 16
v_aw = 18.39
N_t = 4
φ = 1.931
P_ae = 83.4
P_fan = 0.02789
Q_tot_air = 6.761
ρ_a = 2704
ρ_cu = 8938
ρ_t = 8938
r_i = 0.00507
σ = 0.5578
S_y = 0.0254
T_ai = 20
T_sat_ex = 5.004
V_ai = 2.5
X_in = 0.13

290 potential unit problems were detected.

Arrays Table: Main

	a _i	b _i	c _{i,1}	c _{i,2}	c _{i,3}	c _{i,4}	CN _i	CN2 _i	count _{i,1,1}	count _{i,4,1}
1	5	4	2	0	2	0	4	0	2	2
2	0	1	0	2	0	2	1	0	2	2
3	5	4	2	0	2	0	4	0	2	2

Arrays Table: Main

	a_i	b_i	$c_{i,1}$	$c_{i,2}$	$c_{i,3}$	$c_{i,4}$	CN_i	$CN2_i$	$count_{i,1,1}$	$count_{i,4,1}$
4			0	2	0	2	1	0	0	2
5										

Arrays Table: Main

	$count_{i,3,1}$	$count_{i,2,1}$	$d_{i,1}$	$d_{i,2}$	$d_{i,3}$	$d_{i,4}$	$dp_{i,1,1}$ [kPa]	$dp_{i,2,1}$ [kPa]	$dp_{i,3,1}$ [kPa]
1	2	2	1	1	1	1	1.027	1.016	1.065
2	2	2	1	1	1	1	2.178	1.735	1.446
3	2	2	1	1	1	1	2.627	2.812	2.96
4	2	2	1	1	1	1	3.823	3.547	4.726
5									

Arrays Table: Main

	$dp_{i,4,1}$ [kPa]	$dpb_{i,1}$	$dpb_{i,2}$	$dpb_{i,3}$	$dpb_{i,4}$	$\eta_{o,i,4,1}$	$\eta_{o,i,3,1}$	$\eta_{o,i,1,1}$	$\eta_{o,i,2,1}$
1	1.23	0.04782	0.0478	0.04708	0.0432	0.1053	0.1053	0.1053	0.1054
2	1.374	0.06742	0.05985	0.0513	0.04823	0.1053	0.1053	0.1054	0.1053
3	3.566	0.087	0.09646	0.1068	0.1179	0.1053	0.1052	0.1053	0.1051
4	4.257	0.04063	0.04015	0.06539	0.0817	0.1053	0.1053	0.09035	0.1042
5									

Arrays Table: Main

	$f_{i,1,1}$	$f_{i,2,1}$	$f_{i,3,1}$	$f_{i,4,1}$	$h_{i,4,1}$ [kJ/kg]	$h_{i,1,1}$ [kJ/kg]	$h_{i,2,1}$ [kJ/kg]	$h_{i,3,1}$ [kJ/kg]	$h_{a,i,1,1}$	$h_{a,i,2,1}$
1	0.02096	0.02088	0.02081	0.02072	-249.2	-262.6	-257.8	-254.1	26.85	25.56
2	0.02007	0.02021	0.02047	0.02067	-242.8	-215.7	-228.7	-236.4	32.42	28.21
3	0.01967	0.01926	0.01889	0.01845	-147.1	-196.9	-178.5	-163.9	34.77	31.87
4	0.01617	0.01615	0.01685	0.01777	-123.4	-64.36	-75.07	-96.09	41.3	39.85
5										

Arrays Table: Main

	$h_{a,i,3,1}$	$h_{a,i,4,1}$	$h_{ae,i,5,1}$	$h_{ae,i,4,1}$	$h_{ae,i,3,1}$	$h_{ae,i,2,1}$	$h_{ae,i,1,1}$	$h_{ae,i,0,1}$	$h_{ai,i,1,1}$
1	25.52	27.1		25.99	24.83	24.86	25.79		27.91
2	27.16	27.6	30.06	26.35	26.05	26.48	29.34		35.5
3	30.07	33.81		30.06	27.62	28.93	30.94	40.06	38.6
4	37.51	37.56	42.55	32.56	32.47	37.15	40.06		42.55
5									

Arrays Table: Main

	$h_{ai,i,2,1}$	$h_{ai,i,3,1}$	$h_{ai,i,4,1}$	$h_{air,i}$	$h_{ca,i,1,1}$	$h_{ca,i,2,1}$	$h_{ca,i,3,1}$	$h_{ca,i,4,1}$	$h_{cr,i,1,1}$
1	26.26	26.2	28.21	25.59	44.61	44.58	44.77	44.8	2649
2	29.93	28.28	28.84	27.4	44.5	44.86	44.79	44.76	6293
3	34.81	32.52	37.56	30.16	45.27	45.24	44.99	44.61	7012
4	42.55	42.55	42.55	36.34	88.98	47.98	44.61	44.61	1560
5				42.55					

Arrays Table: Main

	$h_{cr,i,2,1}$	$h_{cr,i,3,1}$	$h_{cr,i,4,1}$	$h_{e,i,1,0}$ [kJ/kg]	$h_{e,i,2,2}$ [kJ/kg]	$h_{e,i,3,0}$ [kJ/kg]	$h_{e,i,3,1}$ [kJ/kg]	$h_{e,i,1,2}$ [kJ/kg]	$h_{e,i,2,0}$ [kJ/kg]	$h_{e,i,2,1}$ [kJ/kg]
1	2136	2105	2524	-265.5	-259.7	-255.9	-252.2			-255.9
2	5453	2405	2584				-233.4	-224	-233.4	-224

Arrays Table: Main

	$h_{cr,i,1,1}$	$h_{cr,i,3,1}$	$h_{cr,i,4,1}$	$h_{e,i,1,0}$ [kJ/kg]	$h_{e,i,2,2}$ [kJ/kg]	$h_{e,i,3,0}$ [kJ/kg]	$h_{e,i,3,1}$ [kJ/kg]	$h_{e,i,1,2}$ [kJ/kg]	$h_{e,i,2,0}$ [kJ/kg]	$h_{e,i,2,1}$ [kJ/kg]
3	7326	7685	8000	-207.3	-186.5	-170.5	-157.3			-170.5
4	1712	9407	8952				-82.4	-67.73	-82.4	-67.73
5										

Arrays Table: Main

	$h_{e,i,1,1}$ [kJ/kg]	$h_{e,i,4,0}$ [kJ/kg]	$h_{e,i,3,2}$ [kJ/kg]	$h_{e,i,4,1}$ [kJ/kg]	$h_{e,i,4,2}$ [kJ/kg]	$h_{i,i,4,2}$ [kJ/kg]	$h_{i,i,4,1}$ [kJ/kg]	$h_{i,i,4,0}$ [kJ/kg]	$h_{i,i,3,2}$ [kJ/kg]	$h_{i,i,3,1}$ [kJ/kg]
1	-259.7			-246.2	-252.2		-252.2	-246.2	-252.2	-255.9
2	-207.3	-246.2	-239.5	-239.5		-239.5	-246.2			-239.5
3	-186.5			-136.9	-157.3		-157.3	-136.9	-157.3	-170.5
4	-60.98	-136.9	-109.8	-109.8		-109.8	-136.9			-109.8
5										

Arrays Table: Main

	$h_{i,i,3,0}$ [kJ/kg]	$h_{i,i,2,2}$ [kJ/kg]	$h_{i,i,2,1}$ [kJ/kg]	$h_{i,i,2,0}$ [kJ/kg]	$h_{i,i,1,2}$ [kJ/kg]	$h_{i,i,1,1}$ [kJ/kg]	$h_{i,i,1,0}$ [kJ/kg]	$P_{i,3,1}$ [kPa]	$P_{i,2,1}$ [kPa]	$P_{i,1,1}$ [kPa]
1			-259.7	-255.9	-259.7	-265.5		4008	4009	4010
2	-233.4	-224	-233.4			-224	-207.3	4004	4002	4000
3			-186.5	-170.5	-186.5	-207.3		3992	3995	3997
4	-82.4	-67.73	-82.4			-67.73	-60.98	3980	3975	3972
5										

Arrays Table: Main

	$P_{i,4,1}$ [kPa]	$P_{e,i,4,1}$ [kPa]	$P_{e,i,4,2}$ [kPa]	$P_{e,i,1,0}$ [kPa]	$P_{e,i,2,1}$ [kPa]	$P_{e,i,2,2}$ [kPa]	$P_{e,i,3,0}$ [kPa]	$P_{e,i,1,1}$ [kPa]	$P_{e,i,1,2}$ [kPa]	$P_{e,i,2,0}$ [kPa]
1	4006	4006	4007	4010	4008	4009	4008	4009		
2	4005	4004			4001			3999	4001	4003
3	3988	3986	3990	3999	3993	3996	3993	3996		
4	3984	3982			3974			3970	3974	3977
5										

Arrays Table: Main

	$P_{e,i,3,1}$ [kPa]	$P_{e,i,3,2}$ [kPa]	$P_{e,i,4,0}$ [kPa]	$P_{i,i,1,0}$ [kPa]	$P_{i,i,1,1}$ [kPa]	$P_{i,i,4,2}$ [kPa]	$P_{i,i,3,0}$ [kPa]	$P_{i,i,3,1}$ [kPa]	$P_{i,i,3,2}$ [kPa]	$P_{i,i,4,0}$ [kPa]
1	4007				4010			4008	4007	4006
2	4003	4004	4006	3999	4001	4004	4003	4004		
3	3990				3999			3993	3990	3986
4	3977	3982	3986	3970	3974	3982	3977	3982		
5										

Arrays Table: Main

	$P_{i,i,2,0}$ [kPa]	$P_{i,i,2,1}$ [kPa]	$P_{i,i,4,1}$ [kPa]	$P_{i,i,1,2}$ [kPa]	$P_{i,i,2,2}$ [kPa]	$\dot{q}_{i,1,1}$ [kJ/kg]	$\dot{q}_{i,2,1}$ [kJ/kg]	$\dot{q}_{i,3,1}$ [kJ/kg]	$\dot{q}_{i,4,1}$ [kJ/kg]
1	4008	4009	4007	4009		0.1878	0.1242	0.1208	0.1963
2		4003	4006		4001	0.5454	0.306	0.1973	0.2201
3	3993	3996	3990	3996		0.6795	0.5208	0.4335	0.664
4		3977	3986		3974	0.2205	0.4788	0.8938	0.8856
5									

Arrays Table: Main

	$q_{flux,i,1,1}$	$q_{flux,i,4,1}$	$q_{flux,i,2,1}$	$q_{flux,i,3,1}$	$q_{test,i,1,1}$	$q_{test,i,3,1}$	$q_{test,i,4,1}$	$q_{test,i,2,1}$
--	------------------	------------------	------------------	------------------	------------------	------------------	------------------	------------------

File: Finned tube program - Run 4 working.EES

2013/04/26 11:19:52 AM Page 16

EES Ver. 8.874: #2063: Ten-user license for use only by M-Tech Industrial (Pty) Ltd., Potchefstroom, South Africa

	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]				
1	4.189	4.379	2.769	2.694	0.1878	0.1208	0.1963	0.1242
2	12.16	4.909	6.824	4.401	0.5454	0.1973	0.2201	0.306
3	15.15	14.81	11.62	9.669	0.6794	0.4334	0.664	0.5207
4	4.918	19.75	10.68	19.93	0.2205	0.8938	0.8856	0.4783
5								

Arrays Table: Main

	$Re_{i,1,1}$	$Re_{i,3,1}$	$Re_{i,4,1}$	$Re_{i,2,1}$	$\rho_{i,3,1}$ [kg/m ³]	$\rho_{i,1,1}$ [kg/m ³]	$\rho_{i,2,1}$ [kg/m ³]	$\rho_{i,4,1}$ [kg/m ³]
1	58667	60969	62328	59897	400.6	455.6	423.1	374.8
2	73754	66341	63086	70984	320.5	259.4	294.7	345.6
3	82514	103692	119302	92848	175.8	221.3	193.4	159
4	299908	217741	151442	302501	123.4	103.5	111.4	140.2
5								

Arrays Table: Main

	$Pe_{i,4,1}$ [kg/m ³]	$Pe_{i,3,1}$ [kg/m ³]	$Pe_{i,1,1}$ [kg/m ³]	$Pe_{i,2,1}$ [kg/m ³]	$Pi_{i,4,1}$ [kg/m ³]	$Pi_{i,3,1}$ [kg/m ³]	$Pi_{i,2,1}$ [kg/m ³]	$Pi_{i,1,1}$ [kg/m ³]	RN_i
1	360.4	390.3	435.5	411.3	390.3	411.3	435.5	477.7	0
2	331.9	309.9	241	281	360.4	331.9	309.8	281	1
3	150.4	168.7	204.6	183.4	168.7	183.4	204.6	241	0
4	131.3	116.4	101.2	105.9	150.4	131.3	116.3	105.9	1
5									

Arrays Table: Main

	$T_{i,4,1}$ [C]	$T_{i,1,1}$ [C]	$T_{i,2,1}$ [C]	$T_{i,3,1}$ [C]	$TN_{i,1}$	$TN_{i,2}$	$TN_{i,3}$	$TN_{i,4}$	$T_{a,i,1,1}$	$T_{a,i,2,1}$
1	5.387	5.4	5.385	5.398	1	0	1	0	10.81	9.217
2	5.374	5.305	5.362	5.364	0	1	0	1	18.39	12.83
3	5.194	5.382	5.407	5.335	1	0	1	0	21.2	17.49
4	5.173	12.99	7.133	5.136	0	1	0	1	18.78	22.53
5										

Arrays Table: Main

	$T_{a,i,3,1}$	$T_{a,i,4,1}$	$T_{a,e,i,1,1}$	$T_{a,e,i,2,1}$	$T_{a,e,i,3,1}$	$T_{a,e,i,4,1}$	$T_{a,i,i,4,1}$	$T_{a,i,i,3,1}$	$T_{a,i,i,2,1}$
1	9.135	11.11	7.756	7.147	7.176	8.044	9.673	8.169	8.135
2	11.21	11.75	10.21	8.244	8.027	8.312	10.13	9.62	10.64
3	15.37	20.52	11.27	10.01	9.231	11.03	16.42	12.8	14.2
4	25.49	25.45	17.56	15.62	12.78	12.83	20	20	20
5									

Arrays Table: Main

	$T_{a,i,i,1,1}$	$T_{a,i,i}$ [C]	$T_{a,d,i,4,1}$	$T_{a,d,i,3,1}$	$T_{a,d,i,1,1}$	$T_{a,d,i,2,1}$	$T_{dew,i,1,1}$	$T_{dew,i,2,1}$	$T_{dew,i,3,1}$
1	9.229	7.752	8.585	7.5	8.188	7.447	6.687	6.27	6.193
2	14.42	9.037	8.913	8.527	11.41	8.948	8.357	7.13	6.66
3	16.59	11.14	12.75	10.4	12.84	11.32	8.958	8.033	7.41
4	20	15.46	15.12	15.07	18.78	17.36	9.264	9.264	9.264
5		20							

Arrays Table: Main

	$T_{dew,i,4,1}$	$T_{e,i,4,1}$ [C]	$T_{e,i,3,1}$ [C]	$T_{e,i,2,1}$ [C]	$T_{e,i,1,1}$ [C]	$T_{eq1,i,1,1}$	$T_{eq1,i,2,1}$	$T_{eq1,i,3,1}$	$T_{eq1,i,4,1}$
1									
2									
3									
4									
5									

1	6.567	5.356	5.394	5.402	5.4	10.82	9.216	9.14	11.12
2	6.694	5.342	5.361	5.314	5.376	18.37	12.85	11.22	11.76
3	8.371	5.19	5.273	5.383	5.27	21.33	17.58	15.42	20.52
4	9.264	5.129	5.119	11.05	15.02	20	23.02	25.49	25.45
5									

Arrays Table: Main

	$T_{i,i,1,1}$ [C]	$T_{i,i,2,1}$ [C]	$T_{i,i,3,1}$ [C]	$T_{i,i,4,1}$ [C]	$T_{s,i,1,1}$	$T_{s,i,2,1}$	$T_{s,i,4,1}$	$T_{s,i,3,1}$	$UA_{i,2,1}$	$UA_{i,4,1}$
1	5.4	5.39	5.379	5.368	5.402	5.387	5.389	5.4	32.41	34.29
2	5.31	5.327	5.342	5.356	5.307	5.363	5.376	5.366	40.99	34.52
3	5.288	5.261	5.233	5.203	5.384	5.409	5.196	5.336	43.1	43.32
4	11.05	5.077	5.124	5.166	13	7.139	5.175	5.138	31.07	43.68
5										

Arrays Table: Main

	$UA_{i,1,1}$	$UA_{i,3,1}$	$V_{i,1,1}$	$V_{i,2,1}$	$V_{i,3,1}$	$V_{i,4,1}$	$w_{ae,i,2,1}$	$w_{ae,i,0,1}$	$w_{ae,i,1,1}$
1	34.69	32.33	1.141	1.229	1.298	1.388	0.007027		0.007149
2	41.68	33.77	2.005	1.765	1.623	1.505	0.007226		0.007566
3	42.94	43.18	2.35	2.689	2.959	3.27	0.007484	0.008834	0.007769
4	38.11	43.91	5.026	4.669	4.215	3.71	0.008469		0.008834
5									

Arrays Table: Main

	$w_{ae,i,3,1}$	$w_{ae,i,4,1}$	$w_{ae,i,5,1}$	$w_{ai,i,2,1}$	$w_{ai,i,1,1}$	$w_{ai,i,3,1}$	$w_{ai,i,4,1}$	$w_{air,i}$
1	0.007005	0.007113		0.007184	0.007396	0.007146	0.007335	0.007074
2	0.007142	0.007149	0.00752	0.007627	0.008302	0.007382	0.0074	0.007271
3	0.00728	0.00752		0.008119	0.008652	0.007777	0.00831	0.007513
4	0.007769	0.007785	0.008834	0.008834	0.008834	0.008834	0.008834	0.008214
5								0.008834

Arrays Table: Main

	$w_{wet,i,2,1}$	$w_{wet,i,1,1}$	$w_{wet,i,3,1}$	$w_{wet,i,4,1}$	$x_{i,4,1}$	$x_{i,1,1}$	$x_{i,2,1}$	$x_{i,3,1}$	$x_{e,i,1,1}$
1	0.007027	0.007149	0.007005	0.007112	0.2065	0.1435	0.166	0.1836	0.157
2	0.007224	0.007568	0.007141	0.007148	0.2364	0.3637	0.3026	0.2665	0.4028
3	0.007475	0.007756	0.007275	0.00752	0.6844	0.4515	0.5375	0.6059	0.5002
4	0.008433	0.008834	0.007769	0.007783	0.7951	100	100	0.9221	100
5									

Arrays Table: Main

	$x_{e,i,2,1}$	$x_{e,i,3,1}$	$x_{e,i,4,1}$	$x_{i,i,1,1}$	$x_{i,i,2,1}$	$x_{i,i,3,1}$	$x_{i,i,4,1}$
1	0.1749	0.1924	0.2206	0.13	0.157	0.1749	0.1924
2	0.3246	0.2806	0.2522	0.3246	0.2806	0.2523	0.2206
3	0.5749	0.6369	0.7319	0.4028	0.5002	0.5748	0.6369
4	100	0.9859	0.8583	100	0.9858	0.8583	0.7319
5							