

COMPARISON OF REFRIGERANTS R410A AND R404A FOR USE IN LOW
TEMPERATURE APPLICATIONS:
A COMPUTER MODEL STUDY

BY

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THESIS

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Abstract

The motivation for this thesis is the need for efficient and environmentally friendly refrigerants in low temperature applications. This study provides a perspective for comparison of refrigerant R410a with R404a. As R410a is now widely used commercially, further knowledge is desired on how different an R410a system is from established refrigerant systems and any possibilities for retrofitting. This thesis uses a computer model simulation to specifically compare the performance of R410a and R404a in a supermarket freezer display case system designed for R404a with a standard capacity of 3.42 kW. The computer model is assembled from existing algorithms and correlations for heat transfer, pressure drop, and thermodynamics and run using Engineering Equation Solver (EES). The results are then compared with separate physical experimental results for this exact in-house laboratory refrigeration system. Complimenting the experimental results, the EES model simulates the refrigeration cycle for each refrigerant at four different settings of ambient air temperature into the condenser. The EES model results are compared in graphs and tables to the laboratory results. The model results also show that in this Hussman R404a display case system, R410a still operates more efficiently with a COP generally 0.16-0.19 greater than that of R404a. As expected, R410a operates at a higher compressor discharge pressure and temperature than R404a. After the EES model is verified by comparison to the experimental study, the model is used to predict refrigeration cycle behavior in the case of the heat exchanger component geometry being slightly altered.

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Table of Contents

List of Figures	v
List of Tables	vii
1 Introduction and Literature Review	1
1.1 Introduction.....	1
1.2 Literature Review.....	3
2 Thermodynamic Background.....	9
2.1 Thermodynamic Fundamentals.....	9
2.2 Problem Definition.....	15
3 Methodology	16
3.1 Organization of the EES Code	16
3.2 Heat Exchanger Geometries	25
3.2.1 Designating the Parameters of a Heat Exchanger.....	25
3.2.2 Applying the Geometry Conventions to the System Heat Exchangers.....	30
3.2.3 Tabulation of Heat Exchanger Geometry Parameters	34
3.3 Correlation Modeling Choices	38
4 Data Analysis Methods	40
5 Results and Discussion of Findings	41
5.1 Validation of the EES Model: Comparison of Model and Experimental Data	41
5.2 EES Model Analysis: R404A vs. R410A	52
6 Conclusion	61
Appendix A.....	62
Appendix B: the EES Code.....	70
References.....	91

List of Figures

Figure 1 Basic refrigeration cycle equipment	9
Figure 2 Log P vs. h phase diagram for a generic refrigerant.....	11
Figure 3 P-h diagram with thermodynamic cycle data	11
Figure 4 COP ratio of R410a/propane and R404a/propane	14
Figure 5 Basic contents of the system model code	18
Figure 6 Basic contents of the condenser procedures	19
Figure 7 System model flow diagram for the condenser	21
Figure 8 System model flow diagram for the evaporator	22
Figure 9 Variable flow in procedures within the procedure DP_AIR_SIDE_COND	24
Figure 10 Division of input to procedures within procedure DP_AIR_SIDE_COND.....	24
Figure 11 Width, depth, and height	25
Figure 12 Example for deciding N_row_max and N_row_avg.....	26
Figure 13 Sub heat exchangers	27
Figure 14 Modules stack in parallel.....	27
Figure 15 Tube passes visible from the front face	28
Figure 16 Distinction between modules and circuits per module.....	29
Figure 17 Elements extend through heat exchanger depth	29
Figure 18 Element depth.....	30
Figure 19 Longitudinal and transverse tube spacing	30
Figure 20 EES model representation of the evaporator geometry	31
Figure 21 Trial evaporator geometry	32
Figure 22 EES model representation of the evaporator geometry	33
Figure 23 Internal heat exchangers in ACRC Simulation Models User Guide	37
Figure 24 P-h diagram for R404A at 20 C, model and experimental comparison	41
Figure 25 P-h diagram for R404A at 35 C, model and experimental comparison	42
Figure 26 P-h diagram for R410A at 20 C, model and experimental comparison	44
Figure 27 P-h diagram for R410A at 35 C, model and experimental comparison	44
Figure 28 Compressor discharge pressure	46
Figure 29 Compressor discharge temperature	47

Figure 30	Compression ratio of R410a vs. R404a in EES.....	47
Figure 31	EES and experimental COP comparison.....	48
Figure 32	Model Qevap vs. experimental refrigerant side Qevap.....	49
Figure 33	Model Qevap vs. experimental air side Qevap.....	49
Figure 34	EES model COP and Qevap for both refrigerants.....	50
Figure 35	R404a Cycle Performance Quantities	51
Figure 36	R410a Cycle Performance Quantities	51
Figure 37	EES R404a cycle P-h performance	52
Figure 38	EES R410a cycle P-h performance	53
Figure 39	EES R404a cycle T-h performance	53
Figure 40	EES R410a cycle T-h performance	54
Figure 41	T-h diagram for low Taic	54
Figure 42	T-h diagram for high Taic	55
Figure 43	Mass flux in Evaporator and Condenser	55
Figure 44	LMTD for condenser and evaporator	57
Figure 45	Evaporator U calculation.....	57
Figure 46	Effect of evaporator tube thickness on Q_{evap}	58
Figure 47	Pressure drop limits for evaporator tube diameters.....	59
Figure 48	Effect of evaporator tube thickness on COP	59
Figure 49	P-h diagram for R404a at 25 C, model and experimental comparison	62
Figure 50	P-h diagram for R404a at 30 C, model and experimental comparison	62
Figure 51	P-h diagram for R410a at 25 C, model and experimental comparison	63
Figure 52	P-h diagram for R410a at 30 C, model and experimental comparison	63
Figure 53	Simple condenser schematic.....	68
Figure 54	Simple evaporator schematic.....	68
Figure 55	Evaporator engineering drawing	69

List of Tables

Table 1	Refrigerant properties	13
Table 2	Condenser input	35
Table 3	Evaporator input	36
Table 4	Internal heat exchanger area calculations	37
Table 5	Measured internal heat exchanger dimensions	37
Table 6	Condenser geometry and range allowed by <i>h_plain_Wang_2000</i>	38
Table 7	Evaporator geometry and range allowed by <i>h_plain_Wang_2000</i>	39
Table 8	Verifying EES model for R404a	43
Table 9	Verifying EES model for R410a	45
Table 10	EES mass flux	56
Table 11	R404a pressure enthalpy temperature data	64
Table 12	R410a pressure enthalpy temperature data	65
Table 13	R404a heat exchanger performance quantities	66
Table 14	R410a heat exchanger performance quantities	67

1. INTRODUCTION AND LITERATURE REVIEW

1.1 Introduction

The first known instance of refrigeration was demonstrated by William Cullen at Glasgow University in 1748. Following that, in the early 19th century Michael Faraday compressed ammonia vapor into a liquid, and then followed the invention of the first refrigerator. Since the rise of climate control technologies in the 20th century, only relatively recently has the issue of environmental impact become a major concern. The Clean Air Act Amendments were passed by the U.S. Congress in 1990 following the Montreal Protocol in 1987. The main impetus was of course the ozone depleting potential of the chlorofluorocarbons (CFCs) and hydro-chlorofluorocarbons (HCFCs) commonly used in refrigeration. The HCFCs, including staple R-22, were scheduled for phase-out in 2010, even while bridging the development of refrigerants from early CFCs to modern hydrofluorocarbons (HFCs). The HFCs are accepted because they contain basically zero ozone depletion potential (ODP). They also have an assigned global warming potential (GWP) factor which is drastically lower than that of HCFCs.

Thus the requirement for environmentally friendly, working refrigerants necessitated the invention of refrigerant R-410A by 1991 when it was introduced by Honeywell (then AlliedSignal). Since then R-410A performance has been researched in various applications. Another HFC refrigerant comparable to R-410A is R-404A. Refrigerant R-404A was developed to replace CFC R-502 and HCFC R-22. Refrigerant R-404A is the baseline refrigerant in this study. A computer modeled performance analysis and comparison of these two refrigerants, R-410A and R-404A, will show the advantages of one over the other in a low-temperature application.

Refrigerant R-404A is already known as a suitable replacement for R-22 in low temperature applications. Compressors are available for R-404A in applications of -45 °C through 10 °C evaporation temperature. [1] Honeywell has also found R-410A to be more efficient than R-22 based on university studies which generally give R-410A a 30% higher evaporation heat transfer coefficient and 40% decrease in pressure drop for evaporation and condensation. [2] But the operating pressure for R-410A is generally 60% higher than that of R-22, so R-410A is not a direct replacement for an R-22 system. [2] The analysis of this study will compare R-410A as a direct replacement to R-404A in a supermarket freezer display case system originally designed for R-404A.

The two refrigerants are similar enough with R-410A an azeotropic blend of HFC-32 with HFC-125, and R-404A a near-azeotropic blend of HFC-125, HFC-143a, and HFC-134a. An azeotropic refrigerant acts as a uniform substance with no temperature glide for a constant pressure evaporation or condensation process. For a zeotropic refrigerant, a temperature glide would occur if a component of the refrigerant mixture evaporates or condenses before the other components. Technically R-404A exhibits zeotropic behavior, but the temperature glide is negligible so it is considered a near-azeotrope and comparable to R-410A. Hence R-410A is being considered as a suitable direct replacement for an R-404A system.

The scope of this report is the comparison of R-404A and R-410A low-temperature performance using a computer model for thermodynamic system and heat transfer analysis. The model will use the finite element heat exchanger system model algorithms developed by Dr. Clark Bullard of the University of Illinois ACRC. The model will specifically analyze an approximately 3-ton freezer display case application. This heat transfer model study is complemented by the study of Radko Brock [3] with the in-house laboratory set-up featuring Hussman display cases. Since Brock's experimental work is contained in a separate thesis, this report will not document his calculations or methodology; but it will compare results and data findings to Brock's study. Chapter 2 of this report contains the theoretical and thermodynamic basis for the comparison of R-404A and R-410A in a low temperature application. Chapter 3 details the methodology of the computer simulation using Engineering Equation Solver (EES) software. Chapter 4 presents equations for data analysis, and Chapter 5 presents the results of the performance simulation and compares findings to Brock's experimental study for conclusions about the validity of the model. Additional details about the components of the in-house setup and corresponding model input are in the Appendix.

Known data show that R-410A will be more efficient than R-404A for a range of temperatures spanning low temperature to medium temperature. R-410A in general yields a higher COP and requires lower mass flow rates based on heat transfer characteristics, but it has a significantly higher operating pressure. [3] This project will provide another resource for data analysis between these two refrigerants.

1.2 Literature Review

Kim and Shin [4] carried out a study of R22 and R410A in smooth tube and microfin tube setups. They produced heat transfer coefficients which are consistent with a correlation by Kandlikar. The average evaporation heat transfer coefficients were higher for microfin tubes than smooth tubes. Compared to R22, the heat transfer coefficients for R410A had less variation in value along the circumference of the smooth tubes, and the coefficients were generally higher for R410A at the same test conditions.

Stegou-Sagia et al [5] developed a useful theoretical model of enthalpy and entropy properties of R404a and R410a. To do this they started with the Martin-Hou equation of state and the equation of ideal gas heat capacity at constant pressure, took relevant partial derivatives of pressure, and derived mathematical expressions for exergy losses and COP for heat pump cycles. Their model produced useful diagrams such as the isentropic exponents for differing states of superheated vapor temperature and superheated vapor pressure. Also refrigerant compressibility factor and speed of sound were plotted vs. superheated vapor pressure and superheated vapor temperature. The study also compared the model-calculated R404A and R410A data to accepted literature values by graphing entropy and enthalpy vs. temperature in saturated and superheated regions. They also compared exergy efficiency and COP of R22, R404A, and R410A at constant evaporating temperature (varying condensing temperature).

Hwang et al [6] carried out a study with some similarities to this one, comparing R-290 (propane) with R404A and R410A. They set out to make comparisons in medium temperature and low temperature conditions with small charge. The medium temperature facility was operated at a capacity of approximately 10.7 kW while the low temperature facility was at approximately 3.7 kW. First they used programs “Coil Designer” and “VapCyc” to select optimum evaporator and condenser coils for each facility. Then they obtained the relevant values such as COP and average condenser and evaporator pressures, and finally carried out a life cycle climate performance (LCCP) analysis in terms of power cost and environmental impact of several refrigerant leakage rates.

Stegou-Sagia and Paignigiannis [7] also conducted a study evaluating the efficiency of various mixtures in refrigerating systems focusing on exergy. They studied 10 refrigerants, including R404A and R410A. The study used equations of state, thermodynamic fundamentals and computer simulation. It noted affects of pressure drops in various parts of the cycle,

isentropic compressor efficiency, subcooling, and superheating. They also produced Grassman diagrams which give a pictorial representation of the exergy efficiency.

Geller et al. [8] conducted a study in which they present thermal conductivity data for refrigerants R404A, R410A, R407C, and R507A. The practical nature of this study is obvious as better knowledge of the conductive behavior will aid in refrigeration equipment design. The basis of the test apparatus was a specially constructed measuring cell with a platinum hot-wire made to function as the electric heater and inner resistance thermometer. This wire was inside a capillary tube which was the outer resistance thermometer. The potential difference across the two resistance thermometer was measured and used in the calculation of the thermal conductivity. As motivation for this study Geller et al. stated the need for more available thermal conductivity data in various regions. This study was carried out in vapor and superheated vapor regions in a temperature range from 250K to 400K at atmospheric pressure. Their new data helps fill in some of the missing parts in other available data, which the article references. They mention three studies containing data for thermal conductivity of liquid R410A, and could only find one containing some data for the vapor thermal conductivity of R410A.

Greco and Vanoli carried out an experimental study of the heat transfer coefficient and other parameters for refrigerants in the boiling phase in a smooth horizontal tube. [9] This study compares the refrigerants R22, R134a, R507, R404A, and R410A. The test section of their experimental setup focuses on a stainless steel tube 6 meters in length with an inner diameter of 6 millimeters. The refrigerant evaporates due to an electrical heating device in the test section and continues through the refrigeration cycle which includes a condenser that is part of a secondary loop. The local heat transfer coefficient is calculated from basic heat transfer principles, and a basic uncertainty analysis is applied to the measurements. Mass flux is kept at about 360 kW/m^2 while the pressure is varied from 3-12 bar. As the pressure varies, heat flux also changes through a range of 11-21 kW/m^2 . The results show graphs of heat transfer coefficient versus quality, with multiple trendlines on each graph accounting for the different operating pressures. From each trendline for the different evaporating pressure, the average heat transfer coefficient is calculated and a combination graph is made comparing the average heat transfer coefficient vs. pressure/heat flux for all five refrigerants together. The graph shows average heat transfer coefficient versus evaporator pressure for each refrigerant, in order from highest to lowest, of R134a, R22, R410A, R404A, and R507. This is to be expected and shows

that of the near-azeotropic mixtures R410A and R404A have higher heat transfer coefficients than R507.

The horizontal tube setup of Greco and Vanoli has also been used for other studies, such as the evaporation pressure drop of the previously mentioned refrigerants and one more, R407C. [10] This study maintains an evaporating pressure of 7 bar while observing pressure drop results for varying mass flow rate, taking over 1,000 data points. The results show significantly higher evaporation pressure drop for R410A compared to R404A, as expected. The article references use of a flow map in observing the flow type and explains some of the physical factors behind the graphs of pressure drop versus vapor quality. The pressure drop results are compared in graphs with nine well known pure fluid and refrigerant mixture correlations, such as the Lockhart and Martinelli correlation, Friedel correlation and Baroczy–Chisholm correlation.

Greco and Vanoli have performed further analysis on their steel tube setup with HFC flow boiling, in a two-part writeup. Part I: Experimental Investigations [11] focuses on more experimental tests with R404A and R410A. The first part of these results shows the measurement of heat transfer coefficient at different mass flux settings and multiple evaporating pressure settings per mass flux. The heat transfer coefficient is plotted vs. vapor quality. The graphs distinctly show that the coefficients rise with quality at lower pressures, while at higher pressure they form a local minimum by decreasing and then increasing as quality varies from 0 to 1. This study also looked at the circumferential variation of heat transfer coefficients previously mentioned in Kim and Shin's study [4] Graphs show the variation of R410A heat transfer coefficient with vapor quality along the top, sides, and bottom of the horizontal tube. Greco and Vanoli explain the results in terms of the nucleate boiling contribution and convective heat transfer contribution and their influence varying with the pressure, heat flux, and vapor quality.

Part II: Assessment of Predictive Methods [12] compares the experimental results of the heat transfer coefficients for R404A and R410A with ten different known evaporation heat transfer correlations. First a literature survey documents the developments of these correlations. The best results are clearly obtained when comparing the experimental coefficients with a modified version of Kandlikar's correlation. This modified correlation introduces a pressure correction factor and results in 80% of the R410A and R404A boiling heat transfer coefficients falling in an acceptable range in which they only deviate from the correlation by a maximum of

+23% and a minimum of -16%. Other correlations do not yield such favorable results, such as the Shah correlation where 80% of the coefficients can vary by as much as +51% and -41% from the experimental values.

In yet another study [13] Greco presents graphs which show heat transfer coefficient varying with quality highlighting the effect of varying mass flux. There is also a graph that shows the heat transfer coefficient is dependent on mass flux in terms of an exponent n . Graphs show heat transfer coefficients increasing with quality for R410A, R404A, R407C as well as other refrigerants, and also shows behavior consistent with the previously mentioned study. [12]

More experimental research has been done by Samuel Sami and other professors at the University of Moncton in Canada. Sami and Comeau have studied the affects of liquid injection into refrigerant mixtures flowing in the boiling phase [14] and in the condensation phase [15] for refrigerants R507, R404A, R410A, and R407C. Their lab setup uses a fin-tube evaporator and condenser, both with microfins, and a 3 kW compressor. Basic heat transfer equations were used in the analysis such as heat balance and UA, LMTD relations. The results showed graphs of heat transfer coefficient increasing with Reynolds number, or made comparison with correlations. Depending on the refrigerant, the heat transfer coefficient trend reacted more or less strongly as the percentage of liquid injection varied from 0 to 15%.

In another study, Sami and Comeau [16] studied the relationship between condensation heat transfer coefficients of the newer mixed refrigerants and their transport properties, namely the viscosity and thermal conductivity. They used the same micro-finned cycle setup mentioned above, in heating mode. They used correlations relating Nusselt number and Reynolds number to the transport properties of density and viscosity, and other correlations relating the thermal conductivity to the saturated liquid or saturated vapor temperature. Sami and Kita [17] examined the affects of magnetic field relating to the thermophysical properties of the four refrigerants, including R410A and R404A. The refrigerant loop was fitted strategically with three magnets each with a gauss level of 4000, placed in between the condenser outlet and thermal expansion unit. In this study, R404A actually had a greater evaporator COP and condenser COP than R410A, and the COP of R404A was much more affected by the presence of magnets. The condenser capacity of R410A was greater than R404A but not the evaporator capacity. Measurements of transport properties yielded normal results, however, with R410A having greater specific heat and thermal conductivity but lower viscosity than R404A.

Santos et al. [18] have analyzed a body of relative permittivity data for a number of HCFC's and HFC's. The data were originally gathered in experiments on a cell with work done by two other faculty teams. The work centers mostly on HCFC mixtures but presents a table with experimental and calculated values of the dipole moments of R507, R404A, R410A, and R407A.

Küçüksille et al [19] conducted a study of the effectiveness of ten modeling techniques in predicting refrigerant transport properties. These are mathematical techniques such as regression algorithms and decision trees, which are used to mine and sift through data to establish relationships. The ten techniques were first described, and then their effectiveness compared in predicting the actual heat capacity, viscosity, density, etc. data of R404A, R410A, R134a, and R407C.

Klein et al [20] provide an in-depth study of the impact of liquid suction heat exchangers like the one used in the current study. From analyzing previous results on a variety of refrigerants, they conclude that internal heat exchangers can generally be helpful to the performance of R410A or R404A systems, but have a negative impact on R22 systems. Other studies have been conducted as well which provide information resources on the behavior of the two refrigerants in this study.

Sun and Mishima [21] have studied the two-phase pressure drop of several fluids, including R404A and R410A, using an experimental setup with mini-channel flow and comparing results with known correlations.

Equations of state for refrigerants R-410A, R-404A, R-507A, R-407C have been developed by Lemmon [22] by treating them as “pseudo-pure” fluids and using previous work done on models for the components of R-410A and R-404A (R-32, R-125, R-134a, R-143a). The equations of state yield thermodynamic properties, but only in single phase. One of the objectives was to provide a template for faster calculations.

Nabizadeh and Mayinger [23] have generated data for the viscosity of R-410A, R-404A, R-507A, R-407C in the vapor phase by using an oscillating disk viscometer.

Phase-change behavior near the critical regions is not well understood or documented, so Jiang et al. [24] conducted an experiment on the condensation of R-410A and R-404A in a smooth horizontal tube at high or near-critical pressure. Refrigerant R-404A was observed at

pressures of approximately 2980 kPa and 3350 kPa, while R-410A was at approximately 3920 and 4410 kPa settings.

A capillary tube can be a common expansion method in small scale refrigeration systems, so Yang and Wang [25] developed a model or correlation for predicting the mass flow rate in capillary tubes. This model can extend to several refrigerants including R-410A and R-404A. While research is certainly ongoing, there is already a significant database of knowledge that has been generated for these two refrigerants.

2. Thermodynamic Background

2.1 Thermodynamic Fundamentals

As stated before, this project focuses on the comparison of R-410A with baseline refrigerant R-404A at low temperature. The operating characteristics of the in-house laboratory setup will limit or define the range of thermodynamic data to be analyzed. Below is a diagram illustrating the stages which the refrigerant goes through in the cycle.

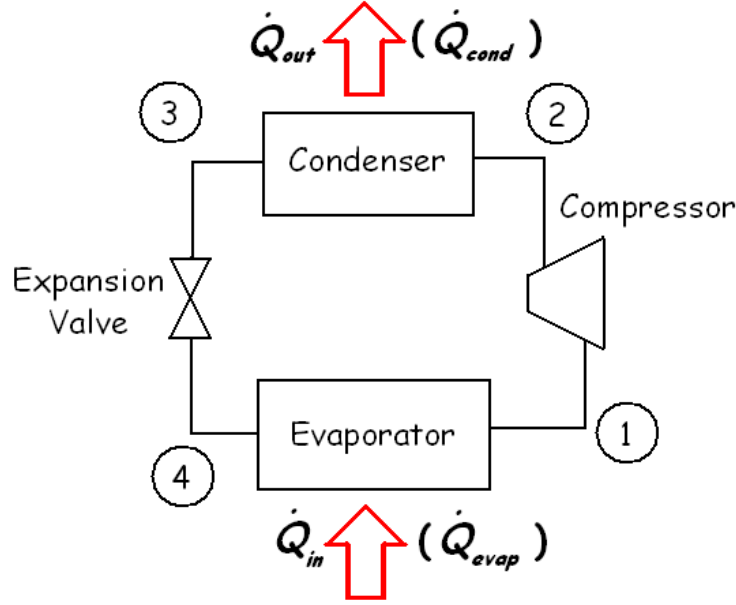


Figure 1 Basic refrigeration cycle equipment

Figure 1 shows a simple refrigeration cycle with 4 stages—Compression (1), Condensation (2), Expansion (3), and Evaporation (4). In the condenser at stage 2, the condensing refrigerant rejects heat Q_{out} to the cold air. In the evaporator at stage 4, the refrigerant absorbs heat Q_{in} from the air circulating in the freezer causing the refrigerant to evaporate. Stage 1 is the compression and stage 3 is constant enthalpy expansion. For both heat exchangers (stage 2 and 4), the first law of thermodynamics applies and we can write the energy balance between the air and the refrigerant.

$$Q - W = \Delta H \quad (\text{Equation 2.1})$$

For the condenser this becomes

$$Q_{air} - W_{cond} = \Delta H_{ref} \quad (\text{Equation 2.2})$$

$$(\dot{m}_{air} h A \Delta T) - (0) = \dot{m}_{ref} (h_{ref,out} - h_{ref,in}) \quad (\text{Equation 2.3})$$

For the evaporator this is

$$Q_{air} - W_{evap} = \Delta H_{ref} \quad (\text{Equation 2.4})$$

with similar terminology.

The energy balance for the entire cycle repeating at a given point will be

$$\sum Q - \sum W = \sum \Delta H \quad (\text{Equation 2.5})$$

$$Q_{evap,in} + (-Q_{cond,out}) + Q_{comp} - (W_{comp} + W_{evap} + W_{cond}) = \Delta H \quad (\text{Equation 2.6})$$

Which reduces to

$$Q_{evap,in} + (-Q_{cond,out}) - W_{comp} = 0 \quad (\text{Equation 2.7})$$

for no net change in energy over one complete cycle.

In this thesis project, $Q_{evap,in}$ or simply Q_{evap} is desired to be kept at constant capacity for comparison of the two refrigerants operating in the equipment. This is because the temperature requirement in the freezer is the same for each refrigerant at -28.8°C . The refrigerant performance will be observed by meeting the same load or temperature in the freezer case while varying the temperature of the outside air where the heat is rejected, in the condenser. In actuality Q_{evap} is kept within a comparable range, since it does vary slightly with the main experimental variable, the condenser air inlet temperature or $T_{air\ in,cond}$. This is the T in equation 2.3, where one can see the variables for the condenser heat transfer. The better the conditions for heat rejection in the condenser, the less work the compressor will have to do to keep the evaporator at constant capacity. Also in equation 2.3, other parameters which could be used to affect the evaporator load are the refrigerant mass flow rate and the air mass flow rate, but in this study they will be kept constant for better comparison of R-404A and R-410A. The refrigerant mass flow rate is lower for R-410A than for R-404A, but it is kept at a constant value for each refrigerant while $T_{air\ in,cond}$ is varied.

As the refrigerant completes one thermodynamic cycle, the changes in refrigerant state can be represented by phase diagrams with respect to changes in pressure or temperature. Shown below is a generic pressure-enthalpy phase diagram (Figure 2). The refrigerant will go through the various phases shown: superheated, saturated vapor, two-phase transition from boiling to liquid, saturated liquid, and sub-cooled refrigerant. Fortunately, the refrigerant will not operate at the critical point as that would decrease system efficiency in the current heat exchanger configurations.

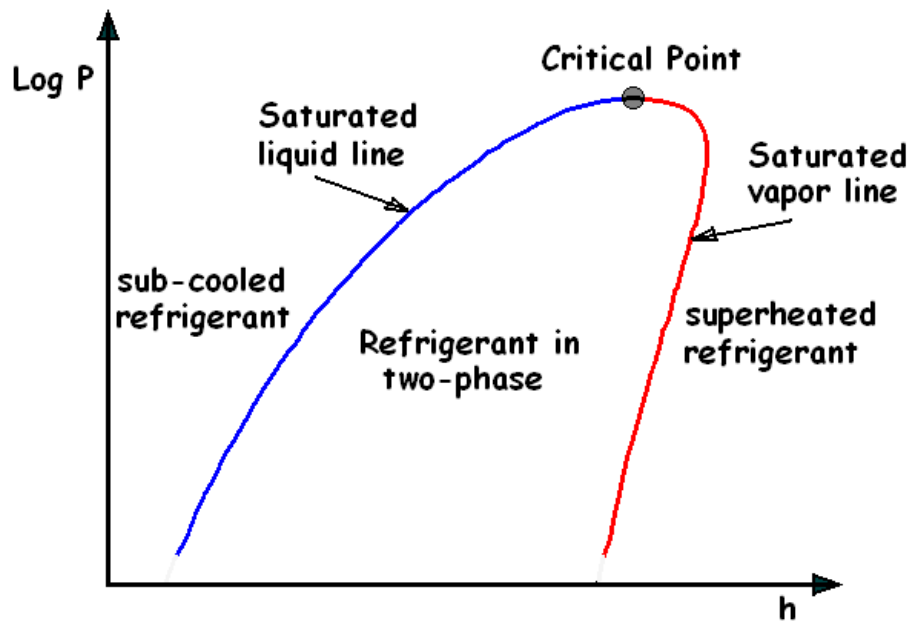
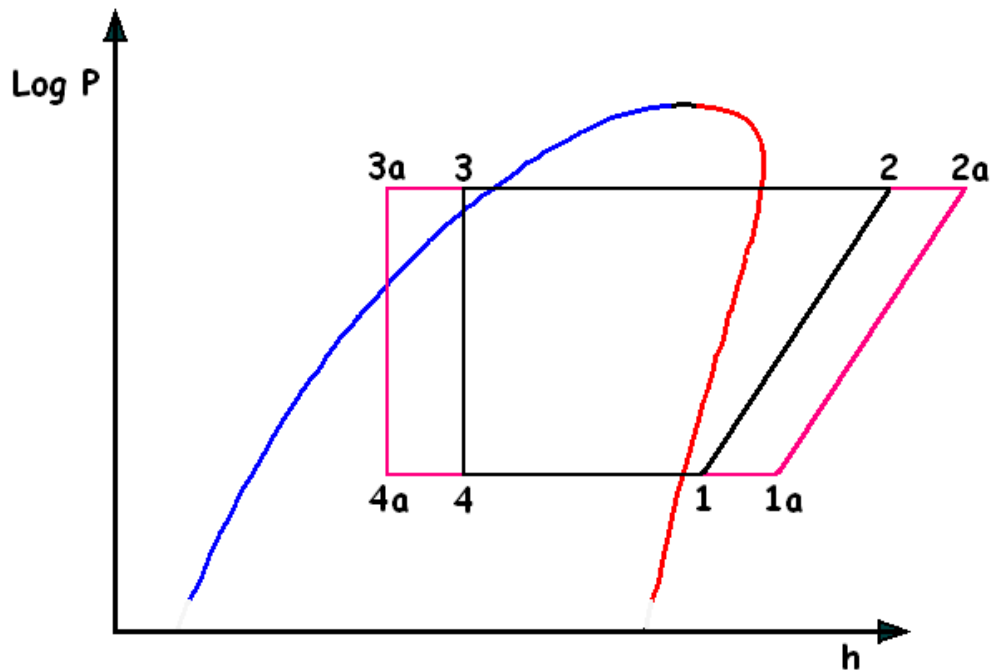


Figure 2: Log P vs. h phase diagram for a generic refrigerant



1-2 Compression 2-3 Condensing 3-4 Expansion 4-1 Evaporating

Figure 3 P-h diagram with thermodynamic cycle data

Figure 3 shows a P-h diagram with actual thermodynamic cycle data. The black line represents a basic cycle with only a compressor, condenser, expansion valve, and evaporator. The

refrigeration cycle in the present study takes advantage of an additional heat exchanger called an internal heat exchanger, which allows for additional evaporation after the fluid leaves the evaporator and additional condensing after the refrigerant leaves the condenser. This is accomplished by attaching a flow tube to the evaporator outlet and running it parallel to another flow tube added to the condenser outlet so that the heated vapor leaving the evaporator transfers heat to the cooled fluid leaving the condenser. After running parallel for this heat exchange, the tubes separate and proceed to the next stage as before. The refrigerant phase change in the internal heat exchanger is shown in Figure 3 by the extended lines from 1-1a and 3-3a, with greater superheat in between the evaporator outlet and compressor inlet (1-1a) and greater subcooling in between the condenser outlet and expansion valve inlet (3-3a). The new change in enthalpy through the evaporator will be from 4a-1, which means the evaporator accommodates a greater load than originally (4-1).

The overall heat transfer and work quantities in the respective heat exchangers can be obtained with simple calculations. Recalling equations 2.1-2.7 above, and consistent with Figure 1 and Figure 3 for a cycle with no internal heat exchanger, the relevant quantities are

$$\frac{Q_{in}}{\dot{m}} = h_1 - h_4 \quad (\text{Equation 2.8}) \quad \text{Heat into evaporator absorbed from product}$$

$$\frac{\dot{W}_c}{\dot{m}} = h_2 - h_1 \quad (\text{Equation 2.9}) \quad \text{Work done by compressor}$$

$$\frac{Q_{out}}{\dot{m}} = h_2 - h_3 \quad (\text{Equation 2.10}) \quad \text{Heat rejected by condenser}$$

The overall measure of cycle efficiency is given by the coefficient of performance

$$COP = \frac{\dot{Q}_{in}}{\dot{W}_c} \quad (\text{Equation 2.11}) \quad \text{Coefficient of performance}$$

which is the ratio of the cooling capacity to the compressor work. These four quantities--evaporator capacity, compressor work, condenser load, and coefficient of performance, will be included in the results section as part of the refrigerant performance and EES model analysis. Of course, how the refrigerant performance data translates onto the phase diagrams, whether through lab experimentation or simulation modeling, will depend on the properties of the

refrigerant. The phase diagrams are merely 2-dimensional projections of a 3-dimensional P-v-T surface which represents the refrigerant behavior with respect to changes in pressure, temperature, and volume. Table 1 shows the properties of R-410A and R-404A which will factor into the different performance of each refrigerant. Table 1 is obtained from Radko's thesis [3] but it is worth including here because it contains fundamental information for the refrigerant comparison.

Table 1: Refrigerant properties [3]

Refrigerant	[Unit]	R410A		R404A	
Component		R125 – R32		R125 – R143a – R134a	
Composition	% by weight	50% - 50 %		52 % – 44 % – 4 %	
Critical temperature	[°C]	71.8		72.1	
Critical pressure	[MPa]	48.9		37.3	
Temperature	[°C]	-30	30	-30	30
Temperature Glide	[K]	0.06	0.11	0.71	0.42
Sat. pressure	[kPa]	270.32	1876.91	204.95	1425.91
Specific heat liquid	[kJ/(kg-K)]	1.40	1.75	1.26	1.56
Specific heat vapor	[kJ/(kg-K)]	0.91	1.51	0.82	1.24
Latent heat	[kJ/Kg]	252.52	197.25	183.56	130.19
Density liquid	[kg/m ³]	1280.6	1035.4	1282.3	1040.7
Density sat. vapor	[kg/m ³]	10.57	76.11	10.98	78.15
Volumetric capacity	[kJ/m ³]	2816.4	-	2155.8	-

The quantities included in Table 1 are essentially the most basic explanation of the expected low temperature application refrigerant performance results. The latent heat of R410a is greater than that of R404a, meaning that the chemical makeup of the substance itself has a greater internal energy. The specific heat of R410a is generally greater than that of R404a, meaning that more energy is required to raise the temperature of 1 kg of the substance by 1

degree. This is consistent with the volumetric capacity numbers which show that R410a can store more energy per m^3 . The R410a will absorb more heat from the air in the evaporator while operating throughout the cycle at a lower mass flow rate than R404a. The increased evaporator efficiency will cause the ratio of capacity to compressor work (coefficient of performance or COP) to be higher for R410a even if the compressor does more work or the operating pressure and discharge temperature of the compressor is higher.

The following graph is taken from the article [6] mentioned in the literature survey. While the actual COP results will be different for this study in terms of magnitude, the results should verify that R410a yields a greater COP in the low temperature display case system than R404a and that cycle COP decreases with increased condensing temperature.

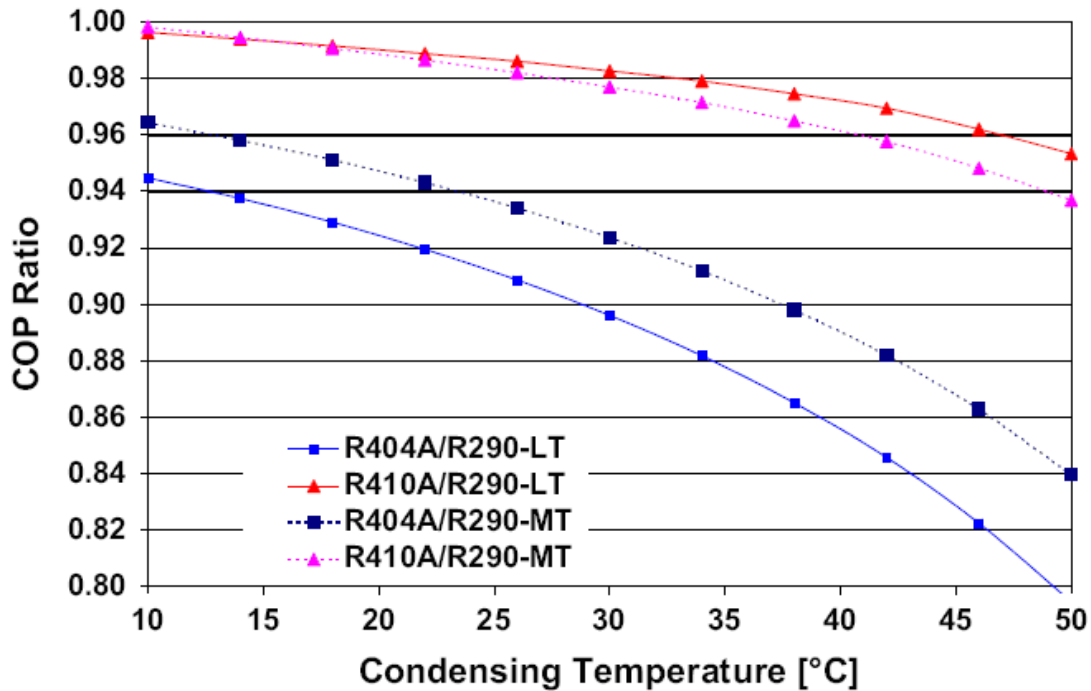


Figure 4 COP ratio of R410a/propane and R404a/propane [6]

2.2 Problem Definition

Specifically for this study, the Hussmann supermarket freezer display cases operate with R-404A at a standard capacity of 3.42 kilowatts, while the operating temperature of the refrigerant as it evaporates is -28.56°C . [3] These parameters are given in the Hussmann evaporator manual and are based on a capacity per door of 0.57 kW. There are two refrigerator display cases connected in series with 3 doors each.

As stated before, the refrigerants R-410A and R-404A are compared by using them separately in this equipment at a variety of ambient air conditions. For a reasonable data set the ambient air is varied in increments of 5°C from 20 – 35°C , resulting in four observable data points for each refrigerant. Thus R-404A system performance will be observed at 20 , 25 , 30 , and 35°C ambient air temperature, and likewise for R-410A.

Now that the independent variables are set, other key parameters will follow from the solution process. The computer model will solve for all the heat transfer and pressure drop information, as well as system performance indicators like compressor work, evaporator and condenser Q 's, and system COP.

As stated before, one difference in the analysis of R-410A and R-404A is that each refrigerant will require a different mass flow rate for the cycle and thus different compressor speeds. The compressor speeds for each refrigerant were determined experimentally by Radko Brock, finding the appropriate speeds of the Bitzer compressor. For R-404A this comes to a value of 22 g/s and for R-410 this comes to about 13 g/s .

3. Methodology

The methodology will be divided into three main sections. The first section will outline how the EES code is organized to calculate all of the thermodynamic performance information of the cycle. The second part explains how the geometry from the actual heat exchangers and equipment in the ACRC lab will be represented in the EES code; this parameter set is fixed once it is entered into the model and does not change no matter how many times it is referenced. The third part will document the choices of correlations used in this particular study.

3.1 Organization of the EES Code

Overview

Quite a large body of equations are contained in the EES code for the simulation of a complete refrigeration cycle. Templates for heat transfer, fluid flow, and thermodynamic relations are all contained in various portions of the code. Programs exist for single component models which are solved in *sequential mode*. That is, codes for single heat exchangers (such as an evaporator) can be analyzed by supplying only a few necessary inputs regarding inlet or outlet states of the air and refrigerant, and the model will proceed to solve logically and step by step for the unknown information. For example, if the inlet state is supplied by the model user, the finite element information within the heat exchanger will be solved for along with the outlet state. This solving will be very quick. When all the component models of a refrigeration cycle are combined for a system model, however, the EES code should be solved in *simultaneous mode*. The heat exchanger inlet states cannot simply be supplied to the model for direct output of the outlet states. Instead the outlet of one heat exchanger becomes the inlet state of the next, as denoted by two connecting equations stating this fact. For example, the outlet enthalpy of compressor is equated to the inlet enthalpy of the condenser, and the outlet pressure of the compressor is equated to the inlet pressure of the condenser in a set of two connecting equations. Thus the system model becomes a large body of equations in an $N \times N$ matrix, or a system of N equations and N unknowns, which is solved by the Newton-Raphson numerical method programmed in EES. The $N \times N$ system model is solved once key variable values are supplied as *initial guess* values rather than direct inputs in defining equations. The entire system model, encompassing all the heat exchangers with their separate portions for geometry, heat transfer and pressure drop information, will take anywhere from 20 seconds to several minutes to solve depending on the thermodynamic properties of the refrigerant used. Since it is such a large

system of equations, dependent on thermodynamic property charts as well as heat transfer and pressure correlations, one can see that the initial guess values are key in getting the code to reach a numerical solution. Initial guess values must be obtained which are thermodynamically feasible for the refrigerant in question to exhibit in the refrigeration cycle. If the guess values are too unrealistic, the numerical method will not be practical and EES will not be able solve. If the guess values are close to realistic, but not close enough, the EES solver can become stuck in an endless loop. For this study, reasonable guess values were obtained from the data of Radko Brock's refrigeration analysis which used the identical system geometry of the laboratory setup. Reasonable or expected thermodynamic cycle data was also estimated from a simple cycle model written in EES to further verify outcomes such as the high side and low side pressures and temperatures.

The Library

Before describing the organization within the EES code model it is important to note that there are actually two separate bodies of code used at the time of analysis. The primary code is the refrigeration cycle system model, while the secondary code is the *library* which must be loaded at the time of analysis. The library simply contains algorithms labeled *Procedures* which the system model needs to choose depending on the situation. The library contains only a few useful algorithms programmed as *Functions* rather than *Procedures* but these are not used in this model and are not important to note here. The Procedures in the library are generally listed in groups. There are Procedures for calculating the properties of a refrigerant for a given state, and also for calculating properties of air wet or dry, or for the Reynolds number of air. The correlations for air side and refrigerant heat transfer coefficients are listed in the library, as well as the air and refrigerant pressure drop correlations.

The Refrigeration Cycle System Model

The following section describes the layout of the system model code. This is the code that contains all N equations and N unknowns. The library, mentioned above, is loaded and available so that any procedures needed by the system model can be called. Figure 5 below makes an important distinction, that the system model itself is divided into two parts, basically 1) the part containing the procedures and 2) the "Main Program" which is very short and calls all the procedures in part 1. (The Main Program also contains input of key constants for the heat exchangers such as air flow rates, ambient temperature and pressure, tube thermal conductivities,

and fan efficiencies). The basic order of contents for both are shown in the figure, identical in order, with the connecting equations in between. The connecting equations insure that the system model is running in sequential mode by replacing inputs of the state information at the inlet of each heat exchanger with equations defining the outlet state of one heat exchanger to be the inlet of the next. As mentioned before, reasonable initial guess values are needed for the key variables to reach a solution. With such a large, thermodynamically sensitive equation matrix there are not enough degrees of freedom to allow a large discrepancy between the initial guess values and a realistic solution.

Element Heat Transfer and Pressure Drop Codes for HX's Compressor Condenser Internal Heat Exchanger Evaporator
Connecting Equations
Main Program: Essential Inputs and Call Statements for HX's Compressor Condenser Internal Heat Exchanger Evaporator

Figure 5 Basic contents of the system model code

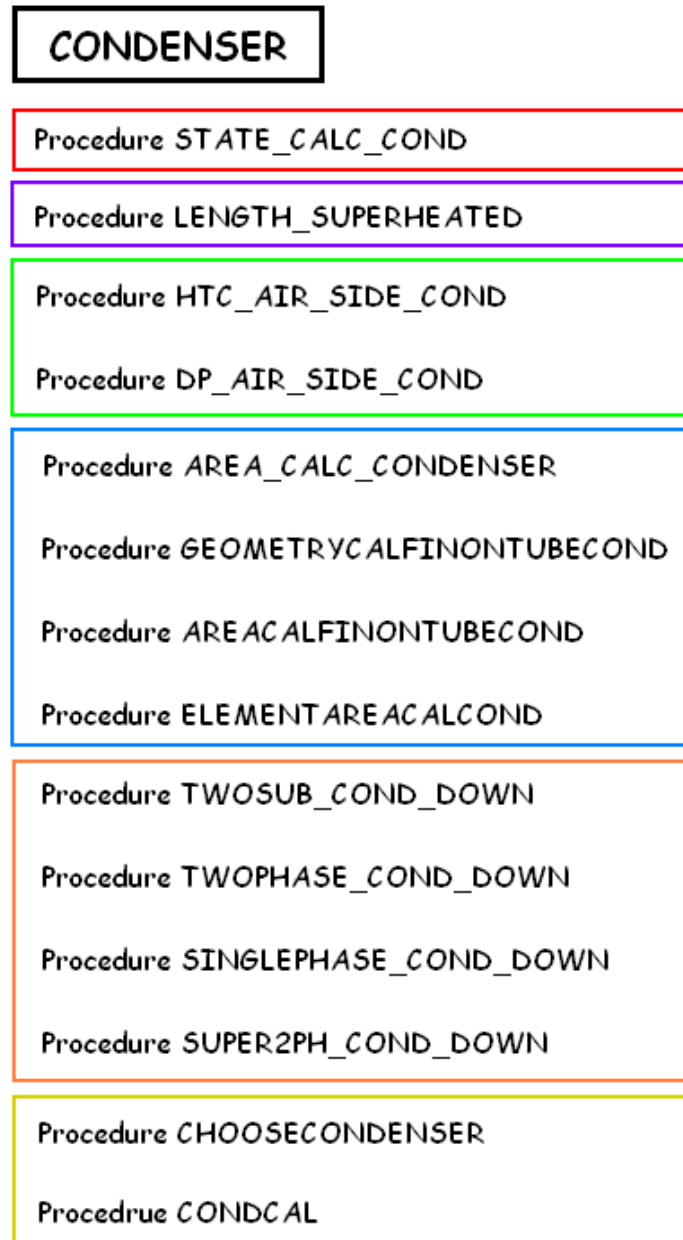


Figure 6 Basic contents of the condenser procedures

Figure 6 on the previous page illustrates the layout of the condenser procedures code. This content view will only be shown for the condenser--the evaporator follows a nearly identical progression (and the internal heat exchanger does as well, only more simplified). This figure merely shows the order in which procedures are listed in the condenser code, not the order in which they are used. The portions for calculating the geometry information are designated in a blue box, while the selection of heat transfer solutions for each refrigerant phase are in the orange portion. Air side heat transfer coefficient and pressure drop calculations are in green. Procedure `State_Calc_Cond` performs the basic step of obtaining the refrigerant state of any element if two of the state variables are known. Procedure `Choosecondenser` actually contains the algorithm for deciding which procedure in the orange segment to use, i.e. the series of if/then statements to determine the phase of the refrigerant entering each element and select the appropriate procedure to calculate the heat transfer data for that element. The illustration of this decision process can be found in the ACRC Simulation Models User Guide. [26] `Condcal` contains commands to solve most of the other procedures in the figure, and is one of the few important procedures called from the Main Program itself. The use of the procedures listed in Figure 6 will be examined in more detail below in Figure 7.

Figure 7 shows how the heat exchanger portion in the Main Program (part 2 of the code) is related to the heat exchanger element procedures (part 1), specifically for the condenser. The Main Program contains only four call statements for each heat exchanger. For each of the procedures colored in orange in Figure 7, there is one call statement in the Main Program that calls the procedure into action. The progression is:

- 1) `State_Calc_Cond` for the refrigerant state applied to the inlet element of the condenser
- 2) `Area_Calc_Condenser` for key heat transfer and pressure drop areas
- 3) `Condcal` for all element heat transfer and pressure drop information
- 4) `State_Calc_Cond` for the refrigerant state applied to the exit element of the condenser

As one can see, the call statements begin with the inlet state of the heat exchanger and end with the outlet state. In between, `Condcal` is responsible for the heat exchanger solution. Note that the procedures within `Condcal` themselves call on other procedures. The procedures called by `HTC_AIR_SIDE_COND` for the air side heat transfer

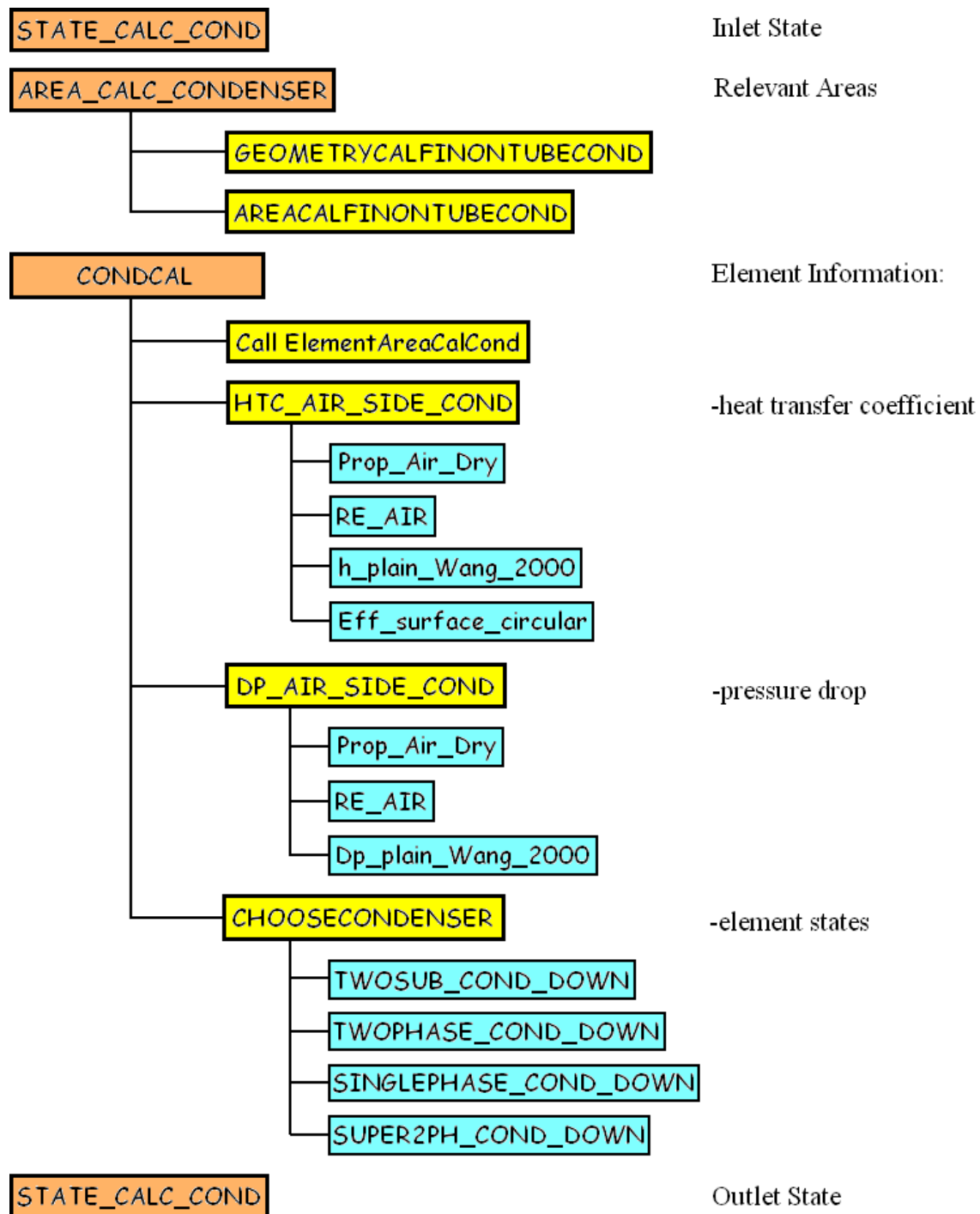


Figure 7 System model flow diagram for the condenser

Figure 7 shows the organization of procedures which results in the necessary information being solved.

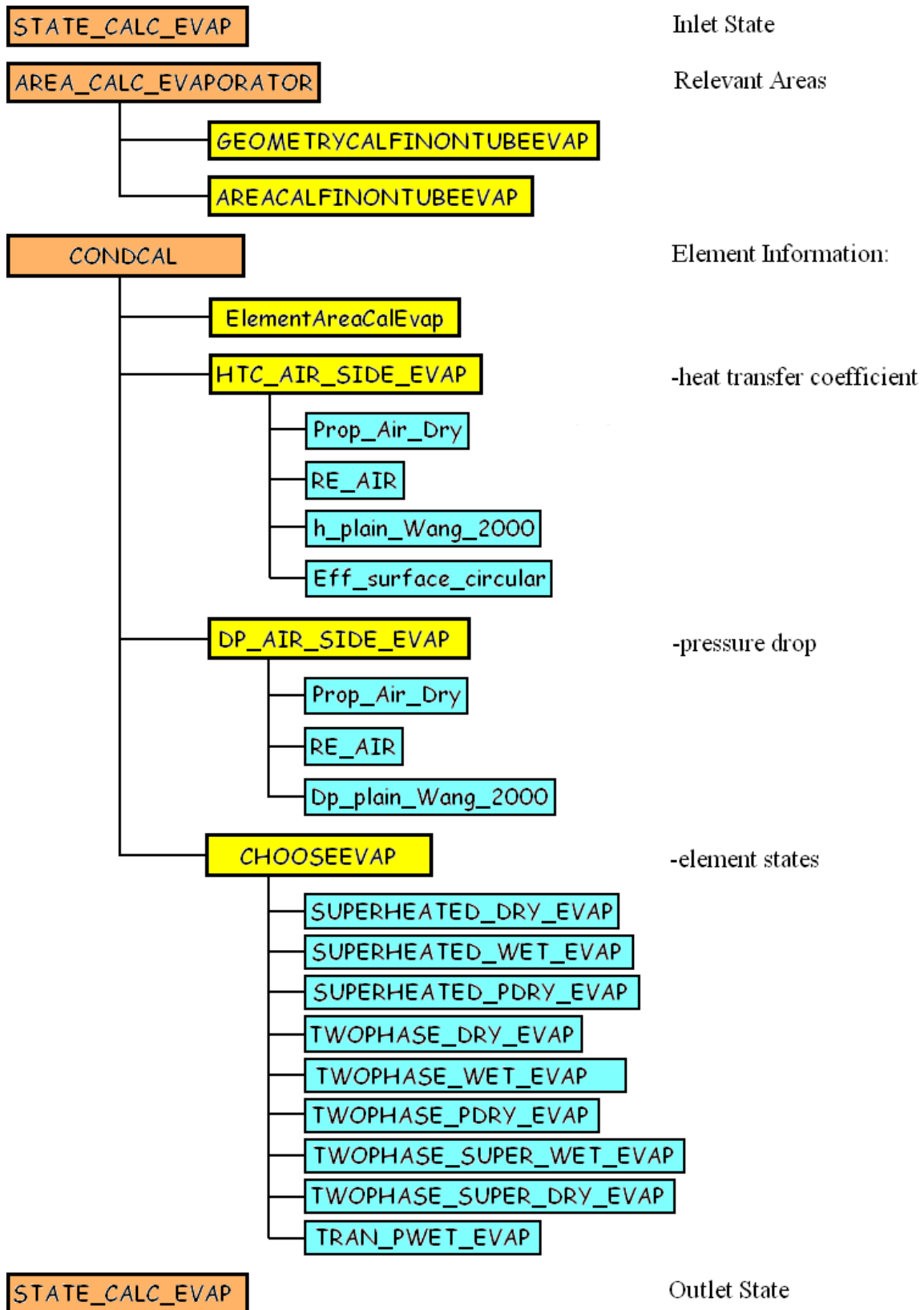


Figure 8 System model flow diagram for the evaporator

coefficient are in the library, as well as the procedures called by DP_AIR_SIDE_COND for the air side pressure drop. The procedures called by CHOOSECONDENSER, however, are listed within the system model as previously shown in Figure 6. All of the major procedures in Condcal yield the solutions for the major heat transfer parameters shown—air side heat transfer coefficients, air side pressure drop, and element refrigerant states. The element refrigerant state procedures listed in Choosecondenser or Chooseevaporator contain call statements to library procedures to obtain the refrigerant side heat transfer coefficient and pressure drops through correlations by Gnielinski, Dobson, and Churchill (for friction).

Figure 8 above shows the similar structure for the evaporator procedures. The evaporator requires more procedures for the phase possibilities of the refrigerant.

Next, Figures 9 and 10 will provide a closer look at the actual exchange of variables within procedures, again looking at the condenser. For a specific example, the procedure DP_AIR_SIDE_COND for the air side pressure drop called within Condcal will be examined.

Figure 9 shows the 3 procedures contained within DP_AIR_SIDE_COND, consistent with Figure 7. The variables needed by this pressure drop procedure include the necessary heat exchanger geometry, the thermodynamic data of the air, and the air mass flow rate. All of the variables that are input into DP_AIR_SIDE_COND are sent into the library pressure drop correlation Dp_plain_Wang_2000. Some of these same inputs are sent into the library procedures Prop_Air_Dry for the dry air properties and RE_AIR for the Reynolds number of the air. One of the dry air properties is also needed for the RE_AIR calculation, and the resulting Reynolds number is in turn needed as an input into the pressure drop correlation. This type of data flow is common in the heat exchanger element solving procedures.

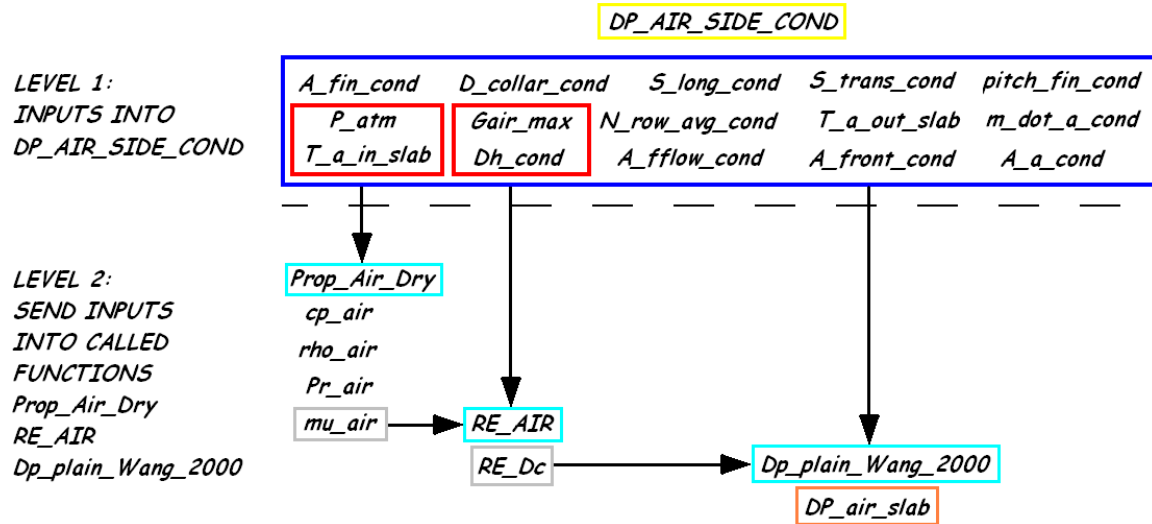


Figure 9 Variable flow in procedures within the procedure DP_AIR_SIDE_COND

This behavior is illustrated again in Figure 10 with emphasis on the number of inputs to each procedure. All 15 of the inputs into DP_AIR_SIDECOND are passed on to the correlation Dp_plain_Wang2000, but two of these 15 are also sent into Prop_Air_Dry and two others are also sent into RE_AIR. Output from Prop_Air_Dry is needed to send one additional input into RE_AIR, which also generates one input useful to Dp_plain_Wang2000 which uses a total of 16 inputs.

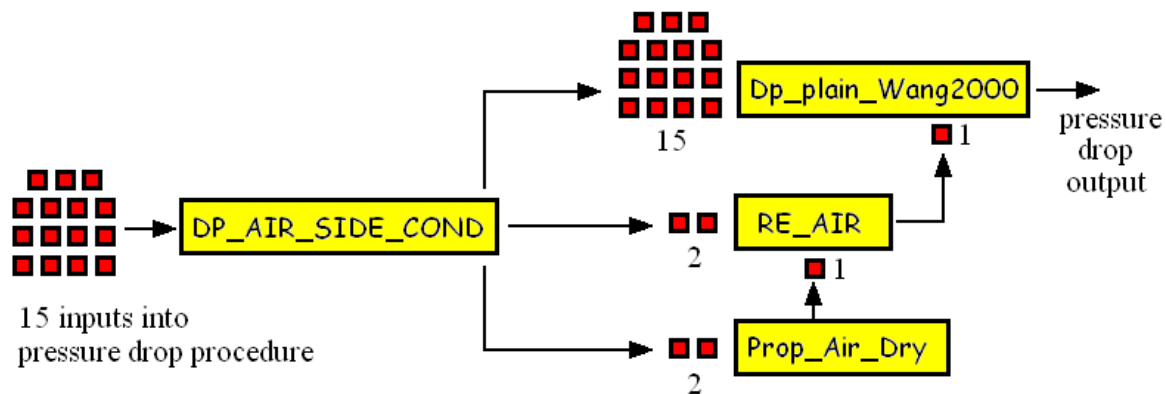


Figure 10 Division of input to procedures within procedure DP_AIR_SIDE_COND

3.2 Heat Exchanger Geometries

3.2.1 Designating the Parameters of a Heat Exchanger

Once the overall EES code structure is understood, it is important to properly input the defining statements of the actual heat exchanger geometry into the Main Program. The following information is an explanation of the instructions found in the ACRC Simulation Models User Guide v.1 [26], written by Gaurav Jain under the supervision of Dr. Clark Bullard.

Ultimately, each heat exchanger is an array of parallel tubes in a space forming a three-dimensional box. The tubes will be broken down into elements and solved one after another for the thermodynamic state of each element. Following are the key variables and considerations in entering the heat exchanger geometry so that it accurately represents the exchanger being modeled while fitting into the natural constraints imposed by the EES model algorithms. For example, once the heat exchanger depth and width is understood, the height is naturally limited to the remaining perpendicular direction.

1) Heat exchanger overall dimensions

These are the conventions for height, width, and depth of the heat exchanger core.

Core *width* = length of one tube pass (horizontal). This is entered directly as input.

Depth = length of HX in air flow direction. Height = remaining perpendicular direction. Figure 11 below shows these basic heat exchanger parameters.

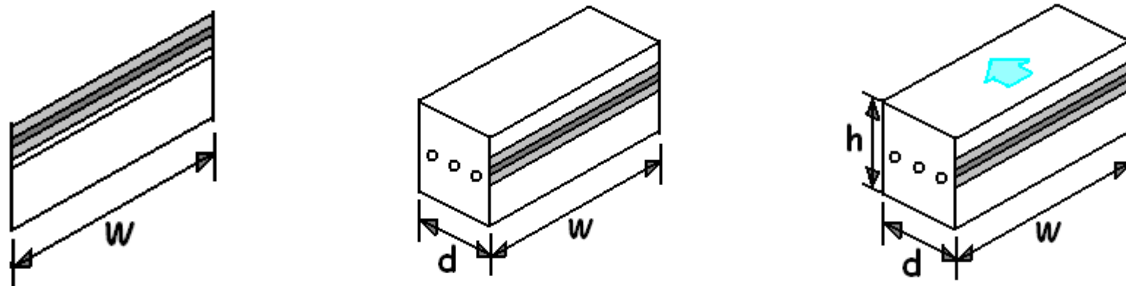


Figure 11 Width, depth, and height

2) Number of slabs

Slabs only apply to identical heat exchangers placed one after another in the air flow direction. This model will only be dealing with one slab per heat exchanger so the issue of slabs is not considered further in this study.

3) Number of rows

Getting into parameters that establish the geometry more specifically, the number of rows is an important input parameter and is always counted in the air flow direction.

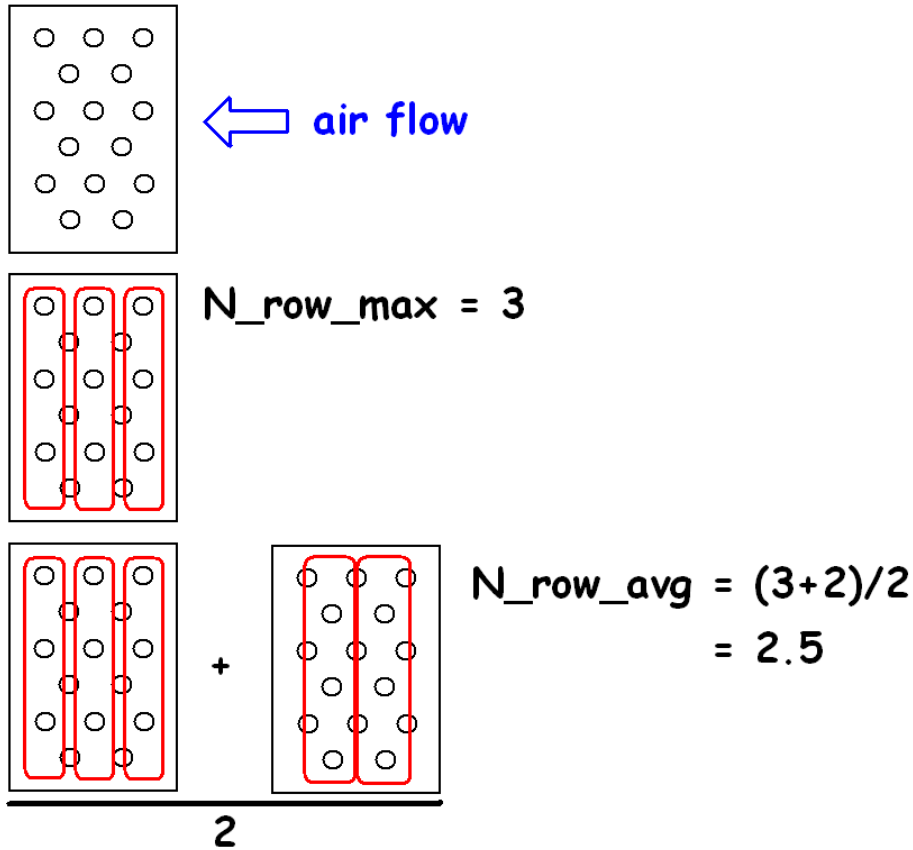


Figure 12 Example for deciding N_{row_max} and N_{row_avg}

Shown in Figure 12 above, the information of $N_{row_max_cond/evap}$ and also $N_{row_avg_cond/evap}$ for alternating row geometry. It is necessarily true that

$$N_{row_max_cond/evap} \geq N_{row_avg_cond/evap}$$

$N_{row_max_cond/evap}$ is used to get HX depth d since it is equal to the summation of the transverse tube spacing in between the rows. $N_{row_avg_cond/evap}$ is needed for heat transfer coefficient correlations and also to account for all the tube passes in the heat exchanger.

4) *Sub-heat exchanger* designates how many heat exchangers are in series.

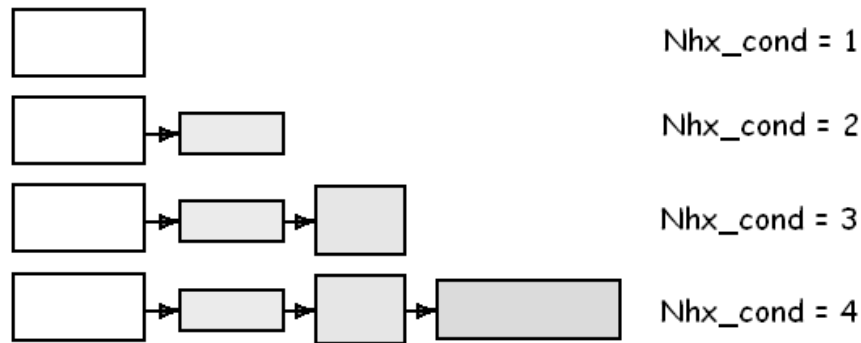


Figure 13 Sub heat exchangers

Our model will only require dealing with one heat exchanger at a time, so $N_{hx_cond} = 1$

5) Modules

5.1) $N_{mod_cond/evap}$

Although not notated as such, these are actually *modules per sub heat-exchanger*.

If there is only one and not a series then it simply means *modules per heat-exchanger*.

Modules simply denotes parallel stacking of identical tube bundles.

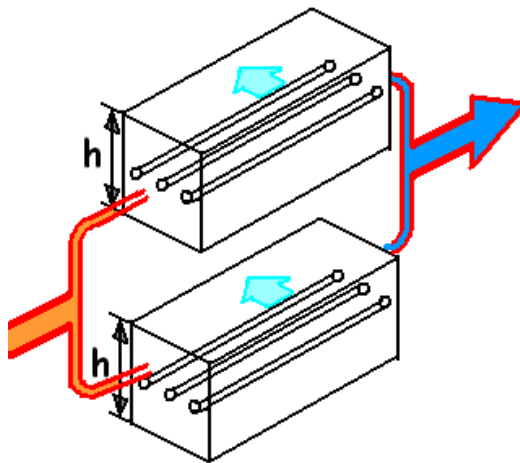


Figure 14 Modules stack in parallel

The illustration shows how modules simply designate identical heat exchanger portions in parallel. Notice that the number of modules in parallel effectively multiplies the height dimension h of each module (two shown in figure). Therefore the height effective for heat transfer of the exchanger shown in Figure 14 would be $2h$. It should be noted that this h is actually determined by the longitudinal spacing between tubes stacked in the h direction. As can

be inferred from the illustration above, the modules have identical inlet and outlet states. In calculating the physical results only one module is computed and the result multiplied by the number of modules. Once modules are designated for a heat exchanger, there are more factors which give the specifics of each module and affect the height and depth of the module.

5.2) $N_{pass_per_mod_evap/cond}$

$N_{pass_per_mod_evap/cond}$ = “the number of tube passes in a module visible when looking at the front face of the sub-heat exchanger.” [26]

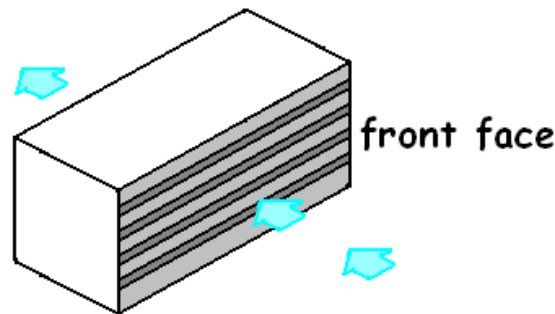


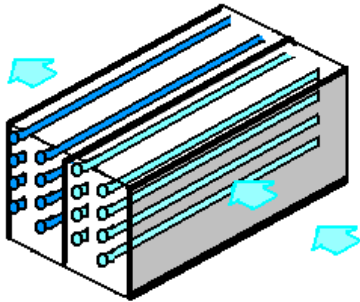
Figure 15 Tube passes visible from the front face

Figure 15 shows 4 passes visible on the front face of a heat exchanger. The air flow direction is always going toward the front face as shown. The limits for this parameter are $N \geq 1$. Each tube pass on the front face extends the height of the module by the longitudinal tube spacing. Another important distinction to make in the use of $N_{pass_per_mod_evap/cond}$ is that these visible tube passes cannot be further divided into parallel “sub-modules”. The next key variable determines how the refrigerant flows if there is any further flow division within the modules.

5.3) $N_{circ_evap/cond}$

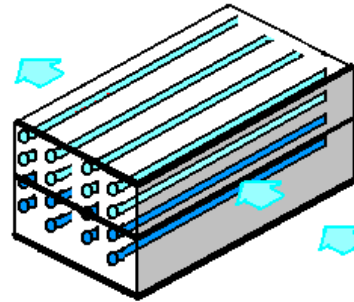
$N_{circ_evap/cond}$ = number of circuits per module. When using this parameter in a module, one must study their heat exchanger carefully to see how the circuitry is arranged. Up until now all the previous variables have set the heat exchanger width, height, and depth. $N_{circ_evap/cond}$ merely divides the existing array of pipes into identical circuits, always dividing in the air flow direction. See Figure 16 on the following page.

N_{circuits} divides evenly
along air flow direction



1 module 2 circuits

$N_{\text{mod_HX}}$ multiplies adds
identical geometry in parallel

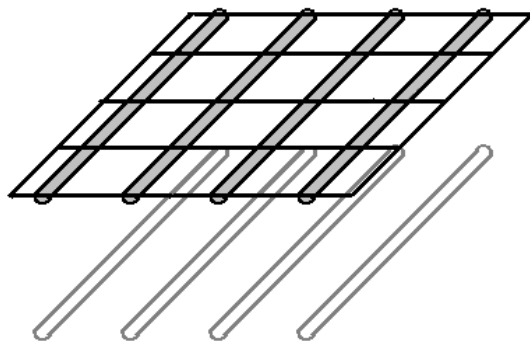


2 modules with 1 circuit each

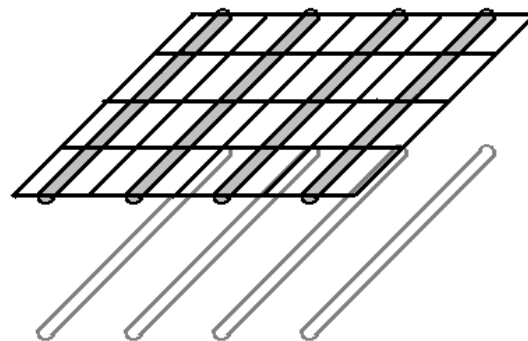
Figure 16 Distinction between modules and circuits per module

6) $N_{\text{elem_perpass_evap/cond}}$ = number of elements in each pass.

This variable fixes this parameter for the heat exchanger to the number of finite elements you want to deal with. An important aspect of the geometry algorithm is the fact that *an element area extends through the entire depth of the heat exchanger* as shown below in Figure 17a and contrary to Figure 17b. It is easy to see that this variable multiplied by the number of total passes visible along one side (in the height direction) will set the number of total elements in the heat exchanger and also the entire heat transfer grid. Obviously more elements will give a more detailed heat transfer profile, but that will also increase computation time. An appropriate selection of $N_{\text{elem_perpass_evap/cond}}$ is needed to accurately calculate the heat transfer without excess computation.



4 ELEMENTS PER LAYER
(17a)



4 X 4 OR 16 ELEMENTS PER LAYER
(17b)

Figure 17 Elements extend through heat exchanger depth

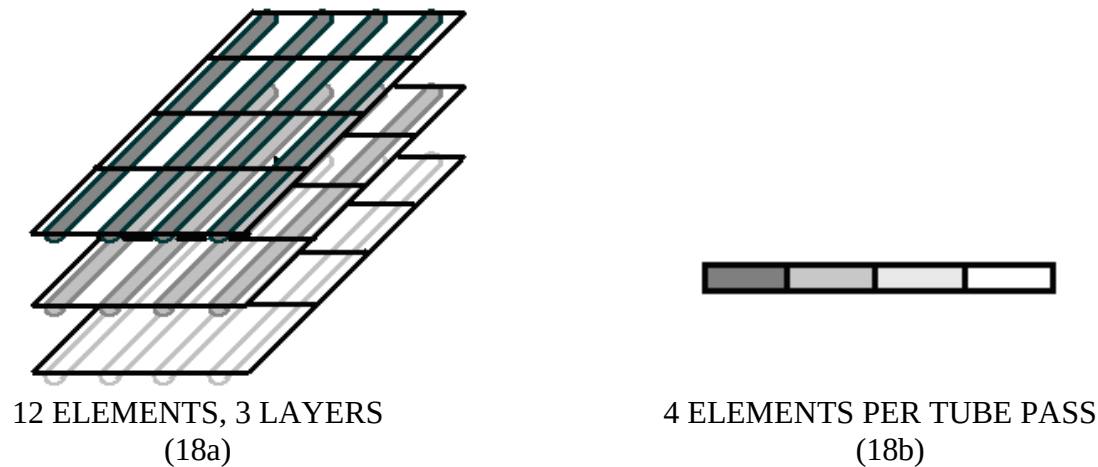


Figure 18 Element depth

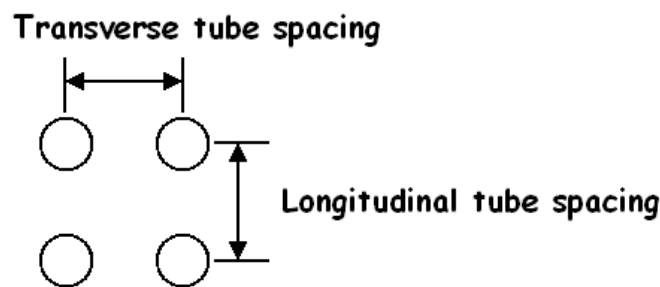


Figure 19 Longitudinal and transverse tube spacing

3.2.2 Applying the Geometry Conventions to the System Heat Exchangers

The following illustrations will show how the variables for the number of rows, modules, circuits, and passes per module were chosen for the condenser and evaporator based on the definitions which have just been explained.

Figure 20 below shows how the conventions were used to model the condenser. The heat exchanger is essentially 1 module with 2 rows in the air flow direction. The module further contains 2 circuits in parallel, but as emphasized above the circuits are divided in the air flow direction. It is a simple slab heat exchanger with 14 tube passes visible from the front face, divided into two circuits for a total of 28 tube passes.

The actual laboratory condenser circuits are not positioned 100% parallel to each other, but this was the simplest representation for the EES model and proved adequate for modeling purposes when comparing the single condenser model with Brock's condenser data.

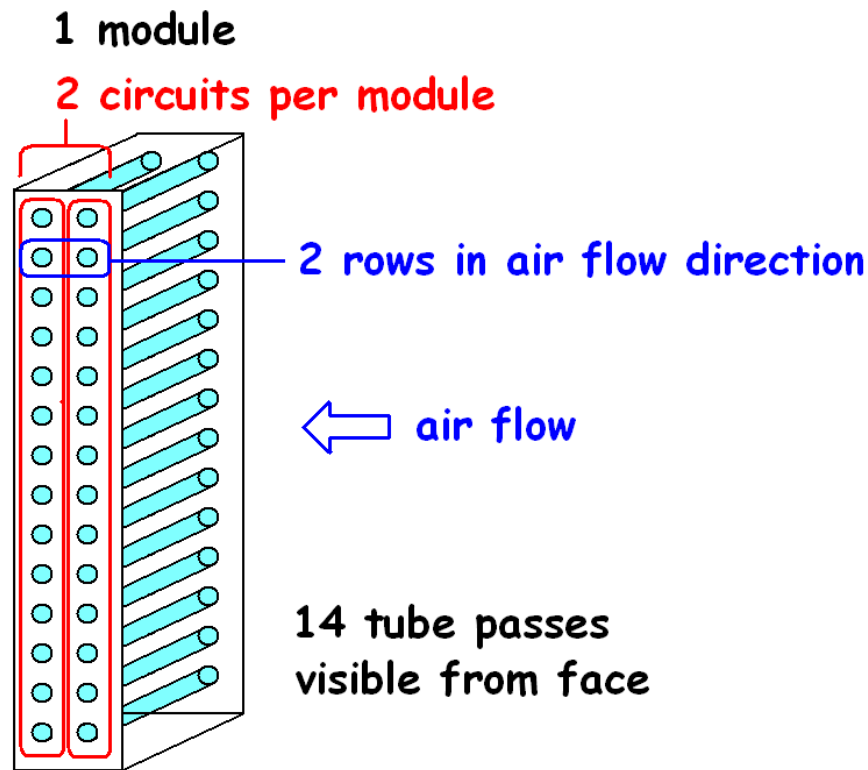


Figure 20 EES model representation of the condenser geometry

Next, Figures 21 and 22 examine the choices that were made to model the Hussmann evaporator geometry. The evaporator geometry is more complex than the condenser and contains tube passes that serpentine along the air flow direction as shown in the engineering drawing (See page 69 of Appendix A). The modeling of this heat exchanger proved more difficult to represent accurately with the EES model conventions. Presented below are two representations that were attempted in the modeling process, with one of them chosen to use for this study based on ease of overall comparison of the model results with Radko Brock's data generated in the laboratory setup. Figure 21 shows what initially seemed like the logical approach to the geometry but Figure 22 is the designation that was chosen to represent the evaporator.

The engineering drawing shows that each display case evaporator is divided into 3 circuits which serpentine along the air flow direction. There are 10 tube passes in each circuit for a total of 30 tube passes per display case. One might suppose that 3 parallel circuits for each evaporator (6 circuits total for both display cases) would best represent the geometry if entered as essentially 6 stacked **modules** with 1 circuit per module each. For the 10 tube passes per

circuit, they could be spaced as 2 x 5 tube passes, or 5 rows in the air flow direction and 2 passes per module front face. This is shown in the figure below.

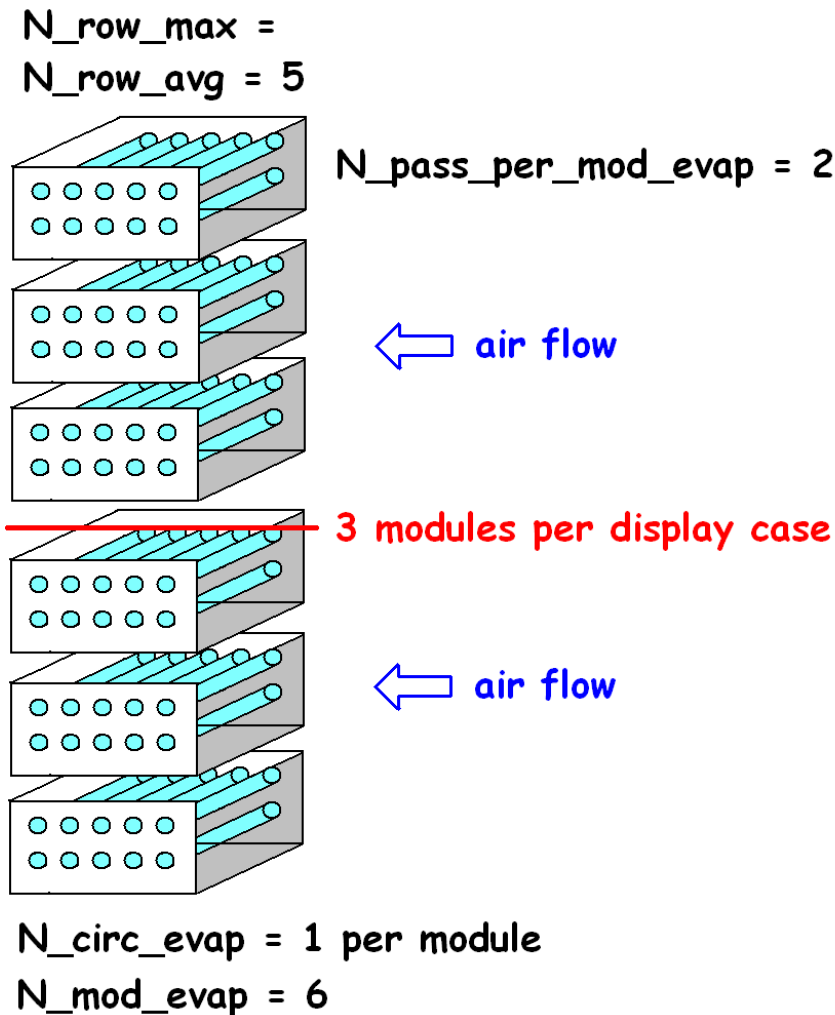
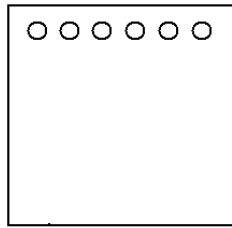
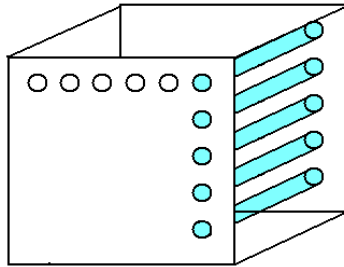


Figure 21 Trial evaporator geometry

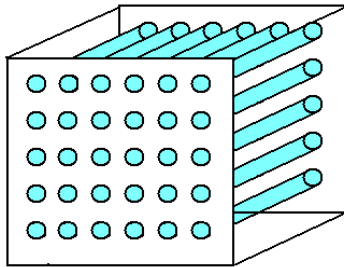
This scheme is not adequately consistent with the engineering drawing partly because it puts 5 tube rows in the air flow direction when there are clearly 6 tube rows in the air flow direction for the Hussmann evaporators. The resulting calculations produced a cooling capacity Q_{evap} that was too low and system model comparison with Brock's data that was unsatisfactory, so further EES estimates of the geometry were attempted. Figure 22 on the following page shows the geometry that was used in this study. It labels each display case evaporator as one of two **modules**, splitting each module into the 3 circuits of 10 passes each. As mentioned before, **circuits** within a module are divided along the air flow direction, perpendicular to those of the actual laboratory evaporators.



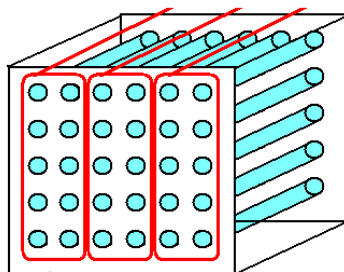
$N_{row_max} = N_{row_avg} = 6$



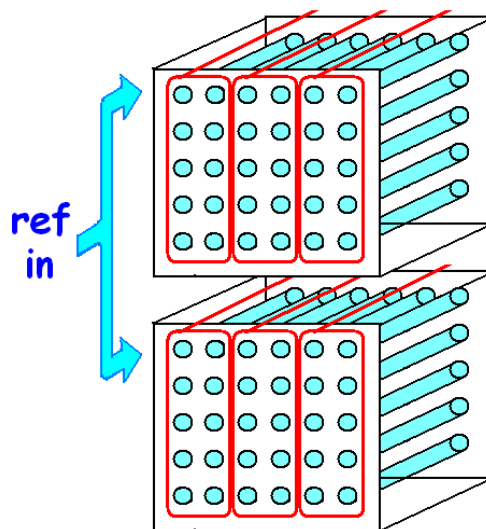
$N_{pass_per_mod_evap} = 5$



← air flow



$N_{circ_evap} = 3$



$N_{mod_evap} = 2$

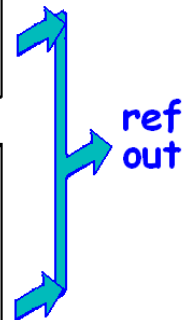


Figure 22 EES model representation of the evaporator geometry

One difference in the modeled heat transfer physics will be the drop in air temperature as it travels through the 3 circuits, whereas the engineering drawing indicates basically identical air temperature inlet into each circuit. This estimation will simulate some of the effect of the refrigerant heat transfer because it accounts for the 3 circuits of 10 passes each in 6 rows in air flow direction, but it is unclear how much the EES grid simulates the actual Hussmann case heat transfer. Nevertheless, the results of this modeling scheme appeared the most suitable for comparing the overall cycle simulation with Brock's data and for further analysis of refrigerants R410a and R404a.

3.2.3 Tabulation of Heat Exchanger Geometry Parameters

Appropriate data entry and accounting of variables must be done for each heat exchanger in the Main Program where the input is placed. This section will document the model parameter choices made to best represent the ACRC laboratory refrigeration loop.

Condenser

Table 2 on the next page lists the Condenser input parameters in the main program. As one can see from the table, the geometry variables are listed designating the number of simple rows and columns (Columns are equal to $N_{\text{pass_per_mod_cond}}$) of each heat exchanger as well as the other defining parameters. The table data is consistent with the condenser drawing in Figure 20 above which shows the split into two circuits. The values for the longitudinal and transverse tube spacing, as well as the condenser width are taken from an initial drawing that was provided in a figure accompanying this study. The dimensions and tube inner and outer diameters were measured with a micrometer caliper.

Evaporator

The variables in Table 3 on page 36 are specified in the evaporator documentation, and are input in the evaporator main program. Inner and outer diameters, as well as the heat exchanger width, were taken from the Hussman engineering drawing included in Appendix A on page 69. Longitudinal and transverse tube spacing is estimated based on the figure dimensions and the chosen modeling scheme. The height of the heat exchanger (also the fin height) is given in the figure, but the relevant heat transfer height is determined in the model from the longitudinal spacing and number of tubes arranged longitudinally ($N_{\text{pass_per_mod_evap}}$), in the height plane.

Table 2 Condenser input

Condenser Geometry Input Parameters		
Model parameter	description	Value (units)
Nhxcond	Number of sub heat exchangers in series	1
N_row_avg_cond	Average number of rows	2
N_row_max_cond	Maximum number of rows	2
N_mod_cond	Number of modules	1
N_circ_cond	Number of circuits per module dividing flow	2
N_pass_per_mod_cond	Number of passes seen from module front face only	14
N_elem_perpass_cond	Number of elements per pass visible on front face	10
width_cond	Tube pass length	34 inches
D_inner_cond	Condenser tube inner diameter	9 mm
D_outer_cond	Condenser tube outer diameter	0.375 inches
S_long_cond	Longitudinal tube spacing	1.083 inches
S_trans_cond	Transverse tube spacing	1.25 inches
Rough_tube_cond	Tube roughness factor	0.000005 m
K_tube_cond	Thermal conductivity of tube material (Aluminum)	0.403 kW/m-k
A_x_inlet_header_cond A_x_exit_header_cond	Header flow areas (diameter 7.1 mm)	0.00003959 m²

The following table lists the Evaporator input parameters in the main program.

Table 3 Evaporator input

Evaporator Geometry Input Parameters		
Model parameter	Description	Value (units)
Nhxevap	Heat exchangers in series	1
N_row_avg_evap	Average number of rows	6
N_row_max_evap	Maximum number of rows	6
N_mod_evap	Modules in parallel	2
N_circ_evap	Number of circuits per module dividing flow	3
N_pass_per_mod_evap	Number of passes seen from module front face only	5
N_elem_perpass_evap	Number of elements per pass visible on front face	4
width_evap	Tube pass length	82.66 inches
D_inner_evap	Evaporator tube inner diameter	0.255 inches
D_outer_evap	Evaporator tube outer diameter	0.485 inches
S_long_evap	Longitudinal tube spacing	1.443 inches
S_trans_evap	Transverse tube spacing	1.666 inches
Rough_tube_evap	Tube roughness factor	0.000005 m
K_tube_cond	Thermal conductivity of tube material (Aluminum)	0.403 kW/m-K
A_x_inlet_header_evap A_x_exit_header_evap	Header flow areas (diameter 7.1 mm)	0.00003959 m²

Internal Heat Exchanger

Before dimensional values can be entered for the internal heat exchanger, the particular geometry type of this portion must be recognized. According to the ACRC refrigeration modeling manual [26], the following types of geometry may be modeled.

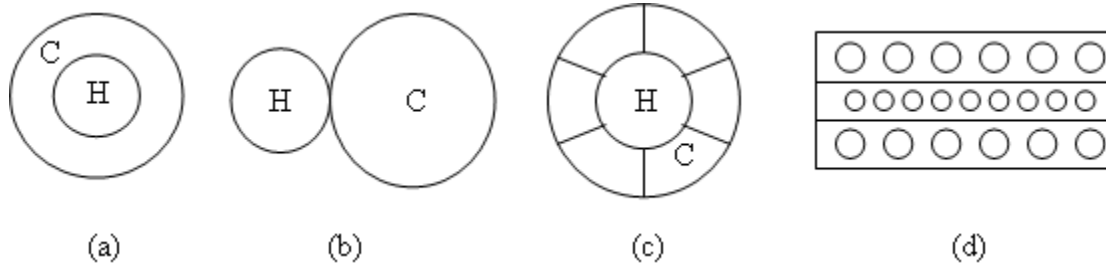


Figure 23 Internal heat exchangers in ACRC Simulation Models User Guide [26]

The type of heat exchanger in this study is that denoted by configuration (b), with the subcooled pipeline running flush with the superheated pipeline. For each geometry the appropriate hydraulic diameter must be selected for the pipes. From the ACRC user guide [26], the appropriate calculation for the hydraulic diameter of configuration (b) is given in table 4 as

Table 4 Internal heat exchanger area calculations

Configuration	Hydraulic Diameters		Flow Area Equation	Heat Transfer Areas	
	Cold	Hot		Cold	Hot
(b)	D_c	D_h	$\frac{\pi D^2}{4}$	$\pi D_c L$	$\pi D_h L$

where D is the measured diameter of the cold or hot tube and L is the total length. The related relevant dimensions for the internal heat exchanger portion of the main program are listed below in table 5.

Table 5 Measured internal heat exchanger dimensions

Parameter	Value
Length	1.32 meters
D_port_c	22.88 millimeters
D_port_h	10.415 millimeters

The length given in table 5 was measured with a meter stick, and the outer diameters of the thick cold tube and thin hot tube were measured with a micrometer caliper.

Compressor and other concerns

The compressor is not extensively modeled, but rather has a small set of defining equations. This takes nothing from the importance of the compressor, however, as it essentially regulates the entire refrigeration cycle performance by determining the refrigerant mass flow rate. An equation basically inputs a volumetric flow rate and thus the mass flow rate consistent with Radko's data for each trial, based on the Bitzer compressor cylinder size.

Another important modeling input in this simulation is the amount of superheat in the condenser and the amount of subcooling in the condenser. These values insured that the combination of individual heat exchangers into the cycle simulation resulted in overall cycle data that was consistent with Radko Brock's data, particularly for the evaporator and condenser outlet states. The values chosen were 2 degrees C in the evaporator and 5 degrees C in the condenser. This seemingly small factor was important in limiting the EES cycle data in a manner consistent with the ability to regulate the experimental heat exchanger outlet states in the laboratory trials.

3.3 Correlation Modeling Choices

Condenser: The correlation used for calculating the air side heat transfer coefficient is Chang and Wang's correlation for plain fins $h_{plain_Wang_2000}$. Here h is simply

$$h = G_{\max} (c_p)_{air} \text{Pr}^{-0.667} j$$

where the factor j is a function of Reynolds number at the collar diameter and heat exchanger tube spacing geometry. This correlation is appropriately selected to match the condenser geometry, since the dimensions meet the requirements of $h_{plain_Wang_2000}$ as designated in table 6.

TABLE 6 Condenser geometry and range allowed by $h_{plain_Wang_2000}$

Geometry parameter	Condenser value	$h_{plain_Wang_2000}$ range
Tube collar diameter Dc	9.685 mm	6.35-12.7 mm
Longitudinal spacing Pl	27.51 mm	12.4-27.5 mm
Transverse tube spacing Pt	31.75 mm	17.7-31.75 mm
Number of tube rows NP	2	1-6
Fin pitch Fp	2.117 mm	1.19-8.7 mm

The correlation used for the refrigerant pressure drop in the condenser is Chang and Wang's correlation *DP_Plain_Wang_2000*. The working data range for this pressure drop correlation is identical to that of *h_plain_Wang_2000*. The calculation is given as

$$\Delta P = 0.5 * \rho_1 V_{air}^2 * \left((K_c + 1 - \sigma^2) + 2 \left(\frac{\rho_1}{\rho_2} - 1 \right) + f \frac{A_{air}}{A_{ff}} * \frac{\rho_1}{\rho_m} - (1 - \sigma^2 - K_e) * \frac{\rho_1}{\rho_2} \right)$$

where densities ρ_1 and ρ_2 are at the inlet and outlet, respectively and ρ_m is the average of ρ_1 and ρ_2 . The air velocity is V_{air} , K_c is the contraction factor, K_e is the expansion coefficient, and σ is the ratio A_{ff}/A_{fr} where A_{ff} is the free flow area and A_{fr} is the frontal area of the heat exchanger. The factor f is a function of Reynolds number at the collar diameter and heat exchanger tube spacing geometry. The total surface area is A_{air} .

Evaporator: The correlation *h_plain_Wang_2000* was also used for calculating the air side heat transfer coefficient of the evaporator. It was used because it was the closest approximation available of the correlations in the model library, but the Hussmann evaporator geometry does not adequately fall into the range prescribed by *h_plain_Wang_2000*. Therefore some error is expected to be contributed from this calculation.

Table 7 Evaporator Geometry and range allowed by *h_plain_Wang_2000*

Geometry parameter	Condenser value	<i>h_plain_Wang_2000</i> range
Tube collar diameter Dc	12.7 mm	6.35-12.7 mm
Longitudinal spacing Pl	36.6 mm	12.4-27.5 mm
Transverse tube spacing Pt	42.32 mm	17.7-31.75 mm
Number of tube rows NP	6	1-6
Fin pitch Fp	8.467 mm	1.19-8.7 mm

The pressure drop in the evaporator is also calculated with the same correlation used for the condenser, *DP_Plain_Wang_2000*. The range of evaporator dimensions best suited for modeling by *DP_Plain_Wang_2000* is also smaller than the actual Hussman evaporator dimensions.

4. Data Analysis Methods

Overall Heat Exchanger and Cycle Performance Quantities

The work of the EES model will be to start with each heat exchanger's inlet state and calculate the heat transfer and work quantities from one element to the next for each entire heat exchanger. These quantities will be consistent with the thermodynamic equations governing cycle performance listed previously in section 2.1. The quantities Q_{evap} , Q_{cond} , W_{comp} are generated from the comprehensive EES simulation and will be included in the results section for each ambient air condition. The heat transfer in the internal heat exchanger portions Q_{IHx} is also accounted for and calculated in the model but it is a smaller quantity and will not be highlighted in the discussion. The results also make use of the P-h or T-h phase diagrams to represent the thermodynamic data.

Error Analysis

The data analysis in the results section makes use of the following basic experimental error equation for any calculated quantity of interest X:

$$\%Error = \frac{|X_{\text{Theoretical}} - X_{\text{Experimental}}|}{X_{\text{Theoretical}}} * 100 \quad (\text{Equation 4.1) Experimental Error}$$

In this report the EES model value will be taken as the theoretical standard with Brock's laboratory cycle performance data as the experimental value.

LMTD and Heat Exchanger U value

In addition to the above mentioned calculations, the model data was used to obtain heat exchanger LMTD's and evaporator U value just as Brock's thesis did. The results can be compared. The equations for LMTD and U, respectively, are

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \quad (\text{Equation 4.2) Log Mean Temperature Difference}$$

$$U = \frac{LMTD}{Q_{\text{evap}} A_{r,\text{evap}}} \quad (\text{Equation 4.3) Evaporator U Heat Transfer Coefficient}$$

where the temperature differences between the refrigerant and the air in the heat exchanger are designated at the inlet or outlet, and $A_{r,\text{evap}}$ is the refrigerant flow area in the evaporator.

5. Results and Discussion of Findings

5.1 Validation of the EES Model: Comparison of Model and Experimental Data

The data for all simulation model runs can now be compared with the experimental work done by Radko Brock [3] on the in-house setup, in order to confirm the validity of the model. This is done for the four condenser ambient air temperature settings (temperature of air into the condenser, T_{aic}) for both refrigerants, a total of eight runs. The graphs will often show the trend of performance variables as T_{aic} varies left to right at values of 20, 25, 30, and 35 °C. The graphs will also compare phase diagrams for the refrigerants at separate T_{aic} temperature settings. First is the comparison of P-h phase diagrams, beginning with R404A. The EES model result will be compared with the laboratory result for each temperature setting.

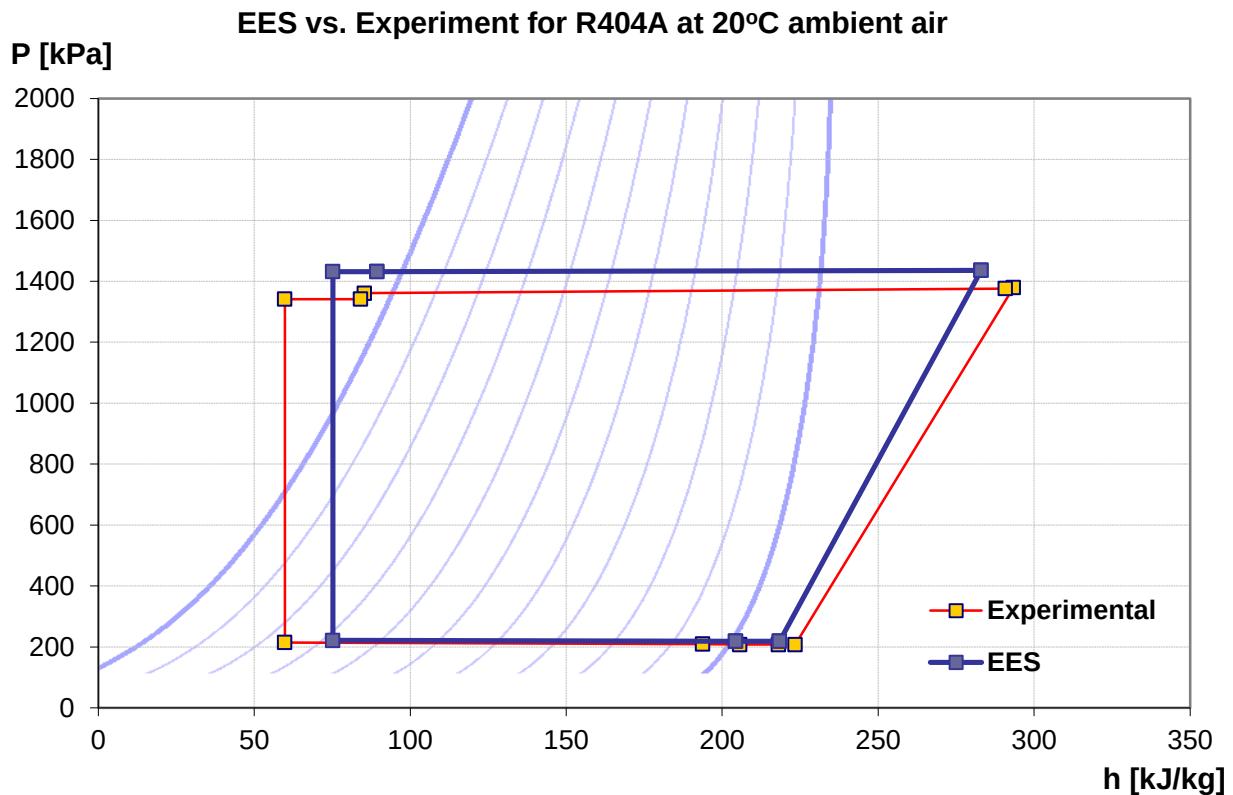
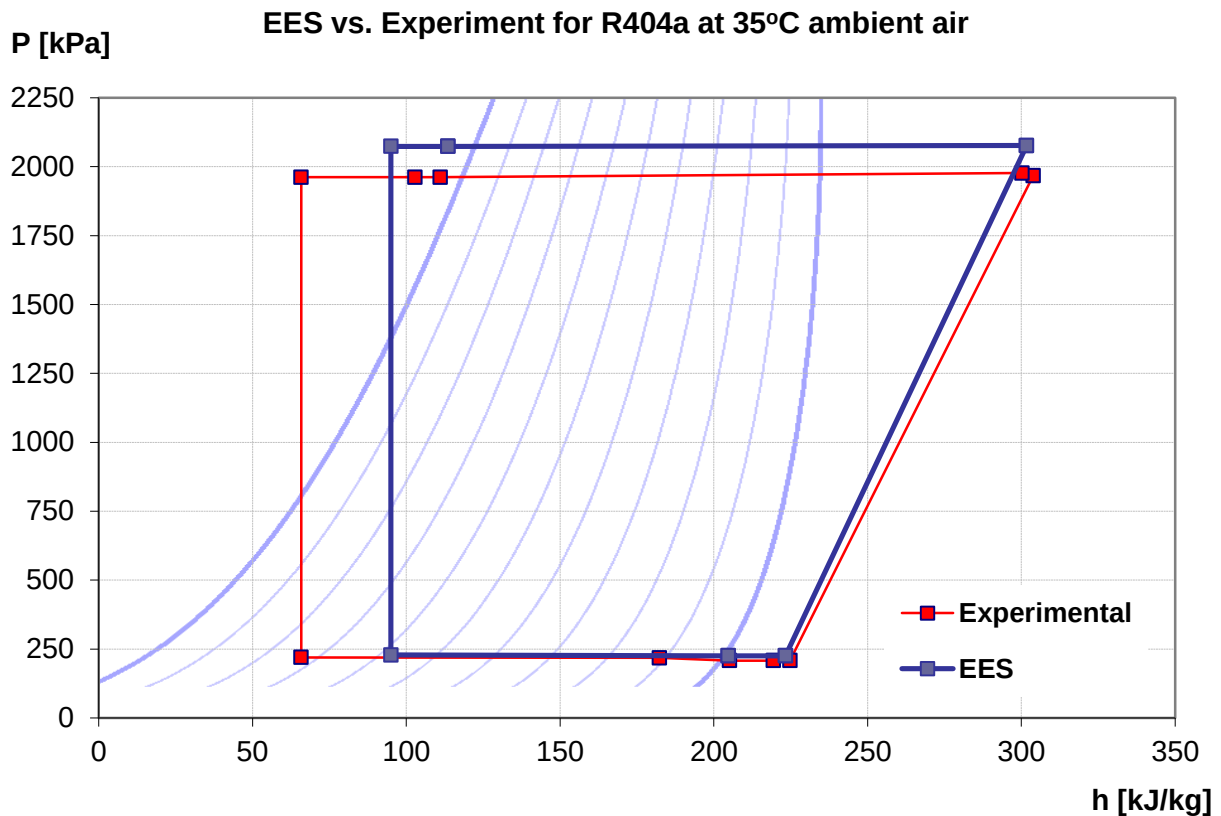


Figure 24 P-h diagram for R404A at 20 C, model and experimental comparison



Figures 24 and 25 above show the agreement of the P-h diagrams for R404a at 20 and 35 °C conditions. On the left side of the graphs, the latter reveals more discrepancy in the isenthalpic expansion, or the enthalpy as the refrigerant leaves the hot internal heat exchanger segment. The actual values of key variables around the P-h diagram is shown below in Table 8. A key agreement between model and experiment is achieved at the evaporator outlet and state for consistent comparison of the evaporator performance and thus the overall system performance.

Table 8 also shows the error results for the discrepancy of EES values vs. the experimental values for the suction pressure into the compressor, the discharge pressure, the discharge temperature, and the compression ratio. It also monitors the discrepancy in key enthalpy values at the evaporator inlet and outlet states. Table 8 shows that the error trends upward or downward as $T_{a,c}$ is varied from 20 to 25, with some values becoming more in agreement and some becoming less in agreement. Following Table 8, the same results are presented for R410a.

Table 8 Verifying EES Model for R404a

Refrigerant R404A Data Comparison, EES and Experiment			
Suction Pressure Psuc (kPa)			
Ta_{ic}	EES	Experiment	Error Psuc
20	218.5	207.833	4.882
25	220.7	207.975	5.766
30	222.9	208.120	6.631
35	225.7	208.167	7.768
Discharge Pressure Pdis (kPa)			
Ta_{ic}	EES	Experiment	Error Pdis
20	1436	1380	3.900
25	1620	1581	2.407
30	1821	1744.6	4.195
35	2077	1967	5.296
Discharge Temperature Tdis (deg C)			
Ta_{ic}	EES	Experiment	Error Tdis
20	76.45	85.443	11.764
25	83.92	91.450	8.973
30	91.29	95.904	5.054
35	99.76	100.667	0.909
Enthalpy into evaporator h_{in_evap} (kJ/kg)			
Ta_{ic}	EES	Experiment	Error h_{in_evap}
20	75.19	59.793	20.477
25	81.31	63.273	22.184
30	87.55	63.918	26.993
35	95.03	65.813	30.745
Enthalpy out evaporator h_{in_IHX_cold} (kJ/kg)			
Ta_{ic}	EES	Experiment	Error_h_{in_IHX_cold}
20	204.3	205.633	0.653
25	204.4	205.525	0.550
30	204.6	205.320	0.352
35	204.7	205.133	0.212
Compression ratio Pdis/Psuc			
Ta_{ic}	EES	Experiment	Error
20	6.572	6.640	1.033
25	7.340	7.556	2.932
30	8.170	8.287	1.442
35	9.202	9.453	2.722

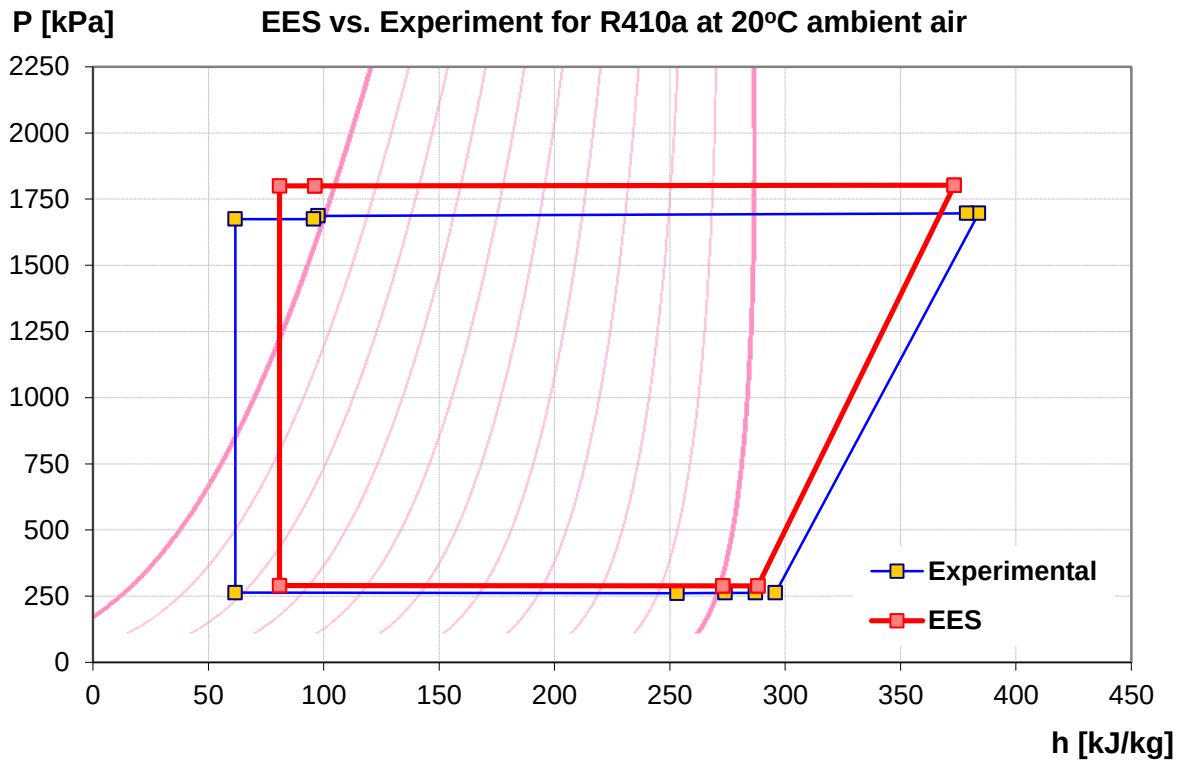


FIG 26 P-h Diagram for R410A at 20 C, Model and Experimental Comparison

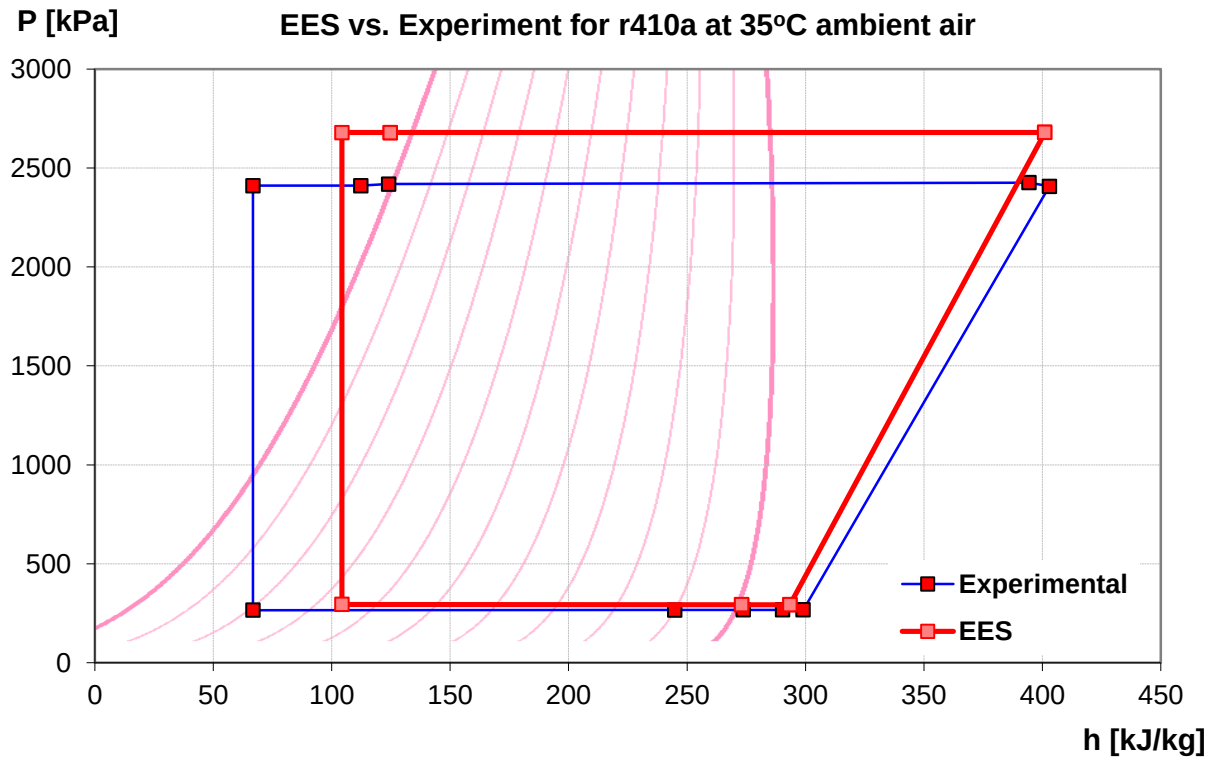


FIG 27 P-h Diagram for R410A at 35 C, Model and Experimental Comparison

Table 9 Verifying EES model for R410a

Refrigerant R410A Data Comparison, EES and Experiment			
Suction Pressure Psuc (kPa)			
Ta_{ic}	EES	Experiment	Error Psuc
20	288.5	262.680	8.950
25	290.4	261.825	9.840
30	292.1	263.080	9.935
35	293.1	266.800	8.973
Discharge Pressure Pdis (kPa)			
Ta_{ic}	EES	Experiment	Error Pdis
20	1802	1696.6	5.849
25	2099	1944	7.384
30	2373	2167.6	8.656
35	2680	2407.5	10.168
Discharge Temperature Tdis (deg C)			
Ta_{ic}	EES	Experiment	Error Tdis
20	102	110.540	8.373
25	114.2	119.575	4.707
30	124.3	126.740	1.963
35	134.6	136.950	1.746
Enthalpy into evaporator h_{in_evap} (kJ/kg)			
Ta_{ic}	EES	Experiment	Error h_{in_evap}
20	80.84	61.666	23.718
25	89.26	62.143	30.380
30	96.53	65.310	32.342
35	104.3	66.740	36.012
Enthalpy out evaporator h_{in_IHX_cold} (kJ/kg)			
Ta_{ic}	EES	Experiment	Error h_{in_IHX_cold}
20	272.9	273.920	0.374
25	273	273.575	0.211
30	273	273.840	0.308
35	273.1	273.800	0.256
Compression ratio Pdis/Psuc			
Ta_{ic}	EES	Experiment	Error
20	6.246	6.462	3.453
25	7.228	7.425	2.730
30	8.124	8.241	1.444
35	9.144	9.659	5.636

The P-h diagrams and Table 9 for R410A at 20 and 35 °C shows that the error is greater for R410a in the high side pressure and the enthalpy into the evaporator. In the P-h diagram, the EES model differs from Radko Brock's data for P_{suc} by about 9% evenly. The discrepancy in P_{dis} varies from approximately 6-10% and in T_{dis} from 8-1%, with minimal error in the compression ratio and once again negligible difference in the state out of the evaporator. This agreement at the evaporator outlet and compressor inlet seems less than for R404a but that is based on the R410a properties working in the model and the model trends are validated by the comparison with Radko's data.

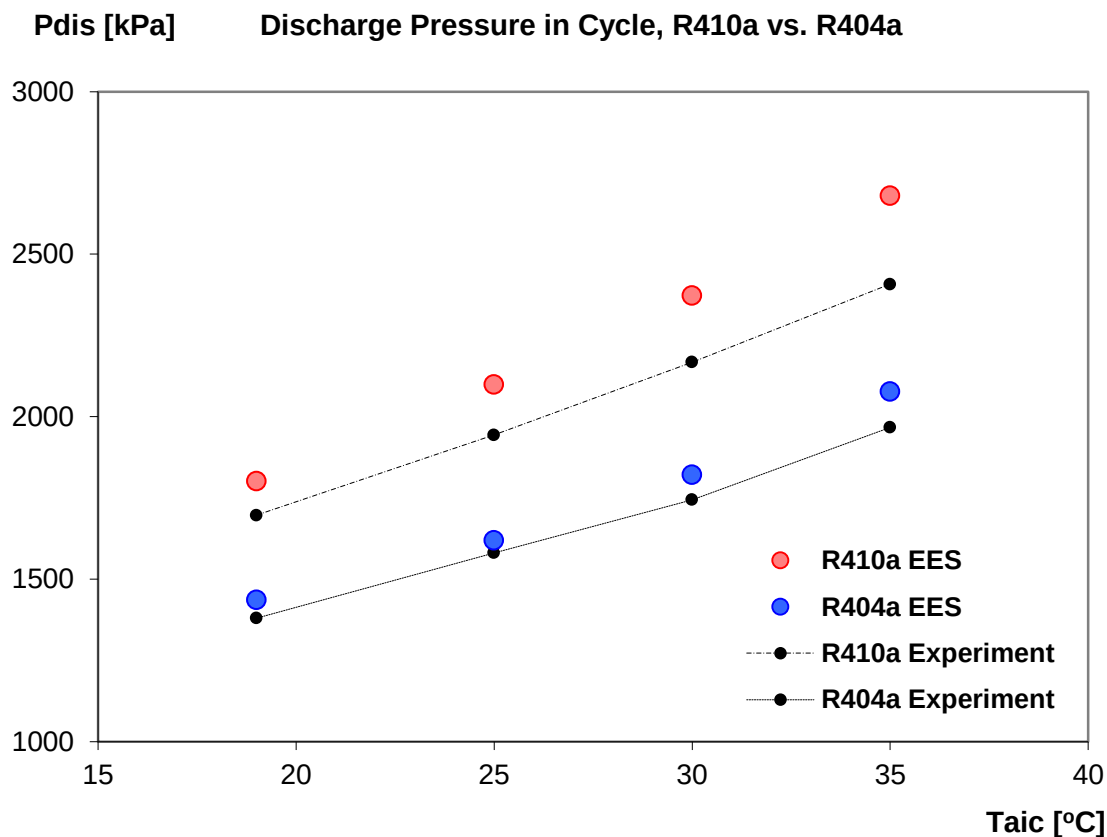


Figure 28 Compressor discharge pressure

The above graph and the next few graphs show more of the trends in tables 8 and 9 for both refrigerants. Figure 28 compares the discharge pressure trends, and as shown R410a diverges with greater discrepancy from the experimental data than R404a.

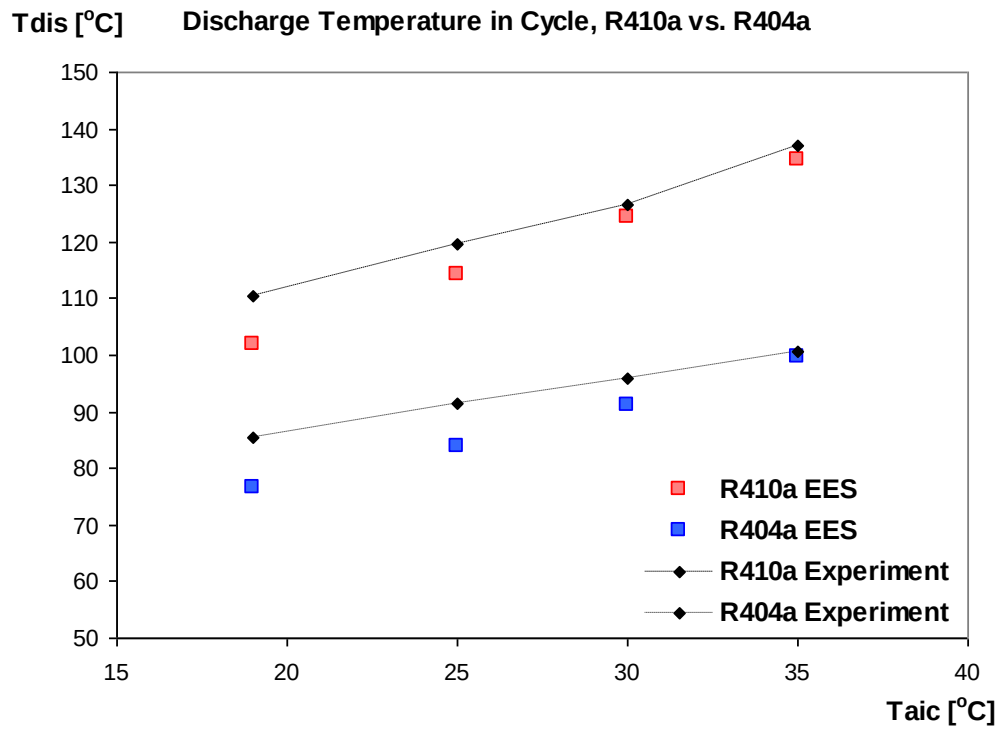


Figure 29 Compressor discharge temperature

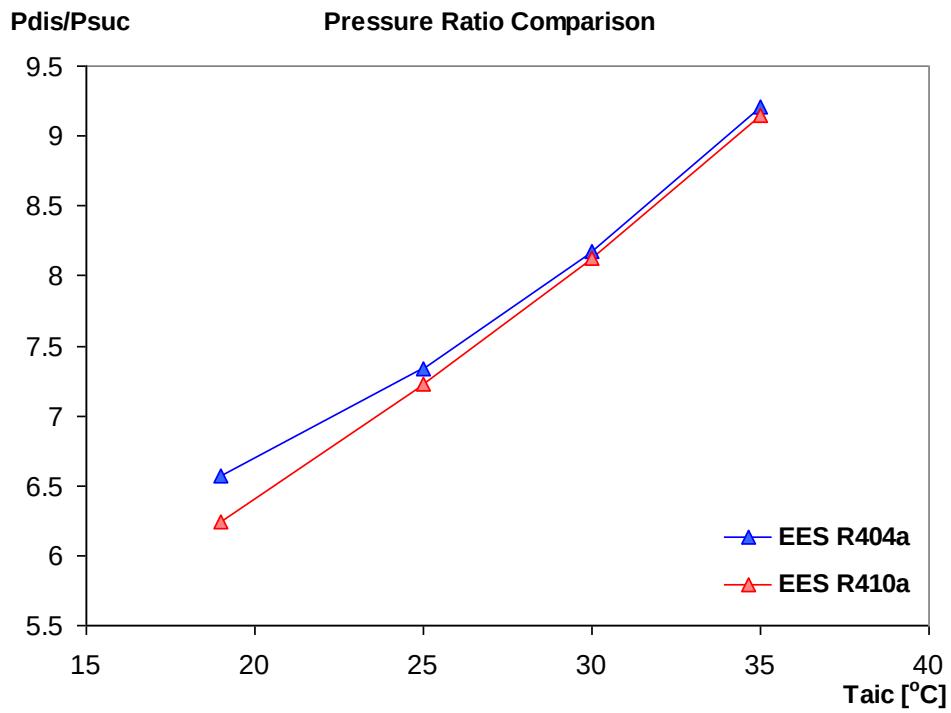


Figure 30 Compression ratio of R410a vs. R404a in EES

Figure 29 above compares the trends in T_{dis} . The EES model has very similar agreement to the experimental values for both R410a and R404a. Figure 30 above plots the pressure ratio for both refrigerants within the EES model. The percent error is shown with regards to the experiment in Tables 8 and 9 but it is negligible and not graphed.

The next several graphs will look at the energy quantities such as COP and Q_{evap} , as well as Q_{cond} and W_{comp} . Figure 31 below shows that R410a has a consistently higher COP than R404a within the EES model, but the difference is not in any substantial agreement with the experimental data except toward the lower end of the ambient air conditions, $T_{a,ic}$ at 20 °C. For the remaining $T_{a,ic}$ conditions, experimental COP data are consistently higher for R410a compared to R404a, but the magnitude of difference is much lower than the model results.

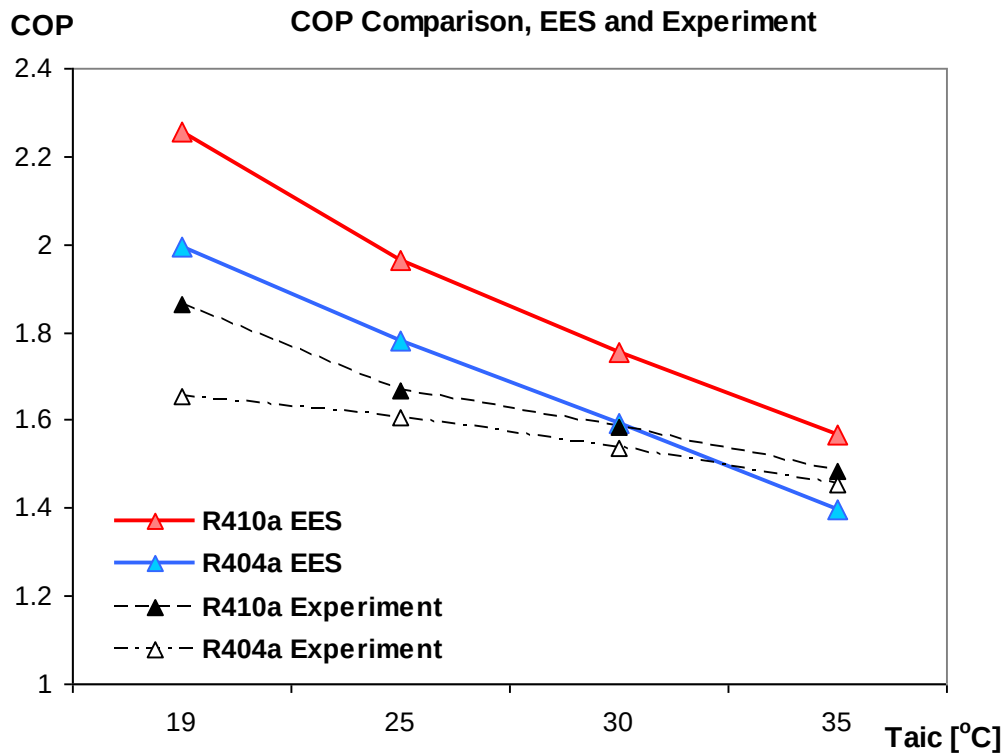


Figure 31 EES and experimental COP comparison

Figures 32 and 33 below compare only the evaporator capacity for the EES model and the experimental data of Brock's study.

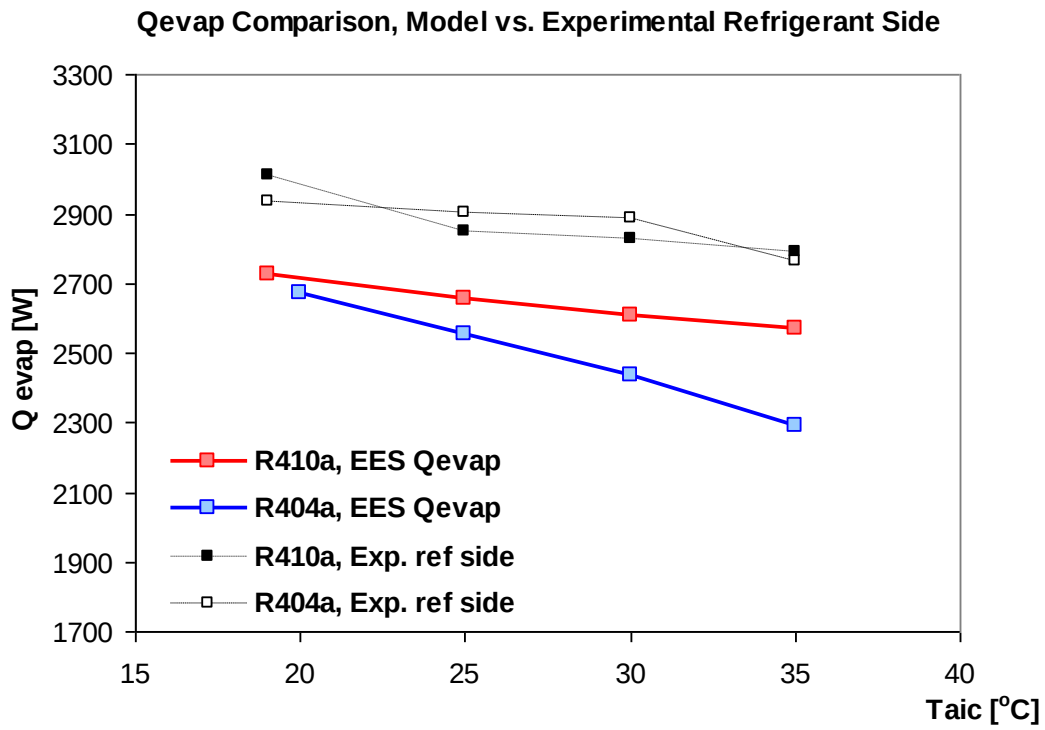


Figure 32 Model Qevap vs. experimental refrigerant side Qevap

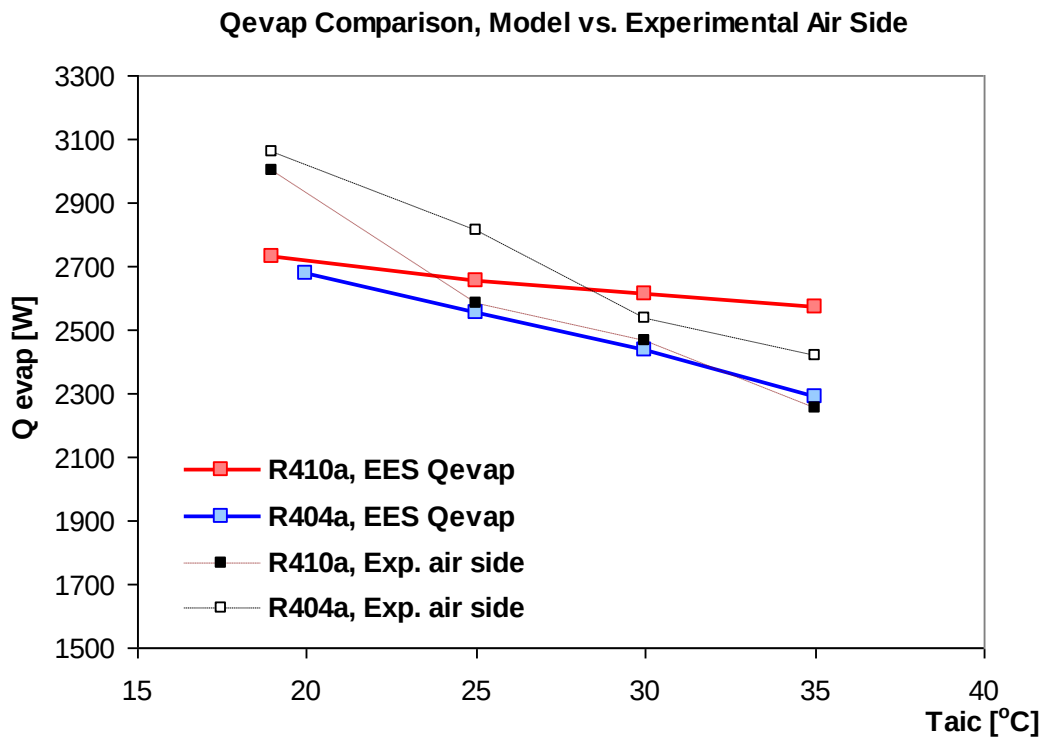


Figure 33 Model Qevap vs. experimental air side Qevap

The comparison with Brock's experimental refrigerant side Qevap shows greater numerical discrepancy with the EES model but closer overall trend slope for R410a. On the other hand, Figure 33 shows greater agreement between the EES value for R404a and Brock's air side Qevap. It is also strange to note that the trends comparing R410a to R404a within Brock's data only in Figure 33 are more similar to the constant spacing of the COP values for the EES model in Figure 31. The EES model Qevap and COP trends are combined in Figure 34 below to aid in viewing the contrast to Figure 31. The EES model shows a nearly constant sloping superiority of R410a to R404a COP but a skewed Qevap trend, while Radko's data shows a nearly constant sloping superiority of R410a to R404a Qevap but a somewhat skewed COP trend. This could indicate that the heat transfer model and experimental data are weighted oppositely with respect to the COP and Qevap.

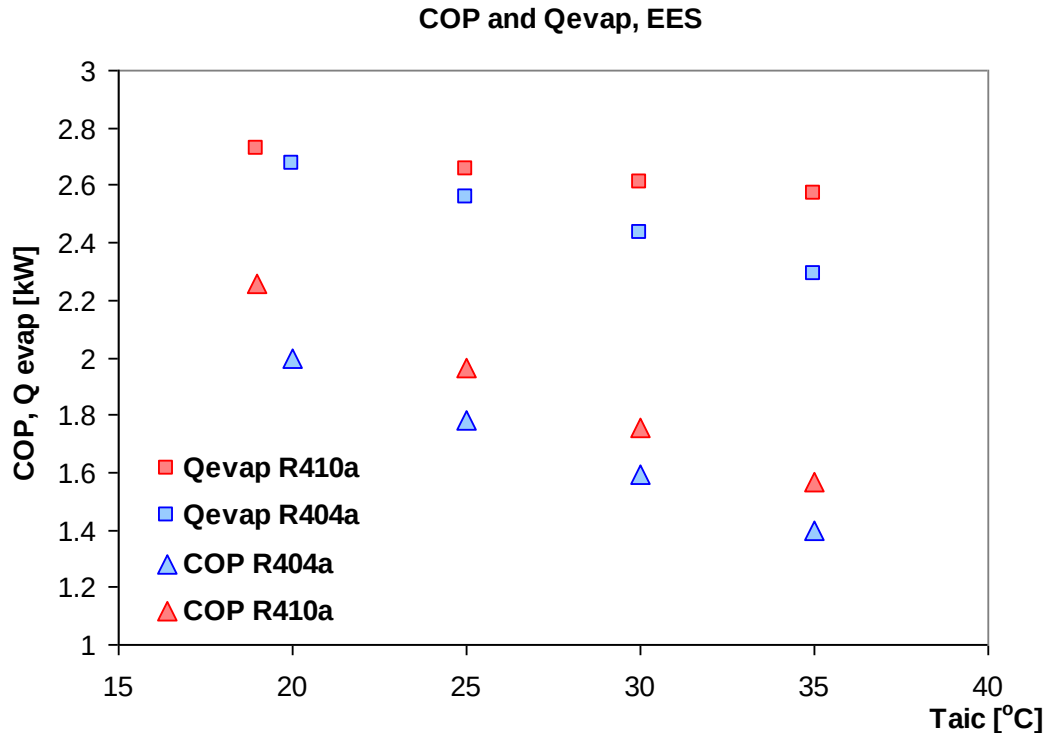


Figure 34 EES model COP and Qevap for both refrigerants

Figures 35 and 36 on the following page will compare the overall heat exchanger performance quantities for the four condenser air inlet conditions. The data for figures 31 through 36 are in appendix A in Tables 13 and 14. The experimental Qcond and Qevap values are from Brock's air side heat transfer calculations in the tables but not the refrigerant side.

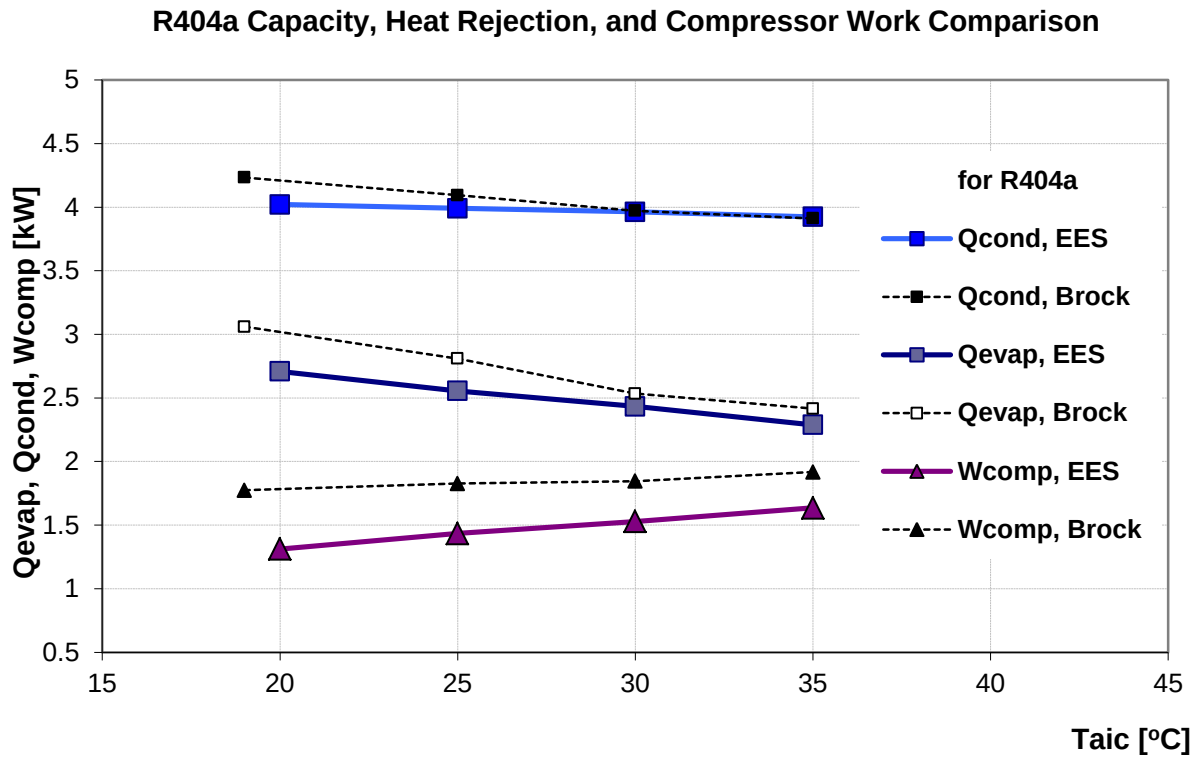


Figure 35 R404a Cycle Performance Quantities

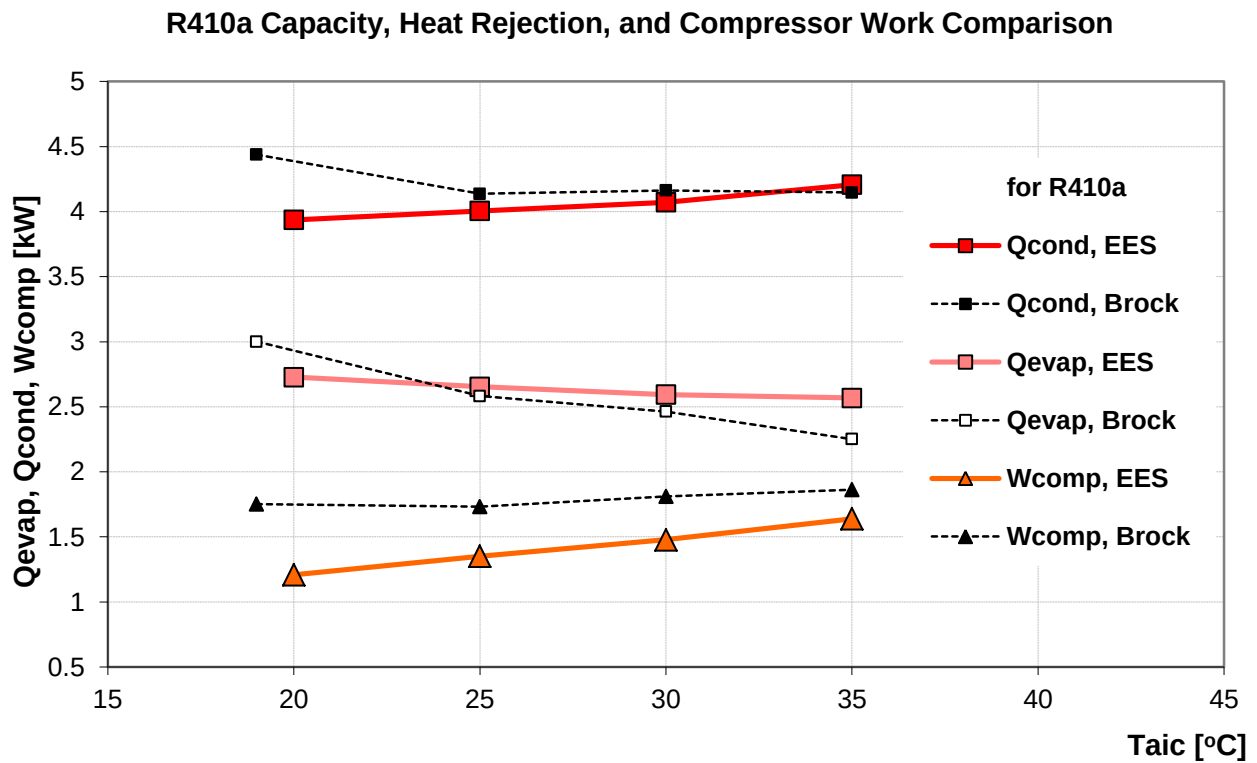
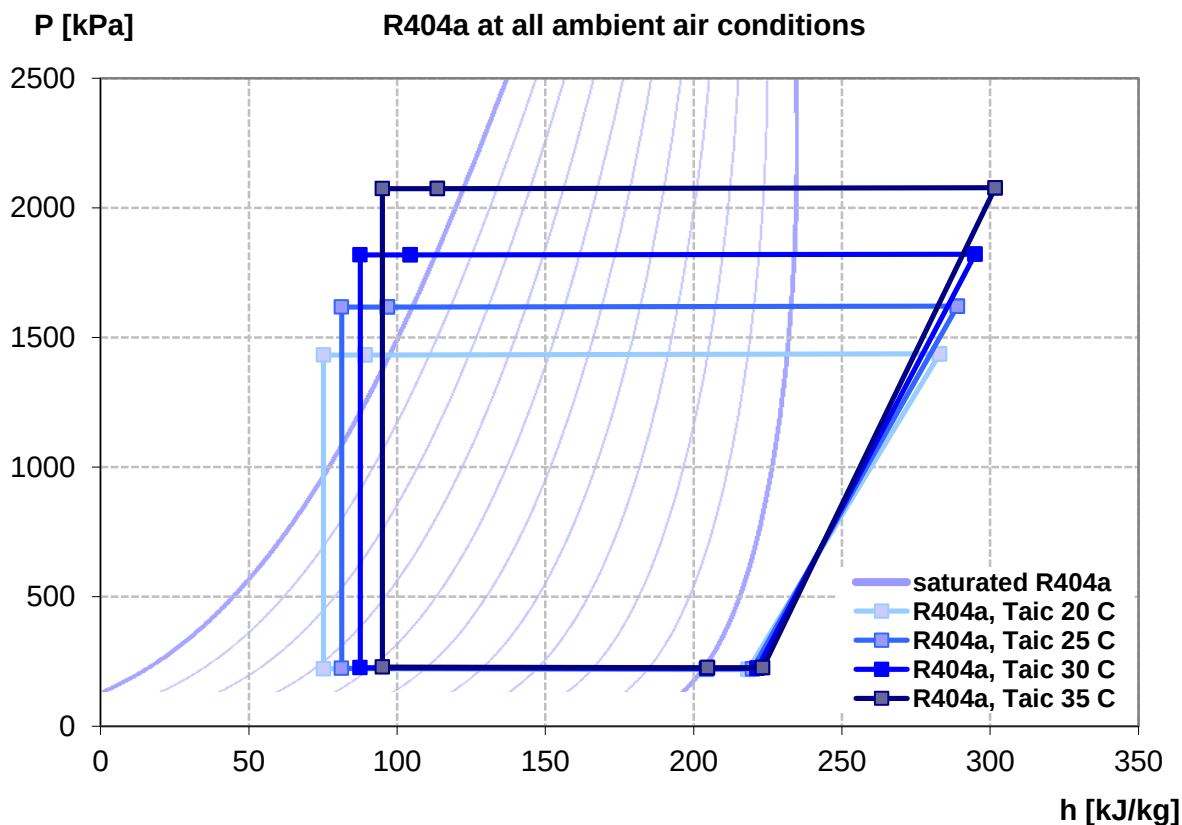


Figure 36 R410A Cycle Performance Quantities

Figures 35 and 36 show that R410a has a steeper reaction to increased Taic trials than R404a, but the overall trends show that the EES model is somewhat reasonable. The biggest discrepancy is in the model and experimental EES values for compressor work W_{comp} , but this is probably due to the difficult nature of accurately representing the bitzer compressor. The template of compressor equations mostly sets the mass flow rate of the cycle but also has inputs for isentropic and volumetric efficiencies. Radko's recorded isentropic compressor efficiencies were approximately 50% and there didn't seem to be any satisfactory way to converge the model consistently with this data. The model convergence and agreement to the experiment was done in part by setting the model isentropic efficiency at an average of 65%.

5.2 EES Model Analysis: R404A vs. R410A

Since the EES model has been validated by comparison to the in-house laboratory setup, the next portion of discussion will make conclusions about R410a performance vs. that of R404a. First is the visual comparison of P-h diagrams for both refrigerants within the EES model, followed by the comparison with T-h diagrams.



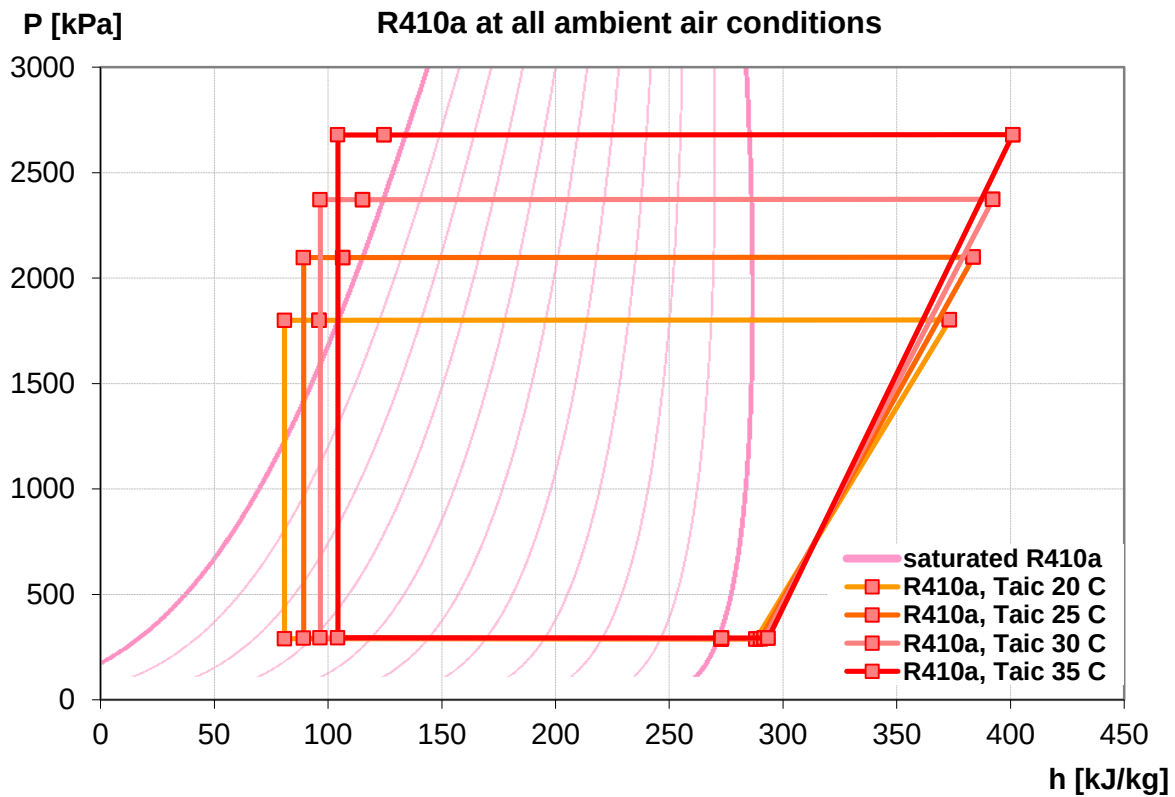


Figure 38 EES R410a cycle P-h performance

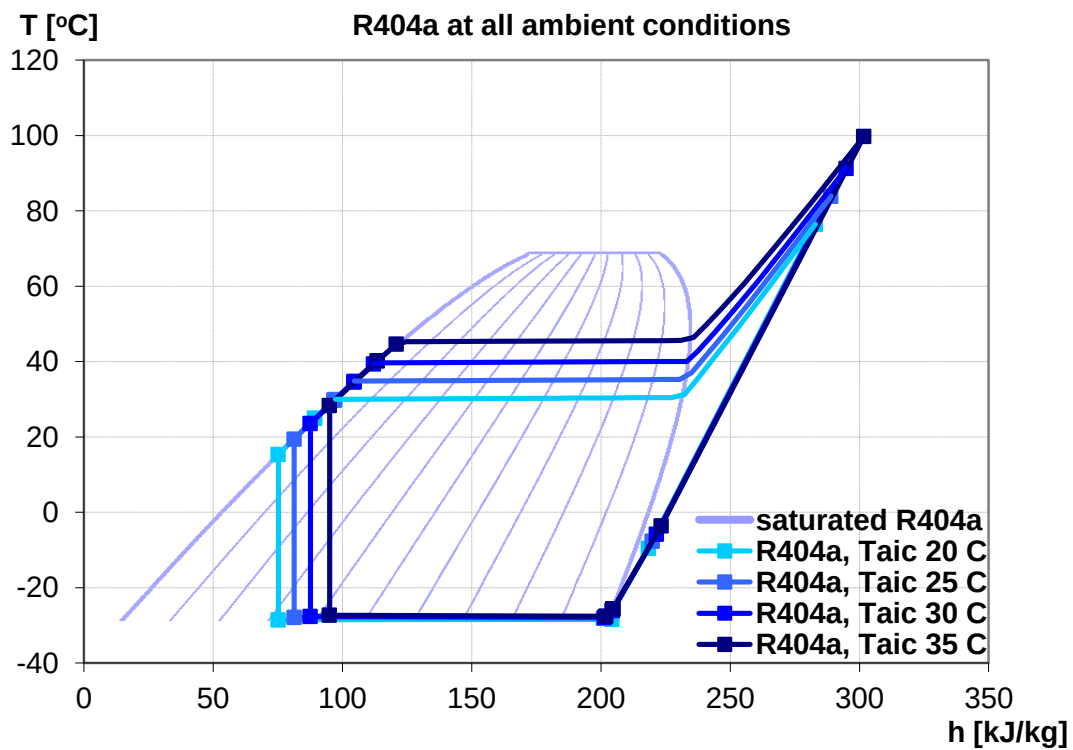


Figure 39 EES R404a cycle T-h performance

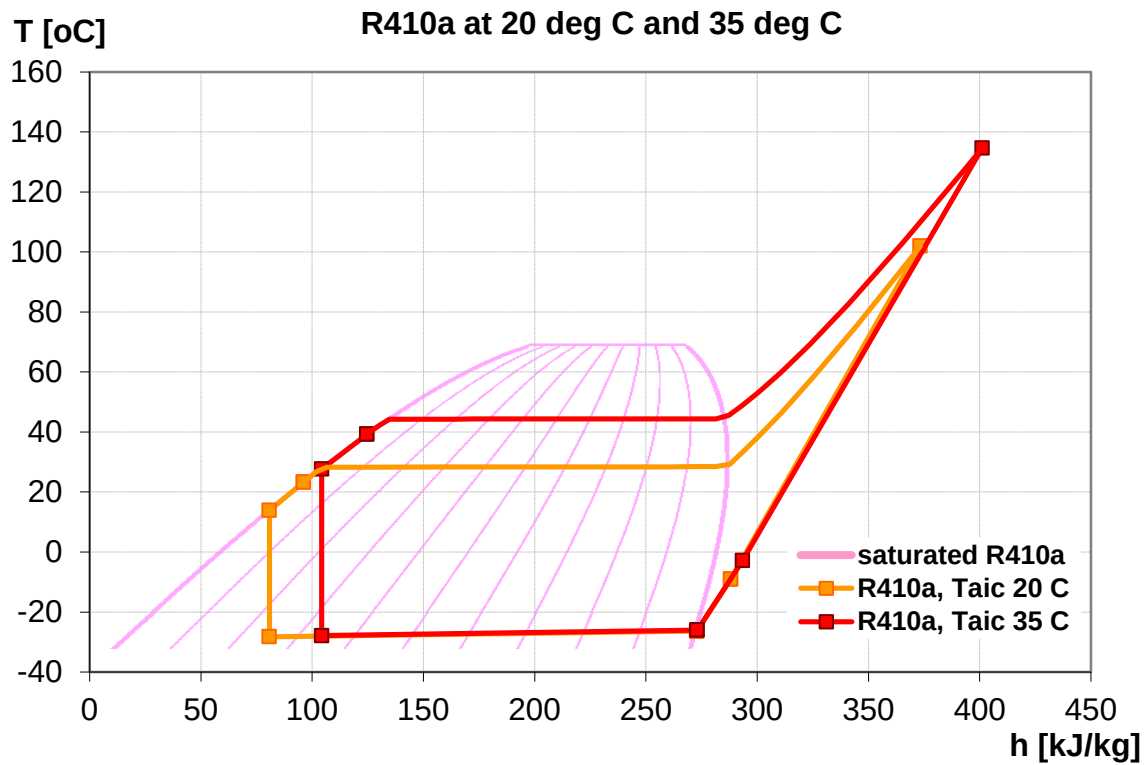


Figure 40 EES R410a cycle T-h performance

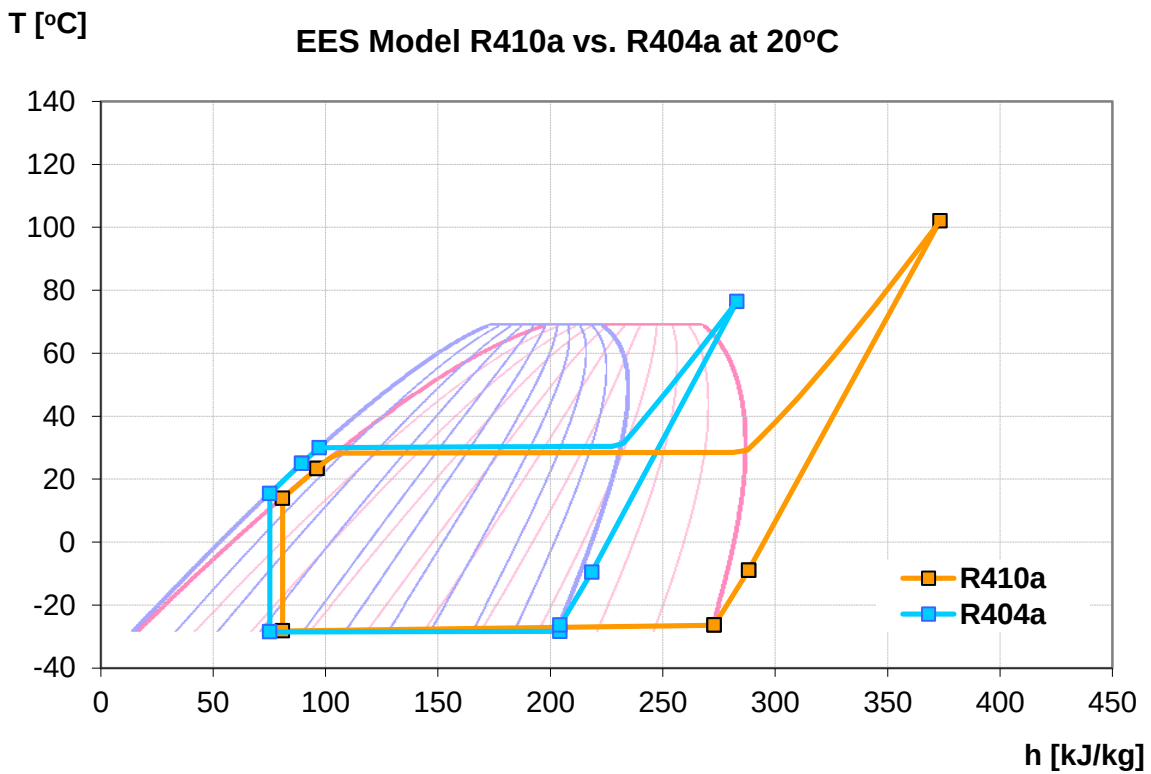


Figure 41 T-h diagram for low Taic

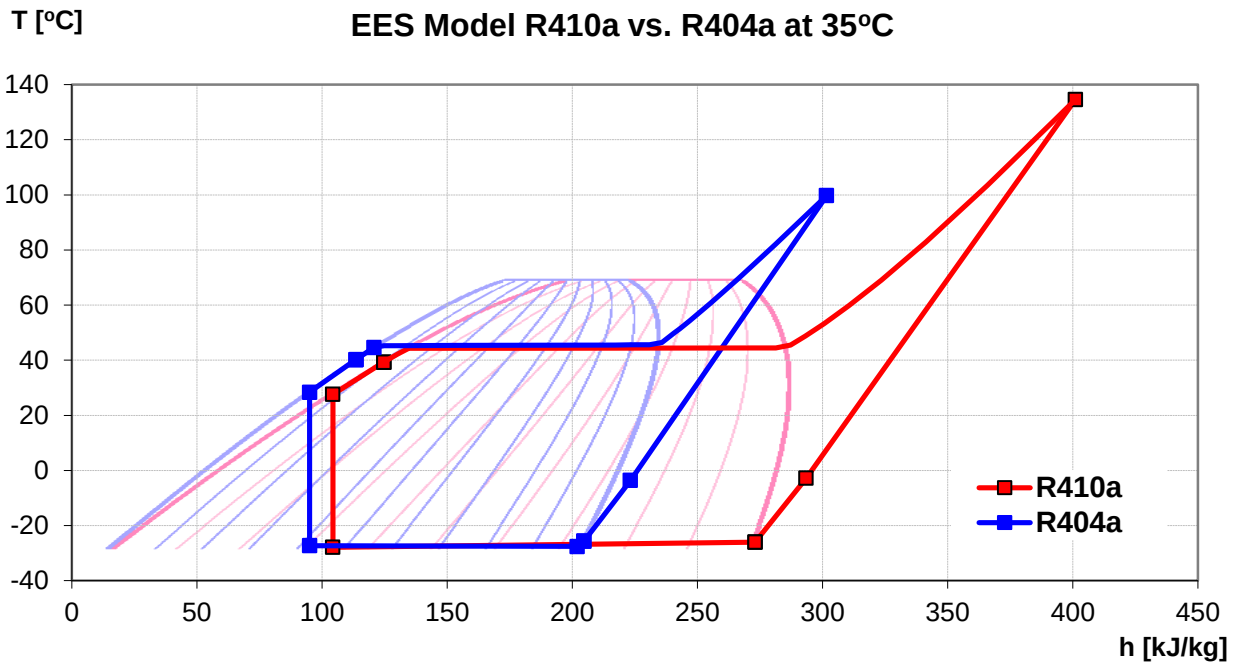


Figure 42 T-h diagram for high $T_{a/c}$

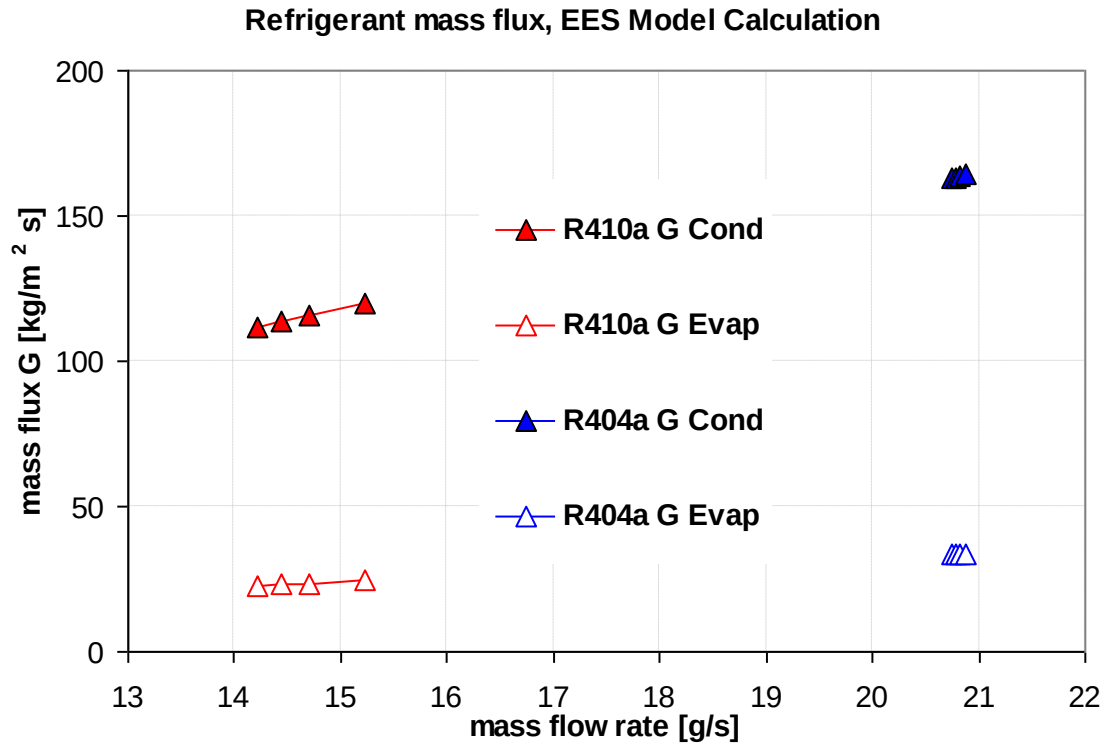


Figure 43 Mass flux in Evaporator and Condenser

The P-h and T-h diagrams of Figures 37 through 42 have essentially confirmed the properties of R410a and R404a. R410a exhibits higher discharge pressures and temperatures than R404a. Figure 43 shows the mass fluxes calculated from the EES model, presented in similar fashion to Brock's experimental results. The left half shows the R410A data for condenser and evaporator portions, while the right half has the R404A data. It shows that the mass flux data are consistently greater for R404a than for R410A, in the evaporator and the condenser. Figure 43 plots the mass flux with varying mass flow rate, but the four mass flow rate points for each trend actually correspond to the four Taic settings from 20-25°C. The EES model values are reasonably close to Radko's graph and are included in table 10.

Table 10 EES mass flux

Refrigerant R404a			
Ta_{ic} (°C)	$\dot{m}$ (kg/s)	G Cond (kg/m² s)	G Evap (kg/m² s)
20	20.74	163	33.24
25	20.78	163.3	33.3
30	20.82	163.6	33.37
35	20.88	164.1	33.47
Refrigerant R410a			
Ta_{ic}	$\dot{m}$	G Cond	G evap
20	14.21	111.7	22.78
25	14.45	113.6	23.17
30	14.71	115.6	23.57
35	15.22	119.6	24.4

Also similarly to Brock's study, the heat exchanger LMTD's and U heat transfer coefficients were calculated using equations 4.2 and 4.3. The EES model produces LMTD trends that are identical to Radko Brock's trends and are in fairly good agreement. The evaporator U value calculations yielded slightly different numbers and a downward trend for both refrigerants. The EES model calculation had a higher U for R410a than for R404a, however, consistent with the refrigerant properties which cause R410a to have a higher heat transfer coefficient. The LMTD and evaporator U plots are in figures 44 and 45 below.

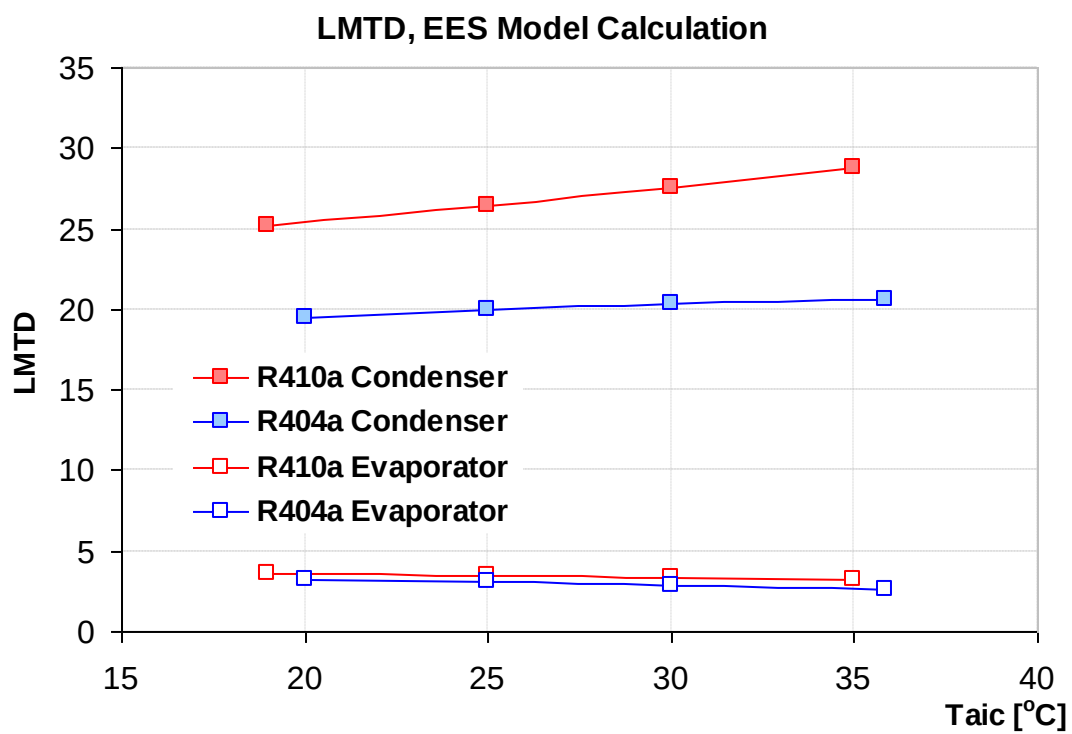


Figure 44 LMTD for condenser and evaporator

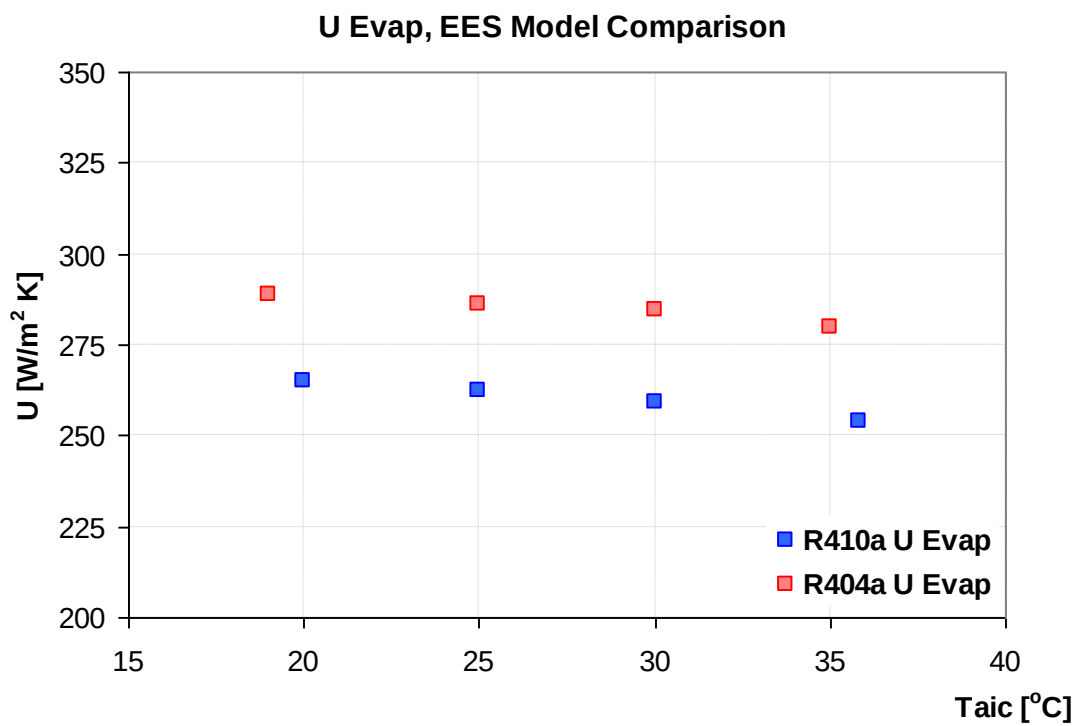


Figure 45 Evaporator U calculation

Using the EES Model as a predictive design tool

Since the validity of the EES model has been confirmed by comparison to Brock's study, it is desirable to show that the EES system model can be used to provide insights into system performance if the geometry is altered. In other words, the model is useful for theoretical conclusions about equipment that isn't available, and can act as a tool for design considerations. Figure 46 shows one such consideration applied to this simulation study. To generate this data, the system model was run for R-410A at a Taic of 25°C, but the evaporator inner tube diameter was varied from 5mm to 12 mm (the lab setup diameter was 6.5 mm).

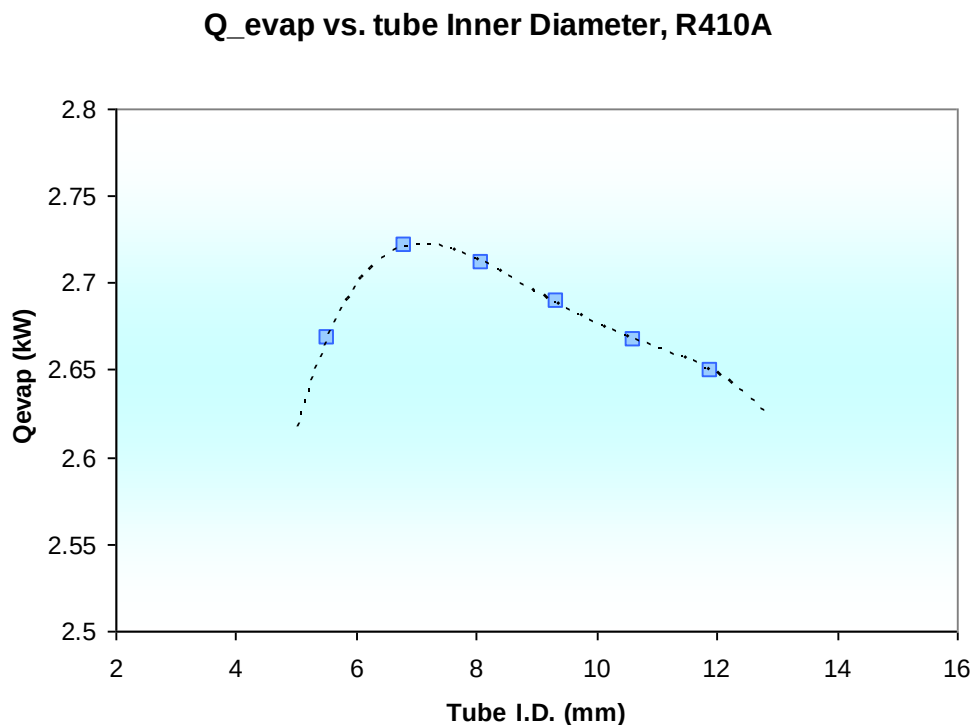


Figure 46 Effect of evaporator tube thickness on Q_evap

From right to left the figure shows the decrease in Evaporator capacity from a maximum of ~2.725 kW to 2.65 kW. An increase in the evaporator pipe inner diameter beyond this maximum, and certainly beyond 12 mm, is unhelpful to the refrigerant capacity performance. On the other hand, decreasing the tube inner diameter beyond the 5 mm mark is not practical either. This is explained by the graph of evaporator pressure drop with evaporator tube inner

diameter, shown in Figure 47. The graph shows that as the tube diameter decreases, the pressure drop in the current system conditions increases exponentially.

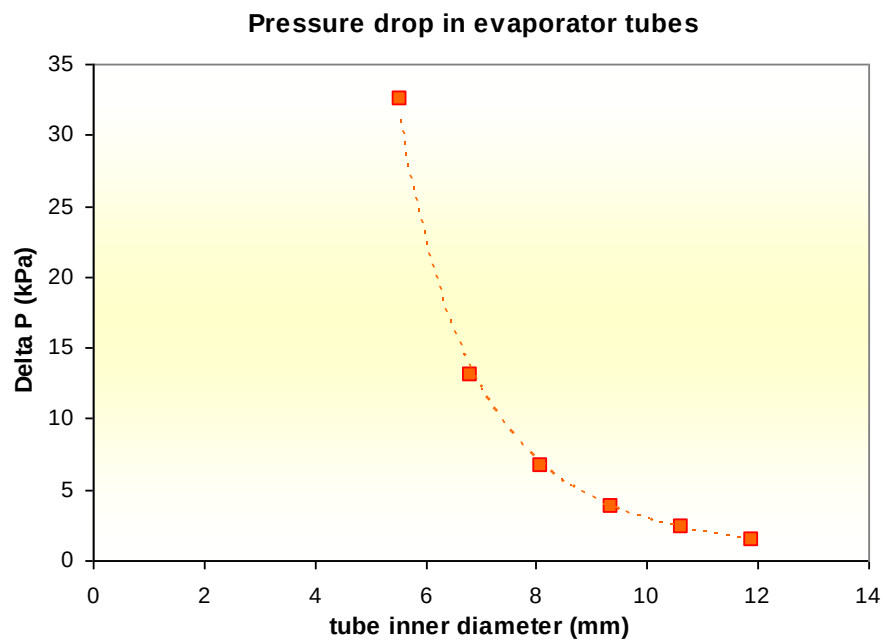


Figure 47 Pressure drop limits for evaporator tube diameters

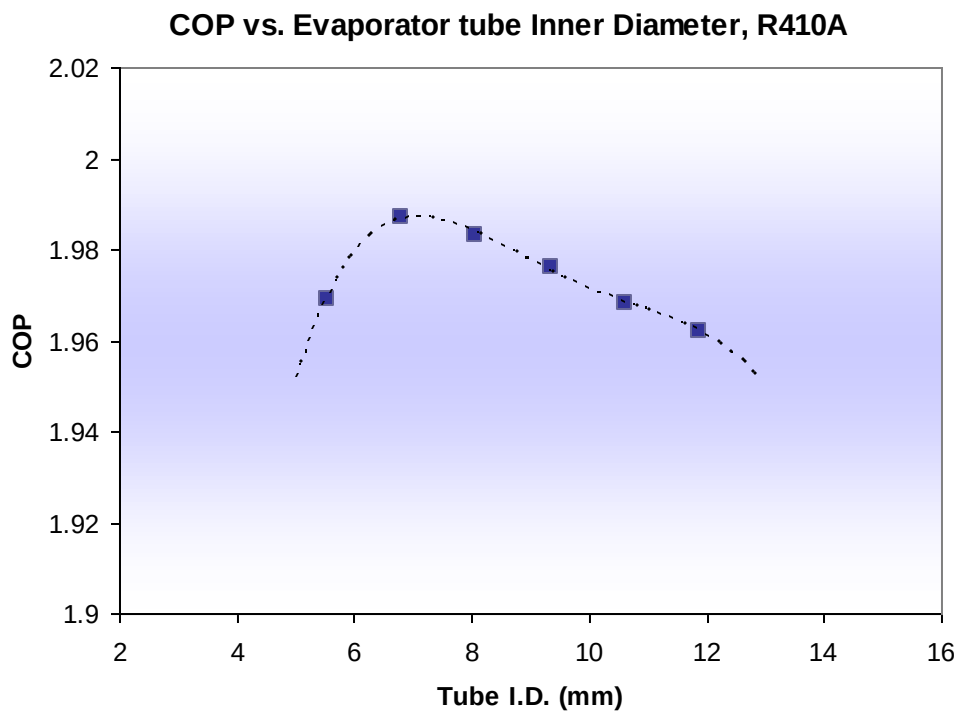


Figure 48 Effect of evaporator tube thickness on COP

Continuing left beyond the 5 mm diameter mark, the pressure losses are so drastic that it is obviously unreasonable to force the refrigerant through such a system geometry. This simulation of new tube geometry for the evaporator yields a COP trend that is nearly identical to that of Qevap, as shown above in Figure 48.

6 Conclusion

The use of the EES model as a comparison to laboratory data and also as a predictive tool has been documented in the preceding chapters. Numerous parameter tabulations showed the viability of the EES heat exchanger simulation models to represent actual or theoretical system geometry. Thermodynamic performance trends of R410a and R404a were exhibited in a variety of standard representations such as P-h and T-h diagrams. The results confirmed known cycle performance behaviors of R410a and R404a in a specific Hussmann display case low temperature application. Refrigerant R410a generally has higher heat transfer coefficients and higher LMTDs than R404a and thus yields more effective heat transfer in the heat exchangers. Yet, as shown, R410a also must be compressed to greater discharge temperature and pressure before the condensation stage. The greater heat capacity of refrigerant R410a allowed it to operate at lower mass flux than R404a and exhibit a higher U value for the heat exchangers. The predictive ability of the model also illustrated the possibility of using the ACRC heat exchanger simulation models in a variety of design efforts.

Appendix A

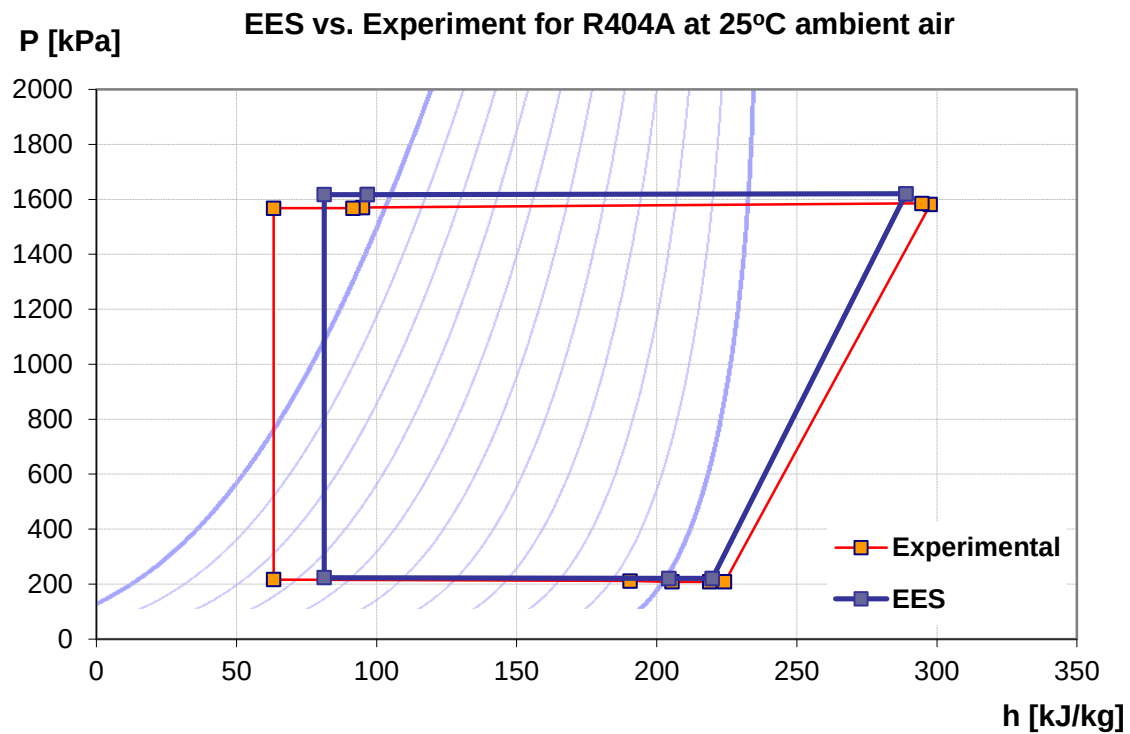


Figure 49 P-h diagram for R404a at 25 C, model and experimental comparison

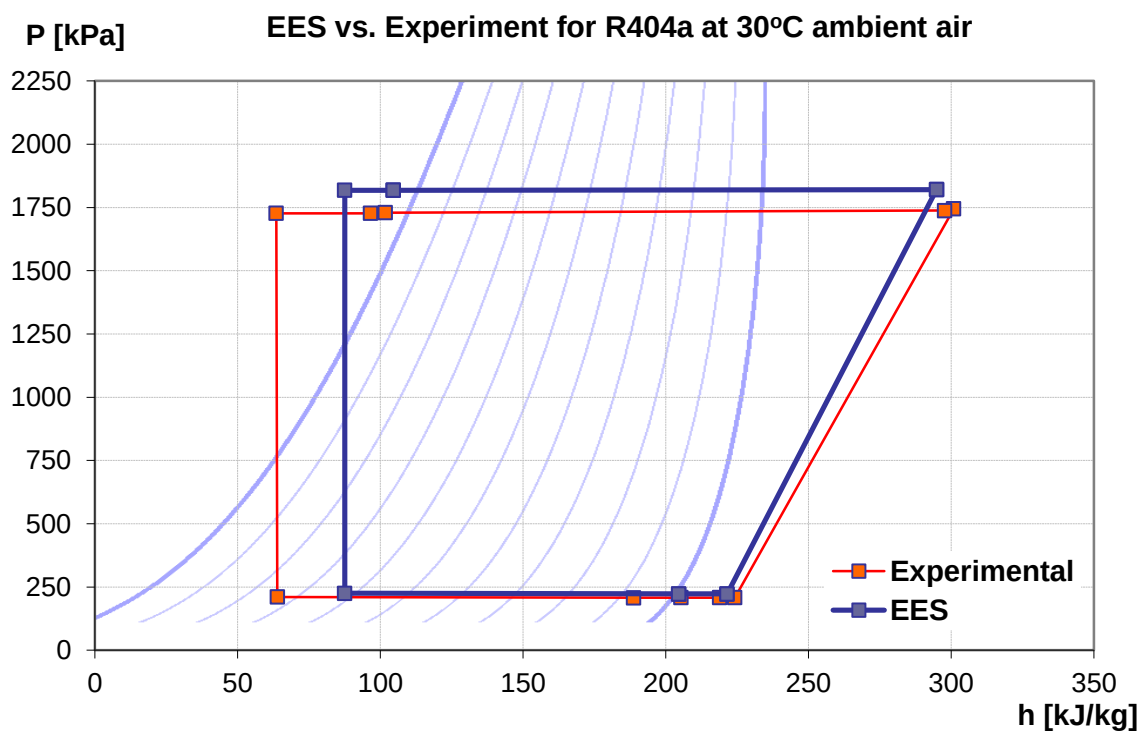


Figure 50 P-h diagram for R404a at 30 C, model and experimental comparison

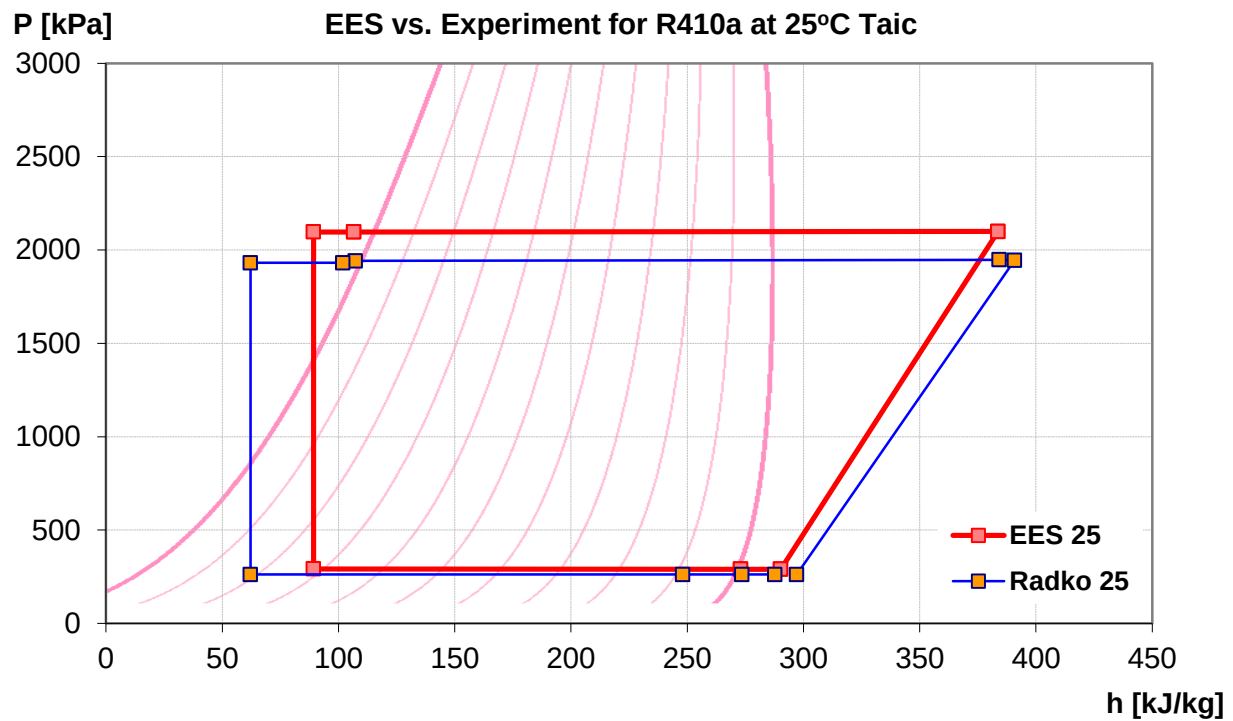


Figure 51 P-h diagram for R410a at 25 C, model and experimental comparison

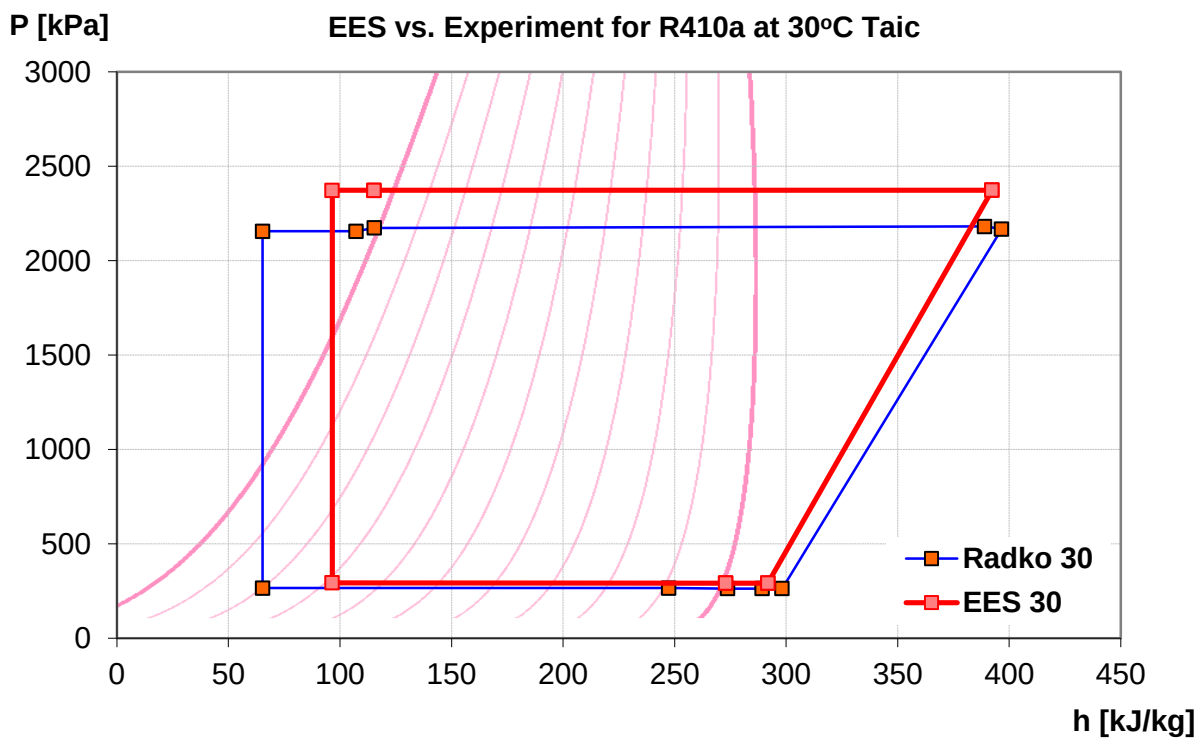


Figure 52 P-h diagram for R410a at 30 C, model and experimental comparison

Table 11 R404a pressure enthalpy temperature data

R404A MODEL P, h, T data				
Temp Taic	20	25	30	35
T actual	19	25	30	35
h (kJ/kg)				
h suc	218.4	219.9	221.5	223.3
h dis	283	289	294.9	301.7
hr in cond	283	289	294.9	301.7
hr out cond	89.31	96.81	104.5	113.6
h h in IHX	89.31	96.81	104.5	113.6
h h out IHX	75.19	81.31	87.55	95.03
hr in evap	75.19	81.31	87.55	95.03
hr out evap	204.3	204.4	204.6	204.7
hc in IHX	204.3	204.4	204.6	204.7
hc out IHX	218.4	219.9	221.5	223.3
P (kPa)				
P suc	218.5	220.7	222.9	225.7
P dis	1436	1620	1821	2077
P r in cond	1436	1620	1821	2077
P r out cond	1432	1617	1818	2074
P h in IHX	1432	1617	1818	2074
P h out IHX	1432	1617	1818	2074
P r in evap	221.1	223.3	225.6	228.4
P r out evap	218.5	220.7	223	225.7
P c in IHX	218.5	220.7	223	225.7
P c out IHX	218.5	220.7	222.9	225.7
T (deg C)				
T suc	-9.57	-7.702	-5.817	-3.594
T dis	76.45	83.92	91.29	99.76
T r in cond	76.45	83.92	91.29	99.76
T r out cond	24.99	29.83	34.65	40.24
T h in IHX	24.99	29.83	34.65	40.24
T h out IHX	15.39	19.47	23.57	28.38
T r in evap	-28.55	-27.84	-27.59	-27.28
T r out evap	-28.4	-26.13	-25.88	-25.58
T c in IHX	-26.38	-26.13	-25.88	-25.58
T c out IHX	-9.57	-7.702	-5.817	-3.594

Table 12 R410a pressure enthalpy temperature data

R410A MODEL P, h, T data				
Temp Taic	20	25	30	35
T actual	19	25	30	35
h (kJ/kg)				
h suc	288.2	290.2	291.8	293.5
h dis	373.3	383.7	392.3	401.1
hr in cond	373.3	383.7	392.3	401.1
hr out cond	96.15	106.5	115.3	124.7
h h in IHX	96.15	106.5	115.3	124.7
h h out IHX	80.84	89.26	96.53	104.3
hr in evap	80.84	89.26	96.53	104.3
hr out evap	272.9	273	273	273.1
hc in IHX	272.9	273	273	273.1
hc out IHX	288.2	290.2	291.8	293.5
P (kPa)				
P suc	288.5	290.4	292.1	293.1
P dis	1802	2099	2373	2680
P r in cond	1802	2099	2373	2680
P r out cond	1800	2097	2372	2679
P h in IHX	1800	2097	2372	2679
P h out IHX	1800	2097	2372	2679
P r in evap	290.1	292.1	293.8	294.9
P r out evap	288.5	290.5	292.1	293.1
P c in IHX	288.5	290.5	292.1	293.1
P c out IHX	288.5	290.4	292.1	293.1
T (deg C)				
T suc	-8.956	-6.633	-4.656	-2.773
T dis	102	114.2	124.3	134.6
T r in cond	102	114.2	124.3	134.6
T r out cond	23.3	29.26	34.23	39.29
T h in IHX	23.3	29.26	34.23	39.29
T h out IHX	13.89	18.94	18.81	27.66
T r in evap	-28.25	18.94	-28.01	-27.84
T r out evap	-26.39	-26.22	-26.08	-26
T c in IHX	-26.39	-26.22	-26.15	-26
T c out IHX	-8.956	-6.633	-6.594	-2.773

Table 13 R404a heat exchanger performance quantities

Refrigerant R404A Data Comparison, EES and Experiment			
Q_evap EES vs. Experimental refrigerant side (kW)			
Ta _{ic}	EES	Experiment	Error Q_evap
20	2.709	2.937	8.404
25	2.556	2.901	13.488
30	2.434	2.886	18.554
35	2.289	2.762	20.679
Q_evap EES vs. Experimental air side (kW)			
Ta _{ic}	EES	Experiment	Error Q_evap
20	2.709	3.060	12.969
25	2.556	2.810	9.947
30	2.434	2.534	4.125
35	2.289	2.416	5.534
Q_cond EES vs. Experimental air side (kW)			
Ta _{ic}	EES	Experiment	Error Q_evap
20	4.021	4.234	5.289
25	3.991	4.094	2.587
30	3.963	3.972	0.222
35	3.925	3.912	0.323
Q_cond EES vs. Experimental refrigerant side (kW)			
Ta _{ic}	EES	Experiment	Error Q_cond
20	4.021	3.951333	1.733
25	3.991	4.12	3.232
30	3.963	3.9482	0.373
35	3.925	3.868667	1.435
Compressor Work W _{cp} (kW)			
Ta _{ic}	EES	Experiment	Error W _{cp}
20	1.312	1.774	35.188
25	1.435	1.827	27.282
30	1.529	1.845	20.693
35	1.636	1.918	17.237
COP			
Ta _{ic}	EES	Experiment	Error h _{in} _evap
20	2.064	1.653	19.929
25	1.781	1.607	9.770
30	1.592	1.535	3.555
35	1.399	1.452	3.812

Table 14 R410a heat exchanger performance quantities

Refrigerant R410A Data Comparison, EES and Experiment			
Q_evap EES vs. Experimental refrigerant side (kW)			
Ta_{ic}	EES	Experiment	Error Q_evap
20	2.728	3.008	10.264
25	2.654	2.848	7.319
30	2.594	2.827	8.982
35	2.568	2.789	8.586
Q_evap EES vs. Experimental air side (kW)			
Ta_{ic}	EES	Experiment	Error Q_evap
20	2.728	3.000	9.985
25	2.654	2.582	2.713
30	2.594	2.463	5.050
35	2.568	2.252	12.305
Q_cond EES vs. Experimental air side (kW)			
Ta_{ic}	EES	Experiment	Error Q_cond
20	3.937	4.438	12.731
25	4.005	4.137	3.302
30	4.072	4.162	2.220
35	4.207	4.147	1.426
Q_cond EES vs. Experimental refrigerant side (kW)			
Ta_{ic}	EES	Experiment	Error Q_cond
20	3.937	4.115	4.521
25	4.005	3.6205	9.600
30	4.072	3.6746	9.759
35	4.207	3.642	13.430
Compressor Work Wcp (kW)			
Ta_{ic}	EES	Experiment	Error Wcp
20	1.209	1.753	44.996
25	1.351	1.733	28.257
30	1.478	1.812	22.571
35	1.639	1.863	13.667
COP			
Ta_{ic}	EES	Experiment	Error h_{in}_evap
20	2.256	1.863	17.402
25	1.964	1.666	15.160
30	1.756	1.586	9.704
35	1.567	1.482	5.424

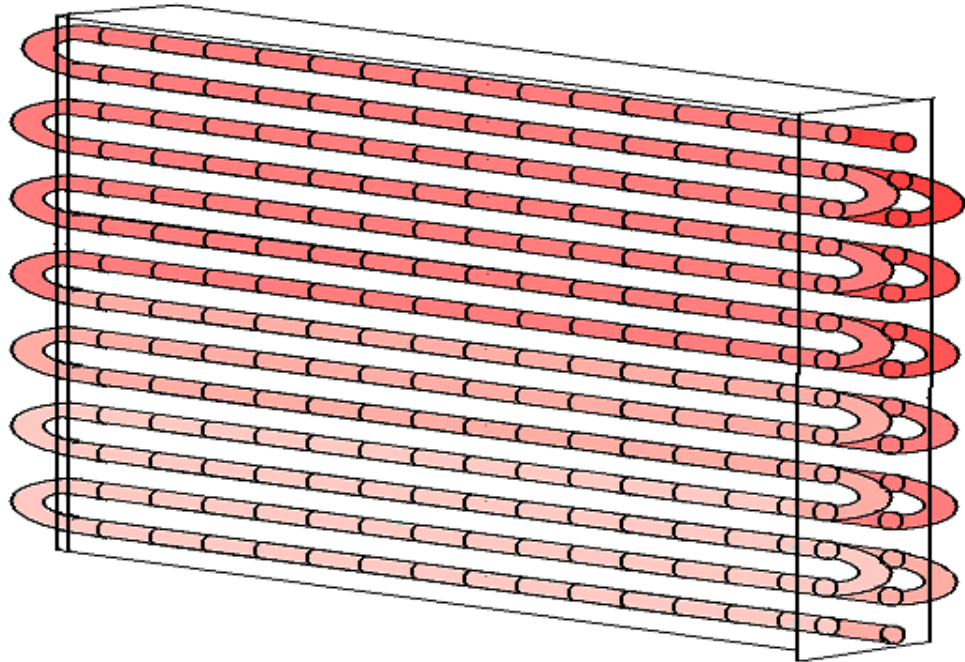


Figure 53 Simple condenser schematic

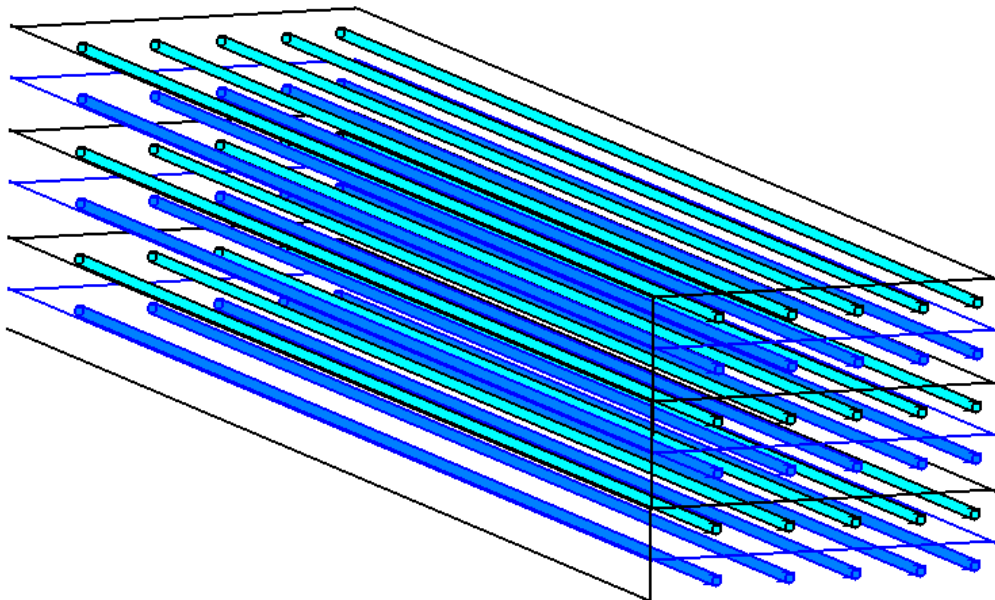


Figure 54 Simple evaporator schematic

Appendix B: The EES Model Code

The EES model code in this section does not contain the entire functioning code because it would add too many pages, and some it is repetitive in nature.

The contents of the EES code that are included will be as follows:

- 1) Compressor Procedure
 - State calculation
 - Compressor effectiveness calculation
- 2) Condenser Procedure

- State calculation
 - Superheated length

- Air side heat transfer correlation
 - Air side pressure drop correlation

Geometry

- Area_calc_condenser
 - Geometrycalfinontubecond
 - Areacalfinontubecond
 - Elementareacalcond

Element Heat Transfer

- Twosub_cond_down
 - Two_phase_cond_down
 - Singlephase_cond_down
 - Super2ph_cond_down

- Choosecondenser
 - Condcal

- 3) Main program with inputs for Condenser, Internal Heat Exchanger, and Evaporator

The individual codes are not included for the internal heat exchanger or evaporator because they follow essentially the same processes as the condenser code.

```
#####COMPRESSOR PROCEDURE#####
```

```
Procedure State_Calc_Comp(fluidname$, P, h : T, x)
```

```
P_crit = P_CRIT(fluidname$)
```

```
if (P>=P_crit) then
```

```
    T = Temperature(fluidname$, P=P, h=h)
```

```
    x = 500
```

```
else
```

```
    h_l = ENTHALPY(fluidname$, P=P, x=0)
```

```
    h_v = ENTHALPY(fluidname$, P=P, x=1)
```

```
    x = (h - h_l)/(h_v - h_l)
```

```
    if ((h<=h_v) AND (h >= h_l)) then
```

```
        T = Temperature(fluidname$, P=P, x=1)
```

```
    else
```

```
        T = Temperature(fluidname$, P=P, h=h)
```

```
    endif
```

```
endif
```

```
end
```

```
Procedure CompCal_eff(fluidname$, sumpside$, h_suc, P_suc, P_dis, m_dot_r_comp, U_comp, A_comp, V_comp, M_oil,  
T_outdoor, T_shell_coeff0, T_shell_coeff1, eta_isen, eta_vol, V_dot_disp : W_dot_comp, h_dis, T_shell, qcomp, Mass_comp,  
Massfrac)
```

```
s_suc=ENTROPY(fluidname$,P=P_suc, h=h_suc)
```

```
"Compressor suction entropy, kJ/kg-K"
```

```
T_suc = Temperature(fluidname$, P=P_suc, h=h_suc)
```

```
h_dis_isentropic=ENTHALPY(fluidname$, P=P_dis, s=s_suc)
```

```
W_dot_comp=(h_dis_isentropic-h_suc)/eta_isen*m_dot_r_comp
```

```
"Iterations begins to determine the compressor discharge temperature; Secant method"
```

```
h_dis_test=(W_dot_comp)/m_dot_r_comp+h_suc
```

```
T_dis_test = Temperature(fluidname$, P=P_dis, h=h_dis_test)
```

```
T_dis[1] = T_dis_test +3
```

```
T_dis[2] = T_dis_test +1
```

```
T_dis_F=CONVERTTEMP(C,F,T_dis[1])
```

```
T_shell_F = T_shell_coeff0 + T_shell_coeff1*T_dis_F
```

```
T_shell=CONVERTTEMP(F,C,T_shell_F)
```

```
qcomp = U_comp*A_comp*(T_shell-T_outdoor)
```

```
h_dis = Enthalpy(fluidname$,T=T_dis[1],P=P_dis)
```

```
h_dis_new=(W_dot_comp-qcomp)/m_dot_r_comp+h_suc
```

```
value[1] = h_dis - h_dis_new
```

```
iter = 0
```

```
Repeat
```

```
T_dis_F=CONVERTTEMP(C,F,T_dis[2])
```

```
T_shell_F = T_shell_coeff0 + T_shell_coeff1*T_dis_F
```

```
T_shell=CONVERTTEMP(F,C,T_shell_F)
```

```
qcomp= U_comp*A_comp*(T_shell-T_outdoor)
```

```
h_dis = Enthalpy(fluidname$,T=T_dis[2],P=P_dis)
```

```
h_dis_new=(W_dot_comp-qcomp)/m_dot_r_comp+h_suc
```

```
value[2] = h_dis - h_dis_new
```

```
T_dis_temp = T_dis[2]
```

```
T_dis[2] = T_dis_temp - value[2]*(T_dis[1] - T_dis_temp)/(value[1] - value[2])
```

```
T_dis[1] = T_dis_temp
```

```
value[1] = value[2]
```

```
iter = iter + 1
```

```
Until(ABS(value[1])<0.00001)
```

```
T_dis = T_dis[1]
```

```
"Compressor charge calculation"
```

```
If (sumpside$='highs') Then
```

```
    Mass_comp=V_comp*DENSITY(fluidname$,T=T_shell,P=P_dis)
```

```
    P_actual = P_dis
```

```
else
```

```
    Mass_comp=V_comp*max(DENSITY(fluidname$,T=T_shell,P=P_suc), DENSITY(fluidname$,P=P_suc, x=1))
```

```
    P_actual = P_suc
```

```
endif
```

```
{Massfrac=0 "Mass fraction of refrigerant in the oil, calculated from oil/refrigerant solubility curve fit, only R22 , R410A, R134a and R404A data is available"}
```

```
"The Solubility and Viscosity of Solutions of HCFC-22 in Naphthenic Oil and in Alkylbenzene at High Pressures and Temperatures, N.A. Van Gaalen, S.C. Zoz, M.B. Pate, ASHRAE Transactions, 1991, pt1, pp101-108"
if (fluidname$='R22') Then
```

```
endif
```

```
"ACRC TR-68 Refrigerant-oil Mixtures and Local Comp. Modeling by Martz and Jacobi, Oil is EMKARATE RL68H, polyol ester"
if (fluidname$='R410A') Then
```

```
Refrigerant$ = 'found'
T=(T_shell+273.15)/293.15
M0=-77.67070771
M1=131.85685
M2=-57.016806
M3=103.15085
M4=-171.68531
M5=72.750827
M6=1.1302177
M7=-1.718839
M8=0.65249827
```

```
Massfrac[1] = 0.4
Massfrac[2] = 0.95*Massfrac[1]
P_new=(M0*Massfrac[1]^2*T^2+M1*Massfrac[1]^2*T+M2*Massfrac[1]^2+M3*Massfrac[1]*T^2+M4*Massfrac[1]*T+M5*Massfrac[1]+M6*T^2+M7*T+M8)*1000
value[1] = P_actual - P_new
```

```
iter = 2
repeat
P_new=(M0*Massfrac[2]^2*T^2+M1*Massfrac[2]^2*T+M2*Massfrac[2]^2+M3*Massfrac[2]*T^2+M4*Massfrac[2]*T+M5*Massfrac[2]+M6*T^2+M7*T+M8)*1000
value[2] = P_actual - P_new
Massfrac_temp = Massfrac[2]
Massfrac[2] = Massfrac[2] - value[2]*(Massfrac[1] - Massfrac[2])/(value[1] - value[2])
Massfrac[1] = Massfrac_temp
value[1] = value[2]
iter = iter + 1
Until (abs(value[1]) < 0.000001)
Massfrac = Massfrac[1]
endif
```

```
"Measurement of viscosity, density, and gas solubility of refrigerant blends in selected synthetic lubricants, Richard C. Cavestri, ASHRAE handbook 2002, the lubricant is EMKARATE RL32S, polyol ester, Data fitted with a 4th order curve in to variables (temperature and percentage massfraction)"
if (fluidname$='R134a') Then
endif
```

```
"Measurement of viscosity, density, and gas solubility of refrigerant blends in selected synthetic lubricants, Richard C. Cavestri, ASHRAE handbook 2002, the lubricant is EMKARATE RL32S, polyol ester, Data fitted with a 4th order curve in to variables (temperature and percentage massfraction)"
if (fluidname$='R404A') Then
```

```
endif
```

```
if (Refrigerant$ <> 'found') then
Massfrac = 0
endif
```

```
if (Massfrac < 0) then
Massfrac = 0
endif
if (Massfrac > 1) then
Massfrac = 1
endif
```

```
Mass_comp=Mass_comp+Massfrac*M_oil
```

```
end
```

```
#####CONDENSER PROCEDURE #####
```

```
Procedure STATE_CALC_COND(fluidname$, P_r_cond, h_r_cond : T_r_cond, x_r_cond)
```

```
P_crit = P_CRIT(fluidname$)
```

```
if (P_r_cond > P_crit) then
```

```
T_r_cond = TEMPERATURE(fluidname$, P=P_r_cond, h=h_r_cond)
```

```
x_r_cond = 500
```

```
else
```

```
h_v = ENTHALPY(fluidname$, P=P_r_cond, x=1)
```

```
h_l = ENTHALPY(fluidname$, P=P_r_cond, x=0)
```

```
if ((h_r_cond <= h_v) AND (h_r_cond >= h_l)) then
```

```
T_r_cond = TEMPERATURE(fluidname$, P=P_r_cond, x=1)
```

```
else
```

```
T_r_cond = TEMPERATURE(fluidname$, P=P_r_cond, h=h_r_cond)
```

```
endif
```

```
x_r_cond = (h_r_cond - h_l)/(h_v - h_l)
```

```
endif
```

```
end
```

{This procedure calculates the length of superheated region for condenser using the Secant method. It is called when splitting of a transition element between superheated and two-phase region is required}

```
Procedure LENGTH_SUPERHEATED(Q_sup, Tai, Tri, UA, Cair, Cref:Cair_new)
```

```
Cmin = MIN(Cair, Cref)
```

```
Cmax = MAX(Cair, Cref)
```

```
NTU = UA/Cmin
```

```
Cair2 = Cair
```

```
Cair1 = 0.95*Cair
```

```
Cmin1 = MIN(Cair1,Cref)
```

```
Cmax1 = MAX(Cair1, Cref)
```

```
Cr1 = Cmin1/Cmax1
```

```
NTU1 = UA/Cmin1*(Cair1/Cair)
```

{Cair1/Cair is the ratio of superheated element area to the whole

element area}

```
epsilon1 = 1 - EXP((1/Cr1)*NTU1^(0.22)*(EXP(-Cr1*NTU1^(0.78)) - 1))
```

```
Ccalc1 = Q_sup/(epsilon1*(Tri - Tai))
```

```
Q_sup_new = Cmin1*epsilon1*(Tri - Tai)
```

```
value1 = Q_sup_new - Q_sup
```

```
iter = 0
```

```
repeat
```

```
Cmin2 = MIN(Cair2,Cref)
```

```
Cmax2 = MAX(Cair2, Cref)
```

```
Cr2 = Cmin2/Cmax2
```

```
NTU2 = UA/Cmin2*(Cair2/Cair)
```

```
epsilon2 = 1 - EXP((1/Cr2)*NTU2^(0.22)*(EXP(-Cr2*NTU2^(0.78)) - 1))
```

```
Ccalc2 = Q_sup/(epsilon2*(Tri - Tai))
```

```
Q_sup_new = Cmin2*epsilon2*(Tri - Tai)
```

```
value2 = Q_sup_new - Q_sup
```

```
Cair_temp = Cair2
```

```
Cair2 = Cair_temp - value2*(Cair1 - Cair_temp)/(value1 - value2)
```

```
Cair1 = Cair_temp
```

```
value1 = value2
```

```
iter = iter + 1
```

```
Until(ABS(value2)<0.0000001)
```

```
Cair_new = Cair2
```

```
end
```

```

Procedure HTC_AIR_SIDE_COND(P_atm, T_a_in_Elem, A_finDIVair_cond, Gair_max, D_outer_cond, D_collar_cond, Dh_cond,
theta_Louver_cond, pitch_Louver_cond, pitch_fin_cond, depth_cond, length_Louver_cond, pitch_MCT_cond, height_Louver_cond,
K_fin_cond, thick_fin_cond, height_fin_cond, S_long_cond, S_trans_cond, N_row_avg_cond : hair_Elem, Eff_Sur_Elem,
Eff_Fin_Elem)
$Common cond_type$

```

```

Call Prop_Air_Dry(T_a_in_Elem, P_atm : cp_air, mu_air, rho_air, Pr_air)

```

```

if (COND_TYPE$ = 'MicroCh') then

```

```

endif

```

```

if (COND_TYPE$ = 'TubeFin') then

```

```

    Call RE_AIR(mu_air, D_collar_cond, Gair_max: Re_Dc)
    Call h_plain_Wang_2000(D_collar_cond, S_long_cond, S_trans_cond, N_row_avg_cond, pitch_fin_cond, Dh_cond,
Re_Dc, Pr_air, Gair_max, cp_air : hair_Elem)
    {
        Call H_LOUVER_WANG_1999( Re_Dc, height_Louver_cond, pitch_Louver_cond, S_trans_cond, S_long_cond,
Dh_cond, D_collar_cond, pitch_fin_cond, N_row_avg_cond, Pr_air, Gair_max, cp_air: hair_Elem)}
    {Call Re_Air(mu_air, 2*pitch_fin_cond, Gair_max: Re_2Fs)
    Call h_wavy_Mirth_1990(Re_2Fs, S_TRANS_cond, pitch_fin_cond, D_collar_cond, depth_cond, Pr_air, Gair_max,
cp_air: hair_Elem)}

```

```

    Call Eff_surface_circular(hair_Elem, S_trans_cond, S_long_cond, K_fin_cond, thick_fin_cond, D_outer_cond,
A_finDIVair_cond : Eff_Sur_Elem, Eff_Fin_Elem)
endif

```

```

end

```

```

Procedure DP_AIR_SIDE_COND(P_atm, Gair_max, T_a_in_slab, T_a_out_slab, m_dot_a_cond, A_ffield_cond, A_front_cond,
A_a_cond, A_fin_cond, D_collar_cond, Dh_cond, theta_Louver_cond, pitch_Louver_cond, pitch_fin_cond, depth_cond,
length_Louver_cond, pitch_MCT_cond, height_Louver_cond, thick_fin_cond, height_fin_cond, S_long_cond, S_trans_cond,
N_row_avg_cond: DP_air_slab)
$Common cond_type$

```

```

Call Prop_Air_Dry(T_a_in_slab, P_atm : cp_air, mu_air, rho_air, Pr_air)

```

```

if (COND_TYPE$ = 'TubeFin') then

```

```

    Call RE_AIR(mu_air, D_collar_cond, Gair_max: Re_Dc)
    A_tube_cond = A_a_cond - A_fin_cond
    Call Dp_plain_Wang_2000 (D_collar_cond, S_long_cond, S_trans_cond, N_row_avg_cond, pitch_fin_cond, Re_Dc, T_a_in_slab,
T_a_out_slab, m_dot_a_cond, A_ffield_cond, A_front_cond, A_a_cond, P_atm : DP_air_slab)

```

```

    {Call DP_LOUVER_WANG_1999}
    {Call Dp_wavy_Mirth_1990_coil1}

```

```

endif

```

```

if (COND_TYPE$ = 'MicroCh') then

```

```

endif

```

```

end

```

```

Procedure AREA_CALC_CONDENSER(Nhxcond, N_row_max_cond, N_row_avg_cond, N_mod_cond[1..Nhxcond],
N_pass_per_mod_cond[1..Nhxcond], N_elem_perpass_cond, N_port_cond, width_cond, D_h_cond, D_w_cond, D_inner_cond,
D_outer_cond, thick_wall_cond, thick_web_cond, thick_end_cond, S_long_cond, S_trans_cond, density_fin_cond, thick_fin_cond,
height_fin_cond, pitch_Louver_cond, theta_Louver_cond : N_totalpass_cond, N_tube_front_cond, N_elem_cond, depth_cond,
Height_cond, Dh_cond, pitch_fin_cond, thick_MCT_cond, height_Louver_cond, pitch_MCT_cond, thick_tubewall_cond,
D_collar_cond, A_front_cond, A_ffield_cond, A_a_cond, A_r_cond, A_x_r_cond, A_airDIVfflow_cond, A_fin_cond,
A_finDIVair_cond)
$Common port_type_cond$, cond_type$

```

```

if (PORT_TYPE_CONDS$ = 'round') then
    thick_MCT_cond = D_inner_cond + 2* thick_wall_cond      "Microchannel tube thickness, [m] "
endif

```

```

if (PORT_TYPE_CONDS$ = 'rectangular') then
    thick_MCT_cond = D_h_cond + 2* thick_wall_cond          "Microchannel tube thickness, [m] "
endif

```

```

D_collar_cond = D_outer_cond + 2*thick_fin_cond
thick_tubewall_cond = (D_outer_cond - D_inner_cond)/2
pitch_MCT_cond = height_fin_cond + thick_MCT_cond      "MCT pitch, m"

"Calculate total number of tube passes, total number of elements and total number of modules in the heat exchanger"
N_elem_cond = 0
N_totalpass_cond = 0
module_total = 0
s = 1
repeat

N_totalpass_cond = N_totalpass_cond + N_mod_cond[s]*N_pass_per_mod_cond[s]*N_row_avg_cond      "Total no. of
tubepasses in the condenser"
N_elem_cond = N_elem_cond + N_pass_per_mod_cond[s]*N_elem_perpass_cond      "Total no. of elements in the heat
exchanger"
module_total = module_total + N_mod_cond [s]      "Total number of modules in
the heat exchanger"

s = s+1

until (s>Nhxcond)

N_tube_front_cond = N_totalpass_cond/N_row_avg_cond      "Number of tubes as seen from front of the heat exchanger"

"Geometry and Area calculations"

If (COND_TYPE$='MicroCh') then

endif

if (COND_TYPE$='TubeFin') then
    Call GEOMETRYCALFINONTUBEcond(N_tube_front_cond, N_row_max_cond, density_fin_cond, pitch_Louver_cond,
    theta_Louver_cond, S_trans_cond, S_long_cond : Height_cond, depth_cond, pitch_fin_cond, height_Louver_cond)

    Call AREACALFINONTUBEcond(N_row_avg_cond, N_tube_front_cond, N_totalpass_cond, depth_cond,
    density_fin_cond, width_cond, Height_cond, S_long_cond, S_trans_cond, D_collar_cond, D_inner_cond, thick_fin_cond:
    A_front_cond, A_fflow_cond, A_a_cond, A_r_cond, A_x_r_cond, A_airDIVfflow_cond, Dh_cond, A_fin_cond, A_finDIVair_cond)
endif

end

"***** Tube-FinCondenser Geometry calculation*****"

PROCEDURE GEOMETRYCALFINONTUBECOND(N_tube_front_cond, N_row_max_cond, density_fin_cond, pitch_Louver_cond,
theta_Louver_cond, S_trans_cond, S_long_cond : Height_cond, depth_cond, pitch_fin_cond, height_Louver_cond)

    Height_cond = S_trans_cond*N_tube_front_cond
    depth_cond = N_row_max_cond*S_long_cond
    pitch_fin_cond = 1/density_fin_cond
    height_Louver_cond = pitch_Louver_cond*TAN(theta_Louver_cond)/2      " louver height , m"

END

PROCEDURE AREACALFINONTUBECOND(N_row_avg_cond, N_tube_front_cond, N_totalpass_cond, depth_cond,
density_fin_cond, width_cond, Height_cond, S_long_cond, S_trans_cond, D_collar_cond, D_inner_cond, thick_fin_cond:
A_front_cond, A_fflow_cond, A_a_cond, A_r_cond, A_x_r_cond, A_airDIVfflow_cond, Dh_cond, A_fin_cond, A_finDIVair_cond)

A_front_cond = width_cond*Height_cond
A_fflow_cond = width_cond* N_tube_front_cond*(S_trans_cond-D_collar_cond)*(1-density_fin_cond*thick_fin_cond)
A_a_cond = 2*N_tube_front_cond*N_row_avg_cond*width_cond*density_fin_cond*(S_trans_cond*S_long_cond-
D_collar_cond^2*PI/4)+N_totalpass_cond*D_collar_cond*PI*width_cond*(1-density_fin_cond*thick_fin_cond)
A_r_cond = PI*D_inner_cond*width_cond*N_totalpass_cond
A_x_r_cond = PI/4*D_inner_cond^2
A_airDIVfflow_cond = A_a_cond/A_fflow_cond
Dh_cond = 4/A_airDIVfflow_cond*depth_cond
A_fin_cond = 2*N_tube_front_cond*N_row_avg_cond*width_cond*density_fin_cond*(S_trans_cond*S_long_cond-
D_collar_cond^2*PI/4)

```

A_finDIVair_cond=A_fin_cond/A_a_cond

end

{ Following two procedure are used to calculate areas for different elements }

PROCEDURE ELEMENTAREACALCOND(width_cond, N_tube_front_cond, A_fflow_cond, A_a_cond, A_r_cond, A_fin_cond, A_x_r_cond, m_dot_r_cond, m_dot_a_cond, N_row_avg_cond, N_mod_cond, N_circ_cond : DL, A_a_elem, A_r_elem, m_dot_r_elem, m_dot_a_elem, G_a_max_elem, G_r_elem, Vol_elem_cond)
\$COMMON N_elem_perpass_cond

DL= width_cond*N_row_avg_cond/(N_ELEM_PERPASS_COND*N_circ_cond) {Length of each element to calculate Pressure drop}
A_a_elem=A_a_cond/N_tube_front_cond/N_ELEM_PERPASS_COND {The air side area element to calculate heat transfer}
{There is a slight overestimation in the fin area of this element, because total no. of fin passes are (N_tube_front_cond + 1), but this is negligible}
A_r_elem=A_r_cond/N_tube_front_cond/N_ELEM_PERPASS_COND {The refrigerant area element to calculate heat transfer}
m_dot_a_elem=m_dot_a_cond/N_tube_front_cond/N_ELEM_PERPASS_COND {The Air side element flow rate}
m_dot_r_elem=m_dot_r_cond/N_mod_cond {Refrigerant flow rate for each element}
== total refrigerant flow rate through each module}
G_a_max_elem=m_dot_a_cond/A_fflow_cond {maximum air flow flux, kg/s-m^2}
G_r_elem=m_dot_r_elem/(A_x_r_cond*N_circ_cond) {The Refrigerant mass flux per element}
Vol_elem_cond=A_x_r_cond*DL*N_circ_cond {Volume of each element to calculate refrigerant charge}

END

Procedure TWOSUB_COND_DOWN(fluidname\$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_Fin, Q_elem_in, Tri, Pri, hri, xi, Tai, mr, mair, Aref, Air, Gair_max, Gref, DL, Vol, L2ph, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro, Tao, xo, Q, M)

\$Common Cond_type\$

{Thermal property calculation}

Call Prop_Air_Dry(Tai, P_atm:cp_air, mu_air, rho_air, Pr_air)
Call Prop_Sat_P(fluidname\$, Pri:T_sat, v_v,v_l,h_v,h_l,mu_v,mu_l,k_v,k_l,cp_v,cp_l,rho_v, rho_l)
Call Prop_2ph_Px(fluidname\$, Pri, xi : v_refi, rho_refi)

Re_LO=Gref*D_inner_cond/mu_l
Call f_1ph_Churchill(Re_LO,Relrough_cond:f)
DL_2ph=DL*L2ph
Vol_2ph = Vol*L2ph
call Dp_2ph_Souza(fluidname\$,Pri, xi, 0, f, DL_2ph, D_inner_cond, Gref:DP_f, DP_acc, DP)
DP=DP*PF_cond
Pro_2ph = Pri - DP
hro_2ph = ENTHALPY(fluidname\$,P=Pro_2ph, x=0)
Tro_2ph = TEMPERATURE(fluidname\$, P=Pro_2ph, x=0)
Q_2ph=mr*(hri-hro_2ph)

{Charge calculation for the two-phase region}

x_ave=(xi+0)/2
Call VoidFraction_Zivi(x_ave,v_v,v_l:raf)
M_2ph=Vol_2ph*(rho_v*raf+rho_l*(1-raf))

Pri_sub = Pro_2ph
hri_sub = hro_2ph-0.1"Make sure it is subcooling"
Tri_sub = Tro_2ph-0.1 "Make sure it is subcooling"
DL_sub = DL*(1-L2ph)
Vol_sub = Vol*(1-L2ph)
xi_sub = -1.5
mair_sub = mair*(1-L2ph)
Aref_sub = Aref*(1-L2ph)
Air_sub = Air*(1-L2ph)
Q_elem_in_new = Q_elem_in*(1-L2ph)

```

Call SinglePhase_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_Fin, Q_elem_in_new, Tri_sub,
Pri_sub, hri_sub, xi_sub, Tai, mr, mair_sub, Aref_sub, Aair_sub, Gair_max, Gref, DL_sub, Vol_sub, Relrough_cond, PF_cond,
EF_cond : Tro, Pro, hro, Tao, xo, L2ph_dummy, Lsup, Q_sub, M_sub)

```

```

Cair=mair*cp_air
Q=Q_sub+Q_2ph
Tao= Tai + Q / Cair
M = M_2ph + M_sub
end

```

```

{This procedure calculates twophase, downstream marching element}
Procedure TWOPHASE_COND_DOWN(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_Fin, Q_elem_in, Tri,
Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro, Tao, xo, L2ph,
Lsup, Q, M)

```

```

$Common Cond_type$, htc_cond$

```

```

{Thermal property calculation}
Call Prop_Sat_P(fluidname$, Pri:T_sat, v_v,v_l,h_v,h_l,mu_v,mu_l,k_v,k_l,cp_v,cp_l,rho_v, rho_l)
Call Prop_2ph_Px(fluidname$, Pri, xi : v_refi, rho_refi)
Call Prop_Air_Dry(Tai, P_atm:cp_air, mu_air, rho_air, Pr_air)
Cair = mair*cp_air
{Refrigerant heat transfer coefficient calculation}
qflux=Q_elem_in/Aref*1000 {Guess value for qflux (W/m2), Q_elem_in is the heat transfer from the previous element}

```

```

{calculation of qflux using secant method}
sigma = SURFACETENSION(fluidname$, T=Tri)
qflux[2] = qflux
qflux[1] = 0.95*qflux
T_surf = Tai + qflux[1]/(hair*Eff_Sur*1000)*Aref/Aair
dT = Tri - T_surf

```

```

if (HTC_CONDS$ = 'Dobson') then
Call h_cond_Dobson(fluidname$, xi,Pri, D_inner_cond,Gref:href)
else
    if (HTC_CONDS$ = 'Cavallini') then
    Call h_cond_Cavallini(fluidname$, Pri, Gref, D_inner_cond, xi, dT, sigma : href)
    endif
endif

```

```

href=href*EF_cond
Call UA_air_ref(hair, Aair, href, Aref, Eff_Sur : UA)
Call epsilon_constTemp(UA, Cair : epsilon)
Q=epsilon* Cair *(Tri - Tai)
value[1] = qflux[1] - Q/Aref*1000
iter = 0

```

```

Repeat

```

```

T_surf = Tai + qflux[2]/(hair*Eff_Sur*1000)*Aref/Aair
dT = Tri - T_surf
if (HTC_CONDS$ = 'Dobson') then
Call h_cond_Dobson(fluidname$, xi,Pri, D_inner_cond,Gref:href)
else
    if (HTC_CONDS$ = 'Cavallini') then
    Call h_cond_Cavallini(fluidname$, Pri, Gref, D_inner_cond, xi, dT, sigma : href)
    endif
endif

```

```

endif
href=href*EF_cond
Call UA_air_ref(hair, Aair, href, Aref, Eff_Sur : UA)
Call epsilon_constTemp(UA,Cair:epsilon)
Q=epsilon* Cair *(Tri - Tai)
value[2] = qflux[2] - Q/Aref*1000
qflux_temp = qflux[2]
qflux[2] = qflux[2] - value[2]*(qflux[1] - qflux[2])/(value[1] - value[2])
qflux[1] = qflux_temp
value[1] = value[2]
iter = iter + 1

```

```

Until(ABS(value[1])<0.00001)

hro= hri - Q / mr
Tao= Tai + Q / Cair

{If the refrigerant outlet is still in twophase zone}
if (hro>h_l) Then

    xo=QUALITY(fluidname$,h=hro,P=Pri)    "approx. quality to calculate DP"
    Re_LO=Gref*D_inner_cond/mu_l
    Call f_1ph_Churchill(Re_LO,Relrough_cond:f)
    call Dp_2ph_Souza(fluidname$,Pri, xi, xo, f, DL, D_inner_cond, Gref:DP_f, DP_acc, DP)
    DP=DP*PF_cond
    Pro= Pri - DP
    h_l_o=ENTHALPY(fluidname$,P=Pro, x=0) "calculate the saturated liquid enthalpy at the outlet pressure"
    if (hro>h_l_o) Then "if the outlet is still in two phase zone"
        xo= QUALITY(fluidname$,h=hro,P=Pro) "Calculate refrigerant outlet quality"
        Tro=TEMPERATURE(fluidname$, P=Pro,h=hro) "Calculate refrigerant outlet temperature based on calculated outlet
pressure"
    else "if the outlet is in subcooling zone"
        xo=-1.5
        Tro=TEMPERATURE(fluidname$, h=hro, P=Pro)
    endif
    L2ph =1
    Lsup = 0
else "the refrigerant outlet is in subcooling zone"
    xo=-1.5    "Set outlet quality is less than 0 to indicate this element is a transition element from twophase to subcooling"
    Pro=Pri    "ignore the refrigerant pressure drop in this transition zone to make calculation easier"
    Q2ph=mr*(hri-h_l)
    L2ph=Q2ph/Q
    Lsup = 0
    Tro=TEMPERATURE(fluidname$, h=hro, P=Pro)

endif

{Charge calculation}
if (xo>0) Then
    x_ave=(xi+xo)/2
    Call VoidFraction_Zivi(x_ave,v_v,v_l:raf)
    M=Vol*(rho_v*raf+rho_l*(1-raf))
else
    x_ave=(xi+0)/2
    Call VoidFraction_Zivi(x_ave,v_v,v_l:raf)
    M=Vol*(rho_v*raf+rho_l*(1-raf))
endif

end

{This element calculate superheated , downstream marching element}
Procedure SINGLEPHASE_COND_DOWN(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_Fin, Q_elem_in,
Tri, Pri, hri, xi,Tai, mr,mair, Aref, Aair, Gair_max, Gref, DL, Vol, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro, Tao, xo, L2ph,
Lsup, Q, M)
$Common Cond_type$

Lsup = 0
L2ph = 0
P_crit = P_CRIT(fluidname$)
if (Pri < P_crit) then
    h_v=ENTHALPY(fluidname$,P=Pri, x=1)
    if (hri > h_v) then
        phase$ = 'super'
    else
        phase$ = 'subcool'
    endif
endif

{Thermal property calculation}
Call Prop_1ph_PT(fluidname$, Tri, Pri : v_refi, mu_refi, k_refi, cp_refi, rho_refi)
Call REPR_1PH(fluidname$,Gref, D_inner_cond, mu_refi, cp_refi, k_refi,Tri, Pri : Re_ref, Pr_ref)
Call Prop_Air_Dry(Tai, P_atm:cp_air, mu_air, rho_air, Pr_air)

```

```

{Refrigerant side heat transfer coefficient calculation}
Call f_1ph_Churchill(Re_ref,Relrough_cond:f)
Call h_1ph_Gnielinski(f, Re_ref,Pr_ref, D_inner_cond, k_refi: Nu, href)
href=href*EF_cond

{UA calculation}
Call UA_air_ref(hair, Aair, href, Aref, Eff_Sur :UA)

{Effectiveness for cross flow}
Cair = mair * cp_air
Cref= mr * cp_refi
Call epsilon_crossflow(UA,Cref, Cair:epson, Cmin, Cmax)

{Heat transfer rate and outlet condition calculation}
Q = epson * Cmin *(Tri - Tai)
hro= hri - Q / mr
Tao= Tai + Q / Cair

if (phase$ = 'super') then
{If the outlet of this element is still in the superheated zone}
if (hro>h_v) Then
Tro=TEMPERATURE(fluidname$, P=Pri,h=hro) "approx. outlet T for DelP calc"
v_refo = VOLUME(fluidname$, P=Pri,T=Tro)
call Dp_1ph(f, DL, D_inner_cond, Gref, v_refi, v_refo : DP)
DP=DP*PF_cond
Pro= Pri - DP
h_v_o=ENTHALPY(fluidname$,P=Pro, x=1) "Calculate saturated vapor enthalpy at the outlet pressure"
if (hro>h_v_o) Then "if the outlet is still in superheated zone"
Tro=TEMPERATURE(fluidname$, P=Pro,h=hro) "Calculate refrigerant outlet temperature based on outlet pressure"
xo=1.5 "Set outlet quality is 1.5 to indicate it is a superheated element"
else "if the outlet is in two-phase zone"
xo= QUALITY(fluidname$,h=hro,P=Pro)
Tro=TEMPERATURE(fluidname$, P=Pro,h=hro)
endif
L2ph = 0
Lsup = 1
rho_refo=DENSITY(fluidname$, P=Pro,h=hro)

else "the outlet is in twophase zone"
Qsup=mr*(hri-h_v)
call Length_Superheated( Qsup, Tai, Tri, UA, Cair, Cref:Cair_new)

Lsup = Cair_new/Cair
L2ph = 1-Lsup
Pro=Pri "ignore the pressure drop in this transition element to make the calcualtion easier"
xo= QUALITY(fluidname$,h=hro,P=Pro)
Tro=TEMPERATURE(fluidname$, P=Pro,h=hro)
rho_refo=DENSITY(fluidname$, P=Pro,h=hro)
goto 10
endif
endif

{pressure drop calculation}
Tro=TEMPERATURE(fluidname$, P=Pri,h=hro) "approx. outlet T for DelP calculation"
v_refo = VOLUME(fluidname$, P=Pri,T=Tro)
rho_refo=1/v_refo
call Dp_1ph(f, DL, D_inner_cond, Gref, v_refi, v_refo : DP)
DP=DP*PF_cond
Pro= Pri - DP
Tro=TEMPERATURE(fluidname$, P=Pro,h=hro) "Calculate refrigerant outlet temperature based on calculated outlet pressure and enthalpy"

if (Pro > P_crit) then
xo = 500
else
xo=-1.5 "Set outlet quality is -1.5 to indicate that this element is in subcooling zone"
endif

```

```

10:
{Charge calculation}
if (xo>1) Then
  M=Vol*(rho_refi+rho_refo)/2
else
  M=Vol*(rho_refi+rho_refo)/2
endif

end

Procedure SUPER2PH_COND_DOWN(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_Fin, Q_elem_in, Tri,
Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Lsup, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro, Tao, xo, Q,
M)
$Common Cond_type$

{Thermal property calculation}
Call Prop_Air_Dry(Tai, P_atm:cp_air, mu_air, rho_air, Pr_air)

{superheat pressure drop}
Call Prop_1ph_PT(fluidname$, Tri, Pri : v_refi, mu_refi, k_refi, cp_refi, rho_refi_1ph)

Call REPR_1PH(fluidname$, Gref, D_inner_cond, mu_refi, cp_refi, k_refi, Tri, Pri : Re_ref, Pr_ref)
Call f_1ph_Churchill(Re_ref, Relrough_cond: f)

Tro_sup=TEMPERATURE(fluidname$, P=Pri, x=1) "approx. outlet T for DelP calc"
v_refo = VOLUME(fluidname$, P=Pri, x=1)
DL_sup=DL*Lsup
Vol_sup = Vol*Lsup
call Dp_1ph(f, DL_sup, D_inner_cond, Gref, v_refi, v_refo : DP)
DP=DP*PF_cond
Pro_sup= Pri - DP
hro_sup=ENTHALPY(fluidname$, P=Pro_sup, x=1) - 0.1"Make sure it is 2 phase"
Tro_sup = TEMPERATURE(fluidname$, P=Pro_sup, h=hro_sup) - 0.1"Make sure it is 2 phase"

Q_sup=mr*(hri-hro_sup)
{Charge calculation}
rho_refi_2ph = DENSITY(fluidname$, P=Pro_sup, x=0.999)
M_sup=Vol_sup*(rho_refi_1ph+rho_refi_2ph)/2

hri_2ph=hro_sup
Pri_2ph=Pro_sup
Tri_2ph = Tro_sup
xi_2ph=0.999

DL_2ph = DL*(1-Lsup)
Vol_2ph = Vol*(1-Lsup)
mair_2ph = mair*(1-Lsup)
Aref_2ph = Aref*(1-Lsup)
Aair_2ph = Aair*(1-Lsup)
Q_elem_in_new = Q_elem_in*(1-Lsup)

call TwoPhase_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_Fin, Q_elem_in_new, Tri_2ph,
Pri_2ph, hri_2ph, xi_2ph, Tai, mr, mair_2ph, Aref_2ph, Aair_2ph, Gair_max, Gref, DL_2ph, Vol_2ph, Relrough_cond, PF_cond,
EF_cond : Tro, Pro, hro, Tao, xo, L2ph, Lsup_dummy, Q_2ph, M_2ph)

Cair=mair*cp_air
Q=Q_sup+Q_2ph
Tao= Tai + Q / Cair
M = M_sup + M_2ph

end

{This procedure choose appropriate elment-solving procedure to calculate each element}
Procedure CHOOSECONDENSER(fluidname$, P_atm, D_inner_cond, D_outer_cond, Pri, hri, Tai, mr, mair, Aref, Aair, Gair_max,
Gref, DL, Vol, Q_elem_in, hair, Eff_Sur, Eff_fin, Relrough_cond, PF_cond, EF_cond : Pro, hro, Tao, Q, M, L2ph, Lsup)

Tri = TEMPERATURE(fluidname$, P=Pri, h=hri)
P_crit = P_CRIT(fluidname$)

```

```

if (Pri > P_crit) then "refrigerant is above the critical point"
xi = 500
call SinglePhase_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_fin, Q_elem_in, Tri, Pri, hri,
xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro, Tao, xo, L2ph, Lsup, Q, M)

else
"initialize state indicator variables"
h_v=ENTHALPY(fluidname$,P=Pri,x=1)
h_l=ENTHALPY(fluidname$,P=Pri,x=0)
xi = (hri - h_l)/(h_v - h_l)

    if (hri > h_v) then
        call SinglePhase_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_fin, Q_elem_in, Tri,
Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro, Tao, xo, L2ph,
Lsup, Q, M)
        if ((Lsup>0) AND (Lsup<1)) Then
            call Super2ph_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_fin,
Q_elem_in, Tri, Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Lsup, Relrough_cond, PF_cond, EF_cond : Tro, Pro,
hro, Tao, xo, Q, M)
        endif
    endif

    if ((hri > h_l) AND (hri<=h_v)) then
        call TwoPhase_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_fin,
Q_elem_in, Tri, Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro,
Tao, xo, L2ph, Lsup, Q, M)
        if ((L2ph>0) AND (L2ph<1)) Then
            call TwoSub_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur,
Eff_fin, Q_elem_in, Tri, Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, L2ph, Relrough_cond, PF_cond, EF_cond : Tro,
Pro, hro, Tao, xo, Q, M)
        endif
    endif

    if (hri <= h_l) then
        call SinglePhase_Cond_down(fluidname$, P_atm, D_inner_cond, D_outer_cond, hair, Eff_Sur, Eff_fin,
Q_elem_in, Tri, Pri, hri, xi, Tai, mr, mair, Aref, Aair, Gair_max, Gref, DL, Vol, Relrough_cond, PF_cond, EF_cond : Tro, Pro, hro,
Tao, xo, L2ph, Lsup, Q, M)
    endif
endif

end

Procedure CONDCAL(fluidname$, P_atm, Relrough_cond, A_x_inlet_header_cond, A_x_exit_header_cond, D_inner_cond,
D_outer_cond, D_return_cond, D_collar_cond, Dh_cond, density_fin_cond, pitch_fin_cond, height_fin_cond, thick_fin_cond,
K_fin_cond, pitch_Louver_cond, length_Louver_cond, height_Louver_cond, theta_Louver_cond, S_trans_cond, S_long_cond,
Thick_MCT_cond, pitch_MCT_cond, width_cond, depth_cond, A_front_cond, A_fflow_cond, A_a_cond, A_r_cond, A_x_r_cond,
A_airDIVfflow_cond, A_fin_cond, A_finDIVair_cond, P_r_in_cond, h_r_in_cond, T_a_in_cond, m_dot_r_cond, m_dot_a_cond,
N_elem_cond, N_slab_cond, N_tube_front_cond, N_row_avg_cond, Nhxc_cond, N_circ_cond[1..Nhxc_cond],
N_pass_per_mod_cond[1..Nhxc_cond], N_mod_cond[1..Nhxc_cond], PF_cond, EF_cond, eta_fan_cond : Q_cond, h_r_out_cond,
P_r_out_cond, W_dot_fan_cond, T_a_out_cond, f_sup_cond, f_2ph_cond, f_sub_cond, charge_cond)

$COMMON cond_type$, marching_cond$, N_Elem_perpass_cond, N_element_depth_cond

P_crit = P_CRIT(fluidname$)
T_r_in_Con = TEMPERATURE(fluidname$, P=P_r_in_cond, h=h_r_in_cond)
if (P_r_in_cond >= P_crit) then
x_r_in_Con = 500
else
h_v_in = ENTHALPY(fluidname$, P=P_r_in_cond, x=1)
h_l_in = ENTHALPY(fluidname$, P=P_r_in_cond, x=0)
x_r_in_Con = (h_r_in_cond - h_l_in)/(h_v_in - h_l_in)
endif

{Calculate each element heat transfer area and mass flux}
k=1
i=1
repeat

```

```

T_a_out_Elem[0,i] = T_a_in_cond
sum_elem=0      {'sum_elem' keeps track of the number of elements in a sub heat exchanger}
  j=1
  repeat
    sum_elem = sum_elem + N_pass_per_mod_cond[k]*N_ELEM_PERPASS_COND
    j = j+1
  until (j>k)

  If (i>sum_elem) then
    k = k+1
  endif

  Call ElementAreaCalCond(width_cond, N_tube_front_cond, A_fflow_cond, A_a_cond, A_r_cond, A_fin_cond,
A_x_r_cond, m_dot_r_cond, m_dot_a_cond, N_row_avg_cond, N_mod_cond[k], N_circ_cond[k] : DLC[i], A_air_Elem[i],
A_ref_Elem[i], m_dot_r_Elem[i], m_dot_air_Elem[i], G_air_max_Elem[i], G_ref_Elem[i], Vol_elem_cond[i])

  i = i+1
until (i > N_elem_cond + 0.5) "Added 0.5 to account for round off errors in floating point"

{T_r_in_Elem, P_r_in_Elem, h_r_in_Elem and x_in_Elem are arrays which store the ref. inlet temperature, pressure, enthalpy and
quality respectively for each element}
T_r_Elem[1,0]=T_r_in_cond
P_r_Elem[1,0]=P_r_in_cond
h_r_Elem[1,0]=h_r_in_cond
x_Elem[1,0]=x_r_in_cond
f_2ph_cond = 0
f_sup_cond = 0
charge_head = 0
charge_cond = 0
head_count = 1      "keeps track of the header number"
Q_Elem[1,0] = 0.3   "Guess value of heat transfer to be used later for the calculation of actual heat transfer in an element"
i_slab = 1
repeat
  T_a_out_slab[i_slab]=0      "Initializing air outlet temperature, C"
  Q_slab[i_slab] = 0          "Initializing total capacity, [kW]"
  j=1
  k_Nhx = 1
  repeat
    sum_elem=0
    i_count=1
    repeat
      sum_elem = sum_elem + N_pass_per_mod_cond[k]*N_ELEM_PERPASS_COND
      i_count = i_count+1
    until (i_count>k_Nhx)

    If (j>sum_elem) then
      k_Nhx = k_Nhx + 1
    endif

    T_a_in_Elem[i_slab,j]=T_a_out_Elem[i_slab-1,N_elem_cond+ 1- j]      "Air inlet temperature for each element; for
multislabs, output of one slab is input of the other"

    "Airside heat tranfer coefficient calculation"
    Gair_max=m_dot_a_cond/A_fflow_cond

    Call HTC_AIR_SIDE_COND(P_atm, T_a_in_Elem[i_slab,j], A_finDIVair_cond, Gair_max, D_outer_cond, D_collar_cond, Dh_cond,
theta_Louver_cond, pitch_Louver_cond, pitch_fin_cond, depth_cond, length_Louver_cond, pitch_MCT_cond, height_Louver_cond,
K_fin_cond, thick_fin_cond, height_fin_cond, S_long_cond, S_trans_cond, N_row_avg_cond : hair_Elem[i_slab,j],
Eff_Sur_Elem[i_slab,j], Eff_Fin_Elem[i_slab,j])

    "Calculating charge in the header from where the refrigerant enters a pass"
    If (COND_TYPE$='Microch') then

All for MCH cond

  endif

Call CHOOSECONDENSER(fluidname$, P_atm, D_inner_cond, D_outer_cond, P_r_elem[i_slab,j-1], h_r_Elem[i_slab,j-1],
T_a_in_Elem[i_slab,j], m_dot_r_Elem[j], m_dot_air_Elem[j], A_ref_Elem[j], A_air_Elem[j], G_air_max_Elem[j], G_ref_Elem[j],

```

```

DLC[j],Vol_elem_cond[j],Q_Elem[i_slab,j-1], hair_Elem[i_slab,j], Eff_Sur_Elem[i_slab,j], Eff_Fin_Elem[i_slab,j], Relrough_cond,
PF_cond, EF_cond : P_r_Elem[i_slab,j], h_r_Elem[i_slab,j], T_a_out_Elem[i_slab,j], Q_Elem[i_slab,j], m_Elem, L2ph, Lsup)

    T_r_Elem[i_slab,j]=TEMPERATURE(fluidname$, P=P_r_Elem[i_slab,j], h=h_r_Elem[i_slab,j])
    if (P_r_Elem[i_slab,j] < P_crit) then
        h_v =ENTHALPY(fluidname$, P=P_r_Elem[i_slab,j], x=1)
        h_l = ENTHALPY(fluidname$, P=P_r_Elem[i_slab,j], x=0)
        x_Elem[i_slab,j]=(h_r_Elem[i_slab,j] - h_l)/(h_v - h_l)
    else
        x_Elem[i_slab,j] = 500
    endif

    f_2ph_cond =f_2ph_cond + L2ph/(N_elem_cond*N_slab_cond)
    f_sup_cond =f_sup_cond + Lsup/(N_elem_cond*N_slab_cond)
    charge_cond=charge_cond+N_mod_cond[k_Nhx]*m_Elem
    Q_slab[i_slab]=Q_slab[i_slab]+N_mod_cond[k_Nhx]*Q_Elem[i_slab,j]
    T_a_out_slab[i_slab]= T_a_out_slab[i_slab] + N_mod_cond
[k_Nhx]*T_a_out_Elem[i_slab,j]/(N_ELEM_PERPASS_COND*N_tube_front_cond)

    If (COND_TYPE$='TubeFin') then
        If (MOD(j,N_ELEM_PERPASS_COND) = 0) Then {Determine whether reach the return bend }
            Call DP_Returnbend(fluidname$, MARCHING_COND$, T_r_Elem[i_slab,j], P_r_Elem[i_slab,j],
x_Elem[i_slab,j], h_r_Elem[i_slab,j], D_return_cond, D_outer_cond, G_ref_Elem[j], Relrough_cond, 1 : DPbend, T_r_Elem[i_slab,j],
P_r_Elem[i_slab,j], x_Elem[i_slab,j])
        endif
    endif

    {if (j >4) then
call error(2)
endif}

    j=j+1

"Calculating charge in the header from where the refrigerant leaves a pass"
    If (COND_TYPE$='Microch') then
        If (MOD(j-1,N_ELEM_PERPASS_COND) = 0) Then {Determine whether reach the header }
            All for MCH
        endif
    endif

until(j>N_elem_cond + 0.5)    "Added 0.5 to account for round off errors in floating point"
i_slab = i_slab+1
if (i_slab <= N_slab_cond) then
    T_r_Elem[i_slab,0]=T_r_Elem[i_slab-1,j-1]
    P_r_Elem[i_slab,0]=P_r_Elem[i_slab-1,j-1]
    h_r_Elem[i_slab,0]=h_r_Elem[i_slab-1,j-1]
    x_Elem[i_slab,0]=x_Elem[i_slab-1,j-1]
    Q_Elem[i_slab,0] = Q_Elem[i_slab-1,j-1]
endif
until (i_slab>N_slab_cond)

{Summarize the modeling result}
T_a_out_cond = T_a_out_slab[N_slab_cond]
P_r_out_cond=P_r_Elem[N_slab_cond,N_elem_cond]
T_r_out_cond=T_r_Elem[N_slab_cond,N_elem_cond]
h_r_out_cond=h_r_Elem[N_slab_cond,N_elem_cond]
f_sub_cond = 1 - f_2ph_cond - f_sup_cond
charge_cond = charge_cond + charge_head
"Calculation of pressure drop in air flow direction across the condorator"
DP_air_cond = 0
W_dot_fan_cond = 0
i_slab = 1
T_a_in_slab[1] = T_a_in_cond
Q_cond = 0
Q_sen_cond = 0
repeat
    Q_cond = Q_cond + Q_slab[i_slab]
    "Air side pressure drop"

```

"Refrigerant outlet pressure, [kPa]"
 "Refrigerant outlet temperature, [oC]"
 "Refrigerant outlet enthalpy, [kJ/kg]"

```

Call DP_AIR_SIDE_COND(P_atm, Gair_max, T_a_in_slab[i_slab], T_a_out_slab[i_slab], m_dot_a_cond, A_flow_cond,
A_front_cond, A_a_cond, A_fin_cond, D_collar_cond, Dh_cond, theta_Louver_cond,
pitch_Louver_cond, pitch_fin_cond, depth_cond, length_Louver_cond, pitch_MCT_cond, height_Louver_cond, thick_fin_cond,
height_fin_cond, S_long_cond, S_trans_cond, N_row_avg_cond : DP_air_slab[i_slab])
W_dot_fan_slab[i_slab]=m_dot_a_cond/DENSITY('Air', P=P_atm, T=T_a_in_slab[i_slab])*DP_air_slab[i_slab]/eta_fan_cond/1000
{fan work}
DP_air_cond = DP_air_cond + DP_air_slab[i_slab]/1000
W_dot_fan_cond = W_dot_fan_cond + W_dot_fan_slab[i_slab]
i_slab = i_slab + 1
if (i_slab<=N_slab_cond) then
T_a_in_slab[i_slab] = T_a_out_slab[i_slab-1]
endif
until (i_slab > N_slab_cond)

end

```

{End of Condenser Procedure}

INTERNAL HEAT EXCHANGER PROCEDURE ##### “

EVAPORATOR PROCEDURES

"Evaporator procedure ends here"

{Connecting Equations}

```

P_dis = P_r_in_cond
h_dis = h_r_in_cond

```

```

m_dot_r_cond = m_dot_r_comp

```

```

h_r_in_evap = h_h_out_IHX

```

```

h_r_out_evap = h_c_in_IHX

```

```

m_dot_r_evap = m_dot_r_comp

```

```

P_c_out_IHX = P_suc
h_c_out_IHX = h_suc

```

```

P_c_in_IHX = P_r_out_evap

```

```

COP = Q_evap/W_dot_comp

```

```

DT_sup = 2.0
DT_sub = 5.0

```

```

{Q_evap = 2.87}
"MAIN"

```

***** Compressor *****

```

fluidname$ = 'R410A'

```

```

{P_suc = 292.5}

```

```

{P_dis =1750}

```

```

{T_suc }

```

```

{h_suc =292.3}

```

```

T_outdoor = 25

```

```

"Input compressor size and heat transfer parameters"

```

```

beta_comp=1

```

```

for which mfgr performance map used"

```

```

A_comp=2.56*CONVERT(ft^2, m^2)

```

```

{U_comp=11.7*CONVERT(btu/hr-F-ft^2, kW/m^2-K)
kW/m^2-K"}

```

"Compressor size relative to one

"Compressor shell area, m^2"

"Compressor shell heat transfer coefficient,

```

U_comp=0
T_shell_coeff0 = 41.091                                "Coefficients to calculate
compressor shell surface temperature"
T_shell_coeff1 = 0.3981
V_comp=0.0771*CONVERT(ft^3, m^3)                        "Compressor volume, m^3"
M_oil=2.5*CONVERT(lbm, kg)                              "Mass of oil, kg"

Call State_Calc_Comp(fluidname$, P_suc, h_suc : T_suc, x_suc)
Call State_Calc_Comp(fluidname$, P_dis, h_dis : T_dis, x_dis)

{V_dot_disp = 0.00185}

V_dot_disp = 2*((0.046^2)/4)*0.033*47

"Computational parameters for compressor Procedure"
sumpside$='highs'
{m_dot_r_comp=0.014 [kg/s]}
"Compressor model based on efficiency"
"For R744 system the efficiencies are"
eta_isen = 0.68 {0.8134 - 0.05296*(P_dis/P_suc)}
eta_vol = 0.85 {1.0214 - 0.0338*(P_dis/P_suc)}
nu_suc = Volume(fluidname$, P=P_suc, h=h_suc)
m_dot_r_comp = eta_vol*V_dot_disp/nu_suc*beta_comp
{V_dot_disp = 0.00176 } {2.7*Convert(m^3/hr, m^3/s)}
Call CompCal_eff(fluidname$, sumpside$, h_suc, P_suc, P_dis, m_dot_r_comp, U_comp, A_comp, V_comp, M_oil,
T_outdoor, T_shell_coeff0, T_shell_coeff1, eta_isen, eta_vol, V_dot_disp : W_dot_comp, h_dis, T_shell, qcomp,
Mass_comp, Massfrac)

"***** Condenser *****"

$include C:\Documents and Settings\srajan\My Documents\My Classes\EES_folder\EES learning\New EES\ACRC
Simulation Models v\ACRC Simulation Models v.1\EES Files\Library\ACRC UserLib v.1.LIB

Cond_type$ = 'Tubefin'                                "Type of the condenser; Microchannel (MicroCh) or
Finned-tube (TubeFine)"
{fluidname$='R410A'}                                  "Refrigerant name"
marching_cond$ = 'downstream'                         "Finite element marching"
htc_cond$ = 'dobson'                                  "Two phase correlation to be used for condensation"
{T_outdoor=20}
P_atm=101.35
"Condenser operating condition inputs"
V_dot_a_cond=1100*CONVERT(cfm,m^3/s)                  "Volumetric air flow rate, m3/s" {check this factor
for face velocity}
eta_fan_cond = 0.2                                    "fan efficiency"
T_a_in_cond = 19.0                                    "air inlet temperature"
{T_r_in_cond = 101.9}                                "refrigerant inlet temperature"
{P_r_in_cond = 1750}
{h_r_in_cond = 400}
{DT_sub = 0}
{m_dot_r_cond=0.014}

"Condenser size and circuiting"
Nhxcond=1                                              "No. of sub HX which are connected in series."
N_slab_cond=1                                         "Slab number of condenser"
N_row_avg_cond = 2                                    "Average number of rows in the heat exchanger"
N_row_max_cond = 2                                    "Maximum number of rows in the heat exchanger"
N_circ_cond[1] = 2                                    "No. of circuits in the 1st Sub HX of the series; is 1 for MCH
by definition"for MCH by definition"

```

```

N_pass_per_mod_cond[1]=14          "Number of tube passes per module as seen from the front of the heat
exchanger, in 1st sub hx"

N_mod_cond [1] = 1                  "No. of identical modules in 1st Sub HX of the series; = no. of
MCT in the 1st pass"

"Computational parameters used for condenser model"
N_elem_perpass_cond=10              "Element number per pass."
N_element_depth_cond = 5

width_cond=34*convert(in, m)      "0.667"          "Condenser width [m]"

"Condenser tube inputs"
D_inner_cond= 9/1000                "1.52/1000"          "Tube/Port internal diameter, m "

"Louver dimensions"
theta_louver_cond = 23              "louver angle[deg]"
pitch_louver_cond = 0.99/1000        "louver pitch [m]"
length_louver_cond = height_fin_cond-2/1000 "louver length"
"*****Microchannel only inputs*****"
port_type_cond$ = 'rectangular'      "round or rectangular"
thick_end_cond = 0.4206/1000          " Thickness at tube end, [m] "
thick_wall_cond = 0.4206/1000         " Wall thickness, [m] "
thick_web_cond = 0.3438/1000          " Web thickness, [m]"
height_fin_cond=11.75/1000           "fin height"
N_port_cond= 28                      "Number of ports in a MCT"
D_w_cond = 1.5/1000                  "Microchannel port width"
D_h_cond = 1.5/1000                  "Microchannel port height"
"*****Finned-tube only inputs*****"
D_outer_cond=0.375*CONVERT(in, m)    "Outer diameter of condenser tube [m]"
S_long_cond=1.083*CONVERT(in, m)     "Longnitube tube spacing [m]"
S_trans_cond=1.25*CONVERT(in, m)     "Transversal tube spacing [m]"
D_return_cond=S_trans_cond           "Return bend diameter [m]"

"*****"
Rough_tube_cond=0.000005             "Tube roughness [m]"
k_tube_cond= 0.403                   " k_('Aluminum',40) "
" Conductivity of tube[kW/m-K]"
PF_cond=1 {1.2}                      "pressure drop penalty factor for
microfinned tube "
EF_cond=1 {1.6}                      "heat transfer enhancement
assumed = area ratio of microfinned tube"

"Condenser fin inputs"
density_fin_cond=12*CONVERT(1/in,1/m) "fin number per m"
thick_fin_cond=0.08/1000              "thickness of fin, m "
K_fin_cond = 0.237                    " k_('Aluminum',40) "
"Conductivity of fin[kW/m-K]"

{Header dimension}
A_x_inlet_header_cond = PI*(7.1/1000)^2/4 "inlet header flow area, m2"
A_x_exit_header_cond = PI*(7.1/1000)^2/4   "exit header flow area, m2"

call STATE_CALC_COND(fluidname$, P_r_in_cond, h_r_in_cond : T_r_in_cond, x_r_in_cond) "Ref. inlet
tempeprature and quality"
call State_Calc_cond(fluidname$, P_r_out_cond, h_r_out_cond : T_r_out_cond, x_r_out_cond) "Ref. inlet
tempeprature and quality"

"Calculate variables needed by condenser Procedures"
m_dot_a_cond=V_dot_a_cond/VOLUME(air,T=T_a_in_cond,P=P_atm) "Air flow rate, kg/s"
Relrough_cond=Rough_tube_cond/D_inner_cond                  "Relative
roughness=roughness /Diameter of tube"

```

```
Call Area_Calc_condenser(Nhxcond, N_row_max_cond, N_row_avg_cond, N_mod_cond[1..Nhxcond],
N_pass_per_mod_cond[1..Nhxcond], N_elem_perpass_cond, N_port_cond, width_cond, D_h_cond, D_w_cond,
D_inner_cond, D_outer_cond, thick_wall_cond, thick_web_cond, thick_end_cond, S_long_cond, S_trans_cond,
density_fin_cond, thick_fin_cond, height_fin_cond, pitch_Louver_cond, theta_Louver_cond : N_totalpass_cond,
N_tube_front_cond, N_elem_cond, depth_cond, Height_cond, Dh_cond, pitch_fin_cond, thick_MCT_cond,
height_Louver_cond, pitch_MCT_cond, thick_tubewall_cond, D_collar_cond, A_front_cond, A_fflow_cond,
A_a_cond, A_r_cond, A_x_r_cond, A_airDIVfflow_cond, A_fin_cond, A_finDIVair_cond)
```

```
Call CONDCAL(fluidname$, P_atm, Relrough_cond, A_x_inlet_header_cond, A_x_exit_header_cond, D_inner_cond,
D_outer_cond, D_return_cond, D_collar_cond, Dh_cond, density_fin_cond, pitch_fin_cond, height_fin_cond,
thick_fin_cond, K_fin_cond, pitch_Louver_cond, length_Louver_cond, height_Louver_cond, theta_Louver_cond,
S_trans_cond, S_long_cond, Thick_MCT_cond, pitch_MCT_cond, width_cond, depth_cond, A_front_cond,
A_fflow_cond, A_a_cond, A_r_cond, A_x_r_cond, A_airDIVfflow_cond, A_fin_cond, A_finDIVair_cond, P_r_in_cond,
h_r_in_cond, T_a_in_cond, m_dot_r_cond, m_dot_a_cond, N_elem_cond, N_slab_cond, N_tube_front_cond,
N_row_avg_cond, Nhxcond, N_circ_cond[1..Nhxcond], N_pass_per_mod_cond[1..Nhxcond], N_mod_cond
[1..Nhxcond], PF_cond, EF_cond, eta_fan_cond : Q_cond, h_r_out_cond, P_r_out_cond, W_dot_fan_cond,
T_a_out_cond, f_sup_cond, f_2ph_cond, f_sub_cond, charge_cond)
```

```
T_sat_out_cond = TEMPERATURE(fluidname$, P=P_r_out_cond, x=0)
DT_sub = T_sat_out_cond - T_r_out_cond
```

```
"-----MAIN-----"
```

```
"COLD UPSTREAM AND HOT DOWNSTREAM MARCHING"
```

```
$include C:\Documents and Settings\srajan\My Documents\My Classes\EES_folder\EES learning\New EES\ACRC
Simulation Models v\ACRC Simulation Models v.1\EES Files\Library\ACRC UserLib v.1.LIB
```

```
{fluidname$='R410A'} "Working fluid"
T_c_sat_out_IHX = Temperature(fluidname$, P=P_c_out_IHX, x=1)
{T_c_sat_in_IHX = -28.95}
{DELTAT_sup_IHX = 5}
T_c_sat_in_IHX = Temperature(fluidname$, P=P_c_in_IHX, x=1)
DELTAT_sup_IHX = T_c_out_IHX - T_c_sat_out_IHX
{P_h_in_IHX = 1748}
{Q_evap = (m_dot_r_c/2)*(h_c_in_IHX - h_h_out_IHX)}
Q_evap = 3.4}
{Q_evap_real = 2*Q_evap}
{m_dot_real = 2*m_dot_r_c}
{x_h_in_IHX = 0.1}
{h_h_in_IHX = 111.4}
{m_dot_r_c = 0.014/2}
{x_c_in_IHX = 1.01}
```

```
Q_IHX_real = Q_IHX*2
```

```
{input geometry parameters}
L_Total_IHX = 1.32 "Length of IHX , m"
D_port_c = 22.88/1000 "Cold line port hydraulic diameter, m"
D_port_h = 10.415/1000 "hot line port hydraulic diameter, m"
```

```
Relrough_h = 0 "Relative roughness=roughness /Diameter of tube"
Relrough_c = 0
PF_IHX = 1 "Pressure penalty factor"
EF_IHX = 1 "Refrigerant side heat transfer enhancement factor"
```

```
N_elem_IHX = 5 "Element number for IHX"
m_dot_r_h = m_dot_r_c
```

```

A_x_h_total = 0.00008519      { 0.0000875 }      {0.000025}      "Hot side equivalent
refrigerant flow area"
A_x_c_total = 0.0004111      { 0.00038 }      {0.000049}      "Cold side equivalent refrigerant
flow area"
A_c_total = 0.09129          {0.0898 }          {0.1063}          "Cold side heat transfer area"
A_h_total = 0.04155          { 0.0408 }          {0.1063}          "Hot side heat transfer area"

Call State_Calc_IHX(fluidname$, P_c_out_IHX, h_c_out_IHX : T_c_out_IHX, x_c_out_IHX)
Call State_Calc_IHX(fluidname$, P_h_in_IHX, h_h_in_IHX : T_h_in_IHX, x_h_in_IHX)

{Call the IHX_Calc procedure to solve the internal heat exchanger}
Call IHX_Calc(fluidname$, EF_IHX, PF_IHX, L_total_IHX, D_port_c, D_port_h, A_x_c_total, A_c_total, A_x_h_total,
A_h_total, Relrough_c, Relrough_h, P_c_out_IHX, h_c_out_IHX, P_h_in_IHX, h_h_in_IHX, m_dot_r_c, m_dot_r_h,
N_elem_IHX : P_c_in_IHX, h_c_in_IHX, P_h_out_IHX, h_h_out_IHX, Q_IHX, fraction_sup, charge_IHX)

Call State_Calc_IHX(fluidname$, P_c_in_IHX, h_c_in_IHX : T_c_in_IHX, x_c_in_IHX)
Call State_Calc_IHX(fluidname$, P_h_out_IHX, h_h_out_IHX : T_h_out_IHX, x_h_out_IHX)

"----- main program -----"

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{----- evaporator module-----}

"Computational parameters used for evaporator model"
N_elem_perpass_evap=4          "Number of elements per pass in the evaporator"
N_element_depth_evap=5        "Number of elements along evaporator depth used to separate
sensible and latent heat"

Evap_type$ = 'TubeFin'      {'microch'}          "Type of the heat exchanger; Microch or TubeFin"
{fluidname$='R410A'}          "Working fluid"
marching_evap$ = 'downstream'      "Finite element marching"
T_indoor = -22.25
T_dp = DEWPOINT(AirH2O,T=T_a_in_evap,P=P_atm,R=Rel_Humidity_in_evap)
Rel_Humidity_in_evap=0.5
{P_atm=101.35 }
"evaporator operating condition inputs"
eta_fan_evap = 0.2          " fan efficiency "
V_dot_a_evap=1400*CONVERT(cfm,m^3/s)      " Volumetric air flow rate, m3/s" {check this factor for face
velocity}
T_a_in_evap=T_indoor          " Air inlet temperature"

{Q_evap_real = Q_evap*2}

{P_r_in_evap = 300}
{h_r_in_evap = 93.09}
{T_r_in_evap=-28}          "Refrigerant inlet temperature"
{x_r_in_evap=0.25}          " Refrigerant inlet quality for the evaporator
first element "
{m_dot_r_evap=0.014[kg/s]}
{
m_dot_r_e = m_dot_r_evap*2}

"Evaporator size and circuiting" {This crossflow evaporator has 3 circuits interlaced among 3 rows during their 10
passes across hx}

Nhxevap=1          "Number of sub HX which are connected in series"
N_slab_evap=1      "Number of slabs in the heat exchanger"

```

```

N_row_avg_evap = 6                                "Average number of rows in the heat exchanger"
N_row_max_evap = 6                                "Maximum number of rows in the heat exchanger"

N_mod_evap[1] = 2                                  "Number of identical modules, in 1st sub HX "
N_pass_per_mod_evap[1]=5                          "Number of tube passes per module as seen from the front of the heat
exchanger, in 1st sub hx"
N_circ_evap[1] = 3                                "Number of parallel circuits, in1st Sub HX"

"Evaporator Size"
width_evap=(82.660)*Convert(in,m)

"Evaporator tube inputs"
D_inner_evap= (0.485-0.032)*convert(in,m)  {0.255*Convert(in,m) }   {9/1000}

"Louver dimensions"
theta_Louver_evap=25                                " louver angle[deg]"
pitch_Louver_evap=1/1000                            " louver pitch [m]"
length_Louver_evap=4/1000                          " louver length [m]"

"*****"
"Inputs only for the microchannel evaporator"
port_type_evap$ = {'round'}      'rectangular'      "round or rectangular"
Height_fin_evap=5/1000
thick_end_evap = 0.4/1000          "Thickness at MCT tube end, [m] "
thick_wall_evap = 0.4/1000        "MCT Wall thickness, [m] "
thick_web_evap = 0.3/1000         "MCT Web thickness, [m] "
N_port_evap= 24                   "Number of ports in an MCT"
D_w_evap = 1.5/1000               "microchannel port width"
D_h_evap = 1.5/1000               "microchannel port height"

"*****"
"Inputs only for the finned-tube evaporator"

S_trans_evap = 1.666*convert(in,m)      "Transverse tube spacing"
S_long_evap = 1.443*convert(in,m)       "Longitudinal tube spacing"
D_outer_evap = 0.485*convert(in,m)      "Tube outer diameter"
D_return_evap = S_trans_evap            "Return bend diameter"
"*****"

Rough_tube_evap=0.000005                "Tube roughness [m]"
k_tube_evap=0.403                      "227*CONVERT(Btu/h-F-ft,kW/m-K)"  " Conductivity of tube[kW/m-K]"
PF_evap=1                              " pressure drop penalty factor for
microfinned tube "
EF_evap=1                              " enhancement factor for
microfinned tube " "heat transfer enhancement assumed =
area ratio of microfinned tube"

"evaporator fin inputs"

density_fin_evap=3*CONVERT(1/in,1/m)    " fin number per m"
thick_fin_evap=0.0075*convert(in,m)     " thickness of fin, m "
k_fin_evap=128.3*CONVERT(Btu/h-F-ft,kW/m-K) " conductivity of fin [kW/m-K]"

"Header dimension"
A_x_inlet_header_evap = PI*(7.1/1000)^2/4 "inlet header flow area, m2"
A_x_exit_header_evap = PI*(7.1/1000)^2/4  "exit header flow area, m2"

"Calculate variables needed by evaporator Procedures"
SHR = Q_sen_evap/(Q_evap)
m_dot_a_evap=V_dot_a_evap/VOLUME(air,T=T_a_in_evap, P=P_atm)
" Air mass flow rate, kg/s "
Omega_a_in_evap=HUMRAT(AirH2O,T=T_a_in_evap,P=P_atm,R=Rel_Humidity_in_evap) " Air inlet
humidity, kg/kg "

```

```

call State_Calc_Evap(fluidname$, P_r_in_evap, h_r_in_evap : T_r_in_evap, x_r_in_evap)
    "Refrigerant inlet temperature and quality"
Relrough_evap=Rough_tube_evap/D_inner_evap
    "Relative roughness=roughness /Diameter

    of tube "

Call Area_Calc_evaporator(Nhxevap, N_row_max_evap, N_row_avg_evap, N_mod_evap[1..Nhxevap],
N_pass_per_mod_evap[1..Nhxevap], N_elem_perpass_evap, N_port_evap, width_evap, D_h_evap, D_w_evap,
D_inner_evap, D_outer_evap, thick_wall_evap, thick_web_evap, thick_end_evap, S_long_evap, S_trans_evap,
density_fin_evap, Height_fin_evap, pitch_Louver_evap, theta_Louver_evap : N_totalpass_evap,
N_tube_front_evap, N_elem_evap, depth_evap, Height_evap, Dh_evap, pitch_fin_evap, thick_MCT_evap,
height_Louver_evap, pitch_MCT_evap, Thick_tubewall_evap, D_collar_evap, A_front_evap, A_fflow_evap,
A_a_evap, A_r_evap, A_x_r_evap, A_airDIVfflow_evap, A_fin_evap, A_finDIVair_evap)

Call EVAPCAL(fluidname$, P_atm, Relrough_evap, A_x_inlet_header_evap, A_x_exit_header_evap, D_inner_evap,
D_outer_evap, D_return_evap, D_collar_evap, Dh_evap, density_fin_evap, pitch_fin_evap, height_fin_evap,
thick_fin_evap, k_fin_evap, pitch_Louver_evap, length_Louver_evap, height_Louver_evap, theta_Louver_evap,
S_trans_evap, S_long_evap, Thick_MCT_evap, pitch_MCT_evap, width_evap, depth_evap, A_front_evap,
A_fflow_evap, A_a_evap, A_r_evap, A_x_r_evap, A_airDIVfflow_evap, A_fin_evap, A_finDIVair_evap,
P_r_in_evap, h_r_in_evap, T_a_in_evap, Omega_a_in_evap, m_dot_r_evap, m_dot_a_evap, N_elem_evap,
N_slab_evap, N_tube_front_evap, N_row_avg_evap, Nhxevap, N_circ_evap[1..Nhxevap],
N_pass_per_mod_evap[1..Nhxevap], N_mod_evap [1..Nhxevap], PF_evap, EF_evap, eta_fan_evap : Q_evap,
h_r_out_evap, P_r_out_evap, Q_sen_evap, W_dot_fan_evap, T_a_out_evap, Omega_a_out_evap, f_sup_evap,
charge_evap)

Q_lat_evap = Q_evap - Q_sen_evap
Call State_Calc_Evap(fluidname$, P_r_out_evap, h_r_out_evap : T_r_out_evap, x_r_out_evap) "Refrigerant
outlet temperature and quality"
DT_sup = T_r_out_evap - T_sat_out_evap
T_sat_out_evap = TEMPERATURE(fluidname$, P=P_r_out_evap, x=1)

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