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ABSTRACT

Transient simulations of refrigeration cycles are crucial for evaluating new refrigerants. Previously developed transient simulation tool was capable of simulating refrigeration systems using refrigerants R22 and R134a and was validated with experimental data; however, incorporating non-traditional refrigerants such as R744 (carbon dioxide) change the cycle operation beyond the capabilities of the original code. This paper introduces a new enthalpy based solver that accounts for the transcritical behavior of R744. In the proposed transient solver, enthalpy and pressure are the independent properties being passed between system component models. The simulation software was developed in an object-oriented framework, allowing for easy component inter-changeability and the use of custom control algorithms. This paper presents the system level solution procedure of refrigeration cycles. Results of the current solver along with new heat exchanger model provided coherent results that need to be further validated for future purposes.

1. INTRODUCTION

Environmental concerns associated with the use of synthetic refrigerants have lead to a substantial increase in natural refrigerant research. In pursuit of alternative refrigerants, new artificial refrigerants such as R410A have been developed; nevertheless, the search included previously-used natural refrigerants such as R717 (ammonia) and R744 (carbon dioxide). Of these, carbon dioxide has gained significant interest; earlier investigation results are shown in Hwang (1997) and Hwang and Radermacher (1998). Billard (2002) and Pearson (2005) documented the increase in research carried out in the field of R744 applications. To cope with this growing interest, transient simulation models should be capable of modeling the transcritical behavior of R744 systems. Transient system simulations provide a less costly and more attractive alternative to experimentation for manufacturers, especially during the design and analysis phase of the product development process. Flexible system simulation can be addressed by the use of component based solvers where component models are integrated into the system.

Previous dynamic vapor compression simulations Anand (1999), (Bendapudi *et al.* 2002) incorporated bulk entity transient heat exchanger models (Jacobsen, 1995); further investigation by Willatzen *et al.* (1998) and Pettit *et al.* (1998) gives a complete formulation of two phase flow heat exchangers which includes a lumped three region transient model. Pfafferott and Schmitz (2004) developed a Modelica library for CO2-refrigeration systems; their work also included validation with experimental data.

In the current heat exchanger model, the three regions were considered with moving boundary. The heat exchanger performance is governed by the refrigerant charge and inlet and outlet refrigerant mass flow rates. The moving boundary approach incorporated in the present model represents a hybrid approach between the lumped and the finite volume heat exchanger models; this feature improves the modeling capabilities while maintaining model simplicity. Numerical results shown at the end of this paper show sound results; yet further validation with experimental data needs to be done.

2. COMPONENT BASED SOLVER

The proposed transient simulation software is built in an object oriented environment. To gain additional flexibility, system components are being modeled separately as well as the different system solvers. Currently available components include: compressor, heat exchangers, capillary tube with suction line heat exchanger, orifice, accumulator, damper, refrigerator cabinets, display cabinet, automotive cabin, cold plate evaporator and thermostats. The components are lumped models and currently air and refrigeration side pressure drops on heat exchangers are neglected. In the following section, modeling of individual components is briefly described. For more details the reader is referred to Anand (1999).

2.1. Modeling approach

The system simulation is based on the transient component based solver developed by Anand (1999) and Gercek et al(2005). The solver is continuously enhanced to widen the scope of applications to include automotive climate control systems, supermarket applications and other refrigeration and heat pump applications. The overall goal of the simulation tool is to accurately represent real world system behavior in a reasonable run time. Consequently, component modeling is kept as simple as possible and discretization is avoided. However, solver flexibility allows users to replace the default component models with more complex and custom ones at the expense of simulation run times.

In addition, the component based modeling approach deals with system components as black box objects that communicate with each other through inlet and outlet ports. Component models are developed in accordance with a Component Standard to facilitate system and component communication in a versatile manner. Furthermore, custom control algorithms are implemented. The tool can employ multiple cascaded control algorithms to provide complex control strategies. System configuration, components and the control algorithms can be saved in a portable data format so that it can be opened for later use. NIST REFPROP 7.0 (Lemmon *et al.,* 2002) is used for refrigerant thermophysical properties, along with a custom wrapper to speedup the property computation. Once system components are loaded and initial conditions are set, the program can be executed. After the program starts, the user can monitor the time history of all variables of interest such as the pressures, temperatures, mass flow rates, charge and quality at different state points. Settings of controls and or other input variables can be changed on the fly to simulate interactions as they may occur in the real world or in a laboratory setup (Gercek *et al.,* 2005).

2.2 Solver Description

The component models listed in the previous section form the system model. The system is solved using successive substitution method. During every time step, each component is run consecutively and output state of each component is propagated as an input to the next component. Initial system conditions are input by the user. Input parameters are the component physical properties, charge and initial wall and air temperatures. The system starts solving each time step with calculating the compressor and expansion device mass flow rates and then marches to the next component respectively in refrigerant flow direction. After running all components in the system and updating the newly calculated parameters in the system, the convergence criterion is checked. If the convergence is satisfied, new transient parameters are calculated based on the parameter time derivative and time step. Adaptive time step is applied to limit the pressure change to 1 kPa based on the evaporator and condenser pressure derivatives. The system is solved again for the new charge and pressure distributions at next time step. Convergence criteria are based on the component inlet and outlet enthalpies and mass flow rate; 10^{-3} relative error between successive iterations is used for convergence check.

3. COMPONENT MODELLING

Modified component models were developed to adapt with the new enthalpy based solver. These modifications include refrigerant properties evaluation calls, components inputs and outputs. Brief description of the above mentioned components along with further modification in the compressor and heat exchanger models are discussed below.

3.1 Compressor

Compressor is modeled with two control volumes. The first control volume is the compressor swept volume. It is modeled considering constant isentropic and mechanical efficiencies; the discharge temperature is calculated assuming polytropic compression. The polytropic index is evaluated based on Equation (1) (Hwang and Radermacher, 1997)

$$n_p = \frac{h_{2s} - h_1}{h_{2s} - h_1 - P_2 v_{2s} + P_1 v_1} \quad (1)$$

The second control volume is the compressor wall. In this control volume, the entire wall is assumed to be at a uniform temperature. The wall temperature changes over time due to the heat transfers between the ambient air on the outer surface and the refrigerant on the inner surface. There is no charge storage inside the compressor.

Input parameters for compressor are the refrigerant inlet enthalpy and inlet and outlet pressures. Compressor shell temperature is stored inside the component and updated before each time step. the model calculates outlet enthalpy, mass flow rate and time derivative of the shell temperature.

3.2 Heat Exchanger Model

The proposed heat exchanger model can be used for both the condenser and evaporator. The model incorporates moving boundaries to model the vapor, two-phase and liquid zones separately. Figure 1 below illustrates the heat exchanger with the two moving boundary limits, in addition to the control volumes used. Control volumes I, II, III are for the refrigerant, wall, and air respectively. Unlike previous transient heat exchanger models, the two-phase region void fraction is calculated based on the Martinelli parameter (Rice, 1987). The Martinelli parameter is calculated at a specific quality; hence in order to calculate the void fraction of the lumped two-phase region the parameter is integrated over the whole length from the inlet quality to the exit quality.

The underlying assumptions in the used generic heat exchanger are:

- One dimensional fluid flow
- Equal inlet and outlet mass flow rates in the single phase regions.
- The pressure drop in the heat exchanger is neglected.
- Constant heat transfer coefficients between the refrigerant and the wall for the three regions
- Constant airside heat transfer coefficient
- The charge is governed by the compressor and orifice performance
- Uniform wall temperature

Input parameters for the heat exchanger models are the inlet and outlet mass flow rates, and inlet enthalpy. Pressure, charge and wall temperature are maintained by the component model and updated before each time step. The model calculates outlet refrigerant state and total airside and refrigerant side heat transfer. In addition, wall temperature, internal energy and refrigerant charge gradients are evaluated for the time step updates.

The heat exchanger boundaries are determined based on the inlet refrigerant state and the refrigerant charge inside the heat exchanger. As such, charge limits are calculated for comparison. The heat exchanger charge is compared with these limits to determine the possible number of phases within the heat exchanger based on the inlet refrigerant state. Table 1 below defines the different charge limits calculation for both condenser and evaporator models. For a condenser, the vapor section length is calculated based on the required heat transfer area as shown in Equations (2) and (3). Based on the calculated length and the average vapor density, the charge inside the vapor region is calculated. The remainder of the charge should be in the 2-phase and the liquid regions such that the overall charge is satisfied, i.e. Equations (4) and (5) should be solved simultaneously.

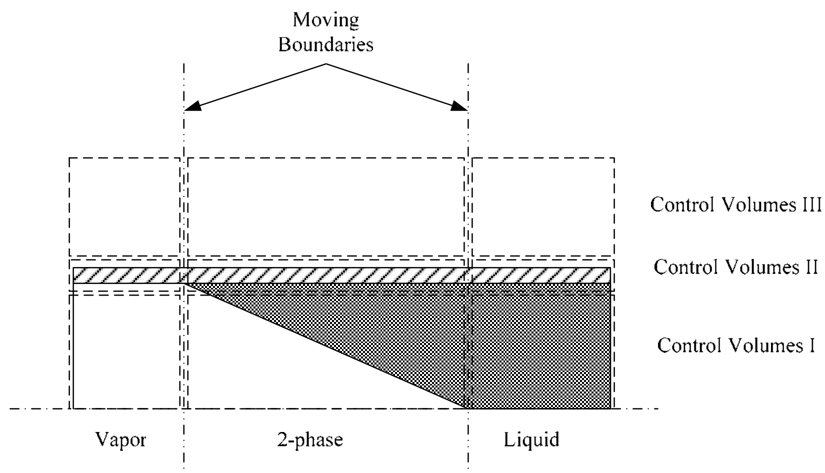

Figure 1: Heat Exchanger Model Description

Table 1: Heat Exchanger Charge Limits.

Charge limit	Symbol	Definition
Ideal 2-phase Charge (from x =0.0 to x = 1.0)	$\overline{M_{fg}}$	$\overline{M_{fg}} = \rho_{fg} \times V$, Where $\rho_{fg} = \alpha \times \rho_g + (1-\alpha) \times \rho_f$
Condenser, all vapor	$\overline{M_g}$	$\overline{M_g} = \dfrac{(\rho_{in} + \rho_g)}{2} \times V$
Evaporator, all vapor	$\overline{M_g}$	$\overline{M_g} = \rho_g \times V$
Condenser, all liquid	$\overline{M_f}$	$\overline{M_f} = \rho_f \times V$
Evaporator, all liquid	$\overline{M_f}$	$\overline{M_f} = \rho_{in} \times V$

$$L_g = \frac{\dot{m}_{in} \times c_p (T_{ref_{in}} - T_{sat})}{U_g \times \pi \times D \times LMTD_g} \tag{2}$$

$$LMTD_g = \frac{T_{ref_{in}} - T_{sat}}{\ln\left(\dfrac{T_{ref_{in}} - T_{wall}}{T_{sat} - T_{wall}}\right)} \tag{3}$$

$$L_{fg} + L_f = L_{H.X.} - L_g \tag{4}$$

$$M_{fg} + M_f = M_{H.X.} - M_g \tag{5}$$

Where

$$M_g = \rho_g \times V \times \frac{L_g}{L_{H.X.}}, \text{ or } M_g = \left(\frac{\rho_{in} + \rho_g}{2}\right) \times V \times \frac{L_g}{L_{H.X.}}$$

$$M_{fg} = \rho_{fg} \times V \times \frac{L_{fg}}{L_{H.X.}}$$

$$M_f = \rho_f \times V \times \frac{L_f}{L_{H.X.}}, \text{ or } M_f = \rho_{in} \times V \times \frac{L_f}{L_{H.X.}}$$

In case there is no subcooled region, the outlet enthalpy is calculated based on the required charge in the 2-phase region. If the subcooled region exists, the outlet temperature is calculated based on the heat transfer calculation as shown in Equations (6) and (7) which are solved simultaneously.

$$T_{ref_{out}} = T_{sat} - \frac{U_g \times \pi \times D \times L_f \times LMTD_f}{m^{\bullet}_{out} \times c_{p_f}} \tag{6}$$

$$LMTD_f = \frac{T_{sat} - T_{ref_{out}}}{\ln\left(\frac{T_{sat} - T_{wall}}{T_{ref_{out}} - T_{wall}}\right)} \tag{7}$$

However, as the cycle march in time, the condenser pressure increases reaching supercritical pressures. At this point, only the vapor section is considered and the outlet temperature is calculated as in Equations (8) and (9) below.

$$T_{ref_{out}} = T_{ref_{in}} - \frac{U_g \times \pi \times D \times L_{H.X.} \times LMTD_g}{m^{\bullet}_{in} \times c_p} \tag{8}$$

$$LMTD_g = \frac{T_{ref_{in}} - T_{ref_{out}}}{\ln\left(\frac{T_{ref_{in}} - T_{wall}}{T_{ref_{out}} - T_{wall}}\right)} \tag{9}$$

The cycle pressures are governed by the performance of the heat exchangers, compressor and orifice. The new time step pressure is evaluated based on refrigerant charge and internal energy inside the heat exchanger. The charge reflects the compressor and orifice performance while the internal energy reflects the heat exchanger performance.

3.3 Orifice Model

A simple adiabatic orifice is used with constant discharge coefficient. If the condenser outlet is single phase, the orifice runs normally. However, if the condenser outlet is 2-phase the orifice mass flow rate is set to minimum value (0.1% of the rated value) in order to simulate blocked conditions. The model also accounts for reversed flow during compressor off-cycles.

3.4 Thermostat Model

A thermostat model is used to control the compressor and fans operation via feedback from user selected sensor. Available sensors are cabinet air and wall temperatures and compressor status “for cascaded control algorithms”. Physical and fictitious actuators are used as required for the component models.

3.5 Cabinet Model

A detailed cabinet model is used. Each compartment within the cabinet model is divided into four control volumes. These are the compartment itself, the ambient air surrounding the compartment, the compartment wall and the air inside compartment, detailed description of the cabinet model can be found in Anand (1999) and Gercek *et al.* (2005). Cabinet heat loss is based on each wall’s heat transfer resistance. Since the fan operation (on or off) will affect the heat convection coefficient inside the compartments, the heat transfer resistance values for each wall are calculated for each of the two cases. After calculating the heat transfer resistance for each wall, the total UA values are calculated. For the door seals, the seal length and the heat transfer coefficient per unit length is specified. Finally, total heat losses from food and freezer cabinets are calculated by summarizing all UA contributions to obtain the overall UA value.

4. NUMERICAL RESULTS

In the current study a small scale top mount R744 refrigerator is modeled using the above mentioned models. The system was first sized using a steady state solver to check the steady state capacity and performance. Figure 2 below describe the refrigeration system used.

The cycle is started from ambient conditions at 298.15 K and allowed to run for a simulation time of 24 hours. The thermostat is used to turn off the compressor and evaporator fans at a temperature of 280 K and then back on when the freezer air temperature is 284 K. Results shown hereinafter are zoomed for the pull-down and two compressor cycles only. In the current results, the new time step pressure is calculated based on the heat exchanger charge and average temperature instead internal energy; this explains the smooth pressure curves. Figures 3 represent the freezer and cabinet wall temperature profiles along one day of simulation. The pull-down time was 132 minutes. That pull-down time is relatively high due to the small evaporator capacity compared with the cabinet size and load. Also the subsequent load cycles are 25 minutes off and 30 minutes on showing a compressor percentage on time of 55%.

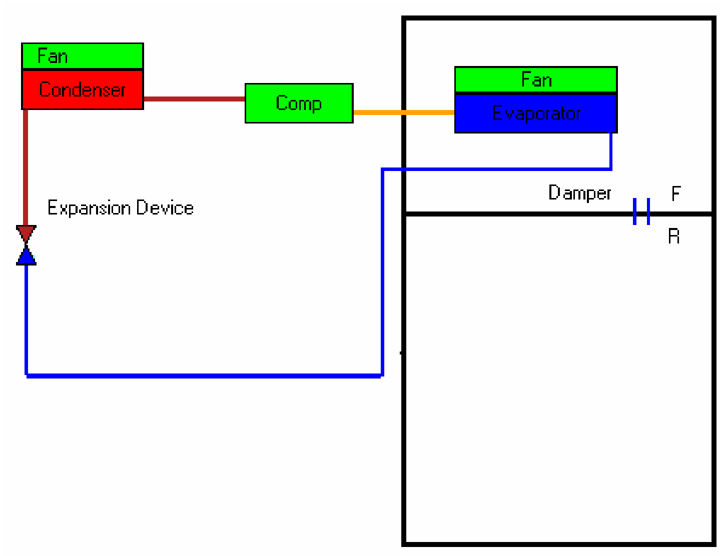

Figure 2: Top mount cabinet system description

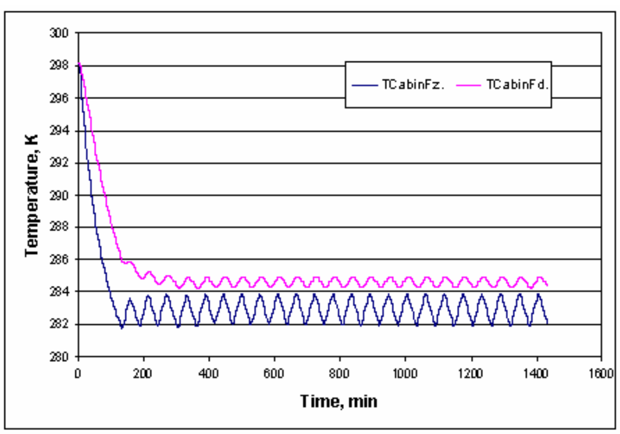

Figure 3: Freezer and cabinet wall temperature profiles along 1 day

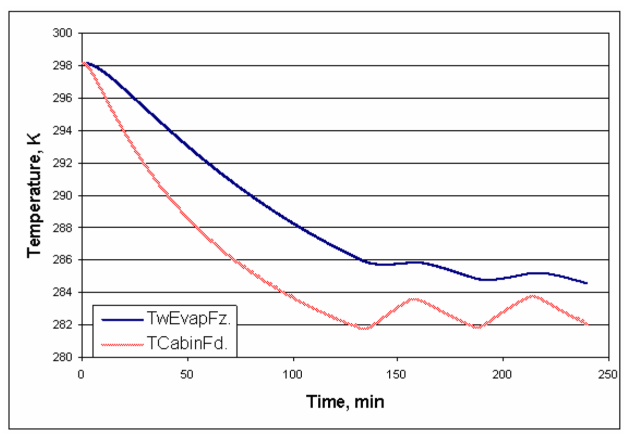

Figure 4: Freezer and cabinet wall temperature profiles for the first 240 minutes

Condenser and evaporator pressures shown in Figure 5 clearly identify the compressor cycling as well as the heat exchanger performance during the off-cycle which is mainly governed by the orifice discharge. Furthermore, charge inventory inside the heat exchangers is illustrated in Figure 6 showing consistent results with the cycle pressures and compressor cycling. The orifice capability to handle reversed flow improved the charge balance especially during the off-cycles.

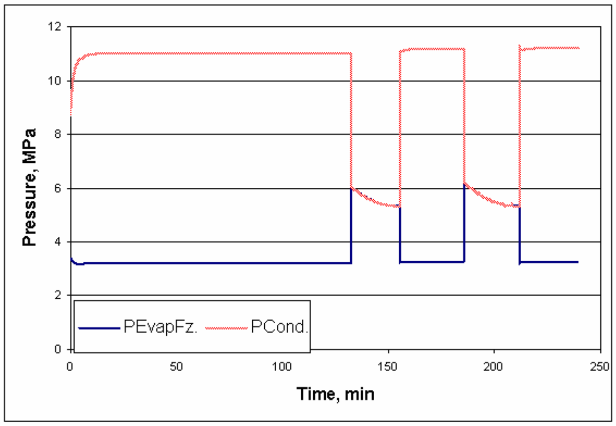

Figure 5: Gascooler and Evaporator pressures for the first 240 minutes

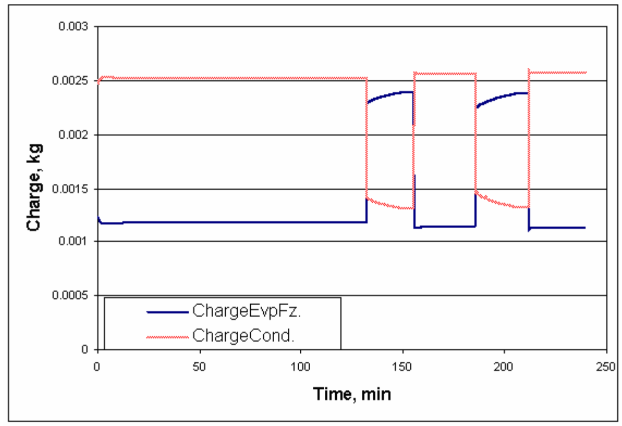

Figure 6: Gascooler and Evaporator charge for the first 240 minutes

The pressure decline during the off-cycle is due to the gascooler wall cool down to ambient; it is important to notice that absolute value of pressure derivative decrease with time as the wall temperatures approach ambient temperature. The current compressor model uses a constant mass flow rate limit. As such the compressor restart is associated with a maximum value of mass flow rate due to the small pressure ratio (≈ 1). This assumption may cause some spikes in the condenser charge and hence condenser pressure. A small spike is shown associated with the second compressor restart.

5. CONCLUSIONS

A new generic heat exchanger model is developed for use in a component based transient simulation tool for vapor compression systems. This generic nature of this heat exchanger model allows for the use of conventional (R134a, R22) and alternate refrigerants (R744, R410A) in the simulation. Additional changes were required for other components to adapt with the new enthalpy based approach. These changes included component input and output parameters, refrigerant property calls. Furthermore, the compressor model incorporated a new equation for polytropic index calculation for R744. Results shown hereinbefore depict the soundness of the new suggested heat exchanger model. The use of three lumped regions for each refrigerant phase reduced the amount of assumptions and allowed for better component modeling. Also, it is shown that the new models along with other components where able to handle the transcritical behavior of R744. The model robustness is also demonstrated by the ability to perform extended real time simulations. Validation with experimental data should be made in the future to insure the simulation accuracy and allow for wide model use.

NOMENCLATURE

c_p	specific heat	(J/kg.K)
D	internal tube diameter	(m)
h	refrigerant enthalpy	(kJ/kg)
L	length	(m)
$LMTD$	logarithmic mean temperature difference	(K)
$\dot{m}$	refrigerant mass flow rate	(kg/s)
M	charge	(kg)
n_p	polytropic compression index	(–)
P	refrigerant pressure	(Pa)
T	temperature	(K)
U	average heat transfer Coefficient	(W/m^2.K)
V	heat exchanger internal volume	(m^3)
v	refrigerant specific volume	(m^3/kg)
x	vapor quality	(–)

Greek Letters

α	Void fraction	(–)
ρ	Density	(kg/m^3)

Subscripts

f	liquid
fg	2-phase
g	vapor
H.X.	heat exchanger
in	inlet
out	outlet
ref	refrigerant
sat	saturation

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