SIMULATION FOR REFRIGERANT CHARGE DIAGNOSTICS IN SUPERMARKET APPLICATIONS

Kiengkrai Assawamartbunlue[1] and Michael Brandemuehl[2]
[1]Sirindhorn International Institute of Technology, Thailand
[2]University of Colorado, Boulder, CO, USA

ABSTRACT

This paper describes a computer program, DRSSIM, that can predict liquid refrigerant level in the receiver of a distributed refrigeration system in a supermarket. DRSSIM has been developed to be further used for refrigerant leakage detection and diagnostic technique in the system. Unlike a simple refrigeration system commonly implemented in HVAC application, the distributed refrigeration system in a supermarket is more complicated. It can consist of several compressors, condensers, and racks of refrigerated cases. An important component that makes the system different from the simple refrigeration system is the presence of a high-side liquid receiver, which affects the balance of the whole system. Modeling this complicated system requires several techniques to reduce calculation time and be suitable for fault detection. The component and system models developed as parts of DRSSIM are steady-state models based on the first-principle engineering equations. DRSSIM can predict the liquid refrigerant level in the receiver with the root mean square error of 1.3%. DRSSIM also has a modular design so that the user can easily add or modify refrigeration components or system configuration.

INTRODUCTION

The refrigeration system implemented in a supermarket application is more complicated than one in an HVAC application. The system can contain several compressors, condensers, racks of refrigerated cases, a subcooler, and an oil separator. In addition, there can be a high-side liquid receiver and a heat recovery system installed into the system. The components are connected with each other using refrigerant pipe that can be as long as 1500 meters. Modeling of this complicated distributed refrigeration system is more difficult than one of a simple refrigeration system, which consists of a compressor, a condenser, an evaporator, and an expansion device. The Distributed Refrigeration System Simulation, DRSSIM, is initially designed for fault detection and diagnosis in the distributed refrigeration system, especially a refrigerant leakage fault. While there are many possible faults in supermarket systems, discussions throughout this paper will focus on the prediction of the refrigerant charge level in the liquid receiver.

Several refrigeration component and system models were developed in the past to predict performance and behavior of such components and system. The first public domain steady-state refrigeration system model for HVAC applications was developed by Hiller and Glicksman in 1976 (Hiller 1976). After that, two public computer models, based on their models, were released in 1983. The first is HPSIM (Domanski 1983) distributed by the National Institute of Standards and Technology. The second is PUREZ (Rice 1983, 1994) distributed by Oak Ridge National Laboratory (ORNL). The capabilities of these two programs have been summarized and compared by Damasceno and Goldschmidt (Damasceno 1989). More recent component models include ACMODEL, developed by Rossi (1995).

This paper describes the development and evaluation of a new computer program, DRSSIM, for simulating a distributed refrigeration system in supermarket applications. DRSSIM is developed after PUREZ (Rice 1994), now usually known as the Heat Pump Design Model (HPDM). Models are developed for each refrigeration component that is normally found in a distributed refrigeration system. The whole refrigeration system is then constructed from these component models. The developed models are steady-state models based on the first-principle engineering equations of system components. Preliminary validation of the system performance can be found in the author's thesis (Assawamartbunlue 2000). One primary goal is to identify the sensitivity of the model in predicting the liquid refrigerant level in the receiver at various system operating conditions. By comparing model predictions with measured values, it is possible to detect refrigerant leakage.

The system model is calibrated and validated with data collected from a supermarket store located in Longmont, Colorado. The store was built in 1998 and has a floor area of about 7500 m^2. The supermarket has a large number of refrigerated cases (evaporators) distributed throughout the store, including frozen food cases, ice cream cases, fresh meat cases, vegetable cases, walk-in cases, etc. There are two separate refrigeration systems to independently serve the low temperature and medium temperature cases. For the purposes of this analysis, we focus only on the low temperature system with evaporators below -12^{o}C . The main components consist of three compressors, 16 refrigeration cases, one evaporative condenser, one liquid receiver, one flow-through receiver, one subcooler, and one heat recovery system as shown schematically in Figure 1.

This system consists of five different types of refrigeration cases, which are grouped depending on their operating temperature. The low-temperature system serves two circuits. Circuit 1 has one compressor and serves four groups of refrigeration cases that operate between -34^{o}C to -28^{o}C. The discharge air temperature varies from -28^{o}C to -23^{o}C. The designed load is 29.2 kW and the circuit capacity is 35.8 kW, which is approximately 22% over capacity. Circuit 2 has two compressors and serves eleven refrigeration cases that operate at -28^{o}C to -26^{o}C. The discharge air temperature varies from -23^{o}C to -17^{o}C. The designed load is 70.8 kW and the circuit capacity is 96.8 kW, which is approximately 37% over capacity.

These two circuits have their own suction lines, but use the same discharge lines and condenser. The total length is approximately 1350 meters. The discharge pressure of each compressor is approximately the same because compressors are close to each other. Two compressors in circuit 2 have the same suction header line, thus, the suction pressure is also the same. Hot, high-pressure refrigerants from each compressor mix with each other and flow to an oil separator that is located near the compressors. Lubricant is extracted from the refrigerant in the oil separator and goes to an oil cooler that draws cooling refrigerant from a flow-through receiver located above a main liquid receiver. Cooled lubricant then returns to the compressors and the cooling refrigerants go to the condenser. At the exit of the oil separator, there is a three-way valve operating in an on/off mode to control the direction of refrigerants. Refrigerants

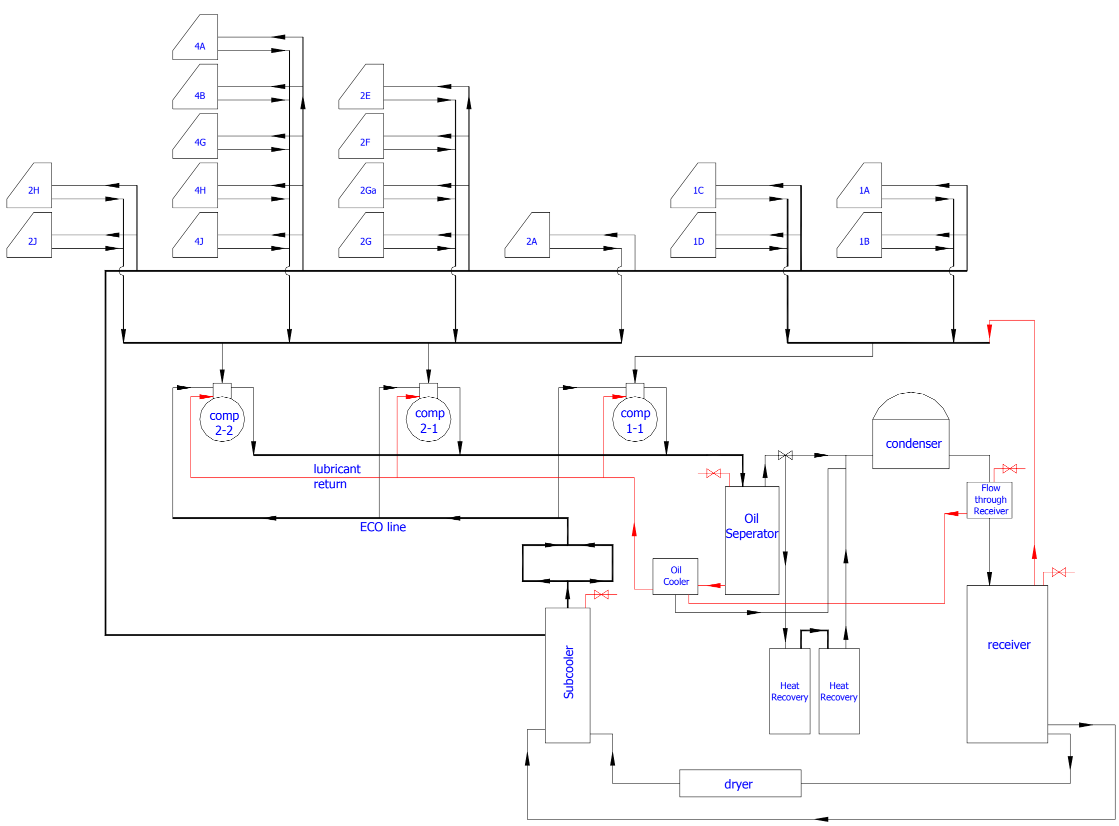

Figure 1: Schematic Diagram of the Prototype Supermarket

leaving the oil separator flow to either the heat recovery equipment or an evaporative condenser. Refrigerants entering the heat recovery system heat domestic hot-water for internal usage within the building, however, the heat recovery system rarely works in the current system setting. Refrigerants then go to the evaporative condenser. As they flow to the condenser, refrigerants are mixed with another refrigerant stream from the oil cooler.

The evaporative condenser is located at the second floor inside the building. Condensed refrigerants from the condenser go to the flow-through receiver and then the main liquid receiver. The flow-through receiver is located above the main liquid receiver to build up pressure for refrigerants that are used for the oil cooler. Two liquid refrigerant streams leave the liquid receiver. One stream is used to cool down the other at the subcooler. The pressure of the cooling refrigerant stream is reduced using an expansion device at the entrance of the subcooler. The subcooler is a flooded-type subcooler with a fixed refrigerant level. The valve is opened to allow more refrigerant if the level is below a setpoint. Inside the subcooler, there is a heat exchanger coil submerged in the cooling refrigerants. The other refrigerant stream from the receiver flows through this heat exchanger to subcool the refrigerant at relatively constant pressure before going to the main liquid line. During the heat transfer process in the subcooler, the cooling refrigerants are evaporated and vaporized, refrigerants are then drawn back to compressors via an economizer port of each compressor. The subcooled refrigerants go to a main liquid line and are then ready to be distributed to each refrigeration case.

Two refrigeration system circuits join the same main liquid line. There are two main distributed branches for circuit 1 and four branches for circuit 2. Each branch serves several refrigerant cases as shown in Figure 1. Each refrigerated case has its own expansion device and liquid solenoid valve or evaporator pressure regulation valve. The expansion device is normally installed at the entrance of the heat exchanger coil inside the refrigerated case and used to control refrigerant superheat. The liquid valve is also installed at the entrance before the expansion device and works as an on/off valve controlled by the case temperature sensor. In some refrigeration cases, there is an evaporator pressure regulation valve instead of a liquid solenoid valve. This valve is installed at the exit of the heat exchanger coil and used to build up pressure within the coil to control discharge air temperature of a case.

In addition to the main heat exchanger coil within each refrigeration case, there is a small built-in liquid-suction heat exchanger that is used to exchange heat between the refrigerant entering and leaving the refrigerated case. The leaving refrigerant from refrigerated cases then flows back to the compressors. As mentioned above, these two circuits operate at different suction pressure, therefore, there is a need for two main suction lines for each circuit, each with a pressure regulating valve.

Since the selected supermarket is an actual operating supermarket, the model development is based on the data available from the system's data acquisition system. Table 1 shows data available from the test site.

Table 1: Data Available at the Test Site

Sensor	Type	Unit
case temperature	Analog	^{o}C
suction pressure	Analog	kPa
compressor speed	Analog	% of full speed
liquid solenoid value on/off	Digital	-
discharge pressure	Analog	kPa
discharge temperature	Analog	^{o}C
liquid temperature	Analog	^{o}C
refrigerant charge level	Analog	% of full tank

MODELING AND SIMULATION

DRSSIM is modeled after PUREZ (Rice 1983). In DRSSIM, individual component subroutines include a compressor, condenser, evaporator, heat exchanger (i.e., subcooler, desuperheater), liquid receiver, and refrigerant lines.

There are two compressor models available in DRSSIM. The first model is a loss and efficiency-based model. The compressor considered is a reciprocating compressor. The modeling is based on the internal energy balances in a reciprocating compressor using user-supplied heat loss and internal efficiency values (Lebrun 1993). The second model is a map-based compressor model based on a set of regression coefficients. These coefficients are typically obtained from data generated during compressor testing under ARI Standard 540 (ARI 1999) and available from the manufacturer.

Heat exchanger performance is calculated using the effectiveness-NTU method and is based on the ASHRAE HVAC2 Toolkit (Brandemuehl 1993). The model is applied directly for the simple single-phase heat exchangers, e.g., subcooler, desuperheater, heat recovery heat exchanger. Two phase heat transfer is modeled with the effectiveness-NTU approach assuming that the two phase component is at a constant temperature. Components with two-phase heat transfer, e.g., condenser and evaporator, are modeled as a collection of single-phase and two-phase heat transfer elements. For example, the condenser is modeled as three sequential lumped masses to account for the desuperheating, condensing, and subcooling processes. The correlations used in PUREZ are adopted here to calculate a single- and two-phase heat transfer coefficient. The refrigerant mass flow rate from the compressor model is divided by the number of the circuits to obtain the flow rate for each circuit. It is assumed that refrigerant pressure drop within each section is small and can be neglected.

Unlike a typical HVAC system, supermarket refrigeration systems usually include a liquid receiver, shown schematically in Figure 2. At steady state, the liquid receiver contains both saturated liquid and vapor. There could be heat transfer between the receiver and the environment around the receiver due to the temperature difference. The refrigerant entering the receiver could be in a subcooled, a saturated, or a two-phase state, depending on the state leaving the condenser and the elevation difference between the receiver and the condenser. If the pressure inside the receiver is higher than condenser pressure, the saturated vapor at the top of the receiver flows back through the condenser via an equalizer line. A check valve is installed to prevent hot, high-pressure refrigerant at the entrance of the condenser from going directly to the receiver. The refrigerant leaves the receiver as a saturated liquid to the evaporators. The receiver is modeled using simple mass and energy balances.

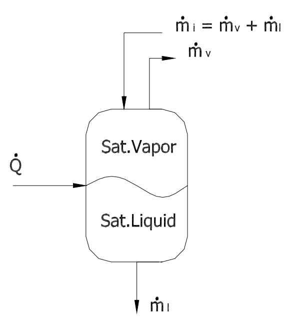

Figure 2: High Pressure Liquid Receiver

Refrigerant lines connect refrigeration components of the cycle together. As mentioned above, the refrigerant lines in supermarket systems are important. They can be as short as a few centimeters to connect the expansion device to the evaporator coil, or as long as several hundred meters, to connect the compressor and the condenser or the evaporator and the compressor. Also, each component may be installed at different elevations. The pressure gains and losses between equipment can dramatically change. The heat transfer between the pipes and the environment around it is significant because of the exposed pipe area. All of the refrigerant-side pressure losses are computed on the basis of equivalent lengths. Thus, the losses of joints and bends have to be converted to equivalent length as an input of the program. The pressure drop is calculated based on Bernoulli equations.

The calculations of refrigerant charge in refrigeration components are divided into two groups based on refrigerant phase. The first group is for single-phase refrigerant, which is either subcooled or superheated refrigerant. The calculations for this group are simple and straightforward. The charge in the components depends on average density and component volume.

$$m = \int_0^L \rho dV = A_c L \frac{\int_0^L \rho dl}{\int_0^L dl} = V\rho_{avg} \qquad 1$$

where ρ is the local single-phase refrigerant density along the tube, A_c is the cross-section area, and L is the length of tube. ρ_{avg} is the average density over the tube length and V is the tube volume.

The second group is for two-phase refrigerant. The charge calculation is more complicated and depends on void fraction and heat flux models. As with the first group, the charge is computed from density and volume, but accounts for the difference between liquid and vapor densities.

$$m = V\left[\frac{\rho_g \int_0^L \alpha dl + \rho_f \int_0^L (1-\alpha) dl}{\int_0^L dl}\right] \qquad 2$$

where ρ_g is the density of saturated vapor, ρ_f is the average density of saturated liquid, and α is the void fraction. The void fraction, α, defined as the fraction of the local flow area occupied by vapor, is generally represented as some function of refrigerant quality, x. The tube length variable, l, must be related to mass quality, x, in some manner. This relationship is obtained from an assumption regarding the heat flux variation, dQ, with differential length, dl, in the two-phase region. There are ten void fraction and two heat flux models available in DRSSIM. More details of each void fraction and heat flux model can be found in Assawamartbunlue (2000) and Rice (1987).

Refrigerant thermodynamic and transportation properties are directly calculated by REFPROP6 (McLinden 1998) that was developed by the National Institute of Standards and Technology (NIST). REFPROP6 can compute the properties for both pure refrigerants and mixture refrigerants. Using REFPROP6 enables DRSSIM to simulate the system regardless of any type of refrigerant. More than 30 pure refrigerants are available. Moist-air psychometric properties are calculated using subroutines developed in HVAC2 Toolkit (Brandemuehl 1993). Unfortunately, there are no subroutines or functions for calculating the transport properties, i.e., thermal conductivity, dynamic viscosity, and specific heat. Thus, three empirical functions are developed for each property using linear regression method.

The system model is constructed by combining component models developed in the previous section. The model inputs are based on what variables are controlled. The controlled variables of the refrigeration system in supermarkets are different from those in HVAC. Typically, the refrigerant pressure in the evaporators is controlled in supermarkets. The air continuously flows through the evaporator coils. The temperature within cases is controlled by turning on/off liquid solenoid valves located at the entrance of the coils. In this test site, the liquid solenoid valves are either fully opened or fully closed. In the evaporative condenser, cooling water is continuously sprayed over the condenser coil at a constant temperature. Loads of the system are met by adjusting compressor speed. Figure 3 shows seven variables that are selected to be inputs of the system model.

The most important assumption involves the method to calculate performance of evaporator coils within the refrigeration cases. The evaporator coils in each circuit are collapsed and treated as one big evaporator coil. The size of this big coil is described by scaling the performance of a representative base evaporator in the system. For instance, if refrigeration circuit has a total rated capacity of 8.22 kW and the base coil has the rated capacity of 2.31 kW, then the equivalent circuit is equal to 8.22/2.31 = 3.55 base coils.

It is further assumed that the base coils are in parallel. Refrigerant and air entering and leaving the base coil are at the same state for each unit. The performance of the base coil is calculated only one time at a given condition and then multiplied by the equivalent unit scaling factor. This same approach is also applied to calculate refrigerant charge inventory in evaporator coils. The charge inventory in the base coil is calculated and then multiplied by the number of equivalent units.

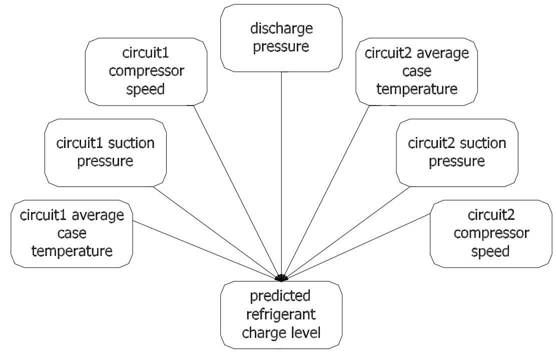

Figure 3: Inputs of the System Model

Because all evaporator coils at a given suction condition are collapsed into one big coil, this implies that each circuit needs only one distributed refrigerant pipe and entering air condition. The equivalent pipe should have the same pressure drop and refrigerant. Unfortunately, the length and diameter of refrigerant pipes from the main liquid pipe to each evaporator coil are not identical, thus, the pressure drop for each pipe is different. The average pressure drop and total volume of refrigerant pipes in a circuit are used to determine the equivalent pipe diameter and length.

MODEL CALIBRATION AND ANALYSIS

The results of the model are compared with the data from the test supermarket. The objective of the analysis is to predict the refrigerant charge level in the liquid receiver as the system operating conditions change over time. The model is initially calibrated to the operations of the test store. There are no validations for each component in the system because of the absence of individual component data collected from the field or provided from manufacturers. Instead, the validations are based on the whole system using available data taken from the field. Subsequent analysis explores the impact of key modeling assumptions and the uncertainty in the comparisons.

Model calibration and component identification are generally limited to the heat exchangers and piping components of the simulation. It is assumed here that the compressor performance as reported by the manufacturer from ARI test procedures is correct. On the other hand, the effectiveness-NTU heat exchanger models require the overall heat transfer coefficient, or *UA*, to be obtained from the test data. The approach here is to identify the *UA* from test data for one typical heat exchanger in each circuit and to scale the size of this representative heat exchanger to account for the capacity of the entire circuit. (In fact, each evaporator has a built-in liquid suction heat exchanger that precools the liquid refrigerant entering the expansion valve by heating the refrigerant leaving the evaporator. The *UA* of this small heat exchanger is estimated by physical calculations.)

Since the evaporator model is a simple sensible heat exchanger, the simulation does not account explicitly for the latent loads on the evaporator or the effect of defrost. The heat exchanger parameters are estimated from the refrigeration load by using data over a complete cycle including defrost. Nevertheless, the identified *UA* is somewhat dependent on the store relative humidity at the time of the parameter identification. Generally, the latent load of the refrigeration case is from the infiltration. The load of the infiltration within the refrigeration case is about 20% of the total load. Approximately, 40% of this infiltration load is the latent load (EPRI 1997). That means the latent load is about 8% of the total load, which is relatively a small quantity. It is expected that the prediction error at other conditions is still very small because of the small latent load.

The overall system model calibration focuses on the prediction of the refrigerant liquid volume fraction in the liquid receiver, which relates to the charge inventory in the receiver and the refrigerant charge in the system components. The measured system quantity is the liquid level in the receiver. The level would change with the different operating conditions of the system and the subsequent changes in the refrigerant charge in the other components. Since the component charge calculations are based on the void fraction and the heat flux assumption models, the calibration task is reduced to identifying the most suitable models. Nine void fraction models and two heat flux assumptions are explored. While the developers of each model claim them to be well matched with experimental data, there is still considerable ambiguity as to which model is most appropriate. It is also known that the different models can have a significant impact on system charge predictions (Assawamartbunlue and Brandemuehl, 2000)

Ten data sets are extracted from the data acquisition system and given in Table 2. These ten data sets are selected in the steady-state period when the change of charge level can be noticed. Among these ten data sets, the suction pressure and the case temperature are relatively close to each other because they are controlled to their setpoints, which are rarely changed. However, the circuit 1 case temperature of data 8 and the circuit 2 case temperature of data 3 and data 10 are slightly different from the others. The major differences are discharge pressure, speed of the compressor, and liquid volume fraction in the receiver.

Dataset #1 is used to calibrate the system charge for each void fraction and heat flux assumption model. Since each model arrives at a different average two-phase refrigerant density, each model will predict a different total system charge. Given the system charge, the liquid volume fraction in the receiver for each other dataset is predicted. More specifically, the change in the predicted charge level relative to the charge level in data point 1 are calculated for each data set and then compared with the change in the measured charge level.

Figure 4 shows the results for the nine void fraction models with the constant heat flux assumption. The general effects of void fraction and heat flux assumption model on the average two-phase density were extensively described by Rice (1987). The heavy line is the change in liquid level of the receiver (in percent of total height) as measured by the data acquisition system at the test store. Each other line is one of the nine void fraction models, which generally follow the trend of the measured data. The predictions by homogeneous void fraction are the closest to the calibration data and are in the range of ±0.5%, except data point 3 and 10.

The error in charge levels predicted for datasets 3 and 10 are much greater than the others. The reason for the larger errors appears to be related to the nonlinearities of average refrigerant density with temperature. Circuit two contains 12 different refrigerated cases, which are modeled as a single evaporator. For the datasets 3 and 10, three of these cases (2E, 2F, and 2G) are operating at much lower temperatures than the other nine cases. The rated capacity of these cases represents about 35%

Table 2: Collected Data from the Test Site

data set	1	2	3	4	5	6	7	8	9	10
date	9/17/99	9/17/99	9/20/99	9/12/99	9/16/99	9/18/99	9/18/99	9/19/99	9/19/99	9/20/99
time	5:00-6:00	16:00-17:00	5:30-6:30	11:30-12:30	18:00-19:00	0:00-1:00	13:00-14:00	0:00-1:00	13:00-14:00	18:00-19:00
discharge pressure (kPa)	1377.35	1284.12	1375.62	1365.55	1292.26	1375.55	1292.26	1359.69	1398.86	1368.17
liquid volume fraction in receiver (%)	32.39	35.21	30.70	33.48	34.59	32.89	35.00	33.92	31.31	30.28
Circuit 1										
suction pressure (kPa)	169.82	161.27	163.76	168.31	158.17	160.24	167.76	164.03	168.93	165.69
% full-load speed	62.19	56.54	33.80	54.21	52.29	87.55	59.44	103.97	61.69	56.54
average case temperature (C)	-24.61	-24.52	-25.27	-24.87	-25.77	-24.44	-24.16	-14.68	-24.26	-24.83
Circuit 2										
suction pressure (kPa)	206.71	204.16	205.20	202.64	203.54	204.85	208.30	209.61	204.16	202.16
% full-load speed	88.09	52.21	66.19	47.34	70.37	65.31	53.52	64.91	96.44	57.72
average case temperature (C)	-20.48	-20.57	-21.91	-20.26	-20.46	-20.34	-20.51	-20.64	-21.32	-22.23

of the total circuit capacity. The average density for this circuit with the two different case temperatures is lower than the density of refrigerant calculated at the capacity-weighted average temperature. The observation further highlights the significance of modeling multiple evaporators with a single effective coil – the multiple evaporators must operate with approximately uniform temperatures.

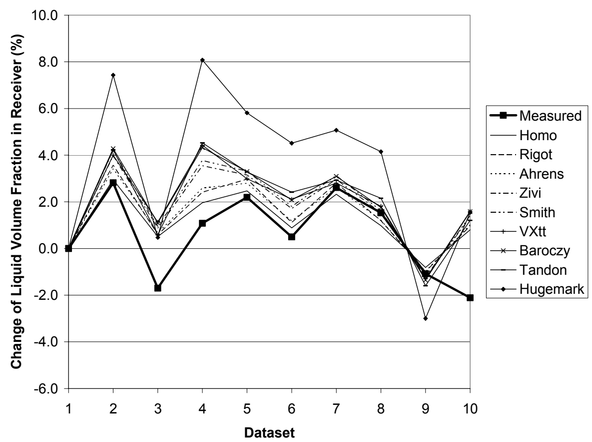

Figure 4: The Prediction of Changes of Liquid Level in the Receiver

Figure 5 shows the prediction using a homogeneous void fraction model with two-heat flux models. The predictions from these two models are very close to each other. Nevertheless, the prediction error suggests that the constant heat flux model is slightly more accurate than the constant wall temperature model. From these calibration processes, a homogeneous model with the constant heat flux assumption is the best combination to predict the refrigerant liquid volume fraction in the receiver. The total installation charge of the system is calculated to be 381 kg.

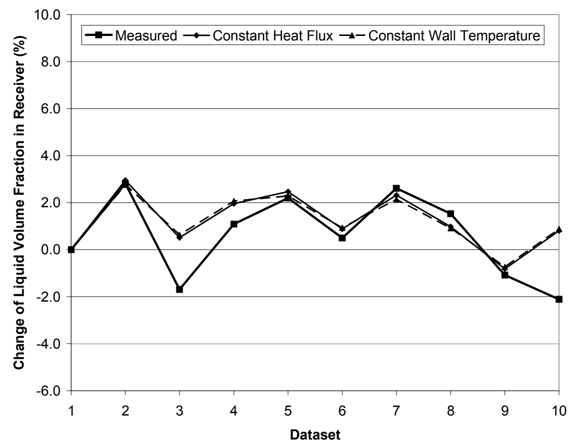

Figure 5: The Effect of Heat Flux Assumption Model Using Homogeneous Model

UNCERTAINTY ANALYSIS

The measured data shown in Table 2 indicate that there are not large variations in the refrigerant charge level in the receiver. The results suggest the importance of evaluating the uncertainty in charge calculations. The uncertainties are influenced by the accuracy of the individual measurements as well as the propagation of those fundamental errors through the model calculations.

Seven inputs are required to predict the liquid volume fraction in the receiver as given in Table 2. The uncertainties in these fundamental measurements are largely determined by the resolution of the data and are described in Assawamartbunlue (2000). Figure 6 compares uncertainties of the actual and predicted liquid volume fraction in the receiver with 99% confidence level using the Homogeneous model. The uncertainties of the predicted liquid volume fraction are in the range of ±1.0%. The uncertainties of the predicted values are slightly greater than the actual values. As expected from previous discussions, the error bars of actual and predicted liquid volume fraction do not overlap datasets 3 and 10.

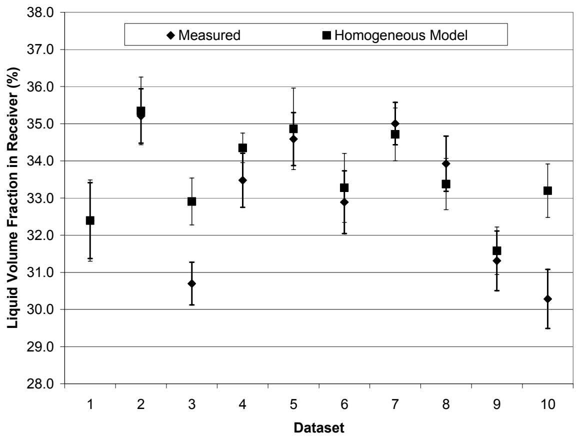

Figure 6: Uncertainties of the Actual and Predicted Liquid Level with 99% Confidence Level

CONCLUSIONS

DRSSIM is a computer program to simulate a distributed refrigeration system for supermarket applications. DRSSIM has a modular design that allows user to easily add and modify the refrigeration component or system configuration. As shown in the results in this paper, DRSSIM accurately predict the liquid volume fraction in the receiver with small amount of prediction errors using the Homogeneous void fraction and the constant heat flux model. However, from this work, there still is a need for more data to verify the accuracy of the model and determine source of prediction error in the program. The capability of each void fraction and heat flux assumption model can be further investigated.

ACKNOWLEDGMENT

This work was prepared with the support and assistance of Mukesh Khattar, PhD, PE, and the Electric Power Research Institute (EPRI), and Lowell Hettinger from King Soopers supermarkets.

REFERENCES

Air-Conditioning & Refrigeration Institute (ARI). 1999. *ARI Standard 540-1999: Positive Displacement Refrigerant Compressors and Compressor Equipment*. Arlington, VA: Air-Conditioning & Refrigeration Institute.

Assawamartbunlue, Kriengkrai 2000. Fault Detection and Diagnosis for Refrigerant Leakage in A Distributed Refrigeration System for Supermarket Applications. Ph.D. Thesis. Department of Civil, Architectural, and Environmental Engineering. University of Colorado at Boulder, Boulder, CO.

Assawamartbunlue, K. and M.J. Brandemuehl. 2000. Effect of void fraction model and heat flux assumption on predicting refrigerant charge level in receivers. Proceedings of 2000 International Refrigeration Conference, Purdue University, W. Lafayette, IN, July 2000.

Brandemuehl, M. J. 1993. HVAC2 Toolkit, Algorithms and Subroutines for Secondary HVAC System Energy Calculations. ASHRAE.

Damasceno, G. S. and Goldschmidt, V. W. 1989. Effects of Refrigerant Charge and Type of Expansion Device on Performance of a Specific Heat Pump: Experimental Data. Ray W.Herrick Laboratories, Purdue University, W.Lafayette, IN, USA. Technical Report HL89-4p.

Domanski, P. and Didion, D. 1983. Computer Modeling of the Vapor Compression Cycle with Constant Flow Area Expansion Valve. U.S.Department of Commerce. National Bureau of Standards, Building Science Series 155.

Grimaldi, C. N. and Mariani, F. 1996. Prediction of the Energy Requirements in a Refrigeration Plant Using Advanced Simulation Techniques. *Proceedings of the 1996 3rd Biennial Joint Conference on Engineering Systems Design and Analysis, Part 1*, July 1-4, 73(1): 237-243.

Hilgendorf, S., Jung, D. and Radermacher, R. 1990. Computer Aided Design of Refrigeration Cycles. *International Journal of Applied Engineering Education* 6(6): 675-686.

Hiller, C. C. and Glicksman, L. R. 1976. Improving Heat Pump Performance via Compressor Capacity cControl - Analysis and Test. MIT Energy Laboratory. Cambridge, MA, USA. Technical Report MIT-EL 76-001.

Jeter, S. M. and Wepfer, W. J. 1986. Analysis and Simulation of Variable Speed Drive Heat Pumps. *ASME, Advanced Energy System Division* 2(2): 71-80.

Lebrun, J. 1993. HVAC1 Toolkit, A Toolkit for Primary HVAC System Energy Calculations. ASHRAE.

McLinden, M. O., Klein, S. A., Lemmon, E. W. and Peskin, A. P. 1998. REFPROP - Thermodynamics and Transport Properties of Refrigerants and Refrigerant Mixtures. NIST Standard Reference Database 23 - Version 6.0. The National Institute of Standards and Technology.

Rice, C. K. and Fischer, S. K. 1983. The Oak Ridge Heat Pump Models: I. A Steady-State Computer Design Model for Air-to-Air Heat Pumps. Oak Ridge National Laboratory. ORNL/CON-81/R1.

Rice, C. K. 1987. The Effect of Void Fraction Correlation and Heat Flux Assumption on Refrigerant Charge Inventory Predictions. *ASHRAE Transactions*. 93(1195): 341-467.

Rice, C. K. and Jackson, W. L. 1994. PUREZ-The Mark V ORNL Heat Pump Design Model for Chlorine-Free, Pure and Near-Azeotropic Refrigerant Alternatives. Oak Ridge National Laboratory.

Rossi, T. M. 1995. Detection, Diagnosis, and Evaluation of Faults in Vapor Compression Cycle Equipment. M. S. Thesis. School of Mechanical Engineering, Purdue University, W.Lafayette, IN.

Smith, T. E., Nelson, R. M. and Pate, M. B. 1987. An Interactive Computer Program for Aanalyzing refrigeration Cycles in HVAC Courses. *ASHRAE Transactions* 93(1): 870-882.

SimBuild2004

I or we, the author(s) of the submitted paper, have read the following Copyright Transfer and Disclaimer, and agree to them by submitting the submitted paper.

(1) The author(s) affirm that the paper has not been published elsewhere and, if the paper is accepted, will not be published elsewhere prior to SimBuild 2004.

(2) If the paper is accepted, the author(s) will automatically grant to IBPSA-USA a nonexclusive, royalty-free, perpetual, worldwide, irrevocable, sub-licensable, and transferable license to publish the paper (in unmodified form) in any fashion (including but not limited to inclusion in the SimBuild 2004 CD proceedings, via electronic means such as the world wide web, and inclusion in future compilations of papers). This "nonexclusive" license means that the author(s) are not restricted as to future use of the material except that exclusive rights cannot be granted to another.

(3) The author(s) affirm that they have the right to grant the license specified in (2), that is, publication by IBPSA-USA or its licensees will not be in conflict with copyright or other restrictions.

(4) The author(s) acknowledge that acceptance of the paper does not imply IBPSA-USA's endorsement of or agreement with the ideas presented. Under no circumstances shall IBPSA-USA be liable for any damages resulting from use information included in the paper.

Figure 2 This text MUST be included in the body for paper submission