2000

The Effect of Void Fraction Models and Heat Flux Assumption on Predicting Refrigerant Charge Level in Receivers

K. Assawamartbunlue
University of Colorado

M. J. Brandemuehl
University of Colorado

THE EFFECT OF VOID FRACTION MODELS AND HEAT FLUX ASSUMPTION ON PREDICTING REFRIGERANT CHARGE LEVEL IN RECEIVERS

Kriengkrai Assawamartbunlue, Ph.D.
Michael J. Brandemuehl, Ph.D, P.E.

Civil, Environmental, and Architectural Engineering
University of Colorado
Engineering Center ECCE 246, Box 428
Boulder, CO 80309-0428
michael.brandemuehl@colorado.edu

ABSTRACT

A computer model has been developed to simulate a vapor compression refrigeration system with a liquid receiver for the purpose of predicting changes in receiver refrigerant charge level during normal system operation. The analysis requires calculation of the amount of refrigerant charge resident in the two-phase heat exchangers of the system, which typically relies on selection of a void fraction model and a relationship between heat flux and tube length. This paper presents the effect of void fraction models and heat flux assumptions on the charge distribution in a vapor-compression system with a liquid receiver. Nine different void fraction models from the literature and two heat flux assumptions are examined over a range of system operating conditions. The various models include different assumptions about velocity "slip" between liquid and vapor phases and the role of total refrigerant mass flux. The results indicate that the prediction of refrigerant charge level is strongly influenced by the choice of void fraction models. The Hugemark model always predicts the highest charge levels and Lockhart-Martinelli predicts the lowest charge. Most of the models show relatively small variations in charge levels with changes in operating conditions. However, the models that depend on mass flux show large changes in component charge levels that are often in the opposite direction compared to other models. The conflicting results indicate the need for additional experimental testing to validate void fraction models.

INTRODUCTION

The performance of a refrigeration or heat pump system can be influenced by the amount of refrigerant charge circulating in the system, especially systems without a high-side liquid receiver. Specifically, systems without liquid receivers are charge-sensitive because excess liquid refrigerant can occupy a significant volume of the condenser, reducing effective condensing heat transfer area. There have been many investigations into the effect of charge on the performance of the these systems [Otaki 1973, Farzad 1990, Damasceno 1991, Robinson 1994, Ding 1996, Goswami 1997]. These previous studies include both experimental and numerical evaluation and conclude that improper charge can reduce system capacity and efficiency and can compromise the reliability of system components.

Given the flexibility and speed of computer simulation, several researchers have developed computer simulations of charge-sensitive refrigeration systems [Domanski 1983, Rice 1983, Otaki 1973, Damasceno 1991, Farzad 1994]. These simulations calculate the required system charge for a given set of design conditions or calculate the performance for a given system charge. To perform the charge inventory calculations, the research must select among alternative models for calculating the average refrigerant density in the two-phase region of the condenser and evaporator. The two-phase density is usually calculated in terms of the "void fraction." Various void fraction models have been proposed by Rigot [1973], Ahrens/Thom [1983], Zivi [1964], Smith [1969], Lockhart and Martinelli [1949], Baroczy [1965], Tandon [1985], and Hugemark [1962].

Unfortunately, the different models give large variations in average two-phase density. Under some operating conditions, Rice [1987] shows that the average density can vary by a factor of 10 among the alternative models. The results of Otaki [1973] and Farzad [1994] indicated that Hugemark correlation provides the best comparison to experimental data, though there has been little recent work to definitively identify the most accurate models based on experiments. However, Rice and other researchers have suggested that performance prediction may not be heavily dependent on the absolute accuracy of these models. Part of the reason for the lower sensitivity to average two-phase density is that the charge-sensitive systems "rebalance" themselves at off-design conditions due to the large difference between vapor and liquid density (factor of 20-60) and the many interactions among system components. In the end, capacity and efficiency may change by about 10%.

The effect of refrigerant charge prediction for the system with liquid receiver is different due to the presence of receiver. As long as there is liquid refrigerant in the receiver, system performance is insensitive to total system charge. In addition, under most operating conditions, the refrigerant leaves the condenser nearly saturated and the fraction of the condenser volume occupied by two-phase refrigerant is greater. Since performance is not affected by charge, there has been no incentive to evaluate the impact of charge calculations for systems with liquid receivers. However, our interest lies in refrigerant leak detection. We propose that leakage can be detected by monitoring the liquid level in the receiver and calculating the refrigerant charge in the piping and components under changing operating conditions. In this case, the uncertainty in the average two-phase refrigerant density has a direct impact on the ability to identify refrigerant leakage.

This paper describes the effect of void fraction models and heat flux assumptions on predicting the refrigerant charge in a refrigeration system with a receiver at various operating conditions. The analysis is performed using a custom computer simulation of a refrigeration system.

SYSTEM MODELING

The system considered in this paper is a simplified representation of a refrigeration system used widely in supermarkets. This system consists of an evaporator, a condenser, an expansion device, a compressor and a liquid receiver as shown in Figure 1. (Actual supermarket refrigeration systems typically have many evaporators and expansion devices in parallel, multiple parallel compressors, and various other heat exchangers.) The liquid receiver is used to collect the excess refrigerant in part load operation. A well designed system will always have both liquid and vapor in the receiver and the refrigerant leaving the receiver to the evaporators is always in the saturated liquid state. The equalizer line connects the liquid receiver to the entrance of the condenser and allows vapor within the receiver can flow back to the condenser through a check valve, if necessary.

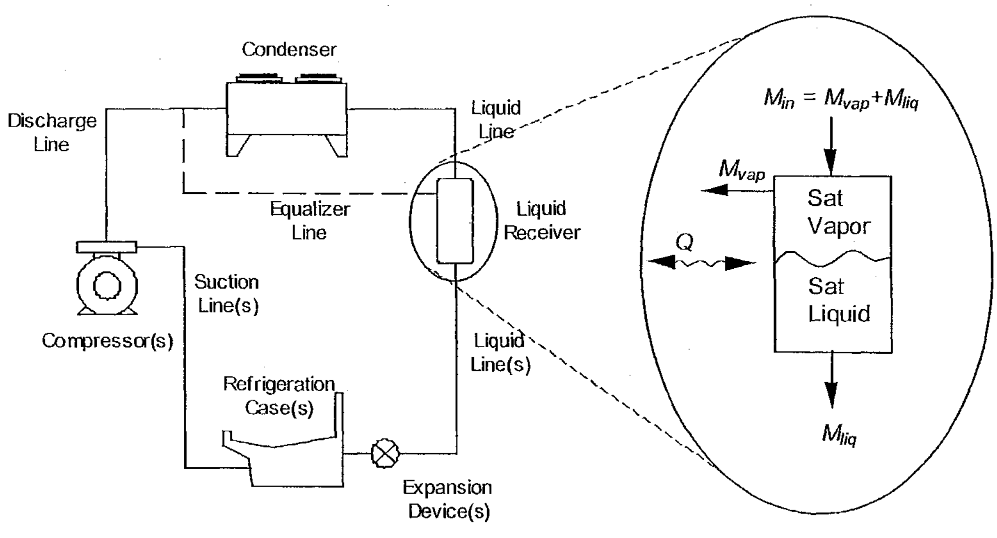

Figure 1 Refrigeration System with Receiver

For the analysis described here, the reciprocating compressor is modeled by a general routine that models the internal energy balances using user-supplied heat loss and internal-efficiency coefficients [Lebrun 1993]. The effectiveness-NTU method is implemented to calculate the performance of the condenser and evaporator. They are air-to-refrigerant heat exchangers and are modeled as equivalent sequential refrigerant circuits. In the condenser, the air first flows through the subcooled section, two-phase section, and then the superheated section before leaving the heat exchanger. Heat transfer coefficients on the refrigerant-side and air-side are drawn from the engineering literature

The thermostatic expansion device is modeled as a one-dimensional, adiabatic, and compressible flow device. The refrigerant line model includes pressure drop and heat transfer calculation, including natural convection and radiation of the outside of the lines [Rice 1983].

Figure 1 also shows a schematic diagram of liquid receiver modeling. Q is the heat transfer rate between the refrigerant inside the receiver and the ambient air. M_{liq} is equal to the refrigerant mass flow rate circulating in the system and M_{vap} is the refrigerant mass that flows back to the condenser through the equalizer line. M_{in} is the total refrigerant mass flow rate leaving the condenser. It is further assumed that there is a check valve installed on the equalizer line to eliminate refrigerant flow directly from the entrance of the condenser to the receiver.

All refrigerant and air thermodynamic and transportation properties are calculated using the REFPROP6 subroutines [McLinden 1998]. Therefore the simulation program can be implemented with all refrigerants available in REFPROP6. In this analysis, only the refrigerant HCFC-22 is simulated to study the effect of void fraction in prediction of charge within the system.

CHARGE INVENTORY CALCULATIONS

Nine void fraction correlations and two heat flux assumptions are presented here [Rice 1987]. The basic equations for charge inventory calculation are given first for both single-phase and two-phase regions of the heat exchanger. The relationship of void fraction correlation and heat flux assumption to these equations are also presented.

The calculation of mass in two-phase portions of the condenser and evaporator is modeled in terms of the void fraction, which is defined as the ratio between the cross-section area occupied by vapor and the total cross-section area, $\alpha = A_g/A_c$. The two-phase refrigerant mass in a length of tubing is obtained by summing the gas, g, and liquid, f, contributions occupying each cross-sectional area over the length of the region. The total mass, M_t, in the two-phase section can be obtained in terms of tube volume, V, and the tube length, L, as

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

The void fraction α is generally represented as some function of refrigerant quality, x, or $\alpha = f_\alpha(x)$. Therefore, to evaluate, M_t, for a given void fraction equation, 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 flow variation, dQ, with differential length, dl, in the two-phase region

$$dQ = \dot{M}_r h_{fg} dx = f_Q(x)dl \tag{2}$$

where $\dot{M}_r$ = refrigerant mass flow rate,
h_{fg} = enthalpy of vaporization of refrigerant
$f_Q(x)$ = assumed heat flux equation.

In terms of the representations $f_Q(x)$ and $f_\alpha(x)$, the total mass of refrigerant in the two-phase region is calculated as

$$M_t = V\left[\rho_g W_g + \rho_f (1-W_g)\right] \tag{3}$$

where

$$W_g = \frac{\int_{x_i}^{x_o} \frac{f_\alpha(x)}{f_Q(x)} dx}{\int_{x_i}^{x_o} \frac{1}{f_Q(x)} dx} \tag{4}$$

and x_i and x_o are inlet and outlet refrigerant qualities. The normalized integral W_g is the refrigerant gas density weighting factor. Thus the evaluation of the two-phase refrigerant mass is reduced to the problem of evaluating the integrals for selected void fraction correlations and heat flux assumptions.

The void fraction is commonly expressed as a function of mass quality, x, and combinations of various properties of the refrigerant. Some correlations also depend on mass flow rate through a Reynolds number dependence. Six of the nine void fraction models discussed here are based on the following form.

$$\alpha = \frac{K_H}{1 + \left[\frac{1-x}{x}\right]\left(\frac{\rho_g}{\rho_f}\right) S} \tag{5}$$

The six models differ in their values or relationships for the parameters K_H, and S, as shown in Table 1. Physically, the slip ratio, S, is defined as the ratio of velocities of the gas and liquid phases, $S = u_g/u_f$. Two other property indices also appear in the various models.

$$PI_1 = \left(\frac{\rho_g}{\rho_f}\right) \qquad PI_2 = \left(\frac{\mu_f}{\mu_g}\right)^{0.2}\left(\frac{\rho_g}{\rho_f}\right) \tag{6}$$

Table 1: Void Fraction Model Parameters - Slip Models

Slip Model Parameters	Homog.	Rigot	Ahrens & Thom	Zivi	Smith	Hughmark
S	1	2	$f(\mathrm{PI}_2)$	$(\mathrm{PI}_1)^{-1/3}$	$f(\mathrm{x},\mathrm{PI}_1)$	1
K_H	1	1	1	1	1	$f(\mathrm{Re},\mathrm{Fr},\alpha)$

The homogeneous model is the simplest, representing the two phases as a mixture with no slip. Rigot [1973] uses a slip ratio of two. Ahrens/Thom [1983] vary the slip ratio as a function of refrigerant pressure through its influence on property index PI_2. Zivi [1964] developed a void fraction model for annular flow based on principles of minimum entropy production under conditions of zero wall friction and zero liquid entrainment. Smith [1969] developed a correlation for slip ratio based on equal velocity heads of a homogeneous mixture center and an annular liquid phase. The Hugemark [1962] model is similar to the homogeneous model except that K_H is an added multiplier dependent on a viscosity-averaged Reynolds number, the Froude number, and the liquid volume fraction.

The Lockhart-Martinelli [1949] model gives the void fraction as a function of the correlating parameter X_{tt}.

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} PI_2^{0.5}\,. \tag{7}$$

This correlating parameter is also used in two other models of the form shown in Table 2. Baroczy [1965] includes a function dependence on the property index PI_2. Tandon [1985] includes the effects of mass flux on void fraction through a Reynolds number dependence.

Table 2: Void Fraction Model Parameters - X_{tt} Models

	Lockhart-Martinelli	Baroczy	Tandon
$\alpha =$	$f(X_{tt})$	$f(X_{tt}, Pl_2)$	$f(X_{tt}, Re_L)$

Note that the Tandon and Hugemark models share the common feature that the void fraction is dependent on mass flux. All other models represent the local void fraction as a function only of refrigerant properties.

As noted in Equations 1-4, the calculation of refrigerant charge in the two-phase region of the heat exchangers requires a model for variation of refrigerant quality with flow length, which is typically expressed in terms of a heat flux assumption. Two different assumptions are typically considered: constant heat flux or constant wall temperature. The assumption of constant heat flux is most commonly applied. This model gives a linear relationship between quality and tube length. In terms of Equation 2, $f_Q(x) = 1$. By comparison, the assumption of constant tube wall temperature results in the heat flux being a function of the local refrigerant-side heat transfer coefficient, U_R. In terms of Equation 2, $f_Q(x) = U_R(x)$.

SIMULATION PROCEDURES AND RESULTS

The alternative charge calculation models have been implemented in a detailed simulation of a refrigeration system. The simulation program is designed to describe systems with multiple evaporators and compressors, as are commonly used in supermarket applications. Each compressor can be controlled to a separate suction pressure and each evaporator pressure can be independently controlled. The simulation program also allows inclusion of desuperheaters, subcoolers, and liquid suction heat exchangers in the system.

For the purposes of this study, the simple system of Figure 1 is evaluated. The system has a single large evaporator and a variable speed compressor with a nominal capacity of 25 kW. Like most distributed supermarket refrigeration systems, the suction and discharge pressures are controlled. Suction pressure is controlled to a setpoint by modulating compressor speed. The nominal suction pressure is 450 kPa, or a saturated suction temperature of –3°C. Evaporator refrigerant temperature is dictated by this controlled suction pressure and the pressure drop in the suction line. (These lines can be very long in supermarkets. In this analysis, lines are assumed to be 24 m.) A thermostatic expansion valve controls superheat leaving the evaporator. A minimum discharge pressure is maintained by modulating condenser airflow. As the outdoor air temperatures increases, the airflow rate will increase until it reaches its maximum (all fans on at 100% speed). Once the condenser fans are at maximum airflow, the discharge pressure floats above the setpoint to balance refrigerant and air-side heat transfer.

A base case analysis has been performed for all combinations of void fraction models and heat flux assumptions. The results of this analysis are summarized in Table 3. The table gives the total refrigerant charge in the system for all components and refrigerant lines except the receiver. This calculate total component charge is the most important indicator of the ability to predict refrigerant level in the receiver, since the remainder of the system charge must reside in the receiver. The results show that the calculated charge varies by a factor of three between the lowest (Lockhart-Martinelli) and the highest (Hugemark). On the other hand, six of the nine models give approximately the same total charge, varying by ±10% from the mean of the six values. The constant wall temperature results are generally about 12-15% greater than the constant heat flux results, though the Lockhart-Martinelli and Hugemark results are very similar for the two heat flux assumptions

Table 3: Effect of Charge Calculation Models on Total Charge in System (kg)

	Homo	Rigot	Ahrens	Zivi	Smith	Lock-M	Baroczy	Tandon	Hugemark
Constant Heat Flux	80.4	90.6	86.6	94.7	91.1	66.1	95.3	124.2	191.0
Constant Wall Temp	91.7	105.1	99.9	109.8	103.4	63.8	107.8	139.4	194.4

While the results of Table 3 show wide variations in calculated system charge, our efforts to detect refrigerant leakage are more affected by *changes* in charge level than in the absolute amount of refrigerant in the system. For example, if the level of charge in the receiver is lower today than it was yesterday, we want to know whether the lower level is due to a leak or due to changes in operating conditions that might cause more of the system charge to reside in the components (e.g., changes in outdoor air temperature, evaporator loading, or system controls). For this reason, a parametric sensitivity analysis has been performed to evaluate the effect of system operating conditions on the total component charge level. The effect of two of these parameters are discussed here.

Figure 2 shows the relative change in system refrigerant mass charge with changes in the air temperature entering the condenser (outdoor air temperature) for the different void fraction models with constant heat flux. The nominal condenser air temperature is 35°C. At very low condenser air temperatures, all models show the same trend of increasing system charge level with increasing condenser air temperature. Recall that at low temperatures, the condenser airflow is modulated to maintain a minimum discharge pressure. Above a condenser air temperature of 17°C, the discharge pressure begins to rise above the minimum setpoint and the condenser airflow is constant at is maximum value. Above this temperature, the Tandon and Hugemark models exhibit an opposite trend compared to the other models. While the other models show a continued increase in charge due to the increased density at higher discharge pressure, the Tandon and Hugemark models show a decrease in charge due to an increase in refrigerant mass flow rate through the system. As the discharge pressure increases, the enthalpy of the refrigerant entering the evaporator increases. Since the evaporator airflow, entering air temperature, and leaving refrigerant enthalpy are fixed, the compressor speed increases to maintain the suction pressure at its setpoint. At higher mass flow rate, the mass flux-dependent models (Tandon and Hugemark) predict that there will be less slip between vapor and liquid phase, leading to less liquid hold-up and a lower average density. Unfortunately, there is not enough evidence in the engineering literature to state definitively which of these two competing trends is correct.

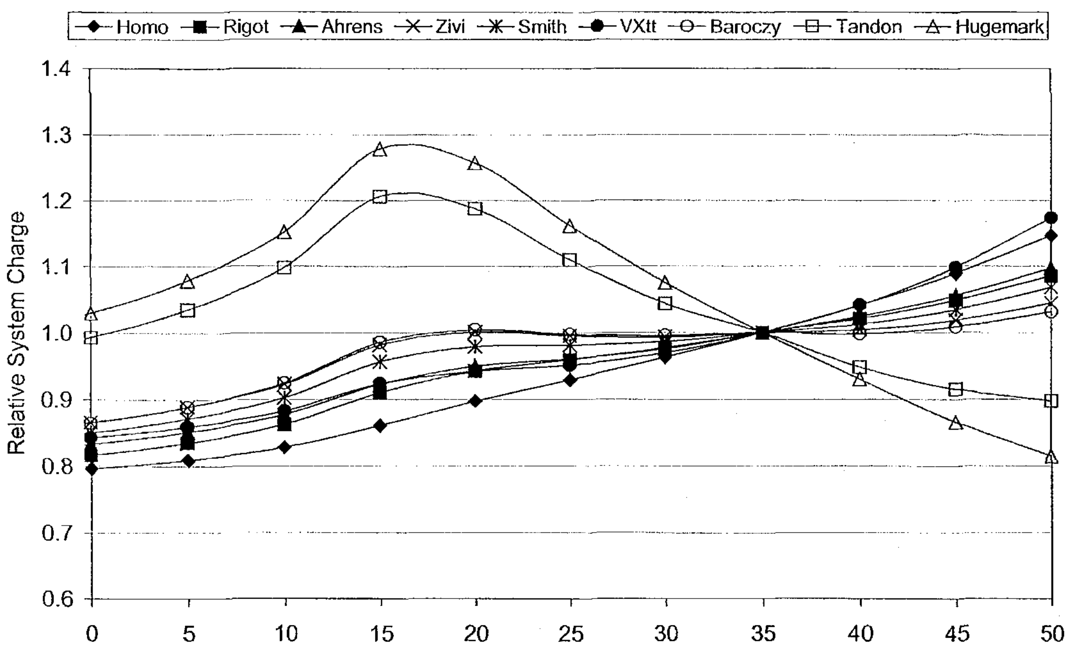

Figure 2 Effect of Condenser Air Temperature on System Charge

Suction pressure control of a refrigeration system can give similar results. Figure 3 shows the relative charge variation with changes in suction pressure. Most of the models show relatively little effect of suction pressure, largely because most models predict very small liquid fractions in the evaporator at mass qualities above about 20%. With fixed evaporator air temperature and flow, the higher suction pressure reduces the load on the evaporator, which leads to lower refrigerant flow rate and greater average density for the Tandon and Hugemark models. The other models do not account for this effect of refrigerant mass flux on void fraction.

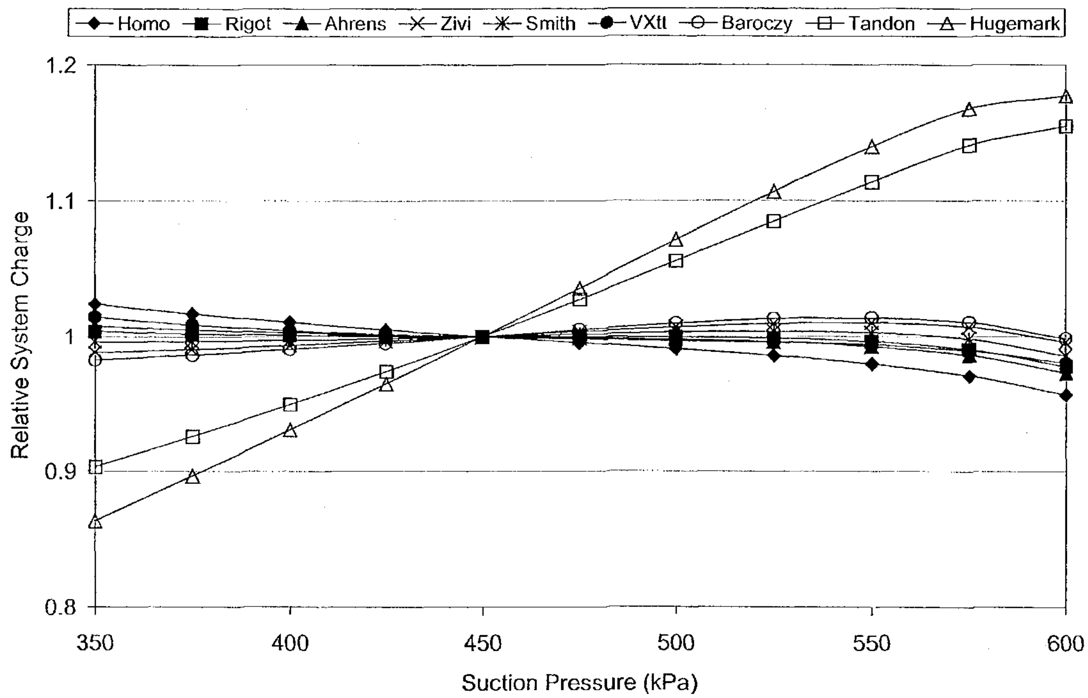

Figure 3 Effect of Suction Pressure on System Charge

CONCLUSIONS

A simulation analysis has been performed to evaluate the impact of refrigerant charge calculation assumptions on the ability to predict the charge level in a vapor compression refrigeration system with a liquid receiver. The analysis evaluated nine void fraction models and two heat flux models. The underlying objective of the work is to develop methods for refrigerant leak detection that are based on measuring the charge level in the receiver. The results of the analysis indicate that there is a large variation in the calculated system charge among the different models. In addition, there are profound differences in the changes to calculated refrigerant charge with changes in typical operating parameters. The main differences are between the mass flux-dependent void fraction models of Tandon and Hugemark and the other models for which void fraction is dependent only on refrigerant properties. In several important cases, the mass flux-dependent models predict an increase in charge level while the other models predict a decrease. While some limited research indicates that the Hugemark model is more accurate for predicting the performance of charge-sensitive systems, there is no insufficient experimental evidence to clearly state which of the two conflicting model conclusions is correct. The results identify the need for additional experimental validation before measurements of receiver charge level can be used to detect refrigerant leakage.

REFERENCE

Ahrens, F.W. 1983. Heat Pump Modeling, Simulation and Design. *Heat Pump Fundamentals. Proceedings of the NATO Advanced Study Institute on Heat Pump Fundamentals, Espinho, Spain, 1980.* J.Berghmans, ed. The Hague, Netherlands: Martinus Nijhoff Publishers.

Baroczy, C.J. 1965. Correlation of liquid fraction in two-phase flow with application to liquid metals. *Chemical Engineering Progress Symposium Series*, **61**(57):179-191

Damasceno, G.S., Domanski, P.A., Rooke, S., and Goldschmidt, V.W. 1991. Refrigerant charge effects on heat pump performance. *ASHRAE Transaction*, **97**(1):304-310.

Ding, G., Zhang, C., and Chen, Z. 1996. Study on the refrigerant charge of refrigerator-freezer. *Proceedings of the 1995 International Conference on Energy and Environment, ICEE May 1995.* NY, p.392-397.

Domanski, P. and Didion, D. 1983. Computer modeling of the vapor compression cycle with constant flow area expansion valve. *National Bureau of Standards, Building Science Series 155,* U.S.Department of Commerce, May 1983.

Farzad, M., and O'Neal, D.L. 1990. Effect of improper refrigerant charging on the performance of an air conditioner with capillary tube expansion. *Energy and Building*, **14**(4) 1990.

Farzad, M., and O'Neal, D.L. 1994. Effect of void fraction model on estimation of air conditioner system performance variables under a range of refrigerant charging conditions. *International Journal of Refrigeration*, **17**(2):85-93.

Goswami, D.Y. et al. 1997. Effect of refrigerant charge on the performance of air-conditioning systems. Proceedings of the 1997 32nd Intersociety Energy Conversion Engineering Conference. Part 3-4, July 27-Aug 1, 1997, v.3-4, Honolulu, p.1635-1640.

Hugemark, G.A. 1962. Holdup in Gas-Liquid Flow. *Chemical Engineering Progress*. **58**(4):62-65.

Lebrun, J. et al. 1993. *HVAC1 Toolkit, A Toolkit for Primary HVAC System Energy Calculaiton.* ASHRAE.

Lockhart, R.W. and Martinelli, R.C. 1949. Proposed correlation data for isothermal two-phase two-component flow in pipes. *Chemical Engineering Progress*, **45**(1):39-48.

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.*

Otaki, T. 1973. Holding refrigerant in refrigeration unit. Proceedings of the XIII International Congress of Refrigeration, Washington, D.C., 1971, **2**:535-544, Westport, CT: Avi Publishing Company, Inc.

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, Aug 1983. 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,* v.93, 1987.

Rigot, G. 1973. Fluid capacity of an evaporator in direct expansion. *Chaud-Froid-Plomberie*, **328:**133-144.

Robinson, J.H., and O'Neal, D.L. 1994. The impact of charge on the cooling performance of an air-to-air heat pump for R-22 and three binary blends of R-32 and R-134a. *ASHRAE Transaction*, **100**(2).

Smith, S.L. 1969. Void fractions in two-phase flow: a correlation based upon an equal velocity head model. *Proc. Instn. Mech. Engrs.*, London, **184**(1)(36):647-664.

Tandon, T.N. et al. 1985. A void fraction model for annular two-phase flow. *International Journal of Heat and Mass Transfer*, **28**(1):191-198.

Zivi, S.M. 1964. Estimation of steady-state steam void-fraction by means of the principle of minimum entropy production. *Transactions ASME, Jour. Of Heat Transfer,* Series C, **86**:247-252.