The Impact Of Modeling Complexity And Two-Phase Flow Parameters On The Accuracy Of System Modeling For Unitary Air Conditioners

T. M. Harms
Purdue University

J. E. Braun
Purdue University

E. A. Groll
Purdue University

THE IMPACT OF MODELING COMPLEXITY AND TWO-PHASE FLOW PARAMETERS ON THE ACCURACY OF SYSTEM MODELING FOR UNITARY AIR CONDITIONERS

Todd M. Harms, James E. Braun*, and Eckhard A. Groll

Purdue University, 1077 Ray W. Herrick Laboratories
West Lafayette, IN 47907, USA
*Author for Correspondence E-Mail: jbraun@ecn.purdue.edu

ABSTRACT

Current air conditioner system models seek to balance the level of code complexity with the speed of execution. Ideally, the simplest code possible that can accurately predict system performance is desired. To further understand this issue, two methods of modeling unitary air conditioning systems are compared: a detailed, local approach and a simplified, global approach. These methods have been applied to two commercial unitary air conditioners from which extensive experimental data was obtained. The global approach predicted system performance as well or better than the local approach.

In a separate analysis, the effects of two-phase modeling parameters on system model predictions were considered. The two-phase friction factors, heat transfer coefficients, and void fraction models were independently varied. The resulting system performance predictions were sensitive to variations in the air-side heat transfer coefficients and the void fraction model, and not as sensitive to variations in the friction factor. Both the void fraction model and the condenser parameters mainly affect the compressor discharge pressure and the compressor power consumption. The evaporator parameters mainly affect the refrigerant mass flow rate and the heat transfer rates in both heat exchangers.

NOMENCLATURE

A: area, m^2
c_1, c_2: constants
c_p: specific heat, kJ/kg-K
$c_{p,m}$: specific heat of moist air, kJ/kg-K
c_s: saturation specific heat, kJ/kg-K
C_{min}: minimum heat capacity rate, W/K
C_r: heat capacity ratio
COP: coefficient of performance
D: diameter, m
f: friction factor
G: mass flux, kg/m^2-s
h: heat transfer coefficient, W/m^2-K
$h_{a,i}$: inlet air enthalpy, kJ/kg
$h_{r,s}$: saturated air enthalpy at T_{ref}, kJ/kg
h_{dis}: compressor discharge enthalpy, kJ/kg
h_{suc}: compressor suction enthalpy, kJ/kg
k: thermal conductivity, W/m-K
K: flow parameter

Greek
α: void fraction
ε: effectiveness
η_o: overall fin efficiency
ρ: density, kg/m^3
ξ: refrigerant-oil solubility

Superscripts
*: wet coil

Subscripts
air: air
blower: evaporator blower
cond: condenser
comp: compressor
dis: discharge
evap: evaporator
fan: condenser fan

L: length, m
m: mass, kg
$\dot{m}$: mass flow rate, kg/s
NTU: number of transfer units
P: pressure, Pa
$\dot{Q}$: rate of heat transfer, kW
S: slip ratio
T: temperature, °C
u: velocity, m/s
UA: overall conductance, W/K
V: volume, m^3
x: quality
$\dot{W}$: power, kW

f: liquid
g: vapor
h: hydraulic
i: inner
o: outer
ref: refrigerant
s: surface
sub: condenser subcooling
suc: suction
super: evaporator superheat
tp: two-phase

INTRODUCTION

Two commonly used system models are HPSIM (Domanski and Didion, 1983) and PUREZ (Rice and Jackson, 1994). These models have been found to under-predict charge inventory. Several issues have been identified as sources of error in the modeling of charge inventory, including incomplete internal volume accounting, neglecting refrigerant-oil diffusion effects, and void fraction modeling assumptions (Damasceno et al., 1991 and Marques and Melo, 1993). Perhaps the most challenging of these is void fraction determination.

At a given cross-section of a tube, the void fraction is defined as the fraction of area occupied by vapor. While mass quality can be determined using conservation equations, in general, void fraction can not be directly calculated and must be modeled in some manner. Rice (1987) presented a comprehensive review of the available void fraction models. The void fraction correlations of Hughmark (1962), Premoli et al. (1971), Tandon et al. (1985), and Baroczy (1965) were recommended, since they yield the highest charge predictions for condensers and the best overall agreement with experimental data. Rice stated that there is insufficient data to recommend one over the others. He admitted that the Hughmark method may over-predict charge in the condenser, yet still yield good agreement with the total charge by way of error cancellation with respect to unaccounted charge elsewhere in the system.

Other studies have compared predictive models with experiments. Damasceno et al. (1991) used HPSIM to compare predicted and measured capacity for various charge levels in a residential air-to-air heat pump. They found that they needed to modify HPSIM to include the Hughmark (1962) correlation to get acceptable results. They did not consider refrigerant-oil diffusion. Marques and Melo (1993) compared the predicted and measured charge inventory of a room air conditioner using HPSIM. They also found that the Hughmark correlation provided the best results. Furthermore, they found that the addition of a refrigerant-oil diffusion calculation was a necessary modification to HPSIM. LeRoy et al. (2000) used PUREZ to compare the predicted and measured performance of ten unitary air conditioners. They sought to reduce modeling errors through the use of two tuning methods. The recommended tuning method is to adjust the heat transfer coefficients to match the cooling capacity. They found that all of the untuned charge inventory results were less than the measured charge. Even the tuned results tended to underpredict charge. Of the eight available void fraction correlations considered, the Hughmark model led to the best agreement between the measured and calculated results. PUREZ does not account for refrigerant-oil diffusion effects. For three of the systems they considered a wider range of tuning parameters, including air-side and refrigerant heat transfer coefficients, pressure drop multipliers, refrigerant charge, superheat, subcooling, refrigerant mass flow rate multipliers, and compressor power multipliers. The tuned parameters were adjusted to match the predicted results with the measured data. The relative importance of each parameter was not considered.

PUREZ and HPSIM differ substantially in complexity. However, these models can not be directly compared for purpose of understanding the effects of model complexity because of their fundamental differences. The literature suggests that the total charge inventory is routinely under-predicted. If the void fraction has been under-predicted in previous studies and void fraction correlations were selected to compensate for unaccounted charge, then the most accurate void fraction correlation available has yet to be identified. The literature also suggests that the effect of the uncertainty in two-phase flow parameters on system model predictions is unclear.

This paper presents a comparison of two system modeling approaches to determine the level of complexity required to obtain accurate system performance predictions. Furthermore, the impacts of various two-phase modeling parameters on performance predictions were carefully investigated. Each parameter under consideration was systematically varied and the resulting predictions are compared. The relative importance of the uncertainty associated with each parameter was evaluated in this manner. At each step the models were validated against experimental data.

EXPERIMENTAL APPARATUS AND PROCEDURE

The scope of the experimental portion of this work encompasses two unitary air conditioners: a 5-ton rooftop unit and 7½-ton split system. Both systems use R-22 and have plate-fin heat exchangers and a thermal expansion valve. Each heat exchanger has multiple, parallel flow circuits to manage pressure drop, and has more than one row of tubes. All of the heat exchangers employ standard microfin tubes, except for the condenser in the 7½ ton split system, which has smooth tubes. The experimental setup is described in detail in Harms et al. (2001). Two operational parameters are considered in the tests: charge inventory and outdoor air temperature. All of the other inputs are fixed between the test cases. The charge inventory was varied through a range of values from well under charged to well over charged. The outdoor air temperature was set to one of three discrete values: 28 °C, 35 °C, or 49 °C.

SYSTEM MODELING

Two modeling approaches are used here: a detailed model and a simplified model. Exactly the same equations and correlations are used in each. Only the manner in which they are applied differs. The detailed simulation model considers each tube separately. Furthermore, each tube is broken into many elements. Both the air states and refrigerant states are tracked from element to element. Depending upon the circuiting arrangement, the solutions to the governing equations may be iterative. The simplified model assumes that each heat exchanger circuit performs identically. Instead of using fixed length elements, the heat exchangers are broken into single-phase and two-phase sections. The length of each section is a variable. An average heat transfer coefficient is utilized in the two-phase sections by integrating with respect to quality. This approach is much less complicated in terms of computational effort. The simplified approach executes on the order of 100 times faster than the detailed approach. To start the solution procedure for either approach, values are guessed for the compressor suction pressure, the compressor discharge pressure, and the evaporator superheat. The guesses are updated according to a third-order Newton's method. The process continues until convergence is achieved. The sub-models are considered in further detail.

Heat Exchanger Modeling

In the condenser, one-dimensional heat transfer was assumed so that,

$$\dot{Q}_{cond} = \varepsilon\, C_{\min}\,(T_{ref} - T_{air}) \tag{1}$$

where C_{min} and ε were determined according to the NTU-ε method (Incropera and DeWitt, 1996).

$$C_{\min} = \min \begin{cases} \dot{m}_{ref}\, c_{p,ref} \\ \dot{m}_{air}\, c_{p,air} \end{cases} \tag{2}$$

The form of the effectiveness depends on the arrangement of the heat exchanger. For single-phase flow in a cross-flow heat exchanger where both fluids are unmixed, the effectiveness is determined as

$$\varepsilon = 1 - \exp\left[NTU^{0.22} C_r^{-1}\left(\exp\left\{-C_r NTU^{0.78}\right\} - 1\right)\right] \tag{3}$$

where,

$$C_r = \min \begin{cases} \dot{m}_{ref}\, c_{p,ref} \,/\, \dot{m}_{air}\, c_{p,air} \\ \dot{m}_{air}\, c_{p,air} \,/\, \dot{m}_{ref}\, c_{p,ref} \end{cases} \tag{4}$$

For two-phase flow in any heat exchanger, the effectiveness is

$$\varepsilon = 1 - \exp(-NTU) \tag{5}$$

The number of transfer units necessary to evaluate effectiveness depends on the overall conductance according to,

$$NTU = UA / C_{\min} \tag{6}$$

where,

$$1/UA = 1/h_i A_i + \ln(D_o / D_i)/2\pi kL + 1/\eta_o h_o A_o \tag{7}$$

The inside heat transfer coefficient in the two-phase region was calculated based on the work of Cavallini and Zecchin (1974) for smooth tubes and Cavallini et al. (2000) for microfin tubes, while the Gnielinski (1976) correlation was used in the single-phase region. The outside heat transfer coefficient was obtained from the equipment manufacturer.

For sections of the evaporator where moisture does not condense, the model is identical to the condenser model so that,

$$\dot{Q}_{evap} = \varepsilon\, C_{\min}\,(T_{air} - T_{ref}) \tag{8}$$

where C_{min} and ε were again determined according to the NTU-effectiveness method. The calculation of the inside and outside heat transfer coefficients follows that of the condenser except that the two-phase inside heat transfer coefficient is based on the work of Kandlikar and Raykoff (1997). If the coil surface temperature of a given element is below the dew point of the air, then a wet coil analysis was used based on Braun et al. (1989),

$$T_s = T_{ref} + \dot{Q}_{evap}\left(1/h_i A_i + \ln(D_o / D_i)/2\pi kL\right) \tag{9}$$

The wet analysis is a mechanistic model of the combined heat and mass transfer. The solution is cast in the form of the NTU-ε method, using enthalpy differences instead of temperature differences,

$$\dot{Q}_{evap} = \varepsilon^* \dot{m}_{air}\left(h_{a,i} - h_{r,s}\right) \tag{10}$$

Again, the form of the effectiveness depends on the phase of the refrigerant, where for single-phase flow,

$$\varepsilon^* = 1 - \exp\left[NTU^{*0.22}\left(\exp\left\{-NTU^{*0.78}\right\} - 1\right)\right] \tag{11}$$

and where for two-phase flow,

$$\varepsilon^* = 1 - \exp(-NTU^*) \tag{12}$$

The number of transfer units for a wet coil is given as,

$$NTU^* = UA^* / \dot{m}_{air} \tag{13}$$

where,

$$1/UA^* = c_s / h_i A_i + c_s \ln(D_o / D_i)/2\pi kL + c_{p,m} / \eta_o^* h_o A_o \tag{14}$$

Casting the equation for conduction in a fin in terms of enthalpy differences results in several unique quantities. The overall fin efficiency for a wet coil, η_o^*, is analogous to the usual form of the overall fin efficiency, η_o, but is additionally a function of the saturation specific heat, c_s, and the specific heat of the air-water mixture, $c_{p,m}$. The saturation specific heat is defined at a constant refrigerant temperature as the derivative of the saturated air enthalpy with respect to temperature.

Pressure drop was calculated in the same manner for both heat exchangers,

$$\Delta P = \frac{G^2}{2\rho}\frac{4 f_{tp} L}{D_h} + G^2\left(\frac{1}{\rho_g} - \frac{1}{\rho_f}\right)\Delta x \tag{15}$$

The two-phase friction factor is calculated according to Choi et al. (1999). When the refrigerant is single-phase the pressure drop is given as,

$$\Delta P = \frac{G^2}{2\rho}\frac{4 f L}{D_i} \tag{16}$$

where the friction factor is calculated according to Haaland (1983).

A typical method of managing pressure drop in heat exchangers is to split the flow into parallel circuits. A rigorous method of determining the mass flow rate in each circuit would be to perform a momentum balance on the system. Since these circuits are typically the same length a simpler approach was employed where the mass flow rate through each circuit was assumed to be the same. While this method is not exact, it does provide accurate results.

Compressor, Expansion Device, and Line Set Modeling

The compressor is modeled using compressor maps supplied by the compressor manufacturer. The maps yield compressor power and refrigerant mass flow rate, where both are empirical functions of compressor suction and discharge pressures. An energy balance yields the discharge enthalpy.

$$h_{dis} = h_{suc} + \dot{W}_{comp} / \dot{m}_{ref} \tag{17}$$

From the experimental testing, the heat loss from the compressors was found to be negligible.

The expansion device in each system is a thermal expansion valve. In general, an expansion valve can be modeled by applying the Bernoulli and continuity equations,

$$\dot{m}_{ref} = c_1 \left(T_{super} - c_2\right)\left(\rho_f \Delta P\right)^{0.5} \tag{18}$$

For thermal expansion valves, the combined expansion coefficient and throat area term is a function of the evaporator superheat. For each system, the constants in Eq. 18 were fit to the experimental data. Inputs of the refrigerant mass flow rate and the evaporator superheat yield the pressure drop across the valve. A constant enthalpy expansion is assumed.

Models of all four line sets are needed to connect the various component models. Each line set is considered to be adiabatic. This is justified in that the surface area of each is small. The pressure drop for each line set is calculated in the manner described for the heat exchangers.

Charge Inventory

The mass of refrigerant in a given volume was determined as,

$$m_{ref} = \rho V \tag{19}$$

For each compressor, the manufacturers gave the volume of the shell occupied by vapor. A temperature measurement on the upper portion of the compressor shell was used to determine the density of the vapor. When two-phase flow is encountered in the system, an apparent density is used,

$$\rho = \rho_g \alpha + \rho_f \left(1 - \alpha\right) \tag{20}$$

where α is the void fraction. Besides in the heat exchangers, two-phase flow is encountered in the line set between the thermal expansion valve and the evaporator.

The mass of refrigerant dissolved in the compressor oil is a function of the solubility of the refrigerant-oil mixture.

$$m_{dissolved} = m_{oil} \frac{\xi}{1-\xi} \tag{21}$$

Solubility data was obtained from the compressor manufacturers as a function of the temperature and pressure of the mixture. A temperature measured at the base of the compressor shell was used as the mixture temperature. The mixture pressure was taken to be equal to the compressor suction pressure. Summing the mass calculations throughout the system gives the total charge inventory.

Void Fraction Correlations

Void fraction models for three flow regimes are considered: separated flow, annular flow, and slug flow. Separated flow is a general classification that includes annular flow and wavy flow, and incorporates two primary assumptions: the flow is separated into continuous liquid and vapor regions and the velocity within each region is uniform and distinct at a given cross-section. The ratio of these velocities is termed the slip ratio because, in general, the vapor phase has a higher velocity.

$$S = u_g / u_f \tag{22}$$

Void fraction can be related to quality in terms of the slip ratio using fundamental definitions.

$$\alpha = \frac{1}{1+\left(\frac{1-x}{x}\right)\frac{\rho_g}{\rho_f}S} \quad (23)$$

Slug flow occurs at low qualities and is characterized by large intermittent bubbles separated by liquid slugs.

The void fraction models examined in this work and the flow regime for each model are listed in Table 1. The models are analytical, semi-empirical, or empirical. Given the complexity of two-phase flow, only very simple models are completely analytical. Here, models are deemed semi-empirical if they are essentially analytical, but utilize empirical closure equations or fitted constants. Two-phase empirical models often assume a functional form or a set of dimensionless parameters, and have constants that are determined from experimental data. As expected of models that are popular for design work, most of the correlations are explicit functions of measurable quantities.

A comparison of the slip ratio predictions for the six void fraction correlations presented here is given in Fig. 1. The predictions were generated for R-22, at 2 MPa, in a 9 mm diameter tube, and at a mass flux of 250 kg/m^2-s. These conditions are typical for a condenser in a unitary air conditioner. In the case of the Baroczy and Taitel correlations, the slip ratio is only an effective value because these correlations are not based on separated flow regimes. An effective slip ratio for these correlations was determined using Eq. 23. The quality at which the slug flow to annular flow transition takes place is indicated in the figure. The deficiency of the slug flow model beyond this transition point is clear as the slip ratio increases to unrealistic levels. At a constant value of one, the slip ratio prediction of the homogeneous model is clearly too low. Lacking any dependence on quality, the Zivi model appears to be too simplistic. The Baroczy, Tandon, and Yashar models have similar slip ratio trends, generally increasing with increasing quality.

RESULTS AND DISCUSSION

The cooling capacity predictions as a function of charge inventory are compared in Figs. 2 and 3 for the two modeling approaches using the Baroczy void fraction correlation. For the 5-ton rooftop unit (Fig. 2), both approaches give reasonable agreement with the measured results. Furthermore, the general trend of cooling capacity with respect to charge is captured by both approaches. The maximum value of cooling capacity occurs at a charge of around 6.5 kg. Of the two modeling approaches, the simplified model has better agreement with the measured results. For the 7½-ton split system (Fig. 3), the maximum relative error for either approach is 2.8%. The detailed model captures the trend of cooling capacity with respect to charge better than simplified model.

The coefficient of performance, COP, is defined as,

$$COP = \frac{\dot{Q}_{evap} - \dot{W}_{blower}}{\dot{W}_{comp} + \dot{W}_{blower} + \dot{W}_{fan}} \quad (24)$$

The COP predictions for the two models are compared in Figs. 4 and 5 using the Baroczy void fraction correlation. For the 5-ton rooftop unit (Fig. 4), both models predict the trend of COP with respect to changing charge, where the maximum COP occurs at a charge of 4.5 kg. The detailed model agrees better at higher charge levels, while the simplified model agrees better at lower charge levels. For the 7½-ton split system (Fig. 5), the detailed model is very accurate, while the maximum relative error for the simplified approach is 2.7%. The detailed model captures the trend of COP with respect to charge much better than the simplified model.

Discharge pressure is given as a function of the outdoor air temperature in Fig. 6 for both systems. The results of both modeling approaches are also given for each system. In each case the Baroczy void fraction correlation is used. Overall, the agreement of the models with the measured data is excellent. Nonetheless, compared to the detailed model, the simplified model has slightly better agreement with the measured results.

The results shown in Figs. 2-6 are only achievable through charge inventory modeling. With these types of models a designer is able to determine the optimum charge for a given system based either on cooling capacity or COP. Also, a designer can determine the maximum pressure that will occur in the system for a given outdoor temperature. Both of these considerations are important steps in the overall design process of an air conditioning system.

Due to the complex nature of two-phase flow, some of the utilized modeling parameters are somewhat uncertain. To better understand the potential effect of these uncertainties, several modeling parameters have been systematically evaluated. In Table 2, the effects of variations in heat transfer coefficients and two-phase friction factors on system performance are considered. These results are for the 5-ton rooftop unit using the simplified model and the Baroczy void fraction correlation. The parameters of interest are the two-phase heat transfer coefficients, the air-side heat transfer coefficients, and the two-phase friction factors. The coefficients for each heat exchanger are treated separately and each parameter is varied by -20% and +20%. Due to the high rate of heat transfer associated with two-phase processes, the air-side heat transfer coefficient is a prominent term in the expression for the thermal resistance from the refrigerant to the air. Therefore, air-side heat transfer coefficient variations have similar but stronger effects on the system performance than do the corresponding two-phase heat transfer coefficients. Altering the heat transfer coefficients in the condenser primarily affects the discharge pressure, and in turn the subcooling. As the air-side heat transfer coefficient decreases by 20%, the discharge pressure must increase by 1.9% to maintain the specified charge inventory. This leads to an increase of 1.8% in the compressor power consumption and a decrease of 1.7% in the COP. In the evaporator, variations in the heat transfer coefficients mainly affect the refrigerant mass flow rate, and in turn the heat transfer rates. As the heat transfer coefficients decrease, the thermal expansion valve control loop decreases the refrigerant mass flow rate to maintain the evaporator superheat. As the air-side heat transfer coefficient increases by 20%, the cooling capacity and rate of heat transfer in the condenser increase by 1.5% and 1.2%, respectively.

Variations in the two-phase friction factors have very little effect on the predicted system performance. The only performance parameter that is altered by a measurable amount is the condenser subcooling. Uncertainty in the refrigerant pressure drop results in variations in the saturation pressure at the condenser outlet and the evaporator inlet, both of which can alter the subcooling.

The effects of various void fraction correlations on system performance are considered in Table 3. These results are for the 5-ton rooftop unit using the simplified model. Recall that as the slip ratio decreases, the void fraction increases and the charge inventory decreases. The void fraction predictions primarily affect the discharge pressure, and in turn the subcooling. As the slip ratio decreases, the pressure must increase to maintain the specified charge inventory. Thus, the homogeneous void fraction model predicts a much higher discharge pressure than the Baroczy correlation. The range of void fraction models is also used in conjunction with a slug flow model. In these cases, the slug flow model proposed by Taitel and Barnea (1990) is used when the slug flow regime is predicted as shown in Fig. 1. Since the slug flow model predicts relatively high slip ratios, these transitional models have lower discharge pressure predictions. This is especially true of the Yashar and Tandon models, which otherwise have very low slip ratio predictions at low qualities.

The Baroczy void fraction correlation leads to the best prediction of the compressor discharge pressure. Nonetheless, the combination of the Yashar model for separated flow and the Taitel and Barnea model for slug flow is perhaps the most accurate description of void fraction of the models considered here (see Harms et al., 2002), and may be the best choice for use in system modeling. If an especially simple void fraction expression is required, then the Zivi model is recommended. The Zivi model is as easy to implement as the homogeneous model, and yet in comparison provides much more accurate predictions of the system performance. Augmenting either the Baroczy correlation or the Zivi model with the Taitel and Barnea model for slug flow is not recommended because the additional computational effort does not improve the system performance predictions.

CONCLUSIONS

Two approaches to air conditioning system modeling have been presented. The first is a detailed model that considers each circuit in the heat exchangers separately. Each tube in a circuit is broken into several tube elements. The second is a simplified model that assumes each circuit to be identical in terms of performance. The representative circuit in broken into single-phase and two-phase sections. Results unique to charge inventory modeling are presented for each approach. The simplified modeling approach is shown to be more than adequate in predicting the performance of air conditioning systems. Charge inventory calculations have been shown to be an important part of air conditioning system models.

Uncertainty in certain correlated two-phase modeling parameters can influence the performance predictions. These effects have been systematically analyzed. Variations in the two-phase friction factors do not appear to be especially important. Uncertainty in the air-side heat transfer coefficients are more important than similar variations in corresponding two-phase heat transfer coefficients. Condenser parameters mainly affect the discharge pressure and the compressor power consumption, while evaporator parameters mainly affect the refrigerant mass flow rate and the heat transfer rates. Void fraction model selection is important and mainly affects the discharge pressure and the compressor power consumption.

REFERENCES

Baroczy, C. J., 1965, "Correlation of liquid fraction in two-phase flow with application to liquid metals," *Chemical Engineering Progress Symposium Series*, Vol. 61, No. 57, pp. 179-191.

Braun, J. E., Klein, S. A., and Mitchell, J. W., 1989, "Effectiveness models for cooling towers and cooling coils," *ASHRAE Transactions*, Vol. 95-2, pp. 164-174.

Cavallini, A. and Zecchin, R., 1974, "A dimensionless correlation for heat transfer in forced convection condensation" *Heat Transfer 1974: Proceedings of the Fifth International Heat Transfer Conference*, Vol. 3, pp. 309-31

Cavallini, A., Del Col, D., Doretti, L., Longo, G. A., and Rossetto, L., 2000, "Heat transfer and pressure drop during condensation of refrigerants inside horizontal enhanced tubes," *International Journal of Refrigeration*, Vol. 23, pp. 4-25.

Choi, J., Kedzierski, M. A., Domanski, P. A., 1999, "A generalized pressure drop correlation for evaporation and condensation of alternative refrigerants in smooth and micro-fin tubes," NISTIR 6333, U.S. Department of Commerce, Washington, D.C.

Damasceno, G. S., Domanski, P. A., Rooke, S., and Goldschmidt, V. W., 1991, "Refrigerant charge effects on heat pump performance," *ASHRAE Transactions*, Vol. 97-1, pp. 304-310.

Domanski, P. and Didion, D., 1983, "Computer modeling of the vapor compression cycle with constant flow area expansion device," National Bureau of Standards, Building Science Series 155, Washington, DC.

Gnielinski, V., 1976, "New equations for heat and mass transfer in turbulent pipe and channel flow," *Int. Chem. Eng.*, Vol. 16, pp. 359-368.

Haaland, S. E., 1983, "Simple and explicit formulas for the friction factor in turbulent pipe flow," *J. Fluids Eng.*, Vol. 105, pp. 89-90.

Harms, T. M., Groll, E. A., and Braun, J. E., 2001, "Charge Inventory Modeling and Validation for Unitary Heat Pumps," Ray W. Herrick Laboratories Report #5288-1, Purdue University, West Lafayette, IN.

Harms, T. M., Groll, E. A., and Braun, J. E., 2002, "A void fraction model for annular flow in horizontal tubes," Ninth International Refrigeration and Air Conditioning Conference at Purdue, Purdue University, West Lafayette, IN.

Hughmark, G. A., 1962, "Holdup in gas-liquid flow," *Chem. Eng. Progress*, Vol. 58, pp. 62-65.

Incropera, F. P. and DeWitt, D. P., 1996, *Fundamentals of Heat and Mass Transfer*, 4th Ed., Wiley, New York.

Kandlikar, S. G., Raykoff, T., 1997, "Predicting flow boiling heat transfer of refrigerants in microfin tubes," *Enhanced Heat Transfer*, Vol. 4, pp. 257-268.

LeRoy, J. T., Groll, E. A., Braun, J. E., 2000, "Evaluating the accuracy of PUREZ in predicting unitary equipment performance," *ASHRAE Transactions*, Vol. 106-1, pp. 200-215.

Marques, M. E. and Melo, C., 1993, "Assessing the capability of the HPSIM program to simulate room air conditioners," *ASHRAE Transactions*, Vol. 99-2, pp. 309-316.

Premoli, A., Di Francesco, D., and Prina, A., 1971, "Una correlazione adimensionale per la determinazione della densità di miscele bifasiche," *La Termotecnica*, Vol. 25, pp. 17-26.

Rice, C. K., 1987, "The effect of void fraction correlation and heat flux assumption on refrigerant charge inventory predictions," *ASHRAE Transactions*, Vol. 93-1, pp. 341-367.

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, Oak Ridge, TN.

Taitel, Y., and Barnea, D., 1990, "Two-Phase slug flow," Advances in Heat Transfer, Vol. 20, pp. 83-132.

Tandon, T. N., Varma, H. K., and Gupta, C. P., 1985, "A void fraction model for annular two-phase flow," *Int. J. Heat Mass Transfer*, Vol. 28, pp. 191-198.

Yashar, D. A., Wilson, M. J., Kopke, H. R., Graham, D. M., Chato, J. C., and Newell, T. A., 2001, "An investigation of refrigerant void fraction in horizontal, microfin tubes," *HVAC&R Research*, Vol. 7, pp. 67-82.

Zivi, S. M., 1964, "Estimation of steady-state steam void-fraction by means of the principle of minimum entropy production," *J. Heat Transfer*, Vol. 86, pp. 247-252.

Table 1: Summary of the void fraction models.

	regime	correlation type	form	notes
Homogeneous		analytical	explicit	S = 1
Zivi (1964)	separated	analytical	explicit	no friction
Baroczy (1965)	separated	empirical	tabular	vertical upflow
Yashar et al. (2001)	separated	empirical	explicit	horizontal
Taitel and Barnea (1990)	slug	semi-empirical	explicit	
Tandon et al. (1985)	annular	semi-empirical	explicit	

Table 2: The effect of variations in heat transfer coefficients and two-phase friction factors on system performance for the 5-ton rooftop unit using the simplified system modeling approach and the Baroczy void fraction model.

	$\dot{Q}_{evap}$ [kW]	$\dot{Q}_{cond}$ [kW]	P_{dis} [kPa]	$\dot{m}_{ref}$ [g/s]	T_{sub} [°C]	$\dot{W}_{comp}$ [kW]	COP
Measured Results	17.62	23.43	1882	113.0	5.98	4.54	2.94
Baseline Model	17.45	23.29	1883	109.9	10.57	4.44	2.96
Condenser, h_i -20%	17.44	23.30	1892	109.9	10.72	4.46	2.95
Condenser, h_i +20%	17.46	23.28	1877	109.9	10.47	4.43	2.97
Evaporator, h_i -20%	17.41	23.20	1882	109.6	10.13	4.44	2.95
Evaporator, h_i +20%	17.48	23.32	1883	110.0	10.57	4.44	2.97
Condenser, h_o -20%	17.40	23.32	1918	109.9	11.03	4.52	2.91
Condenser, h_o +20%	17.49	23.22	1860	109.8	9.82	4.39	3.00
Evaporator, h_o -9%†	17.33	23.16	1879	109.2	10.55	4.44	2.95
Evaporator, h_o +20%	17.71	23.56	1888	111.3	10.58	4.45	3.00
Condenser, f_{tp} -20%	17.45	23.29	1882	109.9	10.62	4.44	2.96
Condenser, f_{tp} +20%	17.45	23.25	1883	109.9	10.08	4.44	2.96
Evaporator, f_{tp} -20%	17.46	23.26	1883	109.9	10.15	4.44	2.96
Evaporator, f_{tp} +20%	17.45	23.28	1882	109.8	10.56	4.44	2.96

† An unstable operating condition exists at h_o –20%

Table 3: The effect of void fraction model selection on system performance for the 5-ton rooftop unit using the simplified system modeling approach.

	$\dot{Q}_{evap}$ [kW]	$\dot{Q}_{cond}$ [kW]	P_{dis} [kPa]	$\dot{m}_{ref}$ [g/s]	T_{sub} [°C]	$\dot{W}_{comp}$ [kW]	COP
Measured Results	17.62	23.43	1882	113.0	5.98	4.54	2.94
Homogeneous	17.53	23.70	2028	109.2	15.66	4.78	2.82
Yashar et al. (2001)	17.52	23.38	1945	109.5	12.11	4.59	2.90
Tandon et al. (1985)	17.51	23.43	1922	109.6	12.31	4.53	2.93
Zivi (1964)	17.49	23.31	1909	109.7	11.06	4.50	2.94
Baroczy (1965)	17.45	23.29	1883	109.9	10.57	4.44	2.96
Yashar et al. + slug flow†	17.50	23.41	1914	109.6	11.98	4.52	2.94
Tandon et al. + slug flow†	17.47	23.27	1893	109.8	10.52	4.47	2.95
Zivi + slug flow†	17.49	23.39	1909	109.7	11.80	4.50	2.94
Baroczy + slug flow†	17.45	23.29	1881	109.9	10.51	4.44	2.96

† Taitel and Barnea (1990)

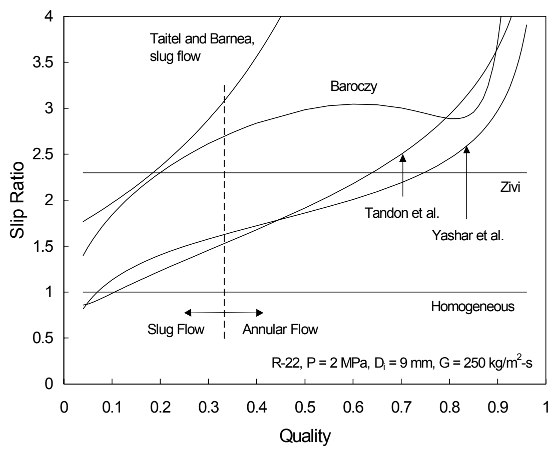

Figure 1: Slip ratio predictions for various void fraction correlations.

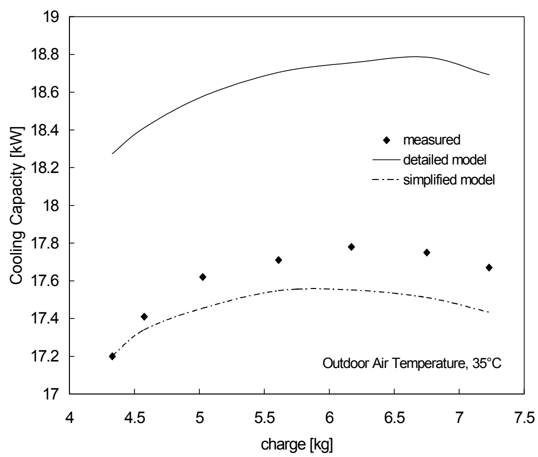

Figure 2: Cooling capacity comparisons for the 5-ton rooftop unit using the Baroczy void fraction correlation.

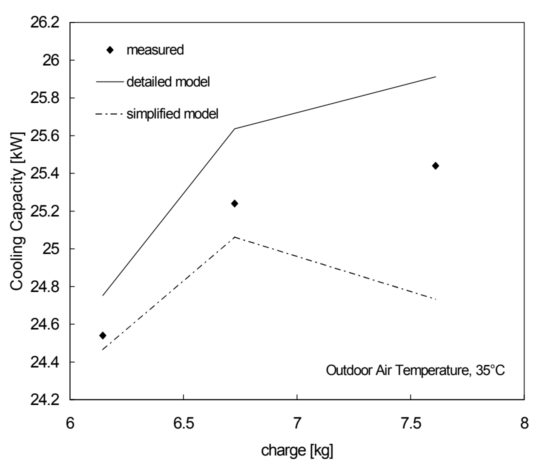

Figure 3: Cooling capacity comparisons for the 7½-ton split system using the Baroczy void fraction correlation.

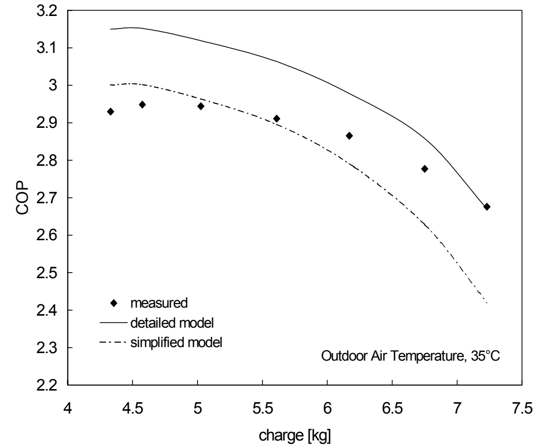

Figure 4: COP comparisons for the 5-ton rooftop unit using the Baroczy void fraction correlation.

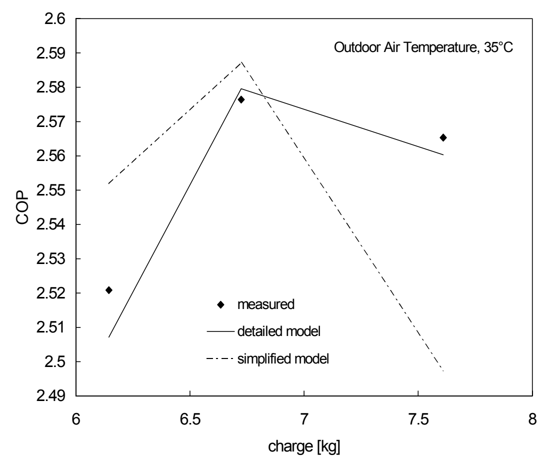

Figure 5: COP comparisons for the 7½-ton split system using the Baroczy void fraction correlation.

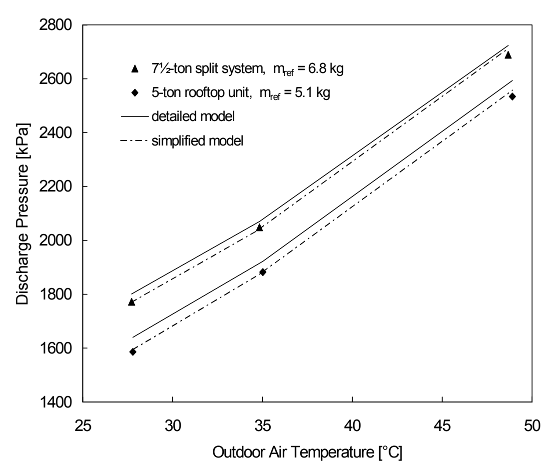

Figure 6: Compressor discharge pressure comparisons with respect to outdoor air temperature using the Baroczy void fraction correlation.