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ABSTRACT

Simulation models for unitary air conditioners and heat pumps usually can not provide accurate predictions in the case of two-phase refrigerant entering the compressor or fixed-area expansion device. This paper focuses on improving modeling methods in these situations. The study utilized extensive laboratory tests and detailed modeling of two unitary air conditioners, one using R-410A and the other using R-407C. Heat exchanger models were used to give upstream qualities of the compressor and fixed orifice. It was found that it is best to use a two-phase suction density correction for compressor maps to predict mass flow rate. However, using a two-phase suction density correction with a compressor map to predict the power consumption leads to significant under-predictions. It is best not to correct the compressor map power consumption for a two-phase entering condition. For the case of two-phase refrigerant entering a fixed-area expansion device, mass flow rate predictions from short-tube orifice models are extremely sensitive to the upstream quality. A two-point charge tuning method is recommended, as it can lead to an accurate prediction of upstream state for the expansion device at all refrigerant charge levels.

1. INTRODUCTION

Leroy et al. (1997) conducted system simulations for ten different R-22 unitary air conditioners operating under off-design conditions. The units had cooling capacities ranging from 2 to 5 tons and different types of expansion devices and compressors. There were two cases where the system simulation results were poor: 1) two split units having a fixed orifice and working at high ambient temperatures and 2) a packaged unit having four parallel fixed orifices and operating at low refrigerant charge.

Based on a detailed analysis of the results, it can be concluded that the poor simulation results of Leroy et al. (1997) were due to two-phase refrigerant entering the compressor and/or fixed-area expansion device. Two-phase refrigerant inlet conditions to a short-tube usually can occur at low charge levels or very high ambient temperatures. Two-phase refrigerant inlet conditions to a compressor tend to occur at high outdoor temperatures, low indoor humidity, low indoor air flow rates, and high charge levels. These off-design conditions are important since they can occur in the field, and can lead to significant errors in predictions from simulation models.

ARI compressor maps (ANSI/ARI 540-99) are typically used within system simulation models to predict compressor mass flow rate and power consumption. The equations are polynomials that are developed for a fixed superheat entering the compressor (e.g., 11.1 K). For the purpose of system simulation, the maps for mass flow rate and power consumption are typically corrected for actual (non-standard) suction superheat. The standard ARI mass flow equation can be corrected for varying superheats using the ratio of the calculated suction gas density to the standard suction gas density corresponding to the ARI standard test condition. Rice (1981) proposed adjusting the suction density using Equation 1.

$$\dot{m}_{ref,actual} = [1 + F_{mass}(\frac{v_{ARI-map}}{v_{act}} - 1)]\dot{m}_{ref,ARI-map} \tag{1}$$

where F_{mass} is an empirical correction factor assigned a value of 0.75, $\dot{m}_{ref,ARI-map}$ and $\dot{m}_{ref,actual}$ are the mass flow rates at the standard and actual suction superheat, and $v_{ARI-map}$ and v_{act} are the specific volumes at the standard and actual superheat.

Rice (1981) also proposed using Equation 2 to correct power consumption predictions for different superheat levels.

$$\dot{W}_{actual} / \dot{W}_{ARI-map} = \frac{\dot{m}_{ref,actual}}{\dot{m}_{ref,ARI-map}} \frac{\Delta h_{is,actual}}{\Delta h_{is,ARI-map}} \tag{2}$$

where $\dot{W}_{ARI-map}$ and $\dot{W}_{actual}$ are the compressor power consumptions for the standard and actual superheat and $\Delta h_{is,ARI-map}$ and $\Delta h_{is,actual}$ are the isentropic enthalpy changes for the compressor for the standard and actual superheat. Equation 2 assumes that the isentropic efficiency doesn't change between the standard and actual suction superheat and accounts for the variations in mass flow rate and isentropic enthalpy change

The Rice (1981) methods for correcting ARI standard compressor maps lead to accurate simulations for a large range of superheated suction states. However, there are no published investigations that have evaluated how well these methods work when two-phase refrigerant enters the compressor. Two-phase refrigerant entering a compressor shell of a hermetic compressor can lead to wet compression, which reduces compressor performance. The actual state of refrigerant entering the compression chamber depends on both the shell inlet state and an energy balance for refrigerant in the shell. For compressor shells that operate at suction pressure, liquid refrigerant is likely to evaporate inside the compressor shell due to heat losses from the motor. This is not the case for shells that operate at discharge pressure. Therefore, it is difficult to know the best suction state to use for correcting compressor mass flow and power predictions in a simple compressor map model when two-phase flow enters the shell. Two simple approaches are considered in this paper: 1) use the calculated two-phase suction density or 2) use saturated vapor density at the suction pressure.

When the inlet condition to a short-tube orifice or capillary tube changes from a subcooled liquid to a two-phase mixture, there is a sudden drop in mass flow rate. The experiments of Kim and O'Neal (1994) for R-22 flowing through short-tube orifices reveal this phenomenon, as depicted in Figure 1. Kim and O'Neal indicated that this tendency is due to a large void fraction for two-phase refrigerant at the entrance. Figure 1 indicates that the curves for different upstream pressures converge to the same point at zero subcooling, meaning that mass flow rate is not sensitive to upstream pressure when the inlet is a saturated liquid. Payne and O'Neal (1999 and 2003) investigated R-410A and R-407C flowing through short-tube orifices and developed separate semi-empirical correlations for mass flow rate within short-tubes for subcooled and two-phase refrigerant inlet conditions. However, in order to obtain accurate results with these correlations, it is very important to have accurate predictions of the inlet state especially when the inlet condition is two-phase or close to the transition between subcooled liquid and two-phase mixture.

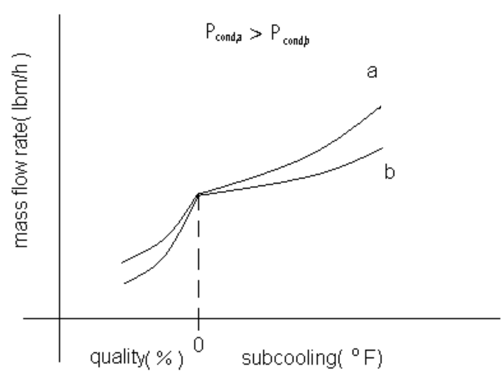

Figure 1: Mass flow rate as a function of subcooling and quality in a sharp-edged short tube (Kim and O'Neal 1994).

2. RESEARCH METHODOLOGIES

Test data and simulation results were used to investigate modeling issues associated with two-phase refrigerant entering compressors and the expansion devices. Two unitary air conditioners were tested, including a 3-ton R-410A packaged unit using a short-tube orifice and a scroll compressor, and a 5-ton R-407C packaged unit using twelve parallel short-tubes and a scroll compressor. The varied operating conditions were outdoor temperature, indoor air flow rate, outdoor air flow rate, indoor relative humidity and charge inventory. Descriptions of the test units, measurements, and test conditions are presented by Shen et al. (2006b).

In the case of two-phase refrigerant entering a compressor or expansion device, pressure and temperature measurements do not provide sufficient information to determine upstream quality. In addition, air-side capacity measurements are not accurate enough to use in calculating refrigerant-side qualities.

In this study, qualities at the entrances to compressors and expansion devices were determined using tuned evaporator and condenser models.,. The evaporator and condenser were modeled using ACMODEL, which is described by Shen et al. (2006b). The heat exchanger models were tuned by adjusting the air side heat transfer coefficients with a multiplier to match the measured cooling and condensing capacity at one design operating condition.

The accuracies of individual component models were assessed independently by employing local measured (or estimated) boundary conditions. For the compressor model, suction pressure, discharge pressure, and suction temperature were inputs and mass flow rate and compressor power were outputs that were compared with measured values. For the fixed-area expansion device model, upstream and downstream states were inputs and mass flow rate was output and compared with measurements. For the heat exchanger models, measured inlet states and flow rates, were inputs and energy transfer rates (capacities) were outputs and compared with measurements determined from refrigerant-side capacities.

The model assessments are presented in terms of relative deviations (i.e., deviation between predicted and measured values divided by the measured value). Table 1 shows mean and maximum (max) deviations between adjusted heat exchanger model capacity predictions and measured values for the cases that did not have two-phase refrigerant entering the compressor and expansion device. A single heat transfer tuning coefficient works very well for a large range of operating conditions, with deviations between predicted and measured capacities smaller than 3.0%. The adjusted heat exchanger models are more accurate than measured air-side capacities and represent a better tool for estimating qualities upstream of the compressor and expansion device for two-phase entrance conditions.

Table 1: Overall accuracy of heat exchanger models.

Heat Exchanger	R-410A, Packaged		R-407C, Packaged	
	Mean	Max	Mean	Max
Condenser	0.7%	2.2%	0.6%	2.7%
Evaporator	0.8%	2.8%	0.4%	1.7%

3. TWO-PHASE REFRIGERANT ENTERING COMPRESSORS

Table 2 presents compressor model assessments for cases having superheated vapor entering the compressor shell. The mass flow rate predictions were determined using the suction density correction given in Equation 1. Compressor power consumption predictions were determined both with and without the suction density correction of Equation 2 and then compared with measurements for the results in Table 2. The compressor map predictions of mass flow rate and power consumption are very accurate for these cases having superheated vapor inlet. It is interesting that the power consumption predictions are a little better if the suction density correction is not utilized. However, the differences associated with using and not using Equation 2 are relatively small. Compressor mass flow rate decreases and isentropic enthalpy difference increases with increasing suction superheat. These effects balance each other so that the correction for power consumption is relatively insensitive to suction superheat.

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Table 2: Compressor model assessments in cases having superheated vapor inlet conditions.

Units		R-410A, Packaged		R-407C, Packaged	
Deviations		Mean	Max	Mean	Max
Mass flow rate	Rice correction (1981)	1.2%	3.1%	0.5%	2.3%
Power consumption	No suction density correction	0.6%	2.5%	0.4%	1.8%
	Rice correction (1981)	1.0%	3.0%	1.3%	4.6%

Both the R-410A and R-407C packaged units had scroll compressors, which can tolerate significant liquid refrigerant without damage. Among the test conditions that were considered, two-phase refrigerant entered the compressor for 26 operating points with the R-410A unit and 23 operating points for the R-407C packaged. These operating points tended to occur at high outdoor temperatures, low indoor humidity, low indoor air flow rates, and high charge levels. The lowest inlet quality encountered was about 90%. When significant two-phase refrigerant entered the compressor, the compressor body temperature dropped drastically.

Deviations between mass flow rate predictions and measurements for the R-410A scroll compressor are shown in Figure 2 for cases with two-phase inlet conditions. Corrections for suction density were determined using either a saturated vapor condition at the measured inlet pressure or a two-phase condition at the measured pressure and quality predicted by an evaporator model. Figure 2 indicates that the use of a saturated vapor density correction tends to give under-predictions of mass flow rate, and the deviations increase with decreasing quality. The use of a two-phase density correction gives more accurate predictions, especially at qualities larger than 93%. Similar results were obtained for the R-407C scroll compressor. Overall statistics for accuracy of the two different correction approaches applied to the R-410A and R-407C compressors with two-phase inlet conditions are given in Table 3.

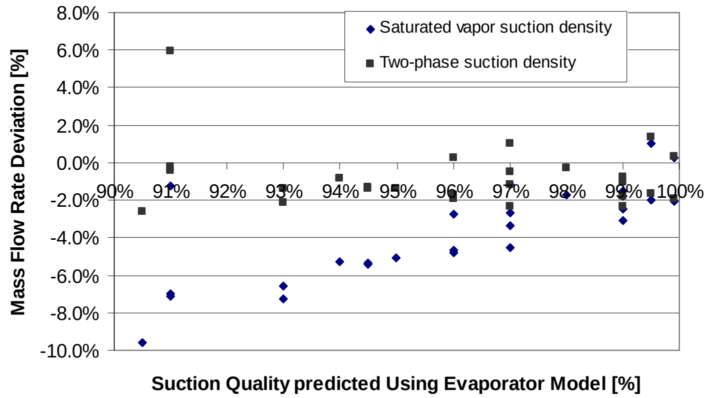

Figure 2: Mass flow rate prediction deviations for cases of two-phase suction with the R-410A scroll compressor

Table 3: Mass flow rate prediction deviations for cases of two-phase suction.

Units	R-410A, Packaged		R-407C, Packaged	
Deviations	Mean	Max	Mean	Max
Two-phase suction	1.5%	6.0%	3.5%	6.5%
Saturated vapor suction	3.8%	9.6%	4.3%	8.5%

Deviations between compressor power predictions and measurements for the 3-ton R-410A scroll compressor are presented in Figure 3 for cases of two-phase flow entering the compressor suction. The predictions were obtained both with and without a correction for suction conditions. The results of Figure 3 indicate that it is best not to correct the map-based power consumption for variations in suction state using Equation 2. The power consumption is nearly independent of inlet quality for these two-phase cases. The use of Equation 2 with two-phase suction state leads to significant underestimation of compressor power. Similar results were obtained for the R-407C compressor. Table 4 shows summary statistics for all two-phase cases with the R-410A and R-407C compressors.

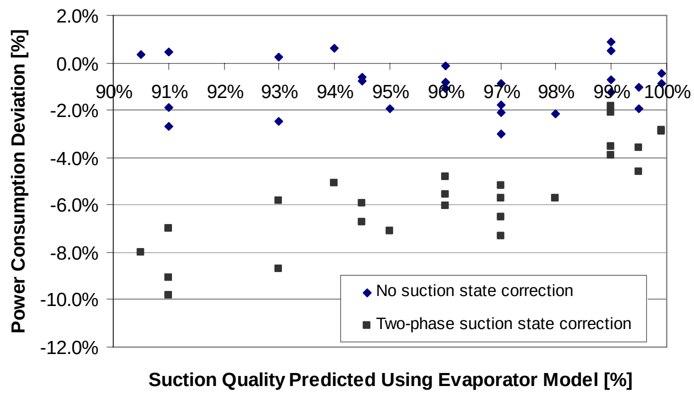

Figure 3: Power consumption prediction deviations in the cases of two-phase suction for the 3-ton R-410A packaged unit.

Table 4: Power prediction deviations in the cases of two-phase suction

	R410A, Packaged		R407C, Packaged	
Deviations	Mean	Max	Mean	Max
No suction state correction	1.2%	3.0%	0.6%	1.9%
Two-phase suction state correction	5.6%	9.8%	1.9%	4.1%

4. TWO-PHASE REFRIGERANT ENTERING FIXED-AREA EXPANSION DEVICE

Table 5 presents assessments of the short tube models of both units for the cases with a subcooled upstream state. For the R-410A packaged unit, the Payne and O'Neal (1999) correlation for pure R-410A was used. For the R-407C unit, the Payne and O'Neal (2003) correlation for pure R-407C was used, and the refrigerant distribution among the twelve parallel orifices was assumed to be uniform.

Table 5: Mass flow rate deviations of short-tube correlations for cases having subcooled upstream states

	R410A, Packaged		R407C, Packaged	
Deviations	Mean	Max	Mean	Max
Mass flow rate	0.6%	1.8%	1.3%	4.7%

Table 5 indicates that the short-tube correlations provide accurate predictions when the upstream state is subcooled. However, there is a sharp drop in refrigerant mass flow rate when the inlet state to a fixed-area expansion device transitions from a subcooled to a two-phase condition, as shown in Figure 1. In this case, model predictions are extremely sensitive to the upstream quality. The inlet condition for a short-tube can become two-phase at low charge levels or very high ambient temperatures. Among the test conditions that were considered, two-phase refrigerant entered the expansion device for 11 operating points with the R-410A system and 8 operating points for the R-407C packaged unit. The highest inlet quality encountered was about 10%.

The accuracies of the short-tube orifice models were evaluated through comparisons of predicted mass flow rates and mass flow rates determined from measurements. Unfortunately, the liquid-line refrigerant flow measurements were unreliable for cases with two-phase exit conditions from the condenser. In these cases, measured mass flow rates were obtained by using the compressor as a virtual sensor. This involved using the tuned compressor map with measured suction conditions and discharge pressure. Table 6 shows validation results for the short-tube orifice model with two-phase inlet conditions for the R-410A packaged unit. In Table 6, X_{cond} is the upstream quality predicted with the tuned condenser model and Dev_{mr} is the deviation between short-tube orifice predictions of mass flow rate and mass flow rates determined using the tuned compressor map model. The deviations in mass flow rate can be quite large for these cases and varied between about 3 and 20%. The largest deviation occurred at a point where the inlet quality was nearly a maximum for the cases considered. In general, the refrigerant inlet qualities and deviations are higher for low refrigerant charge and high ambient temperature.

Table 6: Case studies of two-phase entering short-tube for the R-410A packaged unit.

Conditions	X_{cond} [%]	Dev_{mr} [%]	X_{mr} [%] or $Tsub_{mr}$ [˚F]	Dev between X_{cond} & X_{mr}
Charge<80% of rated;	9.5%	-9.4%	6%	2.3%
82 ˚F ambient;	4.5%	-7.6%	2.2%	1.3%
Dry	0.5%	-13.5%	1.8 ˚F	0.83%
Charge<80%;	5.0%	3.5%	6.0%	1.02%
82 ˚F ambient;	4.5%	-9.9%	1.5%	2.47%
Wet	1.0%	-16.0%	3.8 ˚F	1.67%
Charge<80%;	10.5%	-12.4%	6.0%	3.15%
95 ˚F ambient;	7.0%	-13.9%	2.5%	2.89%
Wet	1.0%	-15.4%	1.8 ˚F	0.88%
Charge<80%;	10.0%	-20.4%	2.8%	4.3%
115 ˚F ambient	3.0%	-14.5%	0.9%	0.98%
Mean Absolute Dev_{mr} [%]		12.4%	Mean Dev between X_{cond} & X_{mr}	2.0%

The upstream qualities or subcooling necessary for the short-tube orifice model to predict the mass flow rates determined with the tuned compressor map model are also presented in Table 6 under the columns labeled X_{mr} and $Tsub_{mr}$. The last column in this table gives deviations between X_{cond} and X_{mr}. These results indicate that mass flow rate predictions from a short-tube orifice model are extremely sensitive to the upstream state under two-phase conditions. For instance, the maximum mass flow rate error of 20.4% only corresponds to a 4.3% deviation in the condenser exit quality. An average difference of 2.0% in exit quality leads to an average error of 12.4% in predicting refrigerant mass flow rates. Therefore, determining an accurate upstream state is the most important issue in obtaining an accurate refrigerant mass flow rate prediction when two-phase flow enters a fixed-area expansion device.

One of the keys in obtaining an accurate inlet state for the expansion device is to adequately account for the effect of refrigerant charge. Typically, refrigerant charge is tuned to account for unknown internal volumes by adjusting the simulated charge level by a fixed amount so that the simulation predicts the correct subcooling at a single operating point. Rossi (1995) presents a method for single-point tuning that was developed for ACMODEL. However, Shen et al. (2006a) demonstrated that a single-point charge tuning method can not provide good simulation results over a large range of charge levels. Shen et al. (2006a) also presented an improved method for adjusting refrigerant charge using a charge correction equation that requires data for two operating points and is called a two-point charge tuning method. The approach associates the variable charge errors with the subcooled liquid length, and considers all other errors to be a constant offsets. The ability of a system model to predict off-design charge effects is significantly improved through the use of this tuning approach. Figure 4 presents example predictions of condenser subcooling from simulations that used the two-point charge tuning method as compared with a one-point charge tuning method. Similar results were obtained for other operating conditions and units. In general, the two-point charge tuning method provides much better predictions of expansion device inlet states than the single-point method..

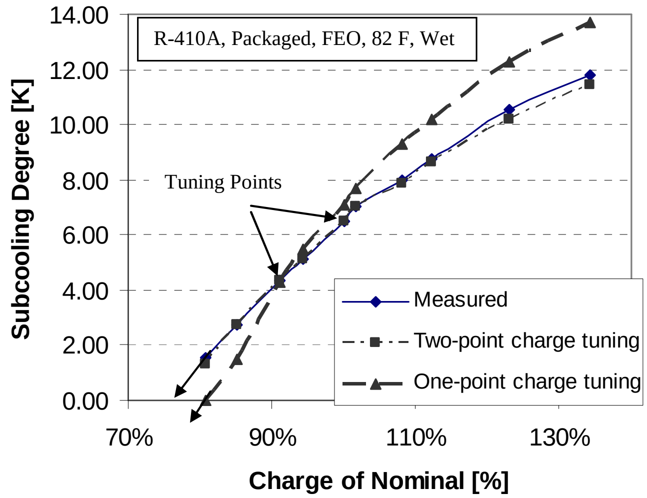

Figure 4: Condenser subcooling as a function of charge mass for the R-410A packaged unit

Table 7 shows the impact on cooling capacity and power predictions of using the two-point charge tuning method as compared with the single-point method for cases with two-phase refrigerant entering the expansion devices for both units. The two-point tuning leads to much better system-level predictions.

Table 7: Simulations of the cases having two-phase state upstream of the fixed-area expansion device.

Prediction Deviations		$Dev_{cooling}$ [%] One-point	$Dev_{cooling}$ [%] Two-point	Dev_{power} [%] One-point	Dev_{power} [%] Two-point
R-410A	Mean	24.1%	8.3%	4.9%	2.8%
	Max	42.0%	22.5%	7.7%	5.6%
R-407C	Mean	13.5%	5.0%	4.0%	0.6%
	Max	24.3%	9.3%	6.8%	1.6%

5. CONCLUSIONS

Typical compressor maps work well for predicting mass flow rate when the entering refrigerant is a superheated vapor. However, they become significantly worse in predicting mass flow rate when the inlet condition is a two-phase mixture. Under these conditions, it is significantly better to use a two-phase suction density correction for mass flow rate predictions as compared with a saturated vapor density correction. For power consumption, it is best to use the compressor map with no correction for the actual superheat (or two-phase) condition. Correcting the suction state for two-phase conditions can lead to large errors in compressor power predictions. Even for superheated conditions, the uncorrected compressor map for power provides slightly better predictions than the correction of Equation 2. It is important to note that the current study was performed only for hermetic, scroll compressors with the shells under suction pressure. Additional work may be necessary to verify the results for other compressor types.

Semi-empirical models for short-tube orifices provide mass flow rate predictions that are extremely sensitive to refrigerant quality for two-phase refrigerant inlet conditions. One of the keys to having accurate predictions of condenser exit states for these cases is to properly account for the effects of refrigerant charge. This can only be accomplished through proper tuning of the simulated refrigerant charge. A single-point charge tuning method is not sufficient. However, a two-point charge tuning method can significantly improve the ability of system simulations to accurately predict cooling capacity and power consumption for situations where two-phase refrigerant enters the expansion device.

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