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ABSTRACT

Models for unitary air conditioners and heat pumps typically have difficulties in providing accurate predictions for the following situations: 1) charge modeling over a large range of charge levels, 2) two-phase refrigerant entering fixed-orifice expansion devices, 3) two-phase refrigerant entering compressors, and 4) non-uniform refrigerant mass flow allocation among evaporator circuits at low indoor air flow rates. This paper provides a detailed evaluation of the impact of non-uniform refrigerant mass flow allocation among evaporator circuits on performance predictions. In addition, it gives brief descriptions and overall assessments of improved modeling methodologies for the other three issues. The study utilizes data obtained from extensive testing at a wide range of operating conditions for two R-410A units and one R-407C unit.

1. INTRODUCTION

Leroy (1997) conducted system simulations for unitary air conditioners working under off-design conditions and considered ten R-22 units with cooling capacities ranging from 2 to 5 tons having different types of expansion devices and compressors. Three system simulation tools were used to perform simulations at conditions that typically lead to inaccurate results: high ambient temperatures, off-design charges and off-design air flow rates. This study concluded that the models should be improved in order to better handle these off-design conditions. In particular, Leroy pointed out that compressor maps may be less accurate at high ambient temperatures, since they are developed at standard ambient conditions. In addition, operation at high ambient temperatures tends to increase the probability of two-phase refrigerant entering the compressor or fixed-area expansion device, which can lead to large errors in refrigerant mass flow rate calculation. Two-phase refrigerant entering the fixed-area expansion device is particularly likely at low refrigerant charge. Leroy also suggested that air-side heat transfer correlations might be less accurate at low air flow rates. Furthermore, low mass flow rates for indoor air and refrigerant can lead to non-uniform distributions of refrigerant among heat exchanger circuits. These off-design conditions are important because they occur in the field and impact overall energy usage of the equipment.

One of the simulation tools considered by Leroy (1997) is ACMODEL, which is the tool utilized in the current study. ACMODEL is a detailed steady-state vapor compression system simulation model developed by Rossi (1995) and enhanced by Shen et al. (2006c). Major modeling approaches and assumptions within ACMODEL include:

1) Finite difference modeling of heat exchangers with each tube divided into small segments. An ε-NTU approach is used for heat transfer calculations within each segment.
2) Use of a method by Braun et al. (1989) to simulate cases of water condensing on an evaporating coil where the driving potential for heat and mass transfer is the difference between enthalpies of the inlet air and saturated air at the refrigerant temperature.
3) Consideration of both refrigerant and air-side heat transfer and pressure drop.
4) Simplification of each air-side fin as an equivalent annular fin.
5) Use of ARI compressor map formulations (ANSI/ARI Standard 540-99) to predict mass flow rate and power consumption and inclusion of a suction density correction and an energy balance to obtain compressor exit enthalpy.
6) Use of the Payne and O'Neal (1994, 1999 and 2003) correlations for modeling fixed-area expansion orifices (FEO). For modeling systems that use a TXV, the superheat of the refrigerant entering the compressor is assumed to be maintained constant.
7) Consideration of detailed charge inventory and allowing specification or prediction of system charge.
8) Capability of modeling actual heat exchanger circuiting patterns and investigating effects of non-uniform refrigerant and air flow.

Rossi (1995) developed a complete tuning approach for ACMODEL that uses measurements at one design point for the refrigerant inlet and outlet states of each component and air-side boundary conditions. For the condenser or evaporator, inlet conditions and measured refrigerant mass flow rate are inputs, and then the air-side and refrigerant-side heat transfer coefficients are tuned with one multiplier to give an outlet enthalpy that matches the value associated with the measured outlet pressure and temperature. After these calculations, the system charge is predicted by integration of the density over all internal volumes. Then the simulated system charge is obtained and the deviation between the simulated charge and actual charge is the unaccounted charge, which can be added as a constant offset at other working conditions. The physical meaning of the unaccounted charge could be the charge mass in unaccounted volumes, such as part of the liquid line. In addition, using upstream and downstream pressure and temperature measurements, the fixed-area expansion device model can be tuned by adjusting a multiplier of the calculated mass flow rate to match the measured refrigerant mass flow rate. The overall tuning approach is called a one-point tuning method, because it uses data from a single test condition. The tuning approach works well in most cases for simulating a system with a constant amount of refrigerant charge. However, it has difficulty in modeling the effects of variations in refrigerant charge over a large range.

Leroy et al. (2000) tuned refrigerant charge for their simulations using a single operating point and then investigated two different methods for adjusting the simulated charge for off-design charge simulations: 1) the tuned charge level was adjusted by the same absolute quantity as was done in the laboratory tests, and 2) the tuned charge level was adjusted by the same percentage as that associated with the laboratory tests. Both approaches did not properly account for the effect of charge on cooling capacity and compressor power. The authors suggested that tuning the system charge at one design condition was not sufficient to allow good performance predictions for other refrigerant charges.

One of the most important issues in charge modeling is to predict an accurate relationship between subcooling degree and system charge inventory. Harms et al. (2002) investigated the issue of charge inventory modeling using data from two R-22 TXV units and one R-407C TXV unit. They evaluated twelve void fraction models and found the Baroczy (1965) model to give the best overall results for charge inventory. The use of the Baroczy (1965) void fraction model led to an improvement in charge modeling, but the model still failed to provide a good relationship between subcooling degree and charge inventory over a large range. There is a need for a better method of system charge modeling that performs well over a large charge range.

The objective of the work described in this paper was to develop and identify methodologies for improving simulation models for unitary air conditioners and heat pumps at off-design conditions. Some of the improvements and detailed validations are presented in other papers (Shen et al. (2006a, 2006b)). The current paper employs all of these improvements, but focuses on the impacts of refrigerant mal-distribution on model accuracy and the overall model accuracy when employing all of the improvements.

2. RESEARCH METHODOLOGIES

2.1 Testing

Three unitary air conditioner systems, two R-410A units and one R-407C unit, were tested in the laboratory over a wide range of operating conditions. The experiments were carried out in ASHRAE standard psychometric chambers. Pressures and temperatures were measured at the flow stream inlets and outlets for each component. Refrigerant mass flow rate was measured using a micro-motion mass flow meter and indoor air flow rate was measured using nozzles downstream of the evaporating coil. In general, refrigerant-side and air-side cooling capacities determined from these measurements were within 6%. Additional measurements were made for specialized purpose of detecting refrigerant mass flow distribution: differential pressure transducers were used to measure the pressure drop across each circuit of a heat exchanger. In addition, the exit superheat degree of each circuit was measured separately using a wire thermocouple soldered on the surface of the exit tube and insulated with foam tape. The circuit pressure drop and superheat degree are indicative of the refrigerant mass flow distribution.

General information describing the test units is given in Table 1. Multiple test series were conducted for each of the three units. In each test series, only one operating condition was varied, while all other conditions were held constant. The varied operating conditions were outdoor temperature, indoor air flow rate, outdoor air flow rate, indoor relative humidity and charge inventory. Table 2 shows the test matrix for the 3-ton R-410A packaged unit that used a fixed-area expansion device. The other two units had similar test matrices. The evaporating coil was “wet” for indoor relative humidities greater 50%, and “dry” for relative humidities below 40%.

Table 1: Test unit information

Information	3-ton R-410A Split Unit	3-ton R-410A Packaged Unit	5-ton R-407C packaged Unit
Expansion	1. TXV; 2. Fixed orifice	Fixed orifice	12 parallel fixed orifices
Compressor	Reciprocating piston	Scroll	Scroll
Condensing coil	Single row; Two parallel circuits combined to one subcooled circuit;	Two rows; Two parallel circuits without subcooled circuit;	Two rows; six parallel circuits combined to one subcooled circuit;
Evaporating coil	Three rows; Five parallel circuits	Three rows; Four parallel circuits	Three rows; twelve parallel circuits

Table 2: Test matrix for 3-ton R-410A packaged air conditioner with a short-tube orifice.

Series No.	# of Tests	Outdoor T [˚F]	Outdoor Air Flow [%]	Indoor T [˚F]	Indoor Air Flow [CFM]	Indoor RH [%]	Charge [%]
No. 1	6	82~125	100%	80	100%	<30%	100%
No. 2	9	82~125	100%	80	100%	51%	100%
No. 3	6	95	100%	80	100%	24~71%	100%
No. 4	9	95	100%	80	50%~120%	<30%	100%
No. 5	10	95	100%	80	40%~120%	51%	100%
No. 6	5	115	100%	80	60%~100%	<40%	100%
No. 7	10	82	100%	80	100%	<30%	60~130%
No. 8	12	82	100%	80	100%	51%	60~130%
No. 9	12	95	100%	80	100%	51%	60~130%
No. 10	6	115	100%	80	100%	<40%	70~120%

2.2 Component and system level validation

Component model validations helped to identify sources of discrepancies, whereas system level model validations were useful for identifying inaccuracies due to charge simulation and for validating the improved modeling methods. To assess the accuracy of component model predictions, measured local boundary conditions (e.g., refrigerant and air states) were input to the component models and then predicted outputs were compared with measured values. In order to simulate a system that uses a TXV, it is necessary to specify the evaporator outlet superheat degree and either the system charge or the condenser outlet subcooling degree. For simulation of a unit that uses a fixed orifice, it is only necessary to specify subcooling degree, superheat degree, or system charge. For system-level model validations, air-side boundary conditions (e.g., temperature and humidity) were provided as inputs and predictions were compared with measured values. In the following sections, the assessments are shown in terms of relative deviation, which is the deviation between the predicted and measured values, divided by the measured value. The maximum (max) and mean deviations are also provided.

3. IMPROVED METHODOLOGIES FOR SIMULATIONS AT OFF-DESIGN CONDITIONS

3.1 Improved Charge Inventory Tuning

Existing models do not accurately predict the effect of off-design refrigerant charge even when tuned at the design operating condition with a method such as the one presented by Rossi (1995) and described in Section 1. Based on the experimental testing and theoretical analyses of the three test units, multiple factors that cause inaccuracies in charge prediction were identified. The factors include unaccounted liquid volumes, refrigerant dissolved in the compressor lubricant, inaccurate void fraction models, inaccurate inside cross-sectional area of tube and an inaccurate estimate of the subcooled liquid length. Some of these factors lead to errors that change with operating conditions and amount of charge. A method was presented by Shen et al. (2006a) for adjusting refrigerant charge using a charge correction equation that requires data for two operating points for parameter tuning, 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 offset, as shown in Equation 1.

$$\Delta M_{liqL} = C + k_{liqL} \times L_{liq} \quad (1)$$

where ΔM_{liqL} is the variable charge inaccuracies associated with the predicted liquid length L_{liq} at any operating condition and C and k_{liqL} are two constant coefficients, determined using measurements at two operating points. The estimated inventory error can be used in two different ways, depending on the goal of the simulation. If the subcooling is specified and the goal is to estimate the required refrigerant charge for a given operating condition, then ΔM_{liqL} can be directly added to the untuned system charge prediction $M_{simulated}$ as shown in Equation 2.

$$M_{corrected,LiqL} = M_{simulated} + \Delta M_{liqL} \quad (2)$$

On the other hand, if the goal of the simulation is to determine the performance of the system at a specified refrigerant charge, then the right-hand side of Equation 2 is set equal to the known charge and the system of equations are solved iteratively to satisfy the charge inventory requirement along with the other modeling equations.

This improved method leads to much improved predictions of the relationship between subcooling degree and charge inventory over a very large range of operating conditions. The tuning approach was tested for all three units and all operating conditions using ACMODEL and compared with both measurements and the results associated with existing tuning methods. Detailed results are presented by Shen et al. (2006a).

3.2 Two-Phase Refrigerant Entering Fixed-Area Expansion Devices

Two-phase refrigerant inlet conditions to a short-tube usually can occur at low charge levels or very high ambient temperatures. In these cases, Shen et al. (2006b) showed that predictions of mass flow rate are extremely sensitive to the upstream quality. Consequently, giving an accurate upstream state is the most important issue in improving refrigerant mass flow rate predictions when two-phase flow enters a fixed-area expansion device. Shen et al. (2006b) showed that the two-point charge tuning method of Section 3.1 can lead to accurate predictions of upstream state of the expansion device as a function of the charge inventory and significantly improves modeling accuracy in cases of two-phase refrigerant entering a fixed orifice.

3.3 Two-Phase Refrigerant Entering Compressors

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. The methods of Rice (1981) for correcting compressor map-predicted mass flow rate and power consumption are well accepted, but no investigations have been done to evaluate the compressor map correction methods in the case of two-phase refrigerant entering the compressor. Shen et al. (2006b) showed that a compressor map for mass flow rate works reasonably well with two-phase flow entering conditions, if the suction density is corrected using the two-phase state condition. This approach is more accurate than what is normally done, which is to assume that the suction condition is a saturated vapor. However, using a two-phase suction density correction to predict power consumption leads to significant under-predictions. It is more accurate under all circumstances to not correct the compressor map for power consumptions using a suction density correction as is typically done.

3.4 Refrigerant Mass Flow Mal-Distribution at Low Indoor Air Flow Rates

It is generally recommended that air and refrigerant-side heat transfer coefficients for evaporators be adjusted using a single multiplier determined at one design operating condition with a tuning method, such as the one employed within ACMODEL. The accuracy of a tuned evaporator model can then be accessed by providing measured entrance enthalpy and exit pressure, air-side inlet temperature, humidity, and flow rate, and refrigerant mass flow rate as inputs, and then comparing predicted cooling capacity with measured cooling capacity for those inputs. This type of comparison was performed for all three test units over all operating conditions considered during testing. Varying air flow rate has the largest impact on performance of an evaporator compared to changing other boundary conditions. Therefore, to assess the accuracies of the evaporator models, cases for nominal and off-design air flow rates were studied separately. The evaporator models were validated using refrigerant-side measurements and test data having two-phase entering the compressor or the fixed-area expansion device were not used. Table 3 shows the mean and maximum deviations between the evaporator model predictions and the measured values for the cases of nominal air flow rates. It seems that a single heat transfer tuning coefficient works very well for a large range of operating conditions at nominal air flow rates. The tuning coefficients are also given in Table 3.

Table 3: Assessing heat exchanger models for cases with nominal air flow rates.

Deviations	R-410A, Split		R-410A, Packaged		R-407C, Packaged	
	Mean	Max	Mean	Max	Mean	Max
Evaporator	0.7%	2.3%	0.8%	2.8%	0.4%	1.7%
Tuning Coefficient	0.80		1.09		0.82	

It is very likely that field units operate at a range of different off-design indoor air flow rates due to variations in the air distribution systems (e.g., duct lengths and diffusers). In order to assess the impacts of variations in indoor air flow rate, four series of tests were conducted with the R-410A split system for the conditions specified in Table 4. The first three series employed a TXV and the fourth series utilized a short-tube orfice. For each series, the indoor air flow was varied over a wide range.

Table 4: Test series with changing indoor air flow rate for the R-410A split system.

Series No.	# of Tests	Outdoor T [˚F]	Outdoor Air Flow [CFM]	Indoor T [˚F]	Indoor Air Flow [CFM]	Indoor RH [%]	Charge [%]
Series I (TXV)	7	95	2800	80	600~1200	<30%	100%
Series II (TXV)	8	95	2800	80	500~1200	50%	100%
Series III (TXV)	5	115	2800	80	900~1700	<40%	100%
Series IV(FEO)	7	95	2800	80	700~1400	50%	100%

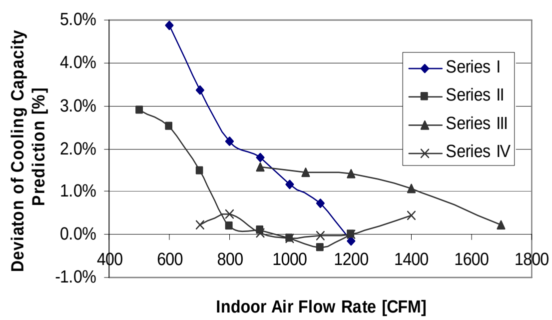

Figure 1: Evaporator model assessments with changing air flow rate for the R-410A split system.

Figure 1 presents deviations between predicted cooling capacity and measured refrigerant-side cooling capacity as a function of the indoor air flow rate for the four test series. The results indicate that the accuracy of the evaporator model predictions for the TXV system degrade at lower indoor air flow rates. It can also be seen from this figure that the variation of the indoor air flow rate does not impact the accuracy of the evaporator model predictions for the short-tube system (Series IV). At the same indoor air flow rate of 700 CFM, the evaporator model within the short-tube system model performs better than the same model within the TXV system model. In addition, the evaporator model within the TXV system model for wet conditions (Series II) is better than the same model for dry conditions (Series I). The difference in humidity is not considered to be the reason for the difference.

The TXV system was found to have a non-uniform refrigerant mass flow rate distribution among the evaporator flow circuits. The refrigerant flow distribution was particularly poor at small indoor air flow rates and dry conditions, since the performance of the distributor ahead of the evaporator is sensitive to mass flow rate. Also, the TXV tends to "hunt" at these low refrigerant flow conditions, which can worsen the refrigerant mass flow mal-distribution. Figure 2 shows measured exit superheat for the five evaporator flow circuits of the R-410A split system for three different tests. Differences in exit superheat are an indication of refrigerant flow mal-distribution. The three test conditions were as follows: 1) TXV, 1200 CFM indoor air flow rate, dry conditions, 2) TXV, 700 CFM, dry conditions, 3) short-tube orifice, 700 CFM, wet conditions. It is apparent from Figure 2 that the mass flow distribution among the evaporating circuits was not uniform for all three tests. The middle circuit (circuit #3) had the largest mass flow rate (i.e., smallest exit superheat). Since the superheat for circuit #3 was the same for all three cases and nearly zero, the exit refrigerant must have been a two-phase mixture. The system with a short-tube orifice had better flow distribution because of higher refrigerant mass flow rates. For the TXV system, the distribution improved with air flow rate. Higher air flow generally leads to increased refrigerant flow.

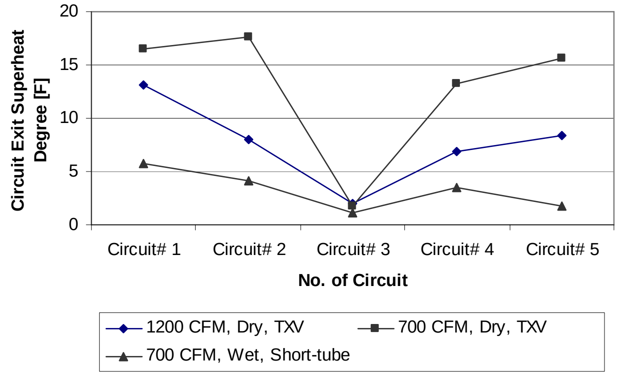

Figure 2: Refrigerant mass flow distribution among evaporating circuits in the R-410A split unit.

The effect of mass flow distribution was investigated using ACMODEL. Since Circuit #3 had the minimum exit superheat degree, it was considered to receive the largest mass flow rate $mr_{\max}$, while the other four circuits were considered to have a uniform mass flow rate $mr_{\min}$. The ratio of the maximum mass flow rate to the minimum mass flow rate (denoted as *R*) is used to characterize the flow mal-distribution as depicted in Figure 3. For different test conditions, the mass flow rate distribution ratio *R* was adjusted within the evaporator model until the model predictions of cooling capacity matched the measured refrigerant-side cooling capacity. Table 5 gives the distribution ratios for different indoor air flow rates for test Series I. The mal-distribution increased dramatically with decreasing air flow rate.

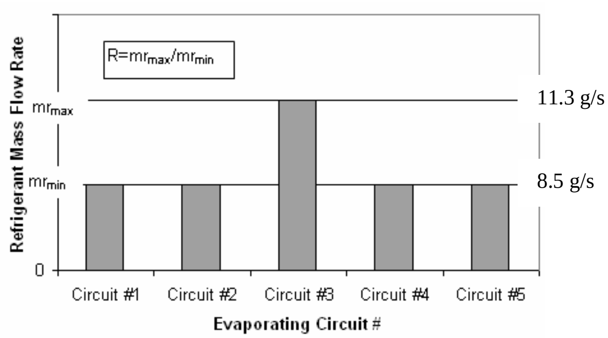

Figure 3: Adjusted refrigerant mass flow distribution in the evaporator of the R-410A split unit.

Table 5: Mass flow rate distribution ratios for test Series I.

Indoor CFM	1200	1100	1000	900	800	700	600
R	1.0	1.13	1.19	1.26	1.33	1.52	1.68

Figure 4 shows predicted circuit exit superheat compared to the measured values for an air flow rate of 800 CFM. The agreement is relatively good. These results indicate that it can be important to include non-uniform mass flow rate distribution when predicting the performance of a TXV system at off-design air flow rates. Figure 5 compares predicted and measured compressor power as a function of the indoor air flow rate. Figure 6 compares the predicted and measured cooling capacity. The predictions were determined using system simulations both with and without the corrections for mal-distribution of refrigerant flow. By considering the refrigerant mal-distribution in the evaporator, the maximum deviation of compressor power was reduced from 4.4% to 1.0%, and the maximum deviation of cooling capacity was reduced from 10% to 0.5%.

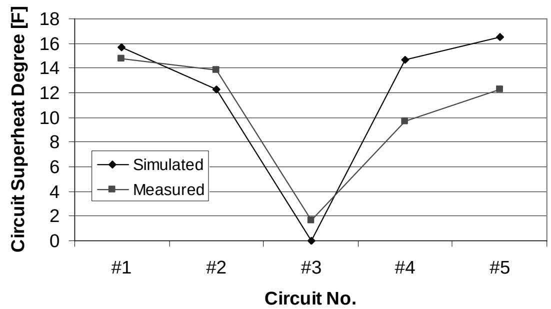

Figure 4: Predicted circuit exit superheat for air flow of 800 CFM with refrigerant flow mal-distribution.

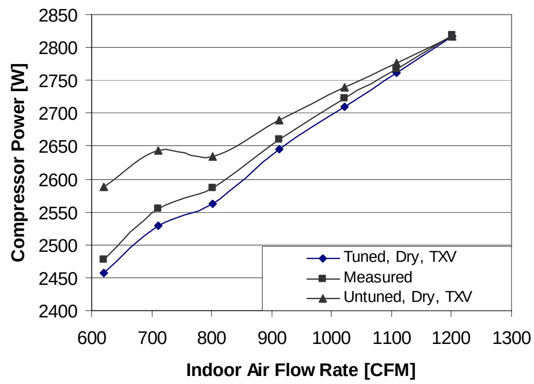

Figure 5: Compressor power predictions for the R-410A split unit using TXV under dry conditions.

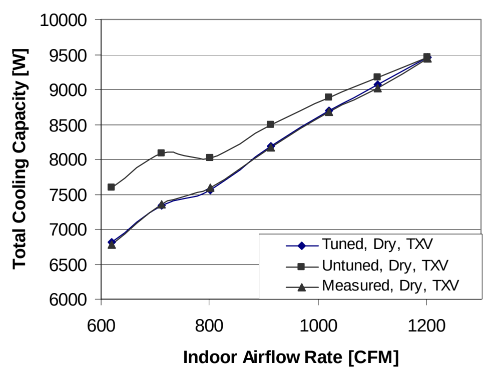

Figure 6: Cooling capacity predictions for the R-410A split unit using TXV under dry conditions.

4. EFFECTS OF IMPROVED METHODOLOGIES

As discussed in Section 3.1, the two-point charge tuning method can improve predicted subcooling degree over a large range of refrigerant charge in comparison to a one-point charge tuning method. This is illustrated with the results of Figures 7, where condenser subcooling is plot as a function of charge mass for the R-407C packaged unit at an outdoor temperature of 82 ˚F and wet conditions for the evaporator. Figure 8 shows errors in system simulation predictions of cooling capacity for the two-point and one-point tuning approaches for a different operating condition than used in determining the tuning coefficients in Equation 1. The two-point charge tuning leads to significantly improved cooling capacity predictions at small charge levels (two-phase entering fixed orifice) compared to the one-point charge tuning. The maximum deviation was reduced from 12.0% to 4.0%.

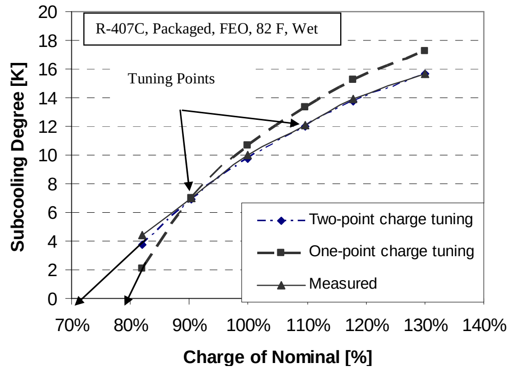

Figure 7: Subcooling degree versus charge mass for the R-407C packaged unit using a short-tube orfice at an outdoor temperature of 82 ˚F and wet condition

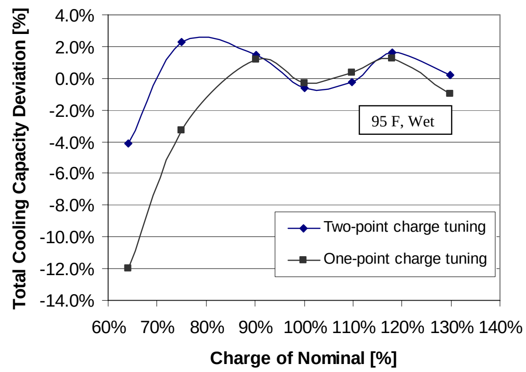

Figure 8: Deviations in predicting cooling capacity as a function of charge mass for the R-407C packaged unit with an outdoor temperature of 95 F and wet condition

Figure 9 presents errors in system simulation predictions of compressor power as a function of charge mass for the R-410A packaged unit at an outdoor temperature of 95 ˚F and wet conditions. Three modeling methodologies were compared. The first used the one-point charge tuning method with the compressor power map correction, the second used the two-point charge tuning method with the compressor power map correction, and the third used the two-point charge tuning method without the compressor power map correction. The two-point charge tuning method with or without the compressor power map correction leads to better simulations at both low and high charge levels, in comparison to the one-point charge tuning method. However, the two-point charge tuning method without the compressor power correction is better than with the correction in the case of high charge levels (two-phase refrigerant entering the compressor.)

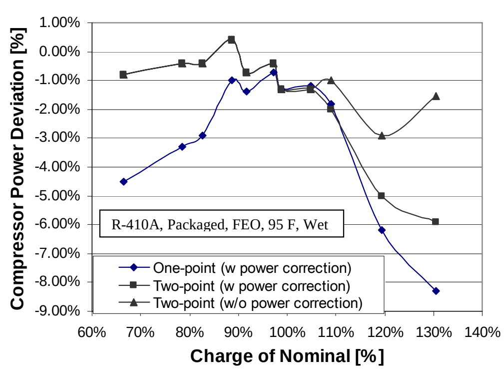

Figure 9: Deviations in predictions of compressor power as a function of charge mass for the R-410A packaged unit at an outdoor temperature of 95 F and wet condition

Overall assessments for accuracy of system simulations are given in Tables 6 for the three units over a large range of off-design indoor air flow rates, charge levels and two-phase refrigerant conditions entering the compressor or fixed orifice. Results with and without implementation of the improved modeling methodologies are compared. The mean, maximum and standard deviations are given in the terms of absolute deviations relative to the measured values. In general, the improvements significantly reduce both the maximum and average deviations.

Table 6: Overall assessments of system simulations with and without improved methodologies

	Cooling Capacity			Compressor Power		
R-410A Split Unit	Mean	STDEV	Max	Mean	STDEV	Max
w/o improved methodologies	4.1%	5.2%	22.2%	2.7%	2.4%	9.4%
w improved methodologies	1.9%	2.9%	13.3%	1.9%	1.5%	6.7%
R-410A Packaged Unit	Mean	STDEV	Max	Mean	STDEV	Max
w/o improved methodologies	6.9%	10.4%	42.0%	1.9%	1.9%	7.7%
w improved methodologies	2.7%	4.6%	22.5%	1.4%	1.2%	5.6%
R-410A Packaged Unit	Mean	STDEV	Max	Mean	STDEV	Max
w/o improved methodologies	5.4%	6.4%	24.3%	2.6%	1.4%	6.8%
w improved methodologies	2.7%	2.9%	9.3%	0.6%	0.5%	2.2%

5. CONCLUSIONS

This paper summarized four improved modeling methodologies for simulating vapor compression equipment that lead to more accurate predictions of cooling capacity and compressor power consumption at conditions that are significantly different than the design operating point. Incorporation of the improved methodologies within ACMODEL led to significant improvements in modeling accuracy for three units that were tested. Detailed descriptions and validations of some of the improvements are presented by Shen et al. (2006a, 2006b). The current paper focused on refrigerant mal-distribution effects and overall assessments. At conditions that lead to low refrigerant flow rate, such as low indoor air flow rates, the refrigerant mass flow distribution within the evaporator can become very non-uniform for systems that employ TXVs. Corrections for mal-distribution effects can lead to improved overall predictions associated with system simulations.

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