Development and Validation of a Room Air Conditioning Simulation Model

C. E. Mullen, B. D. Bridges, K. J. Porter, G. W. Hahn, and C. W. Bullard

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For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

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W. E. Dunn and C. W. Bullard, Principal Investigators

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217 333 3115

Abstract

Component submodels for a room air conditioner computer simulation were developed and validated using data from a 1.5-ton room air conditioner. The room air conditioner was tested over an extremely wide range: both indoor and outdoor temperatures varied from 67 to 115 °F. During the validation, the problems associated with modeling room air conditioners were addressed: nonuniform evaporator and condenser air flow, recirculation and the capillary tube. The evaporator, condenser, capillary tube and compressor power submodels are much more accurate than the compressor mass flow submodel, which is merely a generic curve fit of manufacturers' data.

The entire validation procedure was made easier because the simulation model and solver routine are keep separate by using the Newton-Raphson solution method to solve the 160 simultaneous equations. This separation allows the user to easily "swap" parameters with variables as long as the equation remain independent. The inherent problems associated with the use of the Newton-Raphson routine with highly non-linear and discontinuous equations were overcome by developing automatic-relaxation, variable-checking, and equation-switching methods.

Introduction

This paper describes the process of developing and validating the components of an air conditioner simulation model, through an exhaustive series of experiments conducted over a period of several years using extensively-instrumented room air conditioners. The first objective was to determine experimentally what degree of analytical detail was needed for accurate system simulations. The second was to structure the model in such a way that it could be used efficiently by manufacturers as a design tool.

1.1 Background

The most widely used public-domain simulation models were developed by the US Department of Energy, originally for split systems (Fischer and Rice, 1983) and later modified for room air conditioners (O'Neal and Penson, 1988). These models employ the successive substitution solution method, in which the governing equations are entangled with the solution algorithm. This structure limits their use as design tools, because the input and output variables are predetermined and uniquely associated with the solution algorithm. Therefore our first step was to simply extract the approximately 160 governing equations from the existing model, and solve them using the Newton-Raphson (NR) method (Hahn and Bullard, 1993). The NR method allows the governing equations to be listed in any order (separately from the solution logic), so that they are easier to understand and modify. This method also made it possible to "swap" parameters and variables in the governing equations without changing the solution logic, for example to solve for the condenser or evaporator areas or tube diameters needed to achieve a specified energy efficiency ratio (EER).

Mullen and Bullard (1994) improved the model by changing most of the governing equations, selecting a more general and more robust set of correlations for pressure drop, heat transfer, and void fraction based on experiments conducted with an air-side instrumented 1-ton room air conditioner. Speed enhancements including sparse-matrix Gaussian elimination and sparse-matrix Jacobian calculation were added to the NR solver. Bridges and Bullard (1995) continued the validation, using data from a similarly-instrumented 1.5-ton unit. Most recently, Porter, Jensen and Kirby (Jensen and Dunn, 1996) repeated the experiments for the 1.5-ton unit after Jensen added immersion thermocouples, pressure transducers and venturis to measure refrigerant mass flow. This paper presents the latter data using the modeling methods developed by Mullen and Bridges.

The overall objective was to develop a simulation model that required mainly geometric input parameters plus a few empirical correlations, and to rely only on a few system-specific parameters that, hopefully, could be determined empirically from a few simple measurements taken at a single "standard" rating condition.

While most of the experiments were motivated by the goal of developing a design tool for air conditioning systems in general, this paper focuses on special problems and issues that are to some extent unique to room air conditioners, such as airflow nonuniformities and recirculation, the capillary tube, and nonintrusive instrumentation. Room calorimetry is especially troublesome for low-capacity systems, because of the effect of thermocouple uncertainties over the large surface area of guarded chambers. We avoided these problems by building a unique heavily-insulated environmental chamber that relied on power measurements instead of thermocouple readings to yield very accurate measurements of system capacity (Feller and Dunn 1993, Fleming and Dunn 1993, Rugg and Dunn 1994).

Section 2 provides an overview of the relationship between the model, RACMOD, and the Newton-Raphson solver, which was described in detail by Mullen and Bullard (1994). Section 3 details the experimental test matrix and the system numbering schemes. Experimental validation is described in Section 4 for each component.

Solver-model relationship

The system simulation model now contains over 160 equations which include a three-zone condenser, an adiabatic capillary tube, a two-zone evaporator with dehumidification, and a compressor model based on manufacturer performance data. The "ACRC Solver" is a Newton-Raphson equation solver developed at the University of Illinois Air Conditioning and Refrigeration Center (ACRC) for solving a wide variety of vapor-compression systems. It is specially designed to accommodate, between iterations, changes in the number of governing equations associated with the presence or absence of superheated or subcooled zones in the evaporator and condenser (Mullen and Bullard , 1994). It also performs ASME and Monte Carlo uncertainty analyses and simple sensitivity analyses of the governing equations.

2.1 Structure

The structure and organization of the ACRC room air conditioner model as implemented with the ACRC solver builds upon work described by Porter and Bullard (1992) and is depicted in Figure 1. The separate subroutines for model initialization, checking, and equation-evaluation allow this structure to handle special problems that arise in thermal system simulations. For instance, the boundary checking in RACMOD determines whether the refrigerant at the evaporator exit is currently two-phase or superheated and switches to the modified equation set if the condition has changed since the last iteration. Because the equations are listed separately and in an order-independent fashion, it is relatively easy to modify them or remove a component model and replace it with a new one.

2.2 Swapping parameters and variables

A given variable can become an input parameter if a former parameter simultaneously becomes a variable, as long as the equations remain independent. This allows designers to specify levels of system performance, such as capacity, and solve for the required evaporator size or the length or diameter of the capillary tube. Total system refrigerant charge can be specified, resulting in the calculation of condenser subcooling, or vice-versa.

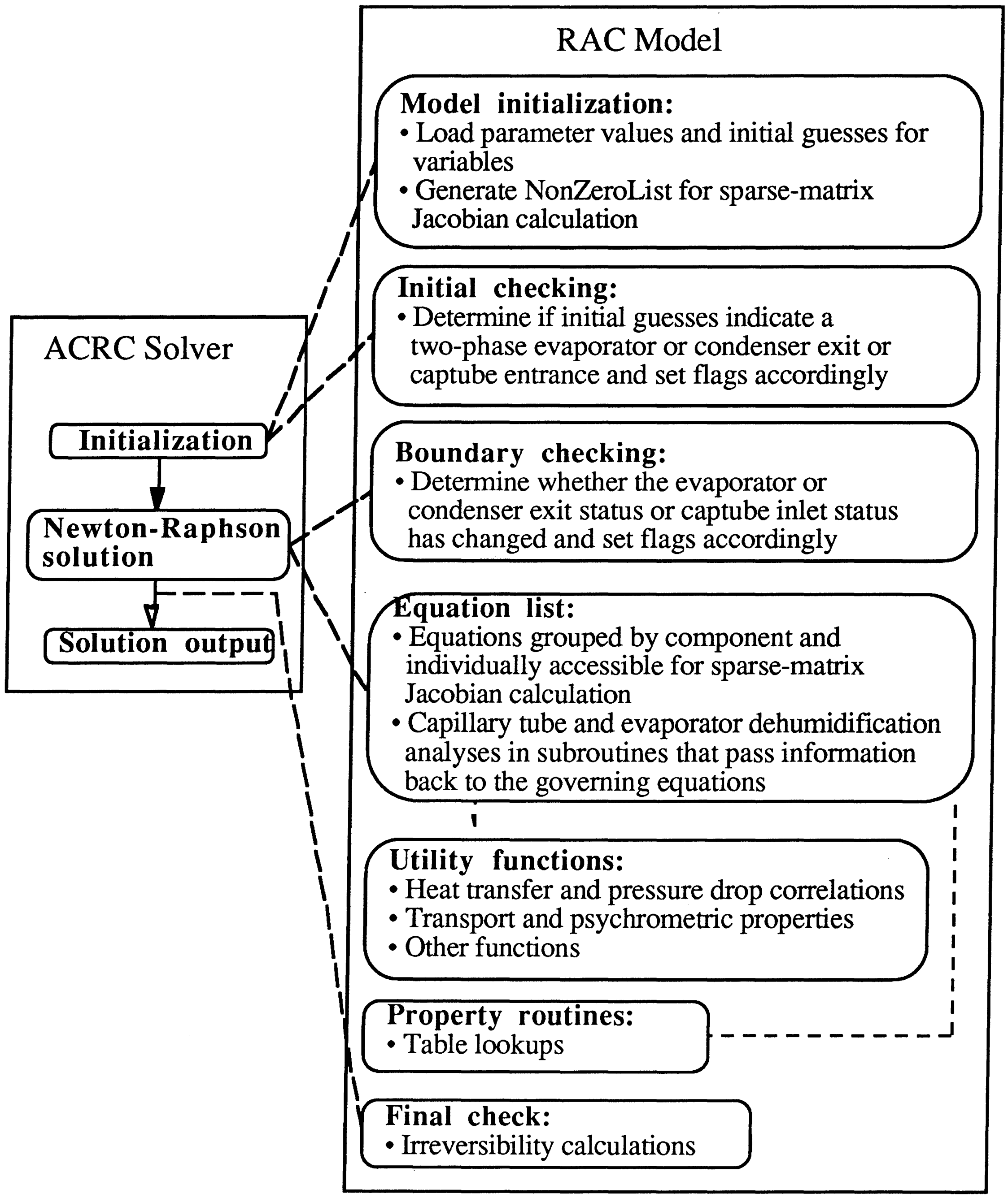

Figure 1. Organization of RACMOD and the ACRC solver

Variable-parameter swapping was also used for parameter estimation, during the validation of this model. For example, the outlet conditions of a heat exchanger may be specified to allow exponents in a heat transfer coefficient to be estimated. Swapping of a parameter and a variable is accomplished easily by changing two flags in the input file. There is no need to change the program or recompile.

2.3 Speed enhancements in the model and solver

Many of the 160 equations contain calls to sometimes lengthy property routines or pressure drop functions. A straightforward evaluation of the Jacobian matrix for the 160 equations would require 160^2 partial derivative calculations and involve considerable execution time. However, most of those equations contain only a few variables, so most of the partial derivatives are always zero. To improve execution time, the solver maps non-zero elements of the Jacobian in advance, to ensure that only the non-zero partial derivatives are evaluated when the Jacobian is calculated. This, together with a sparse-matrix Gaussian elimination routine that uses full pivoting and linked lists (Stoecker, 1989), decreased runtime by a factor of eight.

2.4 Automated step relaxation

Thermal systems are very demanding of the locally-linear Newton-Raphson method, particularly when initial guesses are poor. An iterative step often results in an attempt to evaluate a thermodynamic or transport property outside of its domain or across a discontinuity. Common examples include attempting to calculate a refrigerant quality above the critical temperature or attempting to raise a negative number to a non-integer power (e.g. in a heat transfer or pressure drop correlation). Therefore the algorithm is designed to recognize such occurrences, retrace the step, and reduces the NR step size by half. Details are provided by Mullen and Bullard (1994).

The component models rely on heat transfer and pressure drop correlations. The sources of the correlations are listed in Table 1.

Table 1 Empirical correlations used by RACMOD

	Condenser	Evaporator
Heat transfer		
Subcooled	Dittus and Boelter (1930)	
Two-phase	Dobson *et al* (1994)	Wattelet *et al* (1994)
Superheated	Kays and London (1974)	Dittus and Boelter (1930)
Air-side	Elmahdy and Biggs(1978) Nakayama and Xu (1983)	Xiao and Tao (1990)
Pressure drop - tube		
Single-phase	Colebrook (1939)	same
Two-phase	Souza *et al* (1993)	same
Pressure drop - return bend		
Single-phase	Ito (1960)	same
Two-phase	Christoffersen *et al* (1993)	same

Experimental Validation

To keep the size of the model manageable, several simplifications and assumptions were required: uniform air flow and temperature across the heat exchangers, constant conductances in each zone of the heat exchangers, and treating multiple refrigerant circuits in the evaporator and condenser as identical heat exchangers. Additional assumptions for the individual components are described below.

3.1 Test matrix

Due to certain simplifications and other effects that are not easily modeled, a few empirical parameters were specified as inputs: fan power and airflow rates, air recirculation fractions, condensate spray efficiency, and overall UA's for the compressor shell and connecting lines.

A large test matrix, shown in Figure 2, was employed to validate the simulation model; accuracy at a wide range of off-design conditions is needed for a credible design tool. This test matrix was designed to span the entire range of the compressor performance data within the capabilities of the environmental chambers.

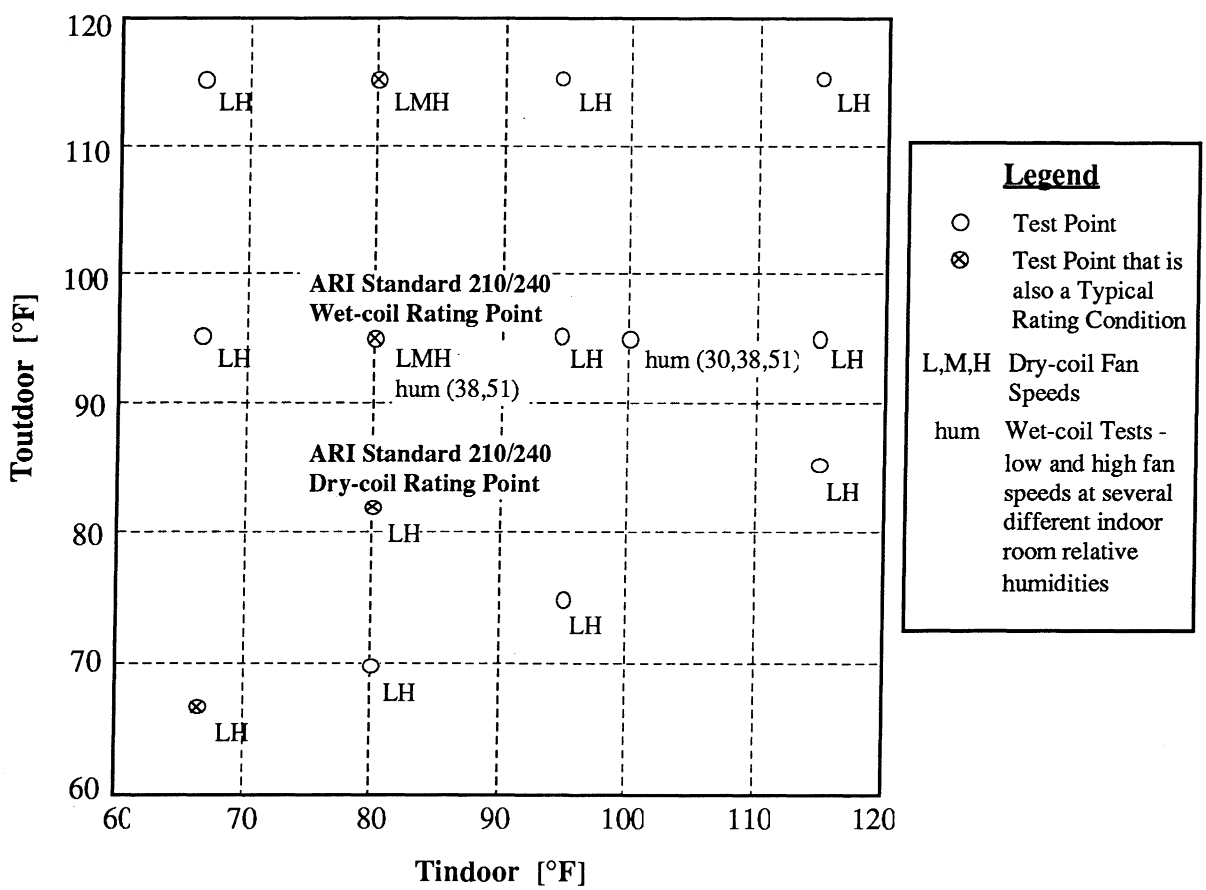

Figure 2. Test matrix

The moist air tests consisted of two fan speeds and either two or three relative humidities. Three dry-coil tests and four wet-coil tests were conducted with an indoor dry bulb temperature of 80 °F and an outdoor dry bulb temperature of 95 °F, while six wet-coil tests were conducted at an indoor temperature of 100 °F and an outdoor temperature of 95 °F.

The test matrix covered a large range of capacities and refrigerant mass flow rates. Evaporator superheat measurements ranged from 45 °F to a two-phase evaporator exit. The measured evaporating and condensing temperatures ranged from 25 to 61 °F and 99 to 156 °F, respectively.

3.2 System description

State points and components are defined in a general form, so the model can simulate split as well as unitary systems simply by specifying different line lengths and other geometric parameters. The system-specific air recirculation patterns encountered in room air conditioners and ductless mini-splits are also shown in Figure 3.

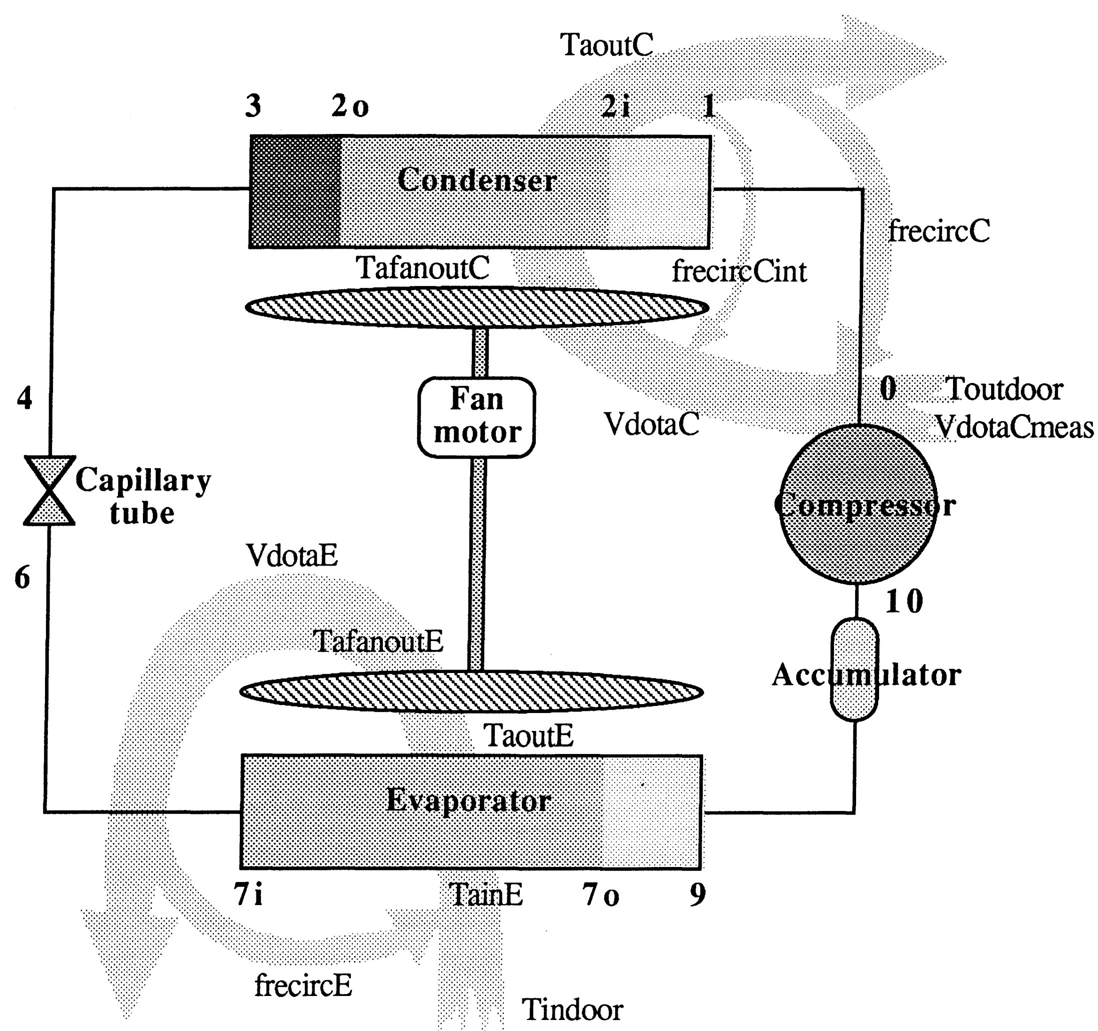

Figure 3. Schematic of room air conditioner model

Component Level Validation

4.1 Evaporator

The evaporator is modeled as a crossflow heat exchanger having uniform inlet air temperature and velocity. Our detailed data sets provided the basis for quantifying the effects of these assumptions, separate from the evaluation of other modeling assumptions. Surface and immersion thermocouples and pressure transducer were installed at the inlet and outlet of each component, and *in situ* compressor calorimetry was accomplished using power transducers and a liquid-line venturi.

4.1.1 Inlet air

A turbine velocity meter was used to measure relative velocities across the evaporator inlet; results are shown in the following contour plot.

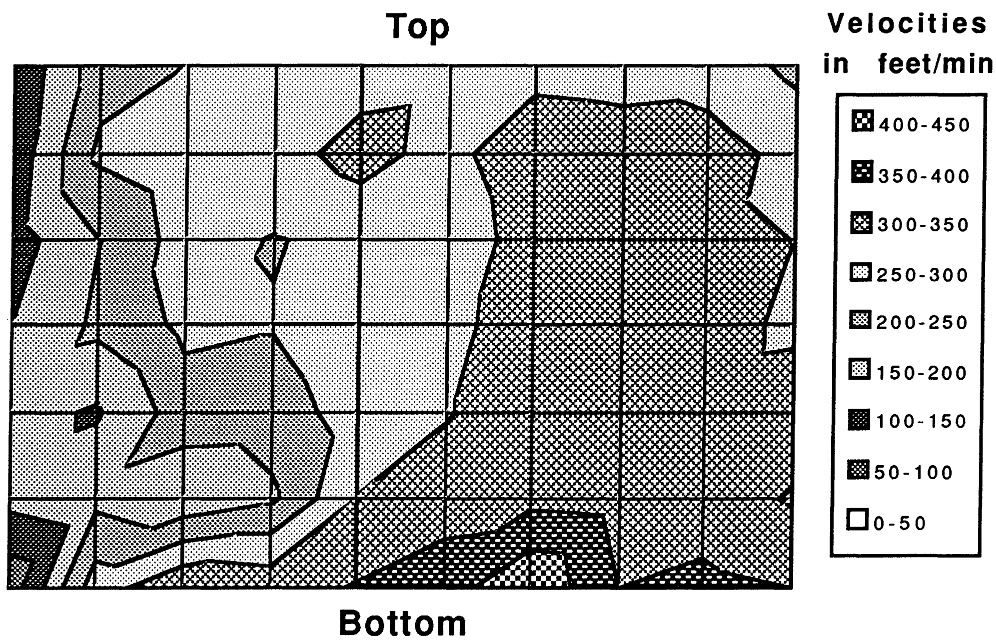

Figure 4. Evaporator inlet velocity profile

Although the squirrel cage blower and uneven edge sealing caused substantial maldistribution, the effect is small since air-side heat transfer depends approximately on Reynolds number to the 0.7 power. If it were linear, heat transfer in areas of high velocity would compensate exactly for the low velocity regions. A more serious concern is the existence of a markedly different velocity in the superheated region where the refrigerant-side heat transfer coefficient is much lower. On average, the superheated region was estimated to occupy about 5%

of the total evaporator area at the top left, where the air velocity was 35% lower. Fortunately even this maldistribution contributes less than a 0.1% error to the total capacity calculation.

4.1.2 Evaporator air recirculation

Since the evaporator exit grille was located directly above the inlet, the average inlet air temperature was decreased about 6°F. Analysis of the initial data set revealed an average recirculation fraction of 0.12, which corresponds to a 5% reduction in EER. A recirculation barrier was therefore installed, reducing the temperature gradient across the evaporator inlet from 20 °F to less than 1 °F for the remainder of the experiments, to focus on validation of heat and mass transfer equations.

4.1.3 Heat transfer and dehumidification

Only two heat transfer zones, two-phase and superheated, are modeled using effectiveness-NTU heat transfer rate equations. Following Fisher and Rice (1983), calculations of refrigerant-side heat transfer coefficients and pressure drop were based on the relatively crude assumption that flow is distributed equally among non-integer numbers of equivalent and parallel circuits. The superheated region is assumed to be dry; while the two-phase region is split into wet and dry subzones in the most general case shown in Figure 5. Each region is modeled by air- and refrigerant-side energy-balance equations, and a heat transfer rate equation.

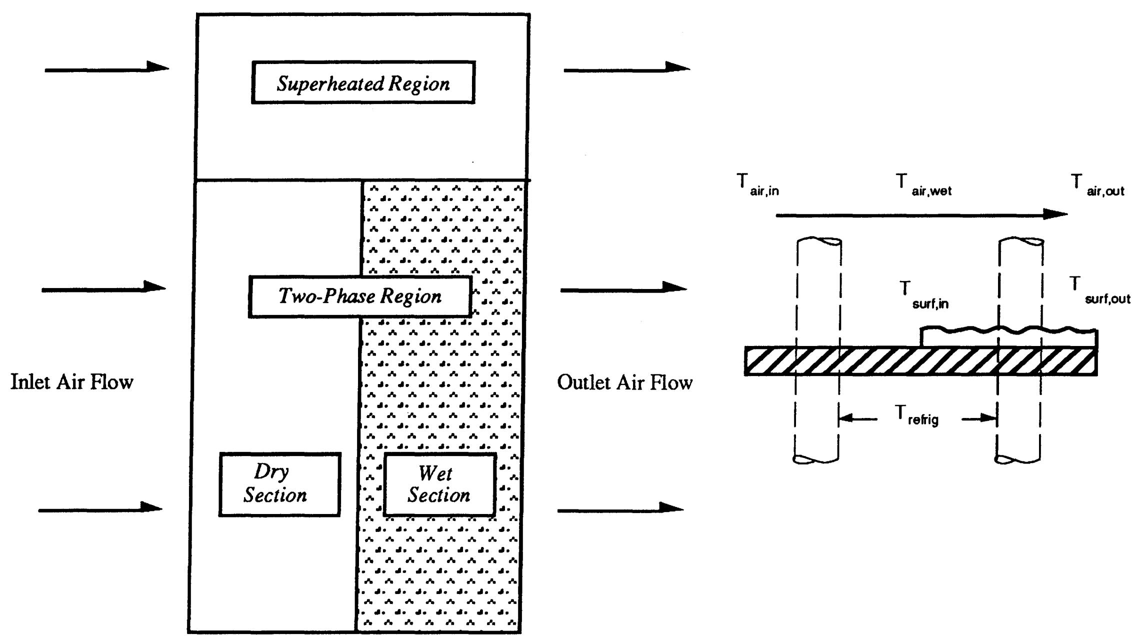

Figure 5. Evaporator submodel geometry

Additional equations for the latent portion include mass transfer, driven by the difference between humidity ratios of the bulk air and wet surface, assuming a Lewis number of unity. The surface temperature at the inlet to the wet region is equal to the dewpoint temperature during partially wet conditions. The water film resistance and its effect on the boundary layer was also assumed to be negligible. These assumptions are only slightly less restrictive than those associated with the enthalpy balance approach and arithmetic mean temperature differences employed by Fisher and Rice (1983), and improved the predictions of evaporator capacity by about 2%.

A fully wet condition can exist if the surface temperature of the leading edge is at or below the dewpoint of the entering air, in which case the equations for the dry two-phase region are ignored by the NR solver in the same manner as for two-phase heat exchanger exits. Similarly, if the surface temperature of the trailing edge of the evaporator is greater than the dewpoint of the entering air, the evaporator is considered to be fully dry.

The evaporative heat transfer correlations developed by Wattelet *et al* (1994) for R-22 and other HCFC's and HFC's were based on smooth-tube data. Using additional microfinned-tube data, Christofferson *et al* (1993) modified the smooth-tube correlation to include the microfinned enhancements for convective and nucleate boiling along with the increased area associated with microfins. These local heat transfer coefficients are integrated over the evaporator quality range by our simulation model.

Various air-side heat transfer correlations were investigated for this model including (Xiao and Tao (1990), Webb (1990) and Elmahdy and Biggs (1983). Xiao and Tao's wavy fin correlation was selected because it was able to accurately predict evaporator capacity using estimated air-side volumetric flow rates only 7% lower than the manufacturer's rated values . The model's ability to predict evaporator superheat is shown in Figure 6.

As shown by Figure 6, the evaporator submodel accurately predicts superheat over an extremely wide range for dry conditions. The superheats shown in Figure 6 represent RMS errors in capacity of 1.3 and 1.7% for dry and wet conditions, respectively. The model tends to over predict superheat and therefore capacity under wet conditions. The possible reduction in the experimental air-side volumetric flow rate during wet conditions could cause this over prediction. Because the bias in only 1.5% of capacity and because we do not have an independent means of verifying this possible reduction in air-side volumetric flow rate, we currently use the same volumetric flow rate to simulate both dry and wet conditions.

Figure 7 compares the evaporator submodel predictions for moisture removal with experimental values accurate to 1%. The RMS error for the moisture removal prediction is 14%.

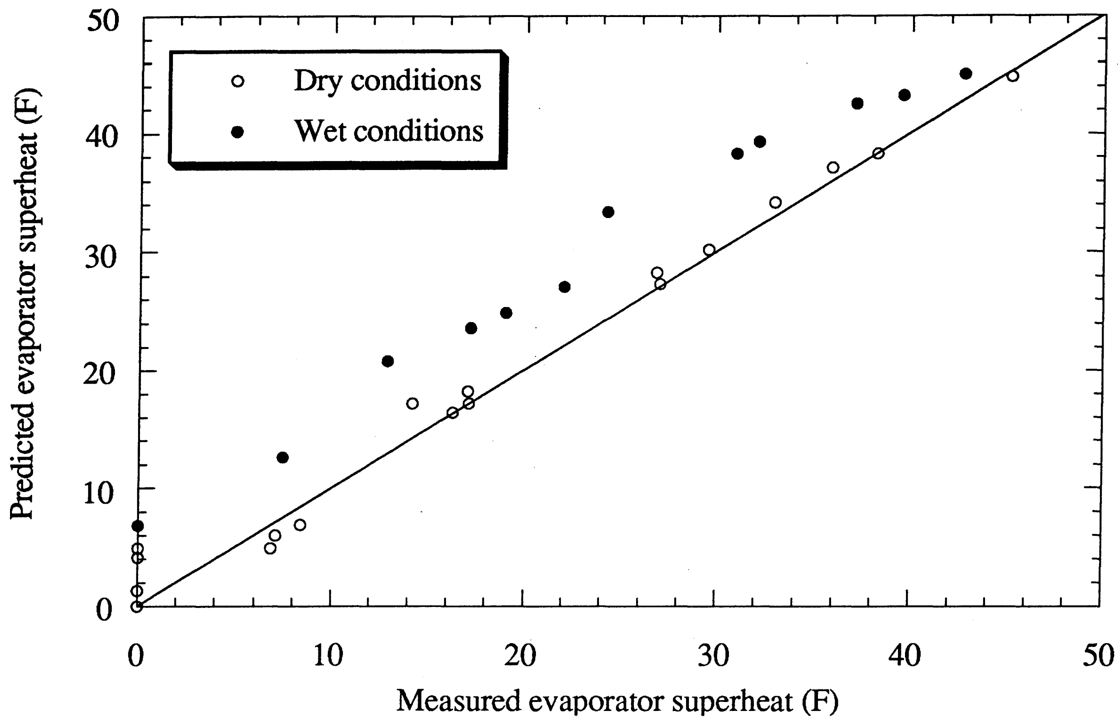

Figure 6. Comparison of predicted and measured evaporator superheat

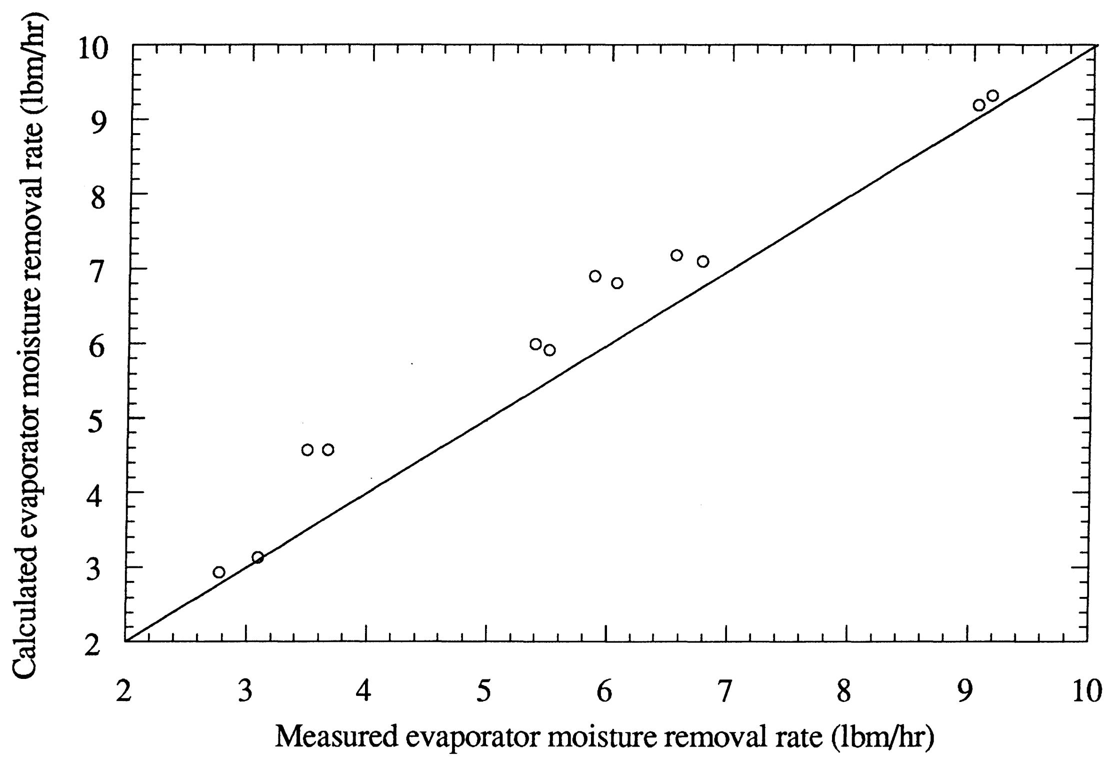

Figure 7. Comparing predicted and measured evaporator moisture removal rates

4.2 Condenser

The condenser is also modeled as a 3-zone crossflow heat exchanger having uniform inlet air temperature and velocity. The air side heat transfer equations were modified to account for heat transfer enhancement due to condensate from the evaporator that is sprayed onto the condenser by the condenser fan's sling ring. The enhancement is based on an effective mass flow rate of water multiplied by the water's heat of vaporization.

4.2.1 Condenser air flow

Due to the proximity of the propeller fan, condenser air inlet velocities are as nonuniform as those in the evaporator, but the effects on heat transfer are similarly small. However the average inlet air temperatures, measured at the louvers in the sides of the cabinet, were found to be 1-2 °F warmer than the outdoor room temperature. This small recirculation fraction (0.04, independent of fan speed) will vary with the size of the test chamber, and must be accounted for when validating other condenser modeling assumptions. Additional recirculation can occur at the condenser exit, if the condenser fins do not span the entire length of the exit grille. Mullen used tufts to observe eddy currents near the tube bends draw hot exit air back into the plenum area upstream of the fan, and estimated the fraction at 0.02, but none was noticeable for the second test unit. Although the magnitude is small, its impact on the validation experiments was potentially important, since such "internal recirculation" passes through the fan twice but escapes detection by the thermocouples at the inlet of the louvers.

4.2.2 Heat transfer and condensate spray enhancement

As in the case of the evaporator, air side heat transfer resistance is dominant, so a large number of correlations were evaluated. Both test units had louvered fins, and in both cases a plain fin correlation (Elmahdy and Biggs, 1978), used with a louver enhancement correlation (Nakayama and Xu, 1983) was found to give the best agreement with the data. The refrigerant side correlation spanned both stratified and annular flow, but the flow was almost always annular. Smooth tube coefficients were modified for 18° helix microfin tubes using a quadratic curve fit of data obtained on similar tubes by Meyer and Dunn (1996) and Schlager (1989), who found that the enhancement factor varied from 100% to 80% of the area ratio, from the low to the high end of the condenser's mass flux range. The area ratio is defined as the ratio of the microfin surface area to the area of a smooth tube having the same maximum diameter. It has been postulated that the enhancement mechanism depends on the microfins protruding through the liquid annulus into the vapor core, to promote condensation (Webb and Yang, 1995). It follows that the enhancement decreases at higher mass fluxes as vapor shear forces dominate the surface tension forces, while liquid trapped near the base of the microfins effectively removes part of the wall surface from the convective heat transfer process.

Besides accounting for air-side geometry considerations, one must also consider condensate spray enhancement for the condenser. Room air conditioners use a sling ring attached to the condenser fan to throw condensate onto the coil, thereby enhancing heat transfer. Water that does not reach the coil either evaporates or blows through the condenser in the form of droplets. In the past, the amount of liquid that blows through the condenser has been considered negligible (Tree and Goldschmidt, 1974). But analyses of our wet coil data, assuming that the sensible heat transfer coefficient for the condenser would be the same as that determined from the dry-coil data, revealed that only 55% of the total water input reached the condenser coil. Rough calculation suggest that less than 10% was likely to have evaporated from the pan, and that adiabatic evaporation of the remaining 45% evaporated would have reduced air temperature only a degree or two, probably undetectable by thermocouples installed upstream and downstream of the sling ring. Therefore it was not possible to determine experimentally much of the condensate's latent heat was lost by droplet evaporation or by spraying onto the hot sheet metal cabinet.

Using the heat transfer correlations described above and the estimated sling ring efficiency of 0.55 for the wet data, Figure 8 compares predicted and measured condenser subcooling. The RMS error for condenser subcooling is 4.8 (F), surprisingly low considering the extremely wide range of operating conditions tested. The scatter associated with Figure 8 corresponds to a RMS error of 2.0% for high-side capacity which includes both discharge-line and condenser heat transfer.

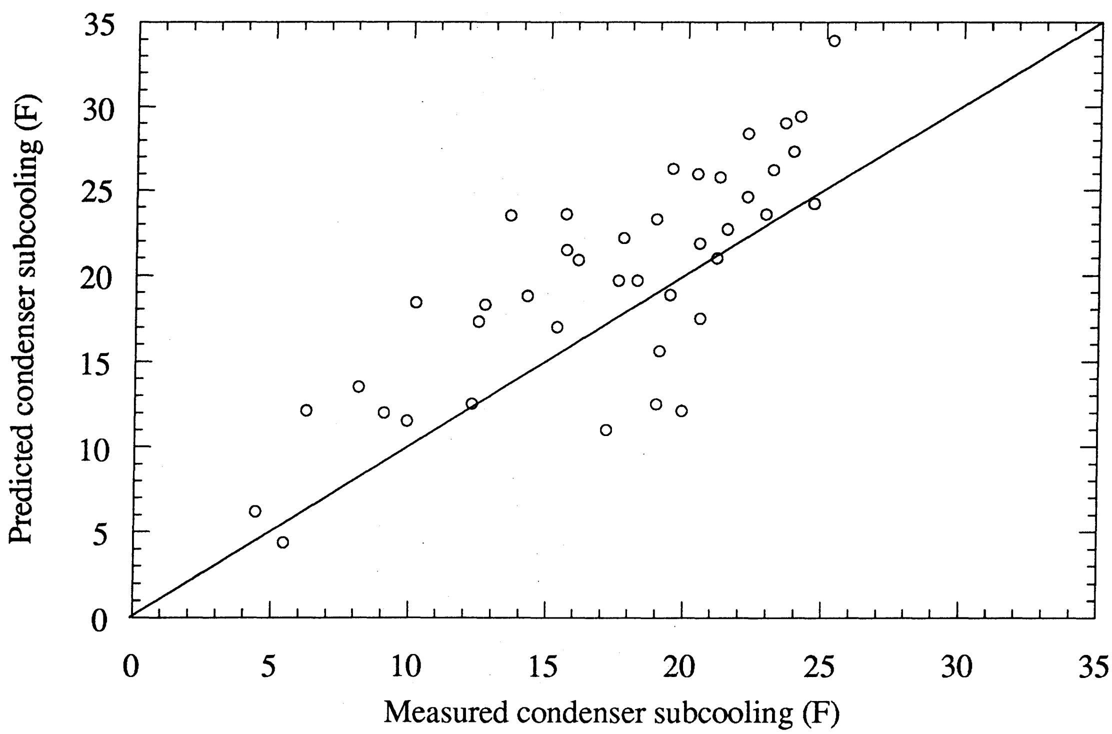

Figure 8. Comparison of predicted and measured subcooling for dry and wet points

4.3 Capillary tube and compressor

In any simulation model, results are highly sensitive to refrigerant flow rate, which is determined by a balance between the compressor and capillary tube. Uncertainties in either component submodel will propagate throughout the system simulation. Since no physical model of the compressor is available, generic manufacturer-supplied curve fits are used. Our tests confirmed that discharge temperature varies linearly with shell temperature, so the model simulates compressor heat rejection and predicts discharge temperature at any operating condition using an overall heat transfer coefficient for the compressor shell and a simple linear curve fit that can be obtained easily from shell and discharge line thermocouple readings at standard rating conditions.

4.3.1 Capillary tube submodel

The finite-difference adiabatic capillary tube model developed by Peixoto and Bullard (1994) has been implemented in system simulation model. Unlike the curve fit of the ASHRAE chart used by Fisher and Rice (1983) and O'Neal and Penson (1988) it uses basic physical equations so alternative refrigerants can be modeled. The Colebrook friction factor correlation is used to calculate the pressure drop in the single-phase portion of the capillary tube. It is modified for the two-phase portion of the capillary tube using Dukler's void-fraction-weighted average of liquid and vapor viscosities, because it is the only method compatible with Reynolds-number scaling of homogeneous two-phase flow (Dukler, 1964). The capillary tube model was placed in a separate subroutine because its more than 200 equations would substantially increase computational time and require too many initial guesses for the Newton-Raphson algorithm. It was necessary to include only five capillary tube equations to be solved simultaneously in the system simulation model, and the five interface variables were sufficient to enable the finite-difference subroutine to be solved sequentially.

The friction factor correlations were selected based on independent measurements of mass flow taken on a capillary tube test stand for a wide range of inlet pressures and refrigerant subcooling (Meyer and Dunn, 1996), using several refrigerants and capillary tubes including one having the same nominal dimensions as the one in the test unit.

4.3.2 Compressor mass flow and power

Curve fits are used in the model to predict refrigerant mass flow and compressor power. These curve fits are empirically-based performance, and obtained from calorimeter measurements for a particular compressor model tested with a suction inlet temperature of 95 °F. Compressor power and refrigerant mass flow are plotted for a range of inlet and outlet saturation temperatures. Manufacturers often claim the data are accurate to within 5%. Polynomial curve fits of the performance data are placed in subroutines in the simulation model.

Due to our wide range of test conditions, the suction inlet temperature varies significantly from the calorimetry value of 95 °F. Therefore in the simulation model, we correct the compressor curve fit for changes in suction-line outlet density experienced under real operating conditions.

4.3.3 Validation

Figure 9 shows the prediction of the finite difference capillary tube model using the Colebrook-Dukler two-phase friction factor correlation. Our reference refrigerant mass flow rate was measured with two venturis which had previously been calibrated *in situ* with a coriolis mass flow meter (Jensen and Dunn, 1996). We used a surface roughness of 30×10^{-6} inches, a value well within Sweedyk's (1981) range of measurements. The capillary tube equations predicted the room air conditioner's mass flow with a RMS error of 3.4% for subcooling inlets between 5 °F to 25 °F and saturation pressures corresponding to condensing temperatures from 98 °F to 155 °F.

Figure 9 also shows the results for our uncorrected and corrected compressor curve fits. The uncorrected curve fit tends to significantly underpredict compressor mass flow rate for low mass flow rates while overpredicting mass flow at high mass flow rates (RMS error = 7.5%). On the other hand, the corrected version tends to overpredict compressor mass flow rate for all mass flow rates (RMS = 7.1%), probably due to heating between the suction inlet and the cylinder. Neither compressor model predicts compressor mass flow rate well as compared to the capillary tube performance. These significant errors will make it difficult to accurately simulate an entire system over a wide range of conditions.

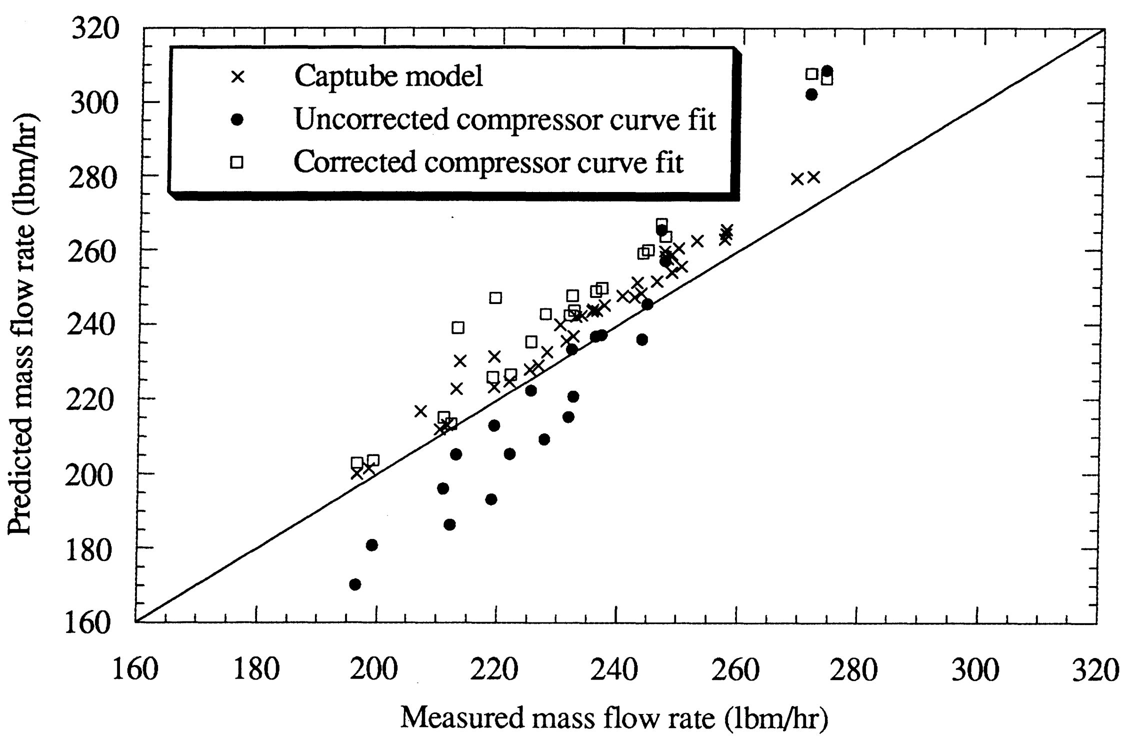

Figure 9. Capillary tube and compressor curve fit mass flow predictions

4.3.4 Compressor power

Figure 10 compares predicted versus measured compressor power for our entire data set. The measured saturation temperatures at the evaporator and condenser were used along with pressure drop correlations to estimate the saturation temperatures at the compressor inlet and outlet. These temperatures were then input into the compressor curve fit. The RMS error for Figure 10 is 3.0%.

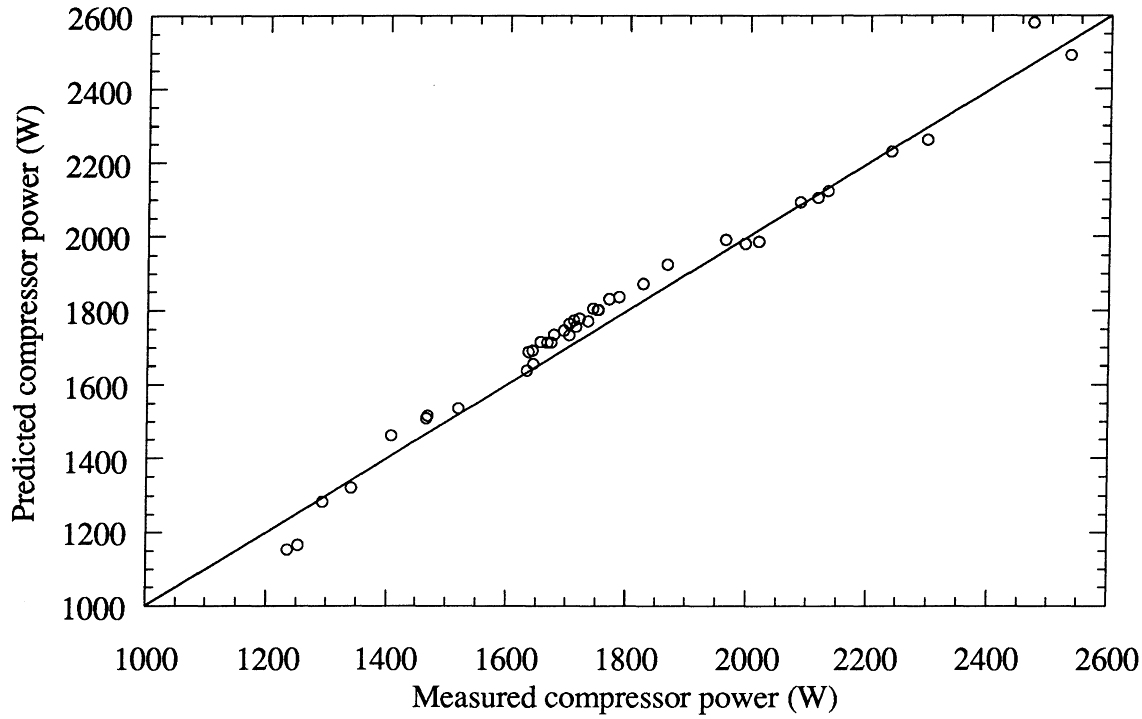

Figure 10. Compressor curve fit power prediction

Conclusion

These experimental results have demonstrated the feasibility of *in* situ component-by-component validation of a room air conditioner simulation model. Specifically, the results have shown that evaporator and condenser models can handle the unique difficulties associated with predicting both evaporator superheat and condenser subcooling for a wide range of dry and wet operating conditions. Empirically determined air-side recirculation factors for both the condenser and the evaporator permit accurate modeling of condenser and evaporator heat transfer despite their highly nonuniform velocity profiles.

The accurate prediction of evaporator exit and suction line superheating is very important because it directly influences the compressor suction inlet density. Even after correcting the generic manufacturer-supplied compressor curve fits for this effect, the remaining uncertainties are quite large. They are even larger than those for the capillary tube submodel, which is based on first principles and physical dimensions. Because refrigerant mass flow rate is such an important variable, almost directly proportional to system capacity, there is little to be gained by refining other component models until the accuracy of the compressor submodel can be improved. It appears to be the largest source of uncertainty affecting accuracy of the overall system simulation.

Separating the simulation model from the solver facilitated the validation process. Without this structure, it would have much more difficult to quantify the effects of parametric uncertainties in each component. The main benefits, however, are the order-independent list of governing equations, which facilitate substitution of component equations, and the ability to select any combination of input and output variables, which increases the model's value as a design tool. After validating the entire model, this structure will allow both designers and researchers to conduct simulates in a variety of different manners in a timely and easy manner.

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