NISTIR 6045

Method for Estimating the Energy Efficiency Ratio of Mixed System Air Conditioners and Heat Pumps

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Technology Administration
National Institute of Standards and Technology

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September 1997

U.S. Department of Commerce
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Prepared for:
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ABSTRACT

In lieu of testing each split system air conditioner and heat pump combination, an empirically based calculation procedure may be used for estimating a unit's Energy Efficiency Ratio at the 35 °C (95 °F) rating condition, EER(95). The procedure accounts for performance changes caused by using different indoor sections with the same condensing unit. The procedure is applicable to all electric units having rated cooling capacities less than 19 kW (65,000 Btu/h) and charged with Refrigerant 22.

To estimate the EER(95) of one or more combinations that use the same condensing unit, a lab-measured EER(95) for one combination must be known. Typically, the known EER(95) will correspond to the indoor section most frequently sold with the given condensing unit–termed the matched system–because the testing of this combination is mandated by federal regulation. In addition to the lab-measured EER(95), select information on the hardware used in the indoor sections of the matched system and the system(s) for which EER(95) is sought, hereafter referred to as the mixed system, must also be known.

Key words: air conditioner, building technology, heat pump, mixed system, rating procedure

ACKNOWLEDGEMENT

This report was prepared for the U.S. Department of Energy, Office of Building Technologies, with Mr. William Noel serving as the project manager. The authors thank Mr. John Bean and Mr. Jonathan Moldover for their help in verifying the rating algorithm with test data.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

FOREWORD

Residential air conditioners belong to the category of appliances for which performance data, according to regulations, must be made available to potential customers. The required data appear on the appliance label and consist of the system capacity at 35 °C (95 °F) rating condition, Q(95), and the Seasonal Energy Efficiency Ratio, SEER. The procedures to obtain these ratings via laboratory tests are described in Part 430, Title 10 of the Code of Federal Regulations[1].

The federal regulations do not require Q(95) and SEER to be obtained by testing for all systems; only the highest-sales-volume combination of a given condensing unit and indoor coil must be tested. Such a system is referred to as a matched system. All other systems involving a given condensing unit and other indoor sections–referred to as mixed systems–can be rated according to an engineering methodology approved by the U.S. Department of Energy (DOE). Several system and coil-only manufacturers have developed their own proprietary method for rating mixed systems, while others use a public-domain rating method developed by NIST[2,3]. The latter has been in use since 1986.

The currently required performance descriptors, Q(95) and SEER, are not adequate for a meaningful rebate program that aims at reducing peak demand for electricity. For a single-speed system, SEER is based on the system's steady-state and cyclic performance the 27.8 °C (82 °F) rating condition. With SEER having a limited relevance to system efficiency at higher temperatures, the Consortium for Energy Efficiency (CEE) included the Energy Efficiency Ratio at the 35 °C (95 °F) rating condition, EER(95), in their new rebate program[4]. For a matched system, EER(95) is known to the manufacturer since the system's capacity and power input are measured during the DOE-required test at 35.0 °C (95.0 °F). However, EER(95) for mixed systems is typically not known, and a methodology for determining it had to be developed.

This report describes the methodology for determining the Energy Efficiency Ratio at the DOE 35 °C (95 °F) rating condition for mixed systems. The main body of the report presents the rating procedure while the appendix describes the system simulations and assumptions that led to the rating algorithm. The proposed algorithm was formulated by applying the same approach that had been employed to formulate the NIST mixed system rating procedure for Q(95) and SEER. This report is compatible with the existing NIST mixed system rating procedure[3]; the same format of presentation is used with some of the sections describing the procedure restrictions being copied from the original document with little or no modification.

1. PURPOSE

The purpose of this report is to establish a procedure for estimating the Energy Efficiency Ratio at the DOE 35 °C (95 °F) test condition, EER(95), for air-source unitary air conditioners and heat pumps consisting of a condensing unit and an indoor section that were not tested together as a system. The procedure relies on independent measurements and calculations made on an outdoor unit in conjunction with a matched indoor coil and a mixed indoor coil. The Energy Efficiency Ratio is the ratio of a system's net cooling capacity and the total electrical power supplied to the unit. The procedure for determining these quantities directly from a single laboratory test is specified in the DOE Test Procedures contained in the Code of Federal Regulations[1].

2. SCOPE

This procedure applies to residential, all-electric, single-phase, air-conditioning and heat pump systems charged with Refrigerant 22, consisting of an indoor air-cooling coil assembly and an outdoor air-source unit whose matched system* rated cooling capacity** is less than 19,000 W (65,000 Btu/h). This procedure does not apply to systems employing multi-speed compressors or systems in which compressor control strategy changes with space cooling load (e.g., cylinder unloads, hot-gas bypass). Additional limitations regarding the applicability of the procedure are given in Sections 5.2.2 and 5.3.2.

3. SYSTEM OF UNITS

In light of transition by the Federal Government to the SI system of units and to comply with pertinent NIST regulations, this document uses SI units as the primary units and British units as secondary units (given in parentheses). Because of differences in these two systems, the calculated value of EER(95) is non-dimensional in SI units but has the dimensions of Btu/(h·W) in the British units. Consequently, values of EER(95) in these two systems of units will be different by a factor of 3.413. The results obtained using the British system of units are compatible with the current definition of EER contained in the DOE Test Procedure[1].

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* Refer to Section 4, Definitions

** As defined by DOE Test Procedure[1]

4. DEFINITIONS

All definitions included in or cited by Title 10, Part 430 of the Code of Federal Regulations[1] shall be considered part of this procedure in addition to the following definitions.

4.1 ARI - Air Conditioning and Refrigeration Institute.

4.2 Air Source Unitary Air Conditioner or Air Source Unitary Heat Pump or Unitary System - an outdoor unit combined with an indoor coil assembly.

4.3 Outdoor Unit - an assembly of refrigerating components designed to compress and evaporate (when space heating) or condense (when space cooling) refrigerant. It consists of a refrigerant vapor compressor, air-cooled refrigerant coil, coil fan and motor, and regularly furnished accessories. A liquid line solenoid valve and other cyclic performance-enhancing devices (excluding the expansion device), if included in an air-conditioner or heat pump, are considered to be a part of the outdoor unit.

4.4 Indoor Coil Assembly - an assembly consisting of a coil, condensate collecting pan, and expansion device, and which may or may not include a blower, motor, and cabinet.

4.5 Matched Coil - an indoor coil assembly that is a part of the matched system.

4.6 Matched System - a unitary system that has been tested and rated in accordance with Appendix M to Subpart B of Title 10, Part 430 of the Code of Federal Regulations[1].

4.7 Mixed Coil - an indoor coil assembly that is used in a unitary system instead of the matched coil.

4.8 Mixed System - a unitary system that is not a matched system.

4.9 Shall - where 'shall' or 'shall not' are used to indicate mandatory provisions if compliance with the procedure is claimed.

4.10 Should, Recommended, or It Is Recommended - 'should', 'recommended', or 'it is recommended' are used to indicate provisions that are not mandatory but which are desirable as good practice.

5. PROCEDURE FOR CALCULATING MIXED SYSTEM $EER_x(95)$

5.1 Rating Correlations

Mixed system Energy Efficiency Ratio at the DOE 35 °C (95 °F) rating condition [$EER_x(95)$] shall be calculated using equations 5.1, 5.2, and 5.3.

$$EER_x(95) = EER_m(95) \, \frac{Q_x(95)}{Q_m(95)} \left[\frac{P_x(95)}{P_m(95)} \right]^{-1} \tag{5.1}$$

$$\frac{Q_x(95)}{Q_m(95)} = \left[1 + \frac{C1 \cdot P_{F,m}}{Q_m(95)}\right] \cdot F_c^{0.37} \cdot F_{ex}^{\alpha} - \frac{C1 \cdot P_{F,x}}{Q_m(95)} \qquad (5.2)$$

$$\frac{P_x(95)}{P_m(95)} = 0.8\ F_c^{\delta}\ F_{ex}^{\beta} + 0.1\ \frac{P_{F,x}}{P_{F,m}} + 0.1 \qquad (5.3)$$

where:

C1=1.0 for SI units, C1=3.413 Btu/(h·W) for British units

$\alpha = 0.0$ and $\beta = -0.15$ for $F_{ex} < 1.0$

$\alpha = -0.15$ and $\beta = 0.0$ for $F_{ex} \geq 1.0$

$\delta = 0.17$ for mixed systems employing a capillary tube or a short tube restrictor

$\delta = 0.15$ for mixed systems employing a TXV.

Other symbols used in equation 5.1, 5.2, and 5.3 are explained below.

$EER_m(95)$= Energy Efficiency Ratio of the matched system at the DOE 35 °C (95 °F) rating condition, dimensionless, (Btu/(h·W))

$EER_x(95)$ = Energy Efficiency Ratio of the mixed system at the DOE 35 °C (95 °F) rating condition, calculated using equation 5.1, dimensionless, (Btu/(h·W))

F_c = indoor coil scaling factor calculated as explained in section 5.2.1, dimensionless

F_{ex} = expansion device scaling factor calculated as explained in section 5.3.1, dimensionless

$P_{F,m}$ = power input to the indoor fan of a matched system as defined in section 5.4.1, W (W)

$P_{F,x}$ = power input to the indoor fan of the mixed system as defined in section 5.4.2, W (W)

$\frac{P_x(95)}{P_m(95)}$ = ratio of power inputs of the mixed and matched systems at the DOE 35 °C (95 °F) rating condition

$Q_m(95)$ = capacity of the matched system at the DOE 35 °C (95 °F) rating condition, as certified by the matched system manufacturer, W (Btu/h)

$\frac{Q_x(95)}{Q_m(95)}$ = ratio of capacities of the mixed and matched system at the DOE 35 °C (95 °F) rating point

5.2 Indoor Coil Scaling Factor

5.2.1 Determination of the Indoor Coil Scaling Factor

The indoor coil scaling factor, F_c, is defined by the following equation:

$$F_c = \frac{Q_{c,x}}{Q_{c,m}} \tag{5.4}$$

where: $Q_{c,x}$ = cooling capacity of a mixed coil when subjected to the air volumetric flow rate of standard air specified for the mixed system, W (Btu/h). The air volumetric flow rate specified for the mixed system shall not exceed the maximum levels specified in the DOE Test Procedure[1]

$Q_{c,m}$ = cooling capacity of a matched coil when subjected to the same the indoor volumetric flow rate of standard air, V_m, [m^3/s (ft^3/min)], as at which matched system capacity, $Q_m(95)$, was measured. If V_m information is not available, the value for the indoor air volumetric flow rate shall be calculated as follows:

$$CFM_m = Q_m(95) \cdot C2 \qquad m^3/s \ \left(\frac{ft^3}{min}\right) \tag{5.5}$$

where: $C2 = 5.703 \cdot 10^{-5}$ m^3/(s·W) for SI system of units ($5.703 \cdot 10^{-5}$ m^3/s per watt of cooling capacity)

$C2 = 425/12000 = 3.542 \cdot 10^{-2}$ ft^3·h/(Btu·min) for British units (425 cubic feet per minute per ton of cooling capacity).

Cooling capacities for the matched ($Q_{c,m}$) and mixed coils ($Q_{c,x}$) shall be obtained using the same verified method (see Section 5.2.3). Coil capacities shall be obtained at the following conditions:

- inlet air conditions: 26.7 °C (80.0 °F) dry bulb and 19.4 °C (67.0 °F) wet bulb
- refrigerant saturation temperature at the evaporator outlet: 7.2 °F (45.0 °F)
- identical refrigerant superheat at the evaporator outlet.

If coil capacities are obtained by means of a catalog or computer simulation, the same catalog or computer simulation shall be used for both coils. Coil material and geometry (e.g., inside tube diameter, tube staggering, fin spacing, fin thickness, fin shape, number of tube rows, inside and outside surface design) shall be accounted for by the method used. That is, the methodology used must have these parameters as independent variables.

5.2.2 Restrictions

The acceptable range of values for the indoor coil scaling factor, F_c, is from 0.8 to 1.2. This rating procedure shall not be used if the indoor coil scaling factor is smaller than 0.8. If the ratio $Q_{c,x}/Q_{c,m}$ results in a value greater than 1.2, the value of the indoor coil scaling factor, F_c, shall be 1.2. If the rating is calculated for a heat pump system able to operate in the heating mode, the following additional conditions shall be satisfied:

a) The internal volume of the mixed indoor coil assembly
   - shall not be smaller than the volume of the smallest indoor coil assembly certified with a given outdoor unit by the outdoor unit manufacturer,
   - shall not exceed the internal volume of the matched coil assembly by more than 20 percent, or shall not exceed the volume of the largest coil certified with a given condensing unit by the condensing unit manufacturer, whichever is larger.

b) The heating capacity of the mixed coil
   - shall not be less than the capacity of the lowest capacity indoor coil certified with a given outdoor unit by the outdoor unit manufacturer, and shall not be less than 85 percent of the matched indoor coil,
   - shall not exceed the capacity of the matched coil assembly by more than 20 percent, or shall not exceed the capacity of the highest capacity coil certified with a given condensing unit by the condensing unit manufacturer, whichever is larger.

   Heating capacities of the matched and mixed coil shall be obtained using the same verified method. The capacities shall be obtained at the following conditions:

   - inlet air dry-bulb temperature: 21.1 °C (70.0 °F)
   - refrigerant saturation temperature at the coil inlet: 43.3 °C (110.0 °F)
   - identical refrigerant subcooling at the coil outlet.

5.2.3 Verification of the Method Used for Determining Coil Capacity

A variety of methodologies exist for calculating the capacity of a coil based on material and geometry data only. For example, several large heat exchanger manufacturing companies published performance curves for their specific products which were commonly used in a generic fashion. That is, the capacities of coils of the same materials and of the same number of rows, tube patterns and diameters, fin spacing, shape, and thickness were assumed to be the same for all manufacturers. More common now is to use a computer simulations which may be based either on regression analysis of laboratory data or on first principles of the heat-transfer phenomena involved. If a specific methodology of either of these categories has at least the independent variables listed in the last paragraph of Section 5.2.1, then the method may be used in this procedure to predict the capacities of the coils (i.e., $Q_{c,x}$ and $Q_{c,m}$). However, the specific methodology chosen must be verified by test, in accordance with ASHRAE Standard 33-88[5] and ARI Standard 410-91[6], to demonstrate that it is sufficiently accurate to simulate the coil line that is being used in the mixed system rating.

Verification requires that, for a given coil line, the capacity range over which the methodology is applied must be straddled by at least two tests which are within 5% agreement with the predicted values. For example, if a manufacturer produces a line of coils with six sizes having capacities ranging from 6000 W (20500 Btu/h) to 10000 W (34000 Btu/h) and uses a single methodology (e.g., computer simulation) to predict the six individual capacity values, the methodology must be within 5% agreement with the test values of the smallest (6000 W) and the largest (10000 W) coils. A similar pair of straddling tests is also required for the line that includes the matched coil if the methodology has not been previously verified for that line.

A coil line is defined as a group of coils which are of the same materials and bonding procedure, configuration (i.e., flat or A-shape), row staggering, fin thickness, fin spacing, fin shape (i.e., flat, wavy, corrugated edge, etc.), tube diameters and internal surface finish. If any of these parameters differ, then a new coil line has been defined and a new verification test pair is required for the methodology.

It is recognized that refrigerant circuitry is not specified as mandatory input for coil capacity predictions; yet it may have a significant effect on performance, all other parameters being held constant. However, the state of the art of coil performance simulation or representation is such that refrigerant circuitry is seldom considered. Some first principle computer simulation programs, such as the one used to develop this standard, do account for these design parameters but these programs are not widely available and are usually so complex that input errors are easily possible. Therefore, to require this degree of sophistication for a coil rating method is deemed unreasonable at this time.

5.3 Expansion Device Scaling Factor

5.3.1 Determination of the Expansion Device Scaling Factor

The expansion device scaling factor, F_{ex}, depends on the type of expansion device used in the matched and mixed systems. F_{ex} shall be determined using Table 1, which provides a value for the scaling factor or refers to the equation by which the scaling factor shall be calculated.

Table 1. Evaluation of the Expansion Device Scaling Factor

Expansion Device		
Matched System	Mixed System	F_{ex}
TXV	TXV*	1.0
Capillary or Short Tube Restrictor	TXV**	1.0
Capillary or Short Tube Restrictor	Capillary or Short Tube Restrictor	equation 5.6

*the mixed TXV shall have equivalent capacity and the same superheat setting as the matched TXV

**the mixed TXV shall have equivalent capacity as the matched expansion device

5.3.2 Restrictions

This rating procedure shall not be used if:

a) The expansion device scaling factor, F_{ex}, is outside the range 0.95-1.35, for air conditioners.
b) The expension device scaling factor, F_{ex}, is outside the range 1.0-1.25, for heat pumps.
c) The mixed system has a combination of capillary tubes or short tube restrictors connected in series.
d) The matched system has a combination of capillary tubes or short tube restrictors connected in series.
e) The matched system has a TXV and the mixed system has either a capillary tube or a short tube restrictor, unless the condensing unit manufacturer also certifies a system in which the matched system TXV is replaced by a capillary tube or short tube restrictor. In such a case, this non-TXV system may be considered as a matched system, and its performance data may be used for estimating the performance of the mixed system.

5.3.3 Equations for Calculating the Expansion Device Scaling Factor for Capillary Tubes and Short Tube Restrictors

The expansion device scaling factor, F_{ex}, is the ratio of summations of refrigerant mass flow rates through the mixed [$m_{x,i}$] and matched [$m_{m,j}$] expansion devices, connected in parallel, at the same operating conditions.

$$F_{ex} = \frac{\Sigma\, m_{x,i}}{\Sigma\, m_{m,j}} = \frac{m_{x,1} + m_{x,2} + \ldots m_{x,i}}{m_{m,1} + m_{m,2} + \ldots m_{m,i}} \tag{5.6}$$

Subscripts x and m refer to mixed and matched expansion devices, and subscripts i and j correspond to the number of parallel connected capillary tubes or short tube restrictors in the mixed and matched systems, respectively. The operating conditions selected for calculation of the expansion device scaling factor are: pressure of 1724 kPa (250.0 psia) and 7.2 °C (13.0 °F) subcooling at the expansion device inlet, and saturation temperature of 7.2 °C (45.0 °F) at the evaporator outlet.

Evaluation of the mass flow rate, m, depends on the type of flow restrictor. For a capillary tube the following equation shall be used:

$$m = C3 \cdot \Phi \qquad \text{kg/s (lb/h)} \tag{5.7}$$

where: $C3 = 1.381 \cdot 10^{-2}$ kg/s for SI units $\quad C3 = 109.6$ lb/h for British units

Φ = flow factor for the capillary tube employed, determined from its geometry with the aid of the ASHRAE Handbook, Refrigeration Volume, 1994, Chapter 44, Figure 43[7].

Refrigerant mass flow rate through a short tube restrictor shall be calculated by the following equations[8]:

$$m = C4 \cdot C_c \cdot D^2 (250 - P_2)^{0.5} \qquad \text{kg/s (lb/h)} \tag{5.8}$$

$$C_c = 1 + 0.104 (C5 \cdot DEPTH)^{0.64} (L/D)^{0.27} \tag{5.9}$$

$$P_2 = 209.92 (1.061 - 0.123\ e^{-0.017 (L/D)^2}) \tag{5.10}$$

where: $C4 = 3.116 \cdot 10^{-3}$ for SI units, $C4 = 15955$ for British units

$C5 = 3.937 \cdot 10^{-2}$ for SI units, $C5 = 1.0$ for British units

D = inner diameter of the short tube restrictor, mm (inch)

L = length of the short tube restrictor, mm (inch)

DEPTH = depth of the inlet chamfer, mm (inch).

The length dimensions required in the prescribed equation shall be measured by methods providing accuracy of $\pm$ 1.5%.

5.4 Power Input to the Indoor Fan

5.4.1 Power Input to the Indoor Fan of the Matched System

Power input to the indoor fan, $P_{F,m}$, shall be measured in accordance with the DOE Test Procedure[1] at the indoor volumetric flow rate of standard air, V_m, at which capacity of the matched system, $Q_m(95)$, was measured. If $P_{F,m}$ is not available, it shall be evaluated by equation 5.11.

$$P_{F,m} = C6 \cdot V_m \qquad \text{W (W)} \tag{5.11}$$

where: V_m = volumetric flow of standard air through the matched indoor coil at which system capacity, $Q_m(95)$ was measured, m^3/s (ft^3/min). If the V_m value i s not available, it shall be calculated by equation 5.5.

$C6 = 773.4$ for SI units, $C6 = 0.365$ for British units.

5.4.2 Power Input to the Indoor Fan of the Mixed System

Power input to the indoor fan, $P_{F,x}$, shall be measured in accordance with the DOE Test Procedure[1] at the indoor volumetric flow rate of standard air, V_x, at which capacity of the mixed system, $Q_x(95)$, is evaluated. If the indoor fan is not supplied with the mixed system indoor unit, $P_{F,x}$ shall be evaluated by the equation:

$$P_{F,x} = C6 \cdot V_x \qquad \text{W (W)} \tag{5.12}$$

where: V_x = volumetric flow of standard air through the mixed indoor coil at which the EER(95) of the mixed system is to be evaluated, m^3/s (ft^3/min).

5.5 Published EER(95) Rating

The Energy Efficiency Ratio of the mixed system shall be expressed in multiples of 0.05. The Energy Efficiency Ratio shall not exceed the $EER_x(95)$ value calculated by equation (5.1).

6. ALTERNATIVE RATING PROCEDURE FOR MIXED SYSTEMS

The large number of variables and the complexities of their interactions associated with an air conditioner always make theoretical or quasi-empirical rating procedures less certain than a whole system test. Therefore, an acceptable alternative to this entire methodology is a formal certification program in which performance of most the mixed systems, all created using indoor coils from a single coil line, are measured.

This rating procedure was developed based on simulations of equipment considered to have characteristics of a "typical" system. Other rating procedures may produce ratings of comparable or better accuracy especially if they were developed for specific production lines and/or utilize more input data on the matched and mixed components, if such data are available. Also some alteration of this rating procedure may be warranted if supported by test data.

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8. APPENDIX

8.1 Development of the rating algorithm

8.1.1 Methodology

Figure A1 shows the main components of an air conditioner based on the vapor compression cycle. Substitution of any of these components will alter the performance of the system if the performance characteristics of the replaced component are different. Quite often the whole indoor section is changed, which means that the new (mixed) system has a new indoor coil, indoor fan, and expansion device. The resulting change in system performance is usually within 10% because the main performance-determining component, the compressor, is the same in the matched and mixed systems. This small change in performance suggests the possibility of determining the mixed system's Energy Efficiency Ratio (EER) by using the matched system's EER as a reference and adjusting it based on the differences between the mixed and matched indoor sections.

The method described here adjusts the matched system's EER to the mixed system's EER by considering the impact of each mixed component individually. This approach carries the implicit assumption that no interactions exist when two or more components are replaced. For this methodology, it is essential to understand how different components affect system performance and what configurations constitute a change of a system component. For example, introducing an indoor coil whose capacity is different from the capacity of the coil supplied with the matched system constitutes a "change of the indoor coil". However, this change can be accomplished either by physical changing the coil or by changing the indoor air flow rate for the existing coil, since capacity changes with air flow rate. By the same principle, physical substitution of the indoor fan will not constitute, in this analysis, a change of the fan if power inputs to the mixed and matched fan are the same. For the purpose of this methodology, the indoor coil is characterized by its capacity at the air flow rate provided by the indoor fan, the indoor fan is characterized by its power input at this air flow rate, and the expansion device is characterized by the refrigerant flow rate when operated at specified conditions.

Considering that a difference in performance between a typical mixed system and matched system is of the same order of magnitude as the expanded uncertainty of a laboratory test (10% versus 3% to 5%), a heat pump simulation model was used to quantify the impact of individual system components on the system's performance. Although a simulation model may not provide results as accurate on the absolute basis as laboratory tests, it is more reliable for evaluating relative performance trends. This study used HPSIM[9], a first-principles-based steady-state heat pump model, which had been employed in the development of the original NIST mixed rating procedure[2]. The model simulates a hermetic, reciprocating compressor. The condenser and evaporator are simulated using a tube-by-tube approach in which each tube of the heat exchanger is simulated separately in a sequence according to the refrigerant path through the coil. HPSIM also performs refrigerant mass inventory on the tube-by-tube basis. This is an important feature because it allows to iterate vapor superheat or quality at the compressor inlet at off-design operating conditions for systems employing a capillary tube, and to simulate condenser subcooling for systems maintaining a constant superheat with a TXV. A capillary tube is modeled in HPSIM using the Fanno flow theory.

8.1.2 Impact of a Mixed Evaporator

The evaporator (indoor coil) is the component that is most often replaced in the matched system. The effect of the evaporator substitution on the system performance depends on refrigerant superheat at the evaporator outlet. For a system employing a constant-superheat expansion device (TXV or electronic valve), vapor superheat at the evaporator exit will not vary significantly. If a constant-flow-area expansion device (capillary tube, short tube restrictor) is used, the superheat will vary with operating conditions. The level of superheat will depend on the charging and optimizing criteria used for the matched system. Thus, mixed system performance was evaluated given different scenarios as to how a matched system employing a capillary tube could be optimized. (For this study, flow characteristics of a capillary tube and a short tube restrictor can be considered as similar).

Before the Seasonal Energy Efficiency Ratio, SEER, was mandated as a performance descriptor in 1979, manufacturers optimized the system capacity and EER at what became the DOE 35 °C (95 °F) rating point (Test A). For a system with a TXV, maximum performance can be obtained by selecting a proper refrigerant charge. If a system employs a capillary tube, a proper combination of refrigerant charge and restrictiveness of the expansion device will produce a desired superheat at the evaporator outlet and maximum capacity and EER.

With the 1979 introduction of the requirement to include Q(95) and SEER on the performance label, matched system manufacturers changed their optimization strategies in response to the significance of the Seasonal Energy Efficiency Ratio (this is particularly true for a system employing a constant-flow-area expansion device in which superheat varies with operating conditions and affects the performance significantly). For single-speed systems, SEER is based on EER(82) obtained at the DOE 27.8 °C (82 °F) rating point (Test B) and on the cyclic degradation coefficient. The characteristics of a vapor-compression machine are such that the capillary tube can be selected to maximize EER(82) or Q(95), but not both. Since EER(82) has the dominant influence on SEER, matched system manufacturers have to compromise between maximum SEER and maximum Q(95). In practice, they may opt for the maximum SEER or maximum Q(95), or may settle for both lower values depending on the SEER and Q(95) threshold levels that they want to attain.

Besides different optimization practices, several manufactures changed their charging procedures with the introduction of SEER. Instead of charging at 35.0 °C (95.0 °F), refrigerant is often charged at 27.6 °C (82.0 °F). This different charging practice also affects the evaporator superheat during Test A and the system response to a change of the indoor section components. The information regarding charging and optimizing criteria applied to a particular system is usually considered proprietary and is not available to the rater of the mixed system. The simulation results presented in this section assess the level of uncertainty caused by the lack of this information.

To evaluate the performance of a given outdoor section with different indoor section components, a residential split system was coded for input to HPSIM, and the matched (original) evaporator or expansion device was replaced by a mixed component, one at a time. For evaluating the indoor coil impact, four mixed systems were derived from the matched system by introducing four different mixed evaporators. Simulations of these five systems served to assess the evaporator impact on the system performance.

The study considered eight cases covering two types of expansion devices and four different charging/optimizing schemes, as listed in Table A1. First simulations were performed for the matched systems for each case in Table A1 to determine the optimum size of the capillary tube or optimum subcooling at the condenser outlet for a TXV-equipped system. For Test A conditions, maximum Q(95) was sought during the optimization process while maximizing EER(82) was the goal of the optimization at Test B conditions. The optimization process required several trial-and-error attempts. In all eight cases, refrigerant superheat at the compressor inlet was set to 5.6 °C (10.0 °F) at the test conditions (A or B) at which a given system was charged. Then simulations at the other test condition followed (with exception of Case 1 and 5 where it was not necessary). The optimum size of the capillary tube or optimum condenser subcooling for the matched system was used in simulations of the mixed systems.

The resulting parameters of interest were Q(95) and P(95) as they are used to determine the system EER(95). The simulation results were fitted by the following correlations using the indoor coil scaling factor, F_c:

$$\frac{Q_{x,g}(95)}{Q_{m,g}(95)} = F_c^{\gamma} \tag{A1}$$

$$\frac{P_{x,comp}(95)}{P_{m,comp}(95)} = F_c^{\delta} \tag{A2}$$

where: $Q_{x,g}(95)$ = mixed system gross cooling capacity (indoor fan heat not included)
$Q_{m,g}(95)$ = matched system gross cooling capacity (indoor fan heat not included)
$P_{x,comp}(95)$ = compressor power when operating in the mixed system
$P_{m,comp}(95)$ = compressor power when operating in the matched system

Table A1. Exponents obtained for equations A1 and A2

Case	Expansion Device	Charging/ Optimizing	γ	δ	$\gamma - \delta$
1	cap. tube	Test A/Test A	0.36	0.17	0.19
2	cap. tube	Test A/Test B	0.37	0.15	0.22
3	cap. tube	Test B/Test A	0.36	0.18	0.18
4	cap. tube	Test B/Test B	0.37	0.16	0.21
5	TXV	Test A/Test A	0.35	0.14	0.21
6	TXV	Test A/Test B	0.36	0.14	0.22
7	TXV	Test B/Test A	0.35	0.14	0.21
8	TXV	Test B/Test B	0.36	0.14	0.22

The indoor coil scaling factor, F_c, is a ratio of a given mixed coil capacity to the capacity of the matched coil:

$$F_c = \frac{Q_{x,coil}}{Q_{m,coil}} \tag{A3}$$

where $Q_{x,coil}$ and $Q_{m,coil}$ are gross capacities (fan heat excluded) of a mixed and matched coil, respectively, when operated as follows: with a 7.2 °C (45.0 °F) refrigerant saturation temperature, a 5.6 °C (10.0 °F) superheat at the coil outlet, a 20% inlet quality, and the recommended air mass flow rate for each coil. To obtain these coil capacities, one matched and four mixed coils were simulated by the evaporator simulating subroutine taken from HPSIM. The indoor coil scaling factors for the four mixed evaporators were 0.89, 1.12, 1.21, and 1.29.

The unknown exponents for equations A1 and A2 were evaluated by the least squares fitting of the simulation results. Table A1 shows the exponents obtained for each charge/optimization scenario. The results for systems with capillary tubes are less consistent than the results obtained for systems using a TXV. Since the charging/optimizing scheme used by the matched equipment manufacturer is not known to the rater of the mixed equipment, single values for γ and δ had to be selected for the rating procedure. For exponent γ the value 0.37 was selected for both expansion devices. The small difference in the simulation results does not justify assigning different values for this exponent. To compensate for the resulting favorable bias for a TXV, the value for δ for TXV-equipped systems was selected 0.15 over using 0.14. The δ value for systems equipped with a capillary tube was selected to be 0.17. Since the difference between γ and δ ultimately affect the EER prediction, the impact of these small manipulations in the proposed exponent values can be shown to be small.

Figure A2 depicts the dependency of EER(95) on the indoor coil scaling factor for two systems equipped with capillary tubes. In this figure, the five points denoted by a circle are Case 3 (see Table A1) simulation results. The solid line represents the least-squares curve fit to these data points. The dotted line is a least-squares fit to the data for Case 2. The spread between these lines pictures the impact of different charging and optimizing schemes on the system performance. The selected values of γ=0.37 and δ=0.15 will place the predictions of EER(95) in the middle. Thus, the inherent uncertainty of predicting the EER change due to replacement of the indoor coil corresponds to half of the spread between the lines.

8.1.3 Impact of a Mixed Constant Flow Area Expansion Device

The effect of an inadequate restriction only has to be discussed for constant-flow-area expansion devices (capillary tubes and short tuber restrictors) because a properly-sized thermostatic expansion valve will open sufficiently in a mixed system to allow appropriate refrigerant mass flow. A convenient measure of the difference in performance between the mixed and matched fixed flow area expansion devices is the expansion device scaling factor, F_{ex}:

$$F_{ex} = \frac{m_x}{m_m} \tag{A4}$$

where m_x and m_m are refrigerant mass flow rates through the mixed and matched expansion devices when operated at the same refrigerant state (pressure and subcooling) at the inlet and the same evaporator pressure. The matched expansion device, by definition, has F_{ex} equal to 1. Also, the value of F_{ex} for a properly sized TXV may be assessed to be equal to 1. The expansion device scaling factor is a useful concept for evaluating the sensitivity of the system to different size expansion devices. Gross capacities and powers of the mixed and matched systems may be calculated using the following exponential correlations:

$$\frac{Q_{x,g}(95)}{Q_{m,g}(95)} = F_{ex}^{\alpha} \tag{A5}$$

$$\frac{P_{x,comp}(95)}{P_{m,comp}(95)} = F_{ex}^{\beta} \tag{A6}$$

Some simulation results reported in reference 10 are applicable to this study and are presented here. To determine the effect of a constant-flow-area expansion device on the system performance, four heat pumps were simulated with varied diameter capillary tubes. The simulations considered systems charged at Test A conditions and optimized for EER(82). Figure A3 shows the capacities obtained for one of the systems. The results are typical of the results obtained for all four systems. As is shown by the figure, a given combination of the compressor, condenser, and evaporator will reach its maximum capacity at outdoor temperature of 35.0 °C (95.0 °F) and 27.8 °C (82.0 °F) at different size expansion devices. The designer is thus confronted with a choice of optimizing the system for maximum Q(95), or maximum Q(82), which corresponds to the maximum EER(82) and SEER[10]. In practice, the selected expansion device should fall in the range bracketed by the expansion devices providing maximum Q(82) and Q(95). If the expansion device is beyond this range, both SEER and Q(95) would be penalized.

Usually, the information about optimizing criteria for the expansion device is not publicly available. Because of this lack of sizing information and different system sensitivities for $F_{ex} > 1$ and $F_{ex} < 1$, individual consideration was given to the undersized and oversized expansion device. Also, because of the sizing uncertainties, the current rating procedure[3] limits the allowable range of the cooling expansion device scaling factor from 0.95 to 1.35.

If the matched system is optimized for Q(95), then Q(95) will decrease for a both undersized and oversized expansion device, with a stronger penalty for a less restrictive expansion device. If the matched system is optimized for Q(82), Q(95) will increase at the beginning and then will decrease with decreasing values of F_{ex}, while $F_{ex}>1$ will cause a gradual Q(95) degradation. Considering this, the mixed system rating procedure assumes that the undersized expansion device does not affect Q(95) for the narrow range of F_{ex} (0.95-1.0) allowed by the procedure. For the under-restrictive expansion device ($F_{ex}>1.0$), the exponent $\alpha = -0.15$ was assumed based on the least squares analysis of the simulation data.

Since simulation results obtained in reference 10 were performed for systems charged at Test A conditions, additional simulations were performed in this study to examine the system sensitivity to the restrictiveness of the expansion device for charging performed at Test B conditions. Figure A4 shows the difference in Q(95) for a system charged during Test B and Test A. The capacity trends are similar. For $F_{ex} > 1$, the regression analysis yielded α=-0.10 and α=-0.12 for the Test B-charged and Test A-charged system, respectively. For this procedure the authors chose to assume α=-0.15 since it is a conservative value and will maintain the compatibility with the existing rating procedure[3].

Estimating the mixed system's power at Test A conditions was not in the scope of the previous publications[2,3,10], so a full range of simulations was performed during this study. Figure A5 presents results obtained for two mixed systems, one equipped with an indoor coil of F_c =0.89 and the other with a coil of F_c=1.21. Both systems were simulated as Test A and Test B-charged systems, which resulted in four data lines presented in Figure A5. For the over-restrictive expansion device, the data fall very close to each other and can be correlated by equation A6 with exponent β = -0.15. A larger difference in results (up to 2%) was obtained for an under-restrictive expansion device. For a given $F_{ex} > 1$, this difference in power depends on the indoor coil scaling factor, on the test condition at which the system was charged, and most likely on the size of the matched coil in relation to the compressor size. Since the last two factors are unknown, the authors decided to accept the conservative solution of giving no credit for the decrease in compressor power when the expansion device is oversized. A conservative estimate for EER(95) results because the procedure accounts for the decrease in Q(95) caused by an oversized expansion device.

8.1.4. Impact of a Mixed Indoor Fan

The impact of the indoor fan can be easily taken into account because the fan is not a part of the thermodynamic cycle and does not affect the compressor power. Consequently, a fan's contribution to the system power can be accounted for in the rating algorithm by an additive term. The indoor fan changes the system capacity by adding heat into the air stream. The procedure assumes that 100% of the fan power is converted into heat.

8.1.5. Derivation of the Rating Correlation

The correlations presented in the previous sections of this appendix calculate compressor power and gross capacity of the evaporator for a mixed system. These correlations can be combined to provide an algorithm for estimating a mixed system EER(95).

The assumption is made that there is no significant interaction between the indoor coil and the expansion device with regard to their influence on the performance of the mixed system. Consequently, the mixed coil gross capacity in a system can be represented by combining equations A1 and A5 in the following form:

$$Q_{x,g}(95) = Q_{m,g}(95)\ F_c^{\gamma}\ F_{ex}^{\alpha} \qquad \text{(A7)}$$

By including the heat added by the mixed and matched system indoor and outdoor fans, equation A7 can be modified to calculate capacity of the mixed system:

$$Q_x(95) = (Q_m(95) + C1 \cdot P_{F,m})\ F_c^{\gamma}\ F_{ex}^{\alpha} - C1 \cdot P_{F,x} \qquad \text{(A8)}$$

where: C1=1.0 for SI units, C1= 3.413 Btu/(h·W) for British units. Then the ratio of capacity for the mixed and matched systems can be calculated by the following equation:

$$\frac{Q_x(95)}{Q_m(95)} = (1 + \frac{C1 \cdot P_{F,m}}{Q_m(95)})\ F_c^{\gamma}\ F_{ex}^{\alpha} - \frac{C1 \cdot P_{F,x}}{Q_m(95)} \qquad \text{(A9)}$$

The ratio $P_x(95)/P_m(95)$ can be derived using equations A2 and A6 and taking into account the total (indoor and outdoor) fan powers of the matched and mixed systems, $P_{Fans,m}$ and $P_{Fans,x}$, respectively.

$$\frac{P_x(95)}{P_m(95)} = (1 - \frac{P_{Fans,m}}{P_m(95)})\, F_c^{\delta}\, F_{ex}^{\beta} + \frac{P_{Fans,x}}{P_m(95)} \tag{A10}$$

The authors assumed that the matched indoor fan power and the outdoor fan power are each equal to 10% of the total power of the matched system. With this assumption, the equation for $P_x(95)/P_m(95)$ takes the following form:

$$\frac{P_x(95)}{P_m(95)} = 0.8\, F_c^{\delta}\, F_{ex}^{\beta} + 0.1\frac{P_{F,x}}{P_{F,m}} + 0.1 \tag{A11}$$

Finally, the mixed system $EER_x(95)$ can be calculated using the following equation:

$$EER_x(95) = EER_m(95)\, \frac{Q_x(95)}{Q_m(95)} \left[\frac{P_x(95)}{P_m(95)}\right]^{-1} \tag{A12}$$

The error resulting from the assumption that there is no interaction between the indoor coil and the expansion device in their influence on the mixed system performance can be now revisited. This assumption was made implicitly when the indoor coil and the expansion device were treated as independent variables in deriving the rating correlations. Figure A6 presents a map with EER(95) for a system charged at Test A and optimized for EER(82) (for this sizing criterion, the maximum EER(95) for F_c=1 corresponds to F_{ex}<1). As can be seen from the figure, if the indoor coil was increased by 20% (F_c=1.2), the current expansion device would have the optimum size for EER(95), while the optimum size for the matched coil (F_c=1.0) would be an expansion device scaling factor F_{ex}=0.93. The difference in EER(95) caused by changing the coil size with the same expansion device would be approximately 0.5 percent. Thus, in the case of the oversized indoor coil, the procedure will be conservative. The opposite will be true for the undersized coil (F_c<1.0). Neglecting this inaccuracy in the rating correlation seems to be justified considering the uncertainty related to charging/optimizing of the matched system and other assumptions of the procedure.

8.2 Verification of the Rating Algorithm

The Air-Conditioning and Refrigeration Institute (ARI) made available to the authors a set performance test results for 19 matched and mixed systems for validation of the rating procedure. This set of data was derived from the ARI data bank which contains results from tests performed under the ARI voluntary certification program. Since system selection for testing does not attempt to specify matched/mixed system pairs, the pool of 19 pairs used for verification of the procedure was assembled by identifying a matched system in the data bank and then finding test data of another system using the same outdoor section and an indoor section made by a different manufacturer. All 19 pairs were tested by the same testing company, and standardized test reports were used as the primary source of information. Table A2 contains a basic description of the systems considered. The tested systems included two tube diameters, smooth and rifled tubes, and a total of 11 types of fins (4 lanced, 6 rippled, and straight). Seven condensing unit manufacturers and five coil manufacturers are represented in the data pool.

Obtaining accurate information on the coils tested (coil and tube dimensions, fin parameters and enhancements) constituted a significant difficulty in this project. The authors took this information from test reports where it is provided as an additional system's identification. Occasional erroneous entries in coil data created significant problems (although these errors are of no consequence for performance reporting purposes within the certification program). These problems usually surfaced in an unrealistic EER(95) predictions for some mixed systems. In cases of questionable data or results, accurate information was obtained directly from the manufacturers.

Proper estimating of the matched and mixed coil capacities has the most significant impact on the calculated EER(95). In this study, the authors used a commercially available simulation model[11] for coil capacity prediction. The model accounted for several inside tube surfaces and close to 200 hundred fin types. To select proper fin options for different products, the authors relied on recommendations provided by the developer[(12)] of the evaporator simulation model. The 25.4 mm x 22.2 mm (1.0 inch x 0.875 inch) center-to-center tube pattern (not covered by the model) was simulated by using the 25.4 mm x 22.0 mm (1.0 inch x 0.866 inch) pattern option.

Figures A7, A8 and Table A3 present validation results performed for two different upper limits of an indoor coil scaling factor, F_c, 1.3 and 1.2. In the first case, if F_c was greater than 1.3, the value of 1.3 was assigned. In the second case, the value of 1.2 was used for F_c exceeding 1.2. In the case of 1.3 limiting value, Figure A7 shows that the prediction error exceeds 5% for 8 out of 19 systems rated. Four out of these eight systems include one coil manufacturer and the same fin design. All but one prediction are within 7% of the tested ERR(95). The largest deviation is 9.1%.

Table A2. Description of matched and mixed coils and expansion devices

	Case	Number of Slabs	Depth Rows	Tube Diamater		Tube Center to Center Distance		Tube Inner Surface	Fin Type	Fin spacing (center to center)		Expansion Device
				mm	inch	mm	inch			mm	inch	
Matched	1	3	1	7.9	5/16	25.4	1.0	Riffled	Rippled(a)	1.3	0.053	TXV
Mixed	1	1	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(1)	2.0	0.077	TXV
Matched	2	2	2	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Lanced(2)	1.8	0.071	TXV
Mixed	2	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(1)	2.8	0.111	TXV
Matched	3	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Rippled(b)	1.4	0.056	Piston
Mixed	3	2	2	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(1)	1.7	0.067	Piston
Matched	4	2	3	9.5	3/8	25.4 x 19.1	1.0 x 0.75	Riffled	Rippled(d)	2.1	0.083	Piston
Mixed	4	2	1	9.5	3/8	25.4	1.0	Riffled	Lanced(3)	1.2	0.045	TXV
Matched	5	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Rippled(c)	2.0	0.077	Piston
Mixed	5	1	2	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(3)	2.0	0.077	TXV
Matched	6	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Rippled(c)	1.7	0.067	TXV
Mixed	6	1	4	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	1.8	0.071	TXV
Matched	7	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Straight	1.8	0.071	Piston
Mixed	7	1	4	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	2.0	0.077	TXV
Matched	8	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Rippled(b)	1.8	0.071	Piston
Mixed	8	1	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	2.0	0.077	Piston
Matched	9	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Rippled(e)	1.7	0.067	TXV
Mixed	9	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	1.6	0.063	TXV
Matched	10	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Lanced(1)	1.8	0.071	Piston
Mixed	10	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	1.6	0.063	TXV
Matched	11	2	4	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Smooth	Rippled(e)	2.0	0.077	TXV
Mixed	11	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(4)	1.5	0.059	TXV
Matched	12	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Rippled(c)	2.0	0.077	Piston
Mixed	12	2	2	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(4)	1.8	0.071	Piston
Matched	13	1	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Straight	1.7	0.067	Piston
Mixed	13	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(1)	2.1	0.083	Piston
Matched	14	1	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Lanced(1)	1.7	0.067	TXV
Mixed	14	1	4	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	1.8	0.071	TXV
Matched	15	1	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Straight	1.8	0.071	Piston
Mixed	15	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Rippled(c)	1.6	0.063	TXV
Matched	16	2	3	7.9	5/16	25.4 x 15.9	1.0 x 0.625	Riffled	Rippled(f)	1.6	0.063	Piston
Mixed	16	2	2	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(3)	1.4	0.056	Piston
Matched	17	2	3	7.9	5/16	25.4 x 15.9	1.0 x 0.625	Riffled	Rippled(f)	1.8	0.071	Piston
Mixed	17	2	2	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(3)	1.4	0.056	Piston
Matched	18	2	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Smooth	Rippled(c)	1.8	0.071	TXV
Mixed	18	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(3)	1.8	0.071	TXV
Matched	19	1	3	9.5	3/8	25.4 x 22.0	1.0 x 0.866	Riffled	Straight	1.8	0.071	Piston
Mixed	19	2	3	9.5	3/8	25.4 x 22.2	1.0 x 0.875	Riffled	Lanced(3)	1.7	0.067	Piston

The limiting F_c value of 1.3 is used in the SEER rating methodology [3]. In comparison, the SEER rating methodology correlated within 3.3% the SEERs of seven systems tested in the same laboratory. In the second round of verification using three manufacturers' proprietary data for 27 systems with $0.8<F_c<1.3$, predictions of the SEER methodology were within 7% for all systems tested[10]. Better predictions of EER(95) were obtained with the 1.2 value for the F_c upper limit (Figure A8). With this more restrictive limit, the prediction error exceeds 5% for only 4 out of 19 systems. All predictions are within 7.1% of the tested EER(95).

The indoor coil scaling factor, F_c, is the most influential parameter in the EER(95) algorithm, and, unfortunately, it is very difficult to evaluate accurately. In the authors' opinion, inaccurate estimates of F_c are most responsible, for the deviations between the tested and calculated EER(95) values. Inaccuracies in F_c may come from two main sources. One may be the inability of the evaporator simulation model to accurately predict the capacity of the mixed coil or matched coil, or both. The task of predicting coil capacities is becoming more challenging with constant development of new enhanced fin and enhanced inner tube surfaces. The other source of error in the F_c value may be a non-uniform air distribution at the coil inlet. Different coils may operate with different air velocity profiles because of different coil sizes and configurations (single slabs, a-shape, v-shape, multi-slabs). This may be a cause of a different level of capacity penalty. Since there is no public-domain information that allows assessing the capacity penalty caused by maldistributed air, capacity predictions using a simulation model may not result in an accurate value of the indoor coil scaling factor, F_c. For the reasons discussed here, the upper limit of the scaling factor has been set to 1.2 in this rating procedure.

A comment should be made on the alternative of not relying on the rating algorithm for EER(95) of a mixed systems, but rather on accepting the EER(95) value of the matched system instead. On average, this would result in somewhat more accurate EER(95) estimates than using the rating algorithm (based on the 19 mixed/matched systems sample applied in this study). Using the matched system EER(95) for the mixed system EER(95) resulted in the average of absolute deviations equal to 2.4%, while 3.4% deviation was obtained for the rating algorithm. However, a counter argument can be made that, this approach would not provide any incentive for the mixed coil producer to offer a system with an improved EER(95).

Table A3. Tested and calculated EER(95) for mixed systems

Case	Matched Coil			Mixed Coil			Tested EER(95)		$\frac{Q_{c,x}}{Q_{c,m}}$	Calculated $EER_x(95)$		Cal. $EER_x(95)/EER_m(95)$	
	No. of Slabs	Capacity, $Q_{c,m}$ W	Capacity, $Q_{c,m}$ Btu/h	No. of Slabs	Capacity, $Q_{c,x}$ W	Capacity, $Q_{c,x}$ Btu/h	Matched System	Mixed System		$F_{e,max}$=1.3	$F_{e,max}$=1.2	$F_{e,max}$=1.3	$F_{e,max}$=1.2
1	3	7577	25860	1	12492	42626	9.82	9.95	1.648	10.60	10.38	1.065	1.043
2	2	19604	66910	4	20860	71178	10.73	11.10	1.064	10.91	10.91	0.983	0.983
3	2	17028	58118	2	16334	55734	9.96	10.08	0.959	9.84	9.84	0.976	0.976
4	2	5959	20338	2	5629	19208	10.49	9.82	0.944	10.33	10.33	1.052	1.052
5	2	14309	48838	1	13618	46466	11.12	10.69	0.951	11.04	11.04	1.033	1.033
6	2	14524	49572	1	12486	42603	12.32	11.85	0.859	12.06	12.06	1.018	1.018
7	2	16321	55704	1	15134	51638	11.56	11.36	0.927	11.33	11.33	0.997	0.997
8	2	19372	66116	1	16575	56555	9.37	9.52	0.855	8.93	8.93	0.938	0.938
9	2	14454	49330	2	15263	52080	10.83	10.46	1.056	11.01	11.01	1.052	1.053
10	2	16114	54996	2	16368	55850	10.31	10.39	1.016	10.35	10.35	0.996	0.996
11	2	18045	61586	2	20099	68582	9.85	9.60	1.114	10.14	10.14	1.056	1.056
12	2	11579	39520	2	10065	34344	10.42	10.34	0.869	10.06	10.06	0.973	0.973
13	1	10912	37242	2	13914	47476	9.61	9.86	1.275	10.28	10.13	1.043	1.027
14	1	12635	43123	1	11692	39896	10.99	10.99	0.925	10.97	10.97	0.998	0.998
15	1	11620	39658	2	14985	51130	8.95	8.89	1.289	9.70	9.52	1.091	1.071
16	2	9306	31760	2	9051	30882	11.41	10.67	0.972	11.32	11.32	1.061	1.061
17	2	11689	39896	2	11170	38112	9.58	9.30	0.955	9.55	9.55	1.027	1.027
18	2	14115	48174	2	19571	66780	11.06	11.17	1.386	11.84	11.6	1.060	1.038
19	1	11620	39658	2	16683	56924	8.95	9.09	1.435	9.62	9.43	1.058	1.037

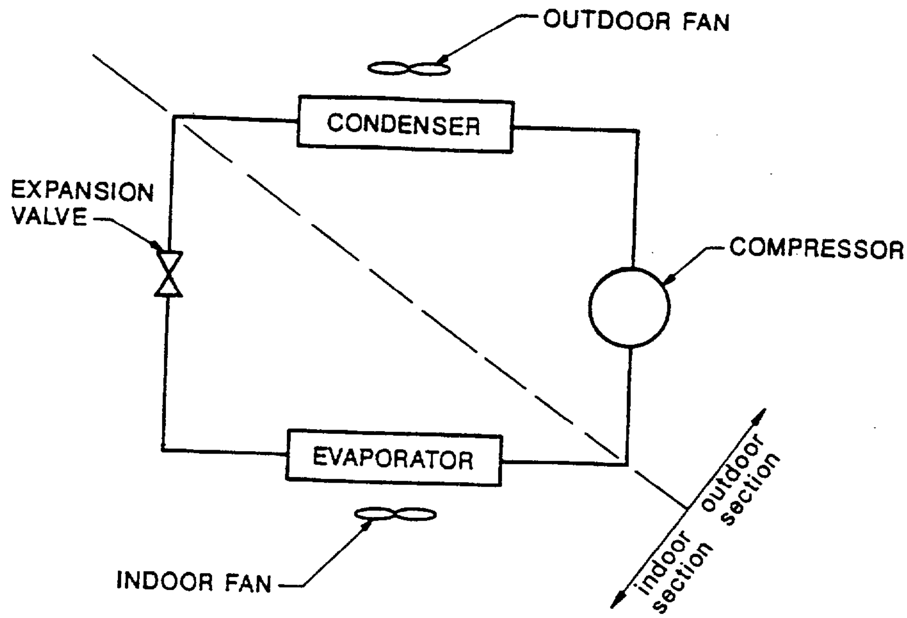

Figure A1. Main components of the vapor compression air conditioner

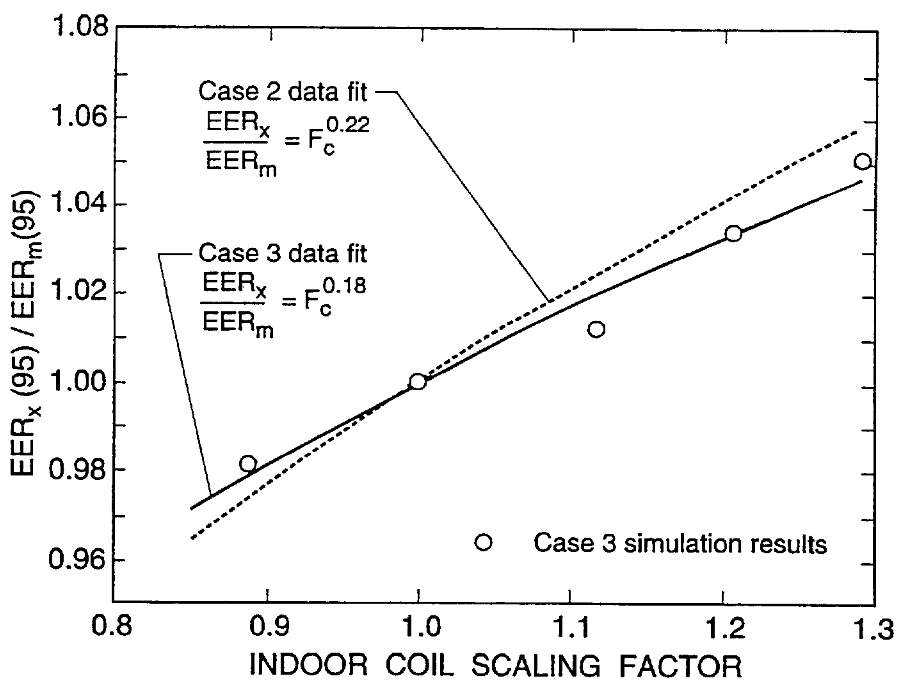

Figure A2. Impact of charging if optimizing on $EER_x(95)$ for a system with a capillary tube

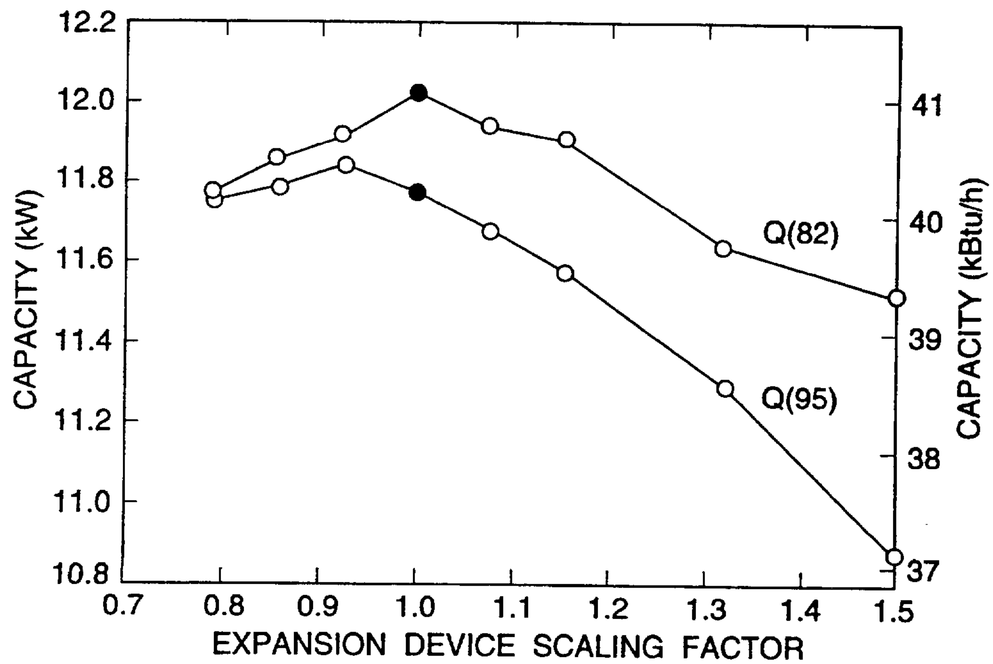

Figure A3. System capacities at Test A and Test B conditions at various expansion device restrictions (charging/optimizing case 2)

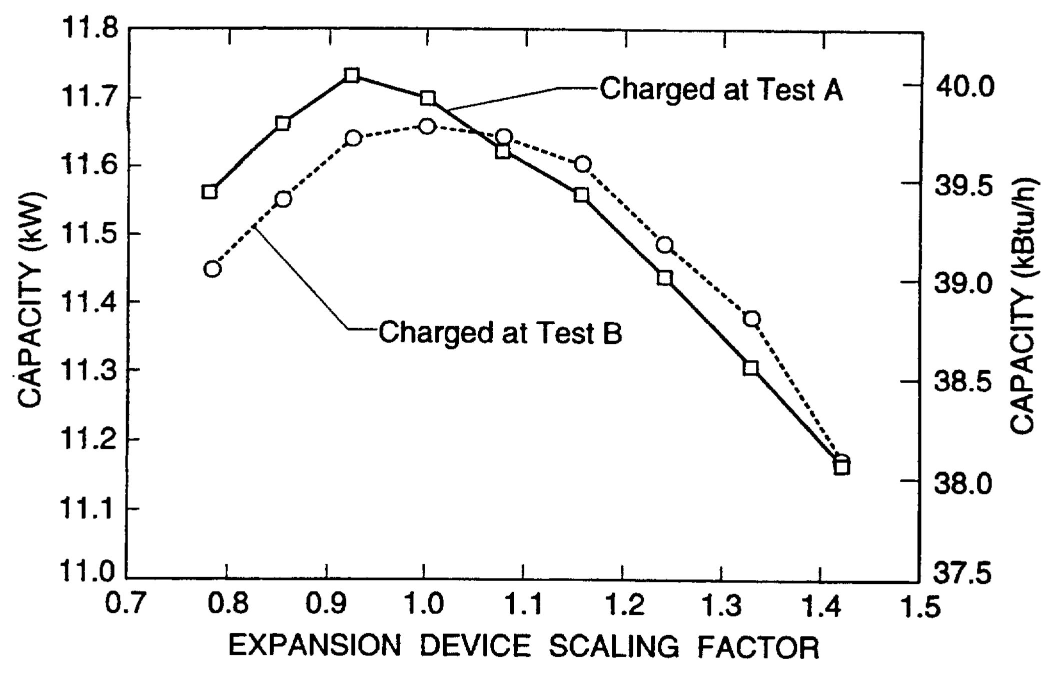

Figure A4. Capacity at Test A conditions if the system is charged at Test A conditions and at Test B conditions

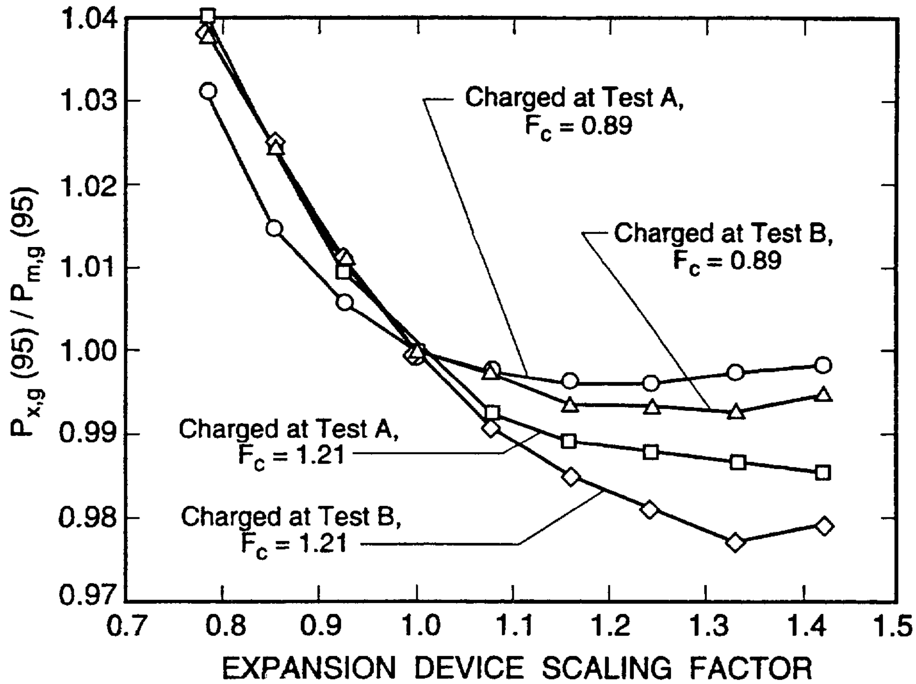

Figure A5. Compressor power at Test A conditions referenced to the compressor power at $F_{ex} = 1.0$

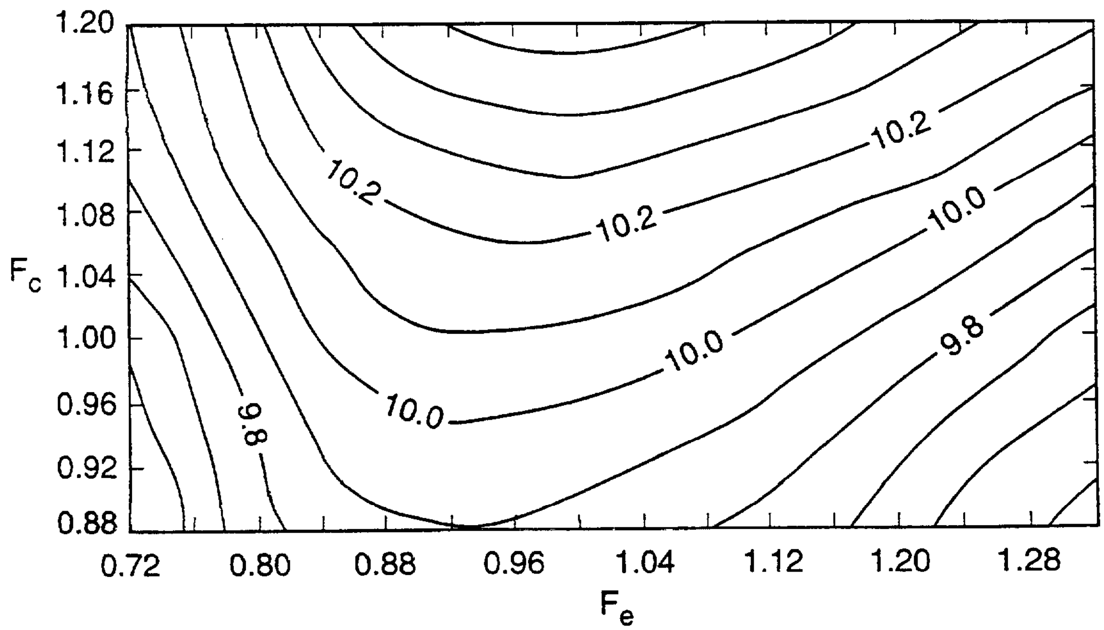

Figure A6. EER(95) map for a matched system charged at Test A conditions and optimized for EER(82)

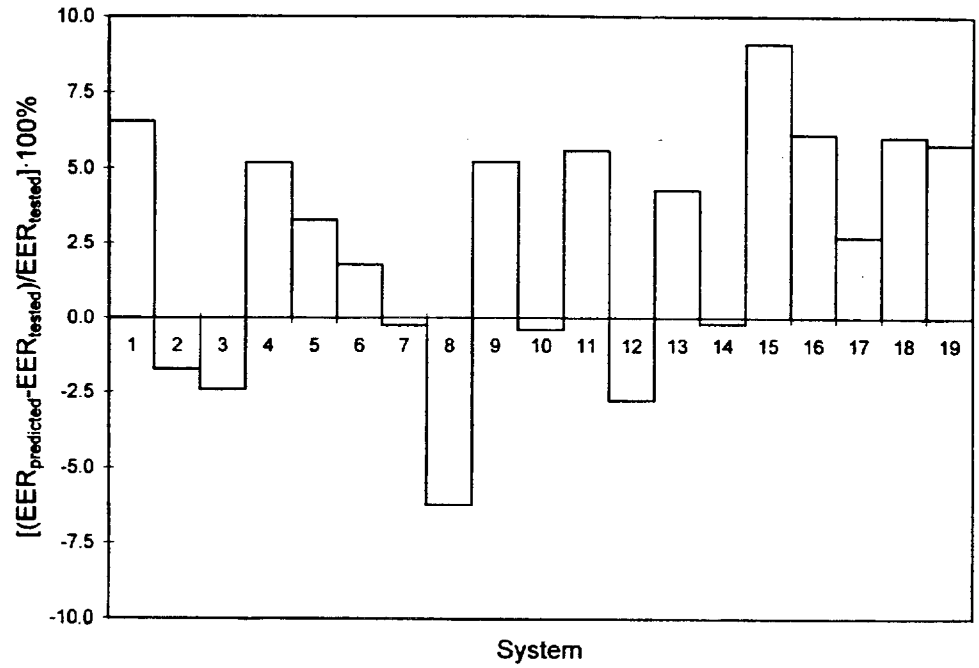

Figure A7. Relative difference between the predicted $EER_x(95)$ and tested $EER_x(95)$ for the 1.3 value of the upper limit of the indoor coil scaling factor, F_c

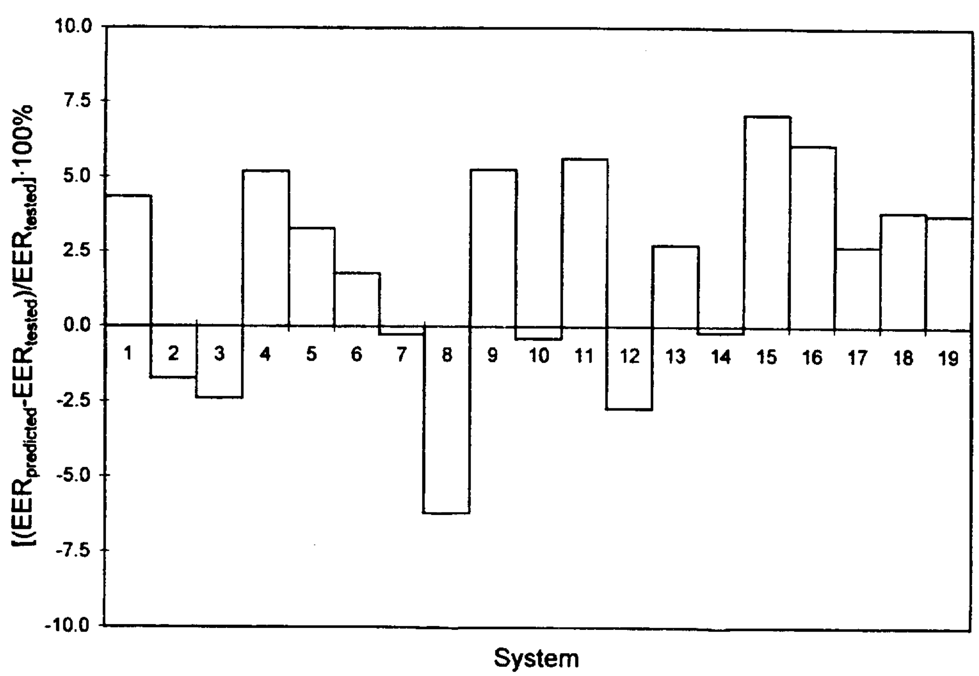

Figure A8. Relative difference between the predicted $EER_x(95)$ and tested $EER_x(95)$ for the 1.2 value of the upper limit of the indoor coil scaling factor, F_c