RATING OF MIXED SPLIT RESIDENTIAL AIR CONDITIONERS

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ABSTRACT

A methodology is presented for rating the performance of mixed, split residential air conditioners. The method accounts for the impact on system performance of the indoor evaporator, expansion device and fan; three major components that are likely to be substituted for the matched components in a mixed system. The method allows calculation of capacity at 95°F rating point and seasonal energy efficiency ratio, SEER, without performing laboratory test of the complete system. Limitations of the procedure, present work, and anticipated improvements are also discussed.

INTRODUCTION

Air conditioners and heat pumps belong to that category of products for which performance data, according to regulations, are required to be made available to a potential customer. In the case of an air conditioner and a heat pump operating in the cooling mode the required performance data consist of system capacity at the outdoor temperature of 95°F, Q(95), and seasonal energy efficiency ratio, SEER. The procedures to obtain these ratings by laboratory tests are described in Part 430, Title 10 of the Code of Federal Regulations[1].

The federal regulations require that manufacturers derive cooling ratings for unitary systems by testing a sample of sufficient size to meet

choose to limit their testing to what they judge to be the highest sales volume combination with that outdoor unit. The highest sales volume combination and any other combination for which tests are conducted on a sample of sufficient size to meet the federal regulations are referred to as matched systems. Other combinations, referred to as mixed systems, may be rated by means of computer simulation or other engineering methodology. Following these rules, a sample of the required size must be tested for at least one combination involving an outdoor unit.

Outdoor units and indoor sections are shipped separately to distributorships. A given outdoor section, for a number of reasons, may be offered for sale and installation with an indoor section other than that specified by the manufacturer of the outdoor section. This generates a need for evaluating performance of such a mixed system.

This paper presents a background of development of the rating procedure formulated at National Bureau of Standards (NBS)[2] and the status of the procedure. The input information required for this procedure are performance data of the matched system which employs the same outdoor unit as the mixed system. The required performance data, Q(95), SEER, and recommended indoor volumetric flow rate of air, are publicly available. The procedure also assumes that the matched indoor section is available for inspection and evaluation of the indoor coil capacity and the indoor fan power.

METHODOLOGY OF DEVELOPMENT OF THE RATING ALGORITHM

The main components of an air conditioner based on the vapor compression cycle principle are shown schematically in Figure 1. Substitution of any of the components may alter system performance. By substituting the indoor section we may, in fact, be changing three components of the system, namely: the indoor coil, the cooling mode expansion device, and the indoor fan. Each of these components will affect system performance in a specific way depending on relative performance characteristics of the newly installed and the original parts. Combination of these effects results in performance of the mixed system being different from that of the original system. The difference is usually within ±10%. This is due to the fact that the main performance-determining component, the compressor, is the same in the matched and mixed systems.

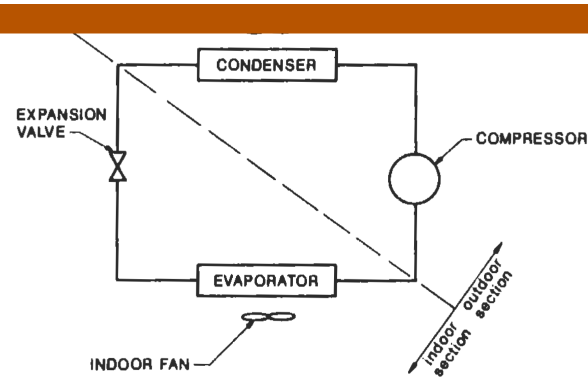

Fig. 1 The Main Components of the Vapor Compression Air Conditioner

The approach taken in this study is to derive ratings of the mixed system using the ratings of the matched system ($Q_m(95)$, $SEER_m$) as a base and adjusting their values to the expected capacity and

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SEER for the mixed system. This adjustment is made for each of the changed components individually. It is important then to assess the effect of each of the components on the system capacity at the outdoor temperature of 95°F, and on capacity and power input at the outdoor temperature of 82°F (Test A and Test B conditions, respectively[1]).

In the individual assessment of performance impact for each of the components, it has to be emphasized that a 'change of the indoor coil' simply means introducing to the system an indoor coil of which the capacity is different from the capacity of the indoor coil supplied as a part of the matched system. Following this definition, a change of the indoor CFM constitutes a change of the indoor coil since capacity of this coil will be different at a new CFM. By the same principle, physical substitution of the indoor fan may not constitute in this analysis a change of the fan if power input to the fan is the same as that of the matched fan. Thus for the purpose of this methodology, the indoor coil is characterized by its capacity at the CFM provided by the indoor fan, the indoor fan is characterized by its power needed to provide this CFM, and the expansion device is characterized by its restrictiveness to the refrigerant flow (explained later).

A computer model of a heat pump was used in this study to evaluate the effect of individual components on performance of the mixed system. A sensitive computer model can properly indicate relative performance trends even if change in performance is small. Although laboratory tests are preferred if absolute values of capacity or power input are needed, they are not as useful if small (in relation to the repeatability scatter) performance changes are investigated unless a significant number of laboratory tests is performed to establish a meaningful data set for developing of required correlations.

A computer model of a heat pump, HPSIM[3], used in this study, is a 'first principles', steady state model developed with emphasis on modeling phenomena taking place in the system on a local basis. The structure of HPSIM is modular. The model consists of 41 subprograms for heat pump component simulation, heat transfer, fluid mechanics, and fluid property calculation. The program totals approximately 5000 Fortran statements.

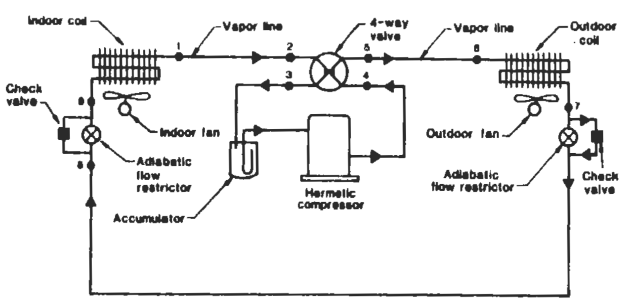


Fig. 2 Schematic of an Air Conditioner Simulated by the Heat Pump Model, HPSIM

System components considered in HPSIM are shown in Figure 2. The model simulates the hermetic, reciprocating compressor. The capillary tube is modeled using Fanno flow theory. Heat exchangers are modeled using a tube-by-tube approach where each tube is analyzed separately. Performance of the tubes is evaluated in the sequence the tubes are circuited yielding capacity of the heat exchanger. HPSIM is able to perform refrigerant mass inventory calculation. Mass inventory for the heat exchanger is also conducted on a tube-by-tube basis.

Among the most important features of HPSIM is the ability to iterate vapor superheat or quality at the compressor inlet at part load operating conditions. The logic of the model is shown in Figure 3.

PERFORMANCE IMPACT OF MIXED COMPONENTS

IMPACT OF A MIXED EVAPORATOR

To evaluate performance of a given outdoor section with different evaporators, a commercially available residential split heat pump was coded for input to HPSIM. The heat pump had a capillary tube as an expansion device optimized for maximum capacity at Test B conditions. This system was assumed as the matched system.

Based on the matched system three mixed systems were created (coded) by substituting three different evaporators for the matched evaporator. Simulation runs of the matched and mixed systems were performed at Test A and Test B conditions.

Simulation runs were also performed for the matched and mixed evaporators as individual components. The evaporators were simulated by the evaporator model subroutine used in the heat pump model. The simulations were conducted for 20% refrigerant inlet quality, 45°F saturation temperature and 10°F superheat at the coil exit, and the same air flow rate as during coil operation as a part of the system.

Individual simulations of the evaporators allowed establishment of the value of the indoor coil scaling factor for each mixed coil, F_c, and to correlate mixed system simulation results in the following form:

$$\frac{Q_{x,g}}{Q_{m,g}} = F_c^{\gamma} \tag{1}$$

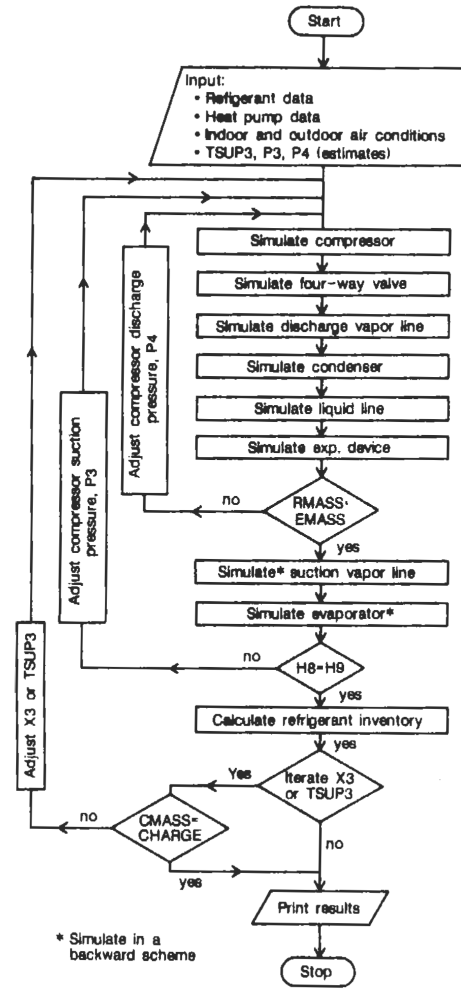
$$\frac{P_{x,c}}{P_{m,c}} = F_c^{\xi} \tag{2}$$

where: $Q_{x,g}$ = mixed system gross capacity (indoor fan heat not included)

$Q_{m,g}$ = matched system gross capacity (indoor fan heat not included)

$$F_c = \frac{Q_{x,coil}}{Q_{m,coil}}, \text{ indoor coil scaling factor} \tag{3}$$

$Q_{x,coil}$, $Q_{m,coil}$ = gross capacity of mixed and matched coils, respectively, at the



Symbols:

EMASS - refrigerant mass flow rate through the expansion device

CHARGE - refrigerant charge in the system (a part of input data)

CMASS - calculated refrigerant charge in the system

H8 - refrigerant enthalpy at the expansion device inlet

H9 - refrigerant enthalpy at the evaporator inlet

P3 - refrigerant pressure at the compressor can inlet

P4 - refrigerant pressure at the compressor can outlet

RMASS - refrigerant mass flow rate pumped by the compressor

TSUP3 - refrigerant vapor superheat at the compressor can inlet

X3 - refrigerant quality at the compressor can inlet

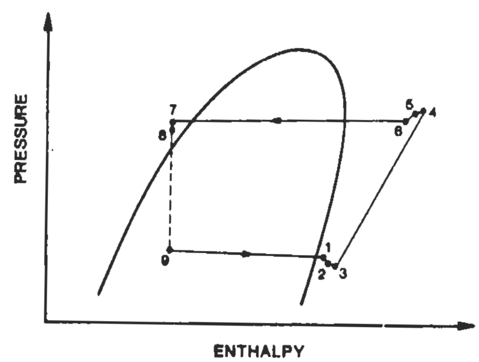


Fig. 3 Logic of the Heat Pump Model, HPSIM

same refrigerant saturation temperature and superheat at the coil outlet, and at the same inlet quality for both coils, and for each coil at the same air mass flow rate as during operation as a part of the system

$P_{x,c}$, $P_{m,c}$ = compressor power input when operating in mixed and matched systems, respectively

The simulation results were curve fitted using the least squares method to evaluate the unknown exponents. The results are shown in Table 1. The graphical representation of Equations (1) and (2) is shown in Figure 4.

IMPACT OF A MIXED EXPANSION DEVICE

We have to consider two cases analyzing the impact of an expansion device on system performance. The first case is when a constant flow area restrictor (capillary tube or short tube restrictor) is employed as the mixed expansion device. The second case involves a thermostatic expansion valve (TXV).

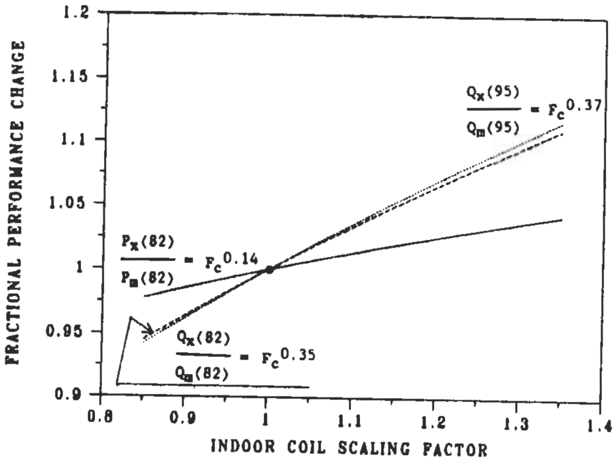


Fig 4. Graphical Representation of Equations (1) and (2)

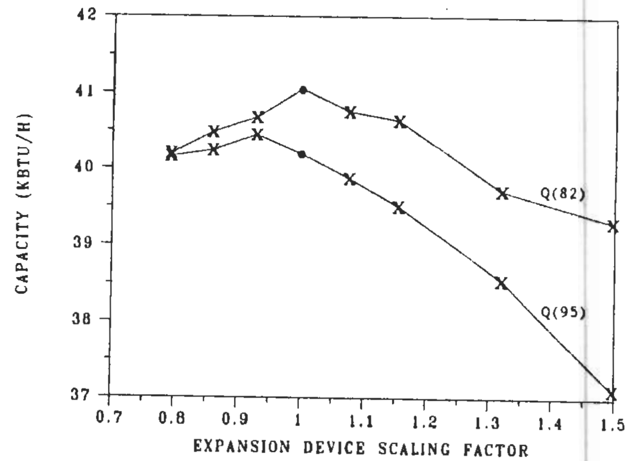


Fig. 5 System Gross Capacity at Various Flow Restrictions

Table 1. Exponents Found for Equations (1) and (2)

Exponent / Test	γ	ξ
A	0.37	n/a
B	0.35	0.14

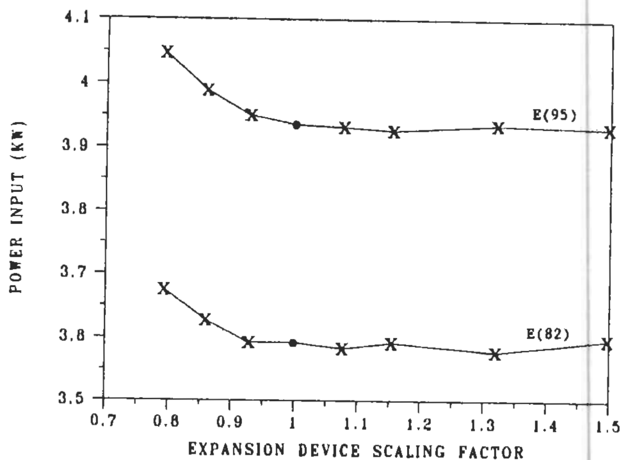


Fig. 6 Compressor Power Input at Various Flow Restrictions

<u>Impact of a Constant Flow Area Restrictor.</u> To determine the effect of a constant flow area expansion device on system performance, four heat pump were simulated at Test A and Test B conditions with varied diameter capillary tubes. Results obtained for one of the systems typical of results obtained for all systems, are shown in Figure 5 and Figure 6. The results are presented as a function of the expansion device scaling factor, F_{ex}, which is defined as a ratio of refrigerant mass flow rates through the mixed and matched expansion devices at the same operating conditions:

$$F_{ex} = \frac{m_x}{m_m} \tag{4}$$

where: m_x and m_m are refrigerant mass flow rates through the mixed and matched expansion devices at the same refrigerant state at inlet and the same evaporator pressure.

The matched expansion device, by definition, has the scaling factor, F_{ex}, equal to 1.

A given combination of compressor, condenser and evaporator will reach its maximum capacity at outdoor temperatures of 95°F and 82°F at different sizes of the expansion device. The figures show that a system designer/manufacturer has a choice of selecting a capillary tube. The selection can be done for the maximum capacity at either Test A conditions or Test B conditions. The latter is the manufacturer's most probable choice since it corresponds to the maximum SEER a system can attain as shown in Figure 7. In any case, the matched expansion device should fall in the range between expansion devices that provide maximum Test A capacity and maximum Test B capacity. Such a selection provides a compromise between capacity at 95°F outdoor temperature and SEER. If the expansion device is beyond this range, both capacity at 95°F outdoor temperature and SEER are penalized.

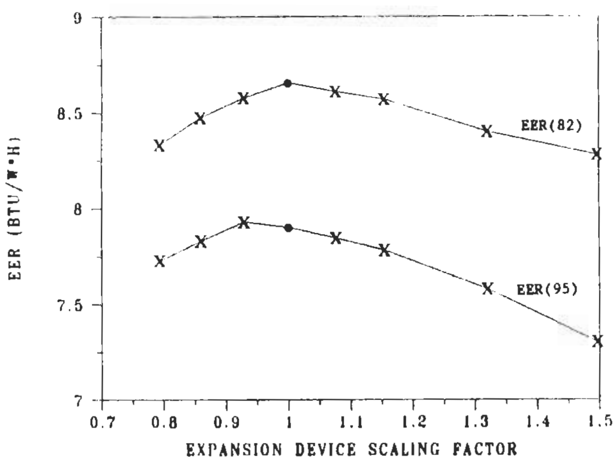


Fig. 7 System EER at Various Flow Restrictions

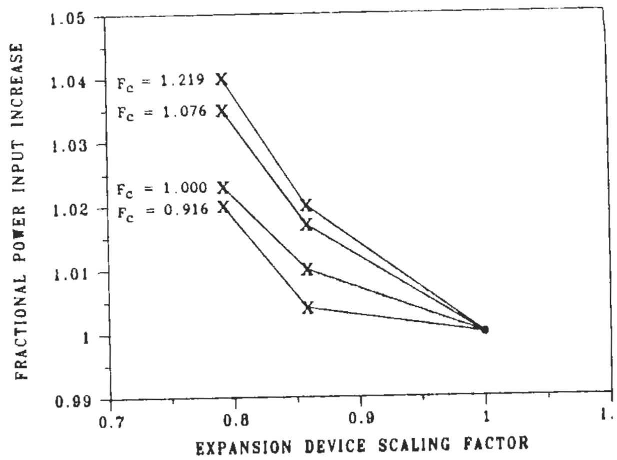


Fig. 8 Compressor Power Input at Various Capacity Evaporators and Various Flow Restrictions

The information about the sizing of the expansion device is treated by manufacturers as proprietary and is not publicly available. It is assumed in this analysis that the matched system employs an expansion device of appropriate dimensions to maximize system capacity at 82°F outdoor temperature.

Because of the expansion device sizing uncertainty and different system sensitivity to the oversized and undersized expansion device, individual consideration has to be given to undersized and oversized mixed flow restrictors.

If an oversized expansion device is provided, capacity gradually degrades with decrease of restrictiveness while the power input remains basically unchanged. Degradation of capacity is similar at both outdoor temperatures of 95°F and 82°F. An undersized expansion device can highly degrade performance of the system at 82°F through capacity decrease and power input increase. System capacity at 95°F may increase slightly with some increase of restrictiveness of the expansion device. After reaching its maximum, Test A capacity decreases along with capacity at Test B conditions.

Performed simulations also demonstrate that the impact the mixed, over-restrictive expansion device has on system performance depends on capacity of the mixed evaporator. This dependency for the power input is shown in Figure 8. The figure shows that over-restrictiveness of the expansion device causes a greater power input increase for more oversized indoor coils. This is understandable since the mixed system with an oversized indoor coil will operate at a higher evaporator pressure than the matched system. A higher evaporator pressure results in a higher vapor density at the compressor suction port and in a higher refrigerant mass flow rate. Consequently, a somewhat less restrictive expansion device is a better match for a mixed evaporator having greater capacity than the matched evaporator.

We assumed the matched expansion device to be sized for the maximum capacity at 82°F outdoor temperature. Since this assumption determines the reference point for evaluation of F_{ex}, some uncertainty exists in determination of the effect of the mixed expansion device on performance. Because this uncertainty is significant for the over-restrictive expansion device, a tight limit has to be imposed for the mixed expansion devices on the over-restrictive side. A relaxed limit is also in order for the under-restrictive side.

In addition to inflicting a problem in prediction of performance of the mixed system, the over-restrictive expansion device may cause serious reliability problems, particulary for heat pumps. If a system with the over-restrictive restrictor is charged to the same superheat as the matched system, significant overcharge of the system may occur. This is displayed in Figure 9. The figure shows a drastic increase of refrigerant charge for F_{ex} less then 1. Since the system compressor and accumulator have predetermined dimensions that allow to accept a limited amount of liquid refrigerant, increased charge may cause liquid refrigerant entering the compressor cylinder and damaging the compressor. This damage may occur during cycling operation and particulary in heat pumps during the defrost cycle.

Substituting an under-restrictive expansion device has also reliability implications. A mixed system charged in the cooling mode will contain less refrigerant than the matched system. This may result in considerable inlet vapor superheat and high discharge temperature at the compressor in the heating mode at part load operating conditions.

The performance simulation results of a system with different flow restriction expansion devices can be represented by the following exponential form correlations:

$$\frac{Q_{x,g}}{Q_{m,g}} = F_{ex}^{\alpha} \qquad (5)$$

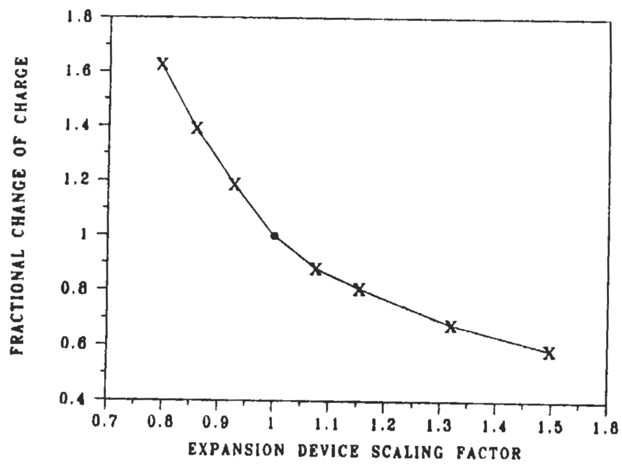


Fig. 9 Change of Refrigerant Charge at Different Flow Restrictions

$$\frac{P_{x,c}}{P_{m,c}} = F_{ex}^{\beta} \qquad (6)$$

where: $Q_{x,g}$, $Q_{m,g}$ = gross capacities of mixed and matched systems, respectively (heat of the indoor fan not included)

$P_{x,c}$, $P_{m,c}$ = compressor power inputs when operating in mixed and matched systems, respectively

Table 2 displays the proposed values of the exponents. Because of variations of the impact on performance of the expansion device scaling factor depending on the mixed coil capacity and the assumption of sizing of the matched restrictor, the value of exponents were estimated using selected simulation cases.

Table 2. Proposed Values for Exponents in Equations (4) and (5).

Exponent	$F_{min} \le F_{ex} < 1$	$1. \le F_{ex} \le F_{max}$
α	0.0	- 0.15
β	- 0.2	0.0

F_{min} = 0.95 and F_{max} = 1.35 for systems operating only in the cooling mode

F_{min} = 1.00 and F_{max} = 1.25 for systems able to operate in both cooling and heating

For the over-restrictive expansion device (F_{ex} < 1), it was assumed that Test A and Test B capacities remain constant while power input decreases. The exponent 0.2 corresponds to Test B results of a system with an oversized coil having the indoor coil scaling factor equal to approximately 1.2. For the under-restrictive expansion device (F_{ex} > 0) no change of the input power is proposed. The value of -0.15 for the exponent α correlating degradation of capacity was obtained applying the least square method to Test A capacities of the system using the matched coil. For the reasons discussed before, tight limits of applicability of the procedure are proposed for the over-restrictive expansion device, as indicated in Table 2.

Impact of a Thermostatic Expansion Valve. The approach taken in the previous section relied on evaluation of the expansion device scaling factor, F_{ex}. This approach does not apply here since a properly sized TXV will open sufficiently to allow appropriate refrigerant mass flow rate. A TXV will not be over-restrictive or under-restrictive for the system. The value of the expansion device scaling factor, F_{ex}, for a property sized mixed TXV may be then assessed to be equal to 1. Obviously, application of a proper size TXV in the mixed system alleviates problems related to an improperly sized constant flow area restrictor, particulary if it is undersized.

A properly sized TXV appears to be a superior expansion device for most mixed systems. The performance improvements will arrive for the system from a controlled superheat at the evaporator outlet during Test B conditions; this applies to the systems charged to a specified superheat at Test A conditions which is the most common manufactures's choice. In such a case, the mixed system capacity at Test A is not affected. However, SEER will improve since it is sensitive to Test B results.

If the TXV is of the non-bleed type, an additional performance improvement may result from reducing the refrigerant migration during the compressor off-time. Decreasing the refrigerant migration improves the cyclic degradation coefficient, C_D, which in turn enhances SEER.

The issue of assessing a numerical value of SEER improvement due to employment of a TXV is controversial. The data we have reviewed had a scatter regarding the effect of refrigerant superheat and migration control on system SEER. Also the industry comments on the proposed rule[4] displayed lack of agreement between different manufacturers. Suggested SEER credits varied from 0 to 3% for a bleed type TXV, and from 0 to 2.5% as an additional credit for a non-bleed feature or a liquid line solenoid valve. The lack of consensus opinion indicates that a TXV may not affect uniformly all systems. The impact also depends on the TXV ability to maintain a constant superheat with changing operating conditions, and on the TXV dynamic response during the start-up period of the system. The system charging criterion may additionally impact the TXV effectiveness.

Table 3 displays the proposed values for a thermostatic expansion valve factor, F_{TXV}. The factor should be applied as a multiplier to SEER. SEER adjustments suggested below are believed to provide on average appropriate correction for TXV employment. The correction proposed for a bleeding TXV over a capillary tube and short tube restrictor is 2.5%. The same correction, 2.5%, is suggested for a non-bleed TXV replacing a bleeding type TXV. A combined correction of 5% applies for a non-bleed TXV replacing a capillary tube or a short tube restrictor. For any not specified above combination, the thermostatic expansion valve factor, F_{TXV}, equals 1.

IMPACT OF A MIXED INDOOR FAN

The performance impact of the indoor fan can be *easy taken* into account. We may assume that 100% of the fan power input is converted into heat. This heat is trans-
fered to the conditioned air decreasing the system capacity. The indoor fan is not a part of the thermodynamic cycle so it does not affect work of the compressor. Contribution of the indoor fan can be then accounted for by additive terms in equations for system power input and capacity.

Table 3. Thermostatic Expansion Valve Factor, F_{TXV}

Expansion Device		F_{TXV}
Matched System	Mixed System	
TXV, no bleed	TXV, no bleed*	1.000
TXV, no bleed	TXV, w/bleed*	0.975
TXV, w/bleed	TXV, no bleed*	1.025
TXV, w/bleed	TXV, w/bleed*	1.000
Capillary or Orifice	TXV, no bleed**	1.050
Capillary or Orifice	TXV, w/bleed**	1.025

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

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

RATING EQUATIONS

Equations for calculation of performance ratings of the mixed system are given below. Derivation of these equations is explained in the Appendix.

Capacity at Test A conditions:

$$Q_x(95)=(Q_m(95)+3.413\cdot P_{m,f})F_c^{0.37}\cdot F_{ex}^{\alpha}-3.413\cdot P_{x,f} \qquad (7)$$

Seasonal Energy Efficiency Ratio, SEER:

$$SEER_x = SEER_m\cdot\frac{Q_x(82)}{Q_m(82)}\left[\frac{P_x(82)}{Q_m(82)}\right]^{-1}\cdot F_{TXV} \qquad (8)$$

$$\frac{Q_x(82)}{Q_m(82)} = \left(1+\frac{3.25\cdot P_{m,f}}{Q_m(95)}\right)F_c^{0.35}\cdot F_{ex}^{\alpha}-3.25\,\frac{P_{x,f}}{Q_m(95)} \qquad (9)$$

$$\frac{P_x(82)}{P_m(82)} = F_c^{0.14}\cdot F_{ex}^{\beta}+0.1\,\frac{P_{x,f}}{P_{m,f}}+0.1 \qquad (10)$$

Exponents α and β are given in Table 2. Values for F_{TXV} are given in Table 3.

These rating equations should be considered as a rating tool which is inferior to testing of a complete system in a laboratory. The accuracy of predictions using these equations is limited by the amount of available matched system data and by a number of assumptions taken during their development.

The equations were derived considering the evaporator and expansion device as being independent variables in the system, such that system performance change due to substitution of these components simultaneously will be equal to the sum of the performance changes resulting from substitution of these devices one at a time.

Another important assumption was that the matched system expansion device is optimized to achieve the maximum capacity at Test B conditions. If the actual matched system is optimized for Test A capacity, the equations will underpredict SEER for $F_{ex} > 1$, and overpredict both Q(95) and SEER for $F_{ex} < 1$. The SEER equation has the embedded assumption that the cyclic degradation coefficient, C_D, is identical for the matched and mixed systems.

The limitations of the equations are also related to the method by which the equations were developed. We have to be aware that by using the proposed rating equations we are extrapolating to all compressors and systems the correlations found for one simulated system. Obviously, also, performance predictions depend directly on accuracy of the required input.

IMPLEMENTATION OF THE PROCEDURE

The procedure requires two types of data as input. The first type are performance data of the matched system, namely: Test A capacity, $Q_m(95)$, and $SEER_m$. This information is publicly available. The second type of data consists of information which describes how much the mixed component differs performancewise from the matched component. This information, needed on the evaporators, expansion devices and indoor fans, has to be developed by the rater.

EVAPORATOR DATA

The biggest impact on the accuracy of performance prediction has the appropriate evaluation of the indoor coil scaling factor, F_c. If capacity of the matched and mixed coils are known with 5% error falling on opposite sides of the true values, the error in the system capacity prediction is 3.8%, and in SEER prediction 3.6%. Five percent discrepancy between capacities obtained by two independent experimental methods is realistic for this type of measurement.

The potential for getting a greater error exists if coil capacity is obtained using third party coil performance catalogs without laboratory verification. The problem becomes evident when we realize that these catalogs, since they represent different detailed design, manufacturing techniques, tooling, etc., provide different capacities for identically specified heat transfer surfaces.

Table 4. Difference of Capacities of Evaporators Obtained from Two Different Catalogs.

Depth of a Coil (rows)	Capacity Difference*	
	Surface I (%)	Surface II (%)
2	13	7
3	12	4
4	10	2
5	9	1

Surface I - 3/8 inch O.D copper tube, 1.00 x 0.866 staggered, corrugated aluminum fins, 0.006 inch thick, 12 fins per inch, face velocity: 450 fpm

Surface II - 0.5 inch O.D. copper tube, 1.25 x 1.08 staggered, corrugated aluminum fins, 0.006 inch thick, 12 fins per inch, face velocity: 450 fpm

*Difference = 100% • (catalog A - catalog B)/ catalog A

Table 4 contains comparison of capacities obtained for two configurations from catalogs of two independent manufacturers[5,6].

The difference in predicting capacity of the same coil by these catalogs was from 1% to 13%, depending on the tube configuration and a number of tube depth rows. At the same time, we should realize, that this comparison does not include consideration for coil circuitry which, if not designed adequately, may be detrimental to coil capacity.

Coil capacity predictions may have even greater error if the air distribution taking place in the real system is not taken into account by the capacity prediction tool. To the best of this author's knowledge, available evaporator simulation models assume a uniform air velocity profile over the heat exchanger face area. Also laboratory tests of coil surfaces for catalog development are most likely performed with a uniform air distribution.

Laboratory test of evaporator coils performed at NBS[7] showed that maldistribution of air may have detrimental effects on coil capacity. Figure 10 presents some of these test results. The tests of a coil were performed in a horizontal duct at vertical and a few slanted coil positions. All test were performed at the same refrigerant inlet quality (20%), the same refrigerant state at the evaporator outlet (45°F saturation temperature and 8°F superheat), and at the same air volumetric flow rate. Observed capacity degradation was as much as 25%. Smoke tests and dynamic pressure measurements indicated that maldistribution of air at different configurations was responsible for this significant loss of capacity.

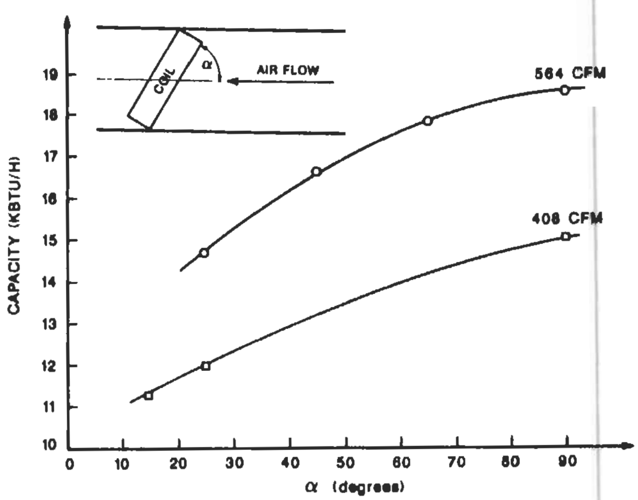


Fig.10 Evaporator Capacity in the Horizontal Duct at Various Configurations

It should be noted that during a standard rating test of a system involving the coil of the most steep angle, the CFM would have to be decreased to meet the industry standard of 37.5 CFM per 1000 Btu/H[8]. The reduction of the air mass flow rate would result in a further capacity decrease.

Since accurate prediction of coil capacities has the greatest relative significance for accuracy of the mixed rating procedure, NBS is developing an evaporator simulation program which will include air distribution as well as coil circuitry as input. At the same time tests are underway for different evaporator configurations to establish common patterns of velocity profiles that could be used with the model.

The use of the model would require verification/tuning of the model with the heat transfer surfaces involved. Once a prediction of the model is related by a correction factor to performance of an evaporator with given refrigerant and air side surfaces, simulations of other evaporators with the same surfaces can be performed. Considering that each manufacturer has usually a limited set of preferred surfaces, the number of required tests should be reasonable.

EXPANSION DEVICE DATA

The refrigerant mass flow rates through the mixed and matched restrictors at the same operating conditions have to be known to calculate the expansion device scaling factor, F_{ex}. An inlet pressure of 250 psia and inlet subcooling of 13°F are considered here as representative conditions for operation at Test A and Test B.

If the mixed expansion device is a thermostatic expansion valve, the expansion device scaling factor F_{ex}, is equal to 1 and an appropriate value for the thermostatic expansion valve factor, F_{TXV}, has to be read from Table 3. If the matched system is equipped with a TXV, the mixed system has also to be equipped in a TXV. Otherwise, the rating procedure cannot be used since the impact of the fixed flow area mixed restrictor cannot be evaluated.

Refrigerant mass flow through a capillary tube can be evaluated based on its geometry with the aid of the ASHRAE Handbook, Equipment Volume[9]. A simplistic correlation for evaluation of the mass flow rate through a short tube restrictor was proposed in[2]:

$$m = \frac{72200 \cdot D^2}{(0.58 + 0.008 \frac{L}{D})^{0.5}} \quad (11)$$

The above equation assumes liquid choking at the restrictor outlet. The equation combines the single pressure drop equation and the pressure drop formula for a flow with a sudden contraction (a slightly beveled entrance is assumed). Currently tests of short restrictors are being performed at NBS to upgrade the short tube restrictor correlation. The variables under investigation are the inlet and outlet pressures, the inlet subcooling, length to diameter ratio, and different tube entrances. It is also planned to explore the possibility of designing a simple testing method for a restrictor using a surrogate fluid to obtain the needed relative performance information of the mixed and matched restrictors. This method would be particulary desirable for restrictors connected in series. Obviously, the refrigerant mass flow rate through a restrictor can be determined by a loop experiment, however, this alternative is too burdensome and would defeat the objective of this procedure.

Any analytical method for evaluation of the refrigerant mass flow rate requires a precise knowledge of restrictor dimensions. The most important datum is the diameter. If the inner diameter is known with an error of 2%, the possible error in evaluating F_{ex} is 8% resulting in 1.5% error in Test B capacity and SEER.

INDOOR FAN DATA

For systems which are supplied without the indoor fan, the indoor fan power may be obtained using the assigned value of 365 watts/1000 CFM[8]. In practice, indoor fans have different efficiencies. Our review of test data supplied by one manufacturer showed ± 20% deviation of tested indoor fan power from the power that was obtained by the above assignment. Another source reported test result in which the indoor fan drew 600 watts/1000 CFM (64% deviation).

The possible error in performance prediction resulting from the use of the assumed wattage is easy to evaluate using Equations (7) and (10). For the matched indoor fan power overestimated by 20%, and the mixed indoor fan power underestimated by 20%, the error in prediction of Test A capacity is 1.5%, and 4% on prediction of SEER.

VERIFICATION OF THE PROCEDURE

Verification of the rating equations was performed by comparing predictions of the Test A capacity and SEER with test results obtained at an independent laboratory. Indoor coil capacities required as input to the correlations were obtained using a third party catalog. Expansion device information was obtained from manufacturers. Indoor fan powers were calculated by assigning 365 watts/1000 CFM since the involved systems were supplied without indoor fans.

Performance of nine units was evaluated regardless of values of the expansion device scaling factor, F_{ex}, falling outside the limits of Table 2. The results are shown in Table 5. Predicted performance of only one system was not at least 95% of the measured performance.

Table 5. Verification of Performance Predictions for Mixed System

System	$\frac{Q(95)_{tested}}{Q(95)_{predict}}$	$\frac{SEER_{tested}}{SEER_{predict}}$	F_c	F_{ex}
1	1.010	1.000	1.38	1.30
2	1.023	1.033	0.83	1.01
3	0.984	0.976	1.10	1.00
4	0.949	0.983	1.16	0.77
5	0.972	1.032	0.78	1.03
6	1.014	1.050	0.85	1.03
7	0.996	1.031	1.22	1.40
8	1.031	1.086	0.76	0.73
9	0.993	0.994	0.98	0.91

Independent from the above, subsequent verification on 36 mixed systems did not provide as good results. This second round verification was done using three manufacturers' proprietary data. Performance of mixed systems was predicted within 7% from the test obtained values for systems satisfying the restriction $0.8 < F_c < 1.3$ (27 systems). The remaining nine systems had the indoor coil scaling factor greater than 1.3. The maximum error of prediction for these systems was as much as 13.9% for Test A capacity and 18.7% for SEER. For only one out of these systems $Q_x(85)$ and $SEER_x$ were predicted within 5% of the test measured values.

The verification results prompted the heat exchanger study[7] which revealed the importance of including the air distribution information at evaluation of the evaporator capacity. Taking into account the air distribution should improve considerably accuracy of coil capacity predictions and prediction of the mixed system performance.

CONCLUDING REMARKS

A methodology is presented for rating the performance of mixed residential air conditioners. The method accounts for impact on system performance of the indoor evaporator, expansion device and fan; three major components that are likely to be substituted for the matched components in a mixed system.

This procedure represents a rating tool for mixed systems in situations where the mixed components are not those specified by the outdoor unit manufacturer and a limited amount of matched system data is available. The limited amount of the matched system data obtainable is the reason for limitations of the procedure. There is not much opportunity for further improvement of the procedure at the present level of data availability or without additional testing.

The presented methodology is still in an evolutionary process. Verification results of the original version of the procedure[2] provided direction for research findings of which will be incorporated in the second version of the procedure. The new features that will be incorporated in the procedure include an evaporator simulation model which will account for refrigerant circuitry and for uneven air distribution. The next version of the procedure will also include a new correlation for prediction of refrigerant mass flow rate through short tube restrictors. Once these enhancements are available, the procedure will undergo a second round of verification.

The large number of variables and the complexities of their interactions always make theoretical or quasi-empirical rating procedures less certain than a whole system test. Therefore, any analytical rating procedure requires a continuous link with laboratory experiment. In the new version of the rating procedure, tests of selected coils, representative of their coil families, will be prescribed. These tests will constitute the required link since the procedure is most sensitive to the value of the indoor coil scaling factor.

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NOMENCLATURE

CFM = volumetric flow rate of air, (ft^3/min)
D = inner diameter of a short tube restrictor (inch)
F_c = $\dfrac{Q_{x,coil}}{Q_{m,coil}}$, indoor coil scaling factor
F_{ex} = expansion device scaling factor as defined by Equation (4)
F_{TXV} = thermostatic expansion valve factor as specified in Table 3
L = length of a short tube restrictor (inch)
m = refrigerant mass flow rate through a short tube restrictor, (lb/H)
P(82), P(95) = system power inputs at Test B and Test A conditions, respectively, (W)
$P_{m,f}$, $P_{x,f}$ = power input to indoor fans, matched and mixed respectively, (W)
P_{out} = power input to the outdoor fan, (W)
Q(82), Q(95) = system capacities at Test B and Test A conditions, respectively, (Btu/H)
$Q_{m,coil}$, $Q_{x,coil}$ = gross capacity of matched and mixed coils, respectively, at the same refrigerant saturation temperature and superheat at the coil outlet, and the same inlet quality for both coils, and for each coil at the same air mass flow rate as during operation as a part of the system
Test A = steady state test at 95°F outdoor temperature, and 80°F dry bulb/67°F wet bulb indoor temperature[1]
Test B = steady state test at 82°F outdoor temperature, and 80°F dry bulb/67°F wet bulb indoor temperature[1]

Superscripts:

α = exponent defined in Table 2
β = exponent defined in Table 2

Subscripts:

g = gross (indoor fan heat not included)
m = matched
x = mixed

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APPENDIX

A. Derivation of correlation for calculation of capacity of a mixed system at Test A conditions, $Q_x(95)$

Assuming that an indoor coil and an expansion device are independent variables in the system their effect on capacity may be represented by combining Equations (1) and (5) in the following form:

$$Q_{x,g} = Q_{m,g} \cdot F_c^{0.37} \cdot F_{ex}^{\alpha} \tag{A1}$$

Including heat added by respective indoor fans, Equation (A1) becomes:

$$Q_x(95)=(Q_m(95)+3.414 \cdot P_{m,f})F_c^{0.37} \cdot F_{ex}^{\alpha}-3.413 \cdot P_{x,f} \tag{A2}$$

B. Derivation of correlation for calculation of Seasonal Energy Efficiency Ratio of a mixed system, $SEER_x$

The Seasonal Energy Efficiency Ratio is defined for single speed units by the following equation[1]:

$$SEER = (1-0.5 \cdot C_D) \frac{Q(82)}{P(82)} \tag{A3}$$

Using this definition, the ratio of SEER for mixed and matched systems is:

$$\frac{SEER_x}{SEER_m} = \frac{(1-0.5 \cdot C_{D,x})}{(1-0.5 \cdot C_{D,m})} \frac{Q_x(82)}{Q_m(82)} \left[\frac{P_x(82)}{P_m(82)}\right]^{-1} \tag{A4}$$

Assuming degradation coefficients $C_{D,x}$ and $C_{D,m}$ are equal (within tolerance of laboratory experiment), Equation (A4) becomes:

$$SEER_x = SEER_m \frac{Q_x(82)}{Q_m(82)} \left[\frac{P_x(82)}{Q_m(82)}\right]^{-1} \tag{A5}$$

The multiplication factor, F_{TXV}, existing in Equation (8) was introduced to adjust system performance rating for performance change associated with employment of the mixed thermostatic expansion valves (see Table 3).

The ratio $Q_x(82)/Q_m(82)$ has to be derived in a few steps. Using Equations (1) and (5), and following the derivation of Equation (A2), we can write:

$$Q_x(82)=(Q_m(82)+3.413 \cdot P_{m,f})F_c^{0.35} \cdot F_{ex}^{\alpha}-3.413 \cdot P_{x,f} \tag{A6}$$

Dividing both sides of Equation (A6) by $Q_m(82)$, we obtain:

$$\frac{Q_x(82)}{Q_m(82)} = \left(1+\frac{3.413 \cdot P_{m,f}}{Q_m(82)}\right)F_c^{0.35} \cdot F_{ex}^{\alpha} - \frac{3.413 P_{x,f}}{Q_m(82)} \tag{A7}$$

Since capacity of the matched system at Test B conditions, $Q_m(82)$, is not publicly available, it is assumed that $Q_m(82)$ is 5% greater than capacity $Q_m(95)$. Implementation of this assumption brings Equation (A7) to the form of Equation (9):

$$\frac{Q_x(82)}{Q_m(82)} = \left(1+\frac{3.25 \cdot P_{m,f}}{Q_m(95)}\right)F_c^{0.35} \cdot F_{ex}^{\alpha}-3.25 \frac{P_{x,f}}{Q_m(95)} \tag{A8}$$

The ratio $P_x(82)/P_m(82)$ can be derived using Equations (2) and (6) and taking into account fans powers $P_{m,fans}$ and $P_{x,fans}$ of matched and mixed systems respectively.

$$P_x(82) = (P_m(82)-P_{m,fans})F_c^{0.14} \cdot F_{ex}^{\beta}+P_{x,fans} \tag{A9}$$

Total fan powers, $P_{m,fans}$ and $P_{x,fans}$, are comprised of indoor and outdoor fan powers. Assuming that the matched indoor fan power and the outdoor fan power are each equal to 10% of the matched system total power, deviding both sides of Equation (A9) by $P_m(82)$ and rearranging we obtain:

$$\frac{P_x(82)}{P_m(82)} = 0.8\ F_c^{0.14} \cdot F_{ex}^{\beta} + 0.1 \frac{P_{x,f}}{P_{m,f}} + 0.1 \tag{A10}$$