BENCHMARK PERFORMANCE ANALYSIS OF AN ECM-MODULATED AIR-TO-AIR HEAT PUMP WITH A RECIPROCATING COMPRESSOR

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ABSTRACT

A benchmark analysis was conducted to predict the maximum steady-state performance potential of a near-term modulating residential-size heat pump. Continuously variable-speed, permanent-magnet electronically commutated motors (ECMs) were assumed to modulate the compressor and the indoor and outdoor fans in conjunction with existing modulating reciprocating compressor technology. A modulating heat pump design tool was used to optimize this ECM benchmark heat pump using speed ranges and total heat exchanger sizes per-unit-capacity equivalent to that used by the highest SEER-rated variable-speed unit presently on the market (SEER = 16.4).

Parametric steady-state performance optimization was conducted at a nominal design cooling ambient of 95°F (35°C) and at three off-design ambients of 82°F (27.8°C) cooling and 47°F and 17°F (8.3°C and −8.3°C) heating. In comparison to the reference commercially available residential unit, the analysis for the ECM benchmark predicted steady-state heating COPs about 35% higher and a cooling EER almost 25% higher at the nominal design cooling condition. The cooling EER at 82°F (27.8°C) was 13% higher than that of the reference unit when a comparable sensible heat ratio of 0.71 was maintained, while an EER gain of 24% at the 82°F (27.8°C) rating point was predicted when the sensible heat ratio was relaxed to 0.83.

An optimal control strategy was defined as a function of compressor speed to generate performance maps vs. speed and ambient temperature for the heating and cooling modes. These maps were used to predict the seasonal and annual performance factors of the ECM benchmark case for an 1,800-ft^2 (167.2-m^2) house in a DOE Region IV city, and the results were compared to those from a similar seasonal analysis for three commercially produced variable-speed heat pumps (based on available manufacturers' data). Conventionally sized and 50%-oversized units were considered. For a conventionally sized unit, the ECM benchmark had an SEER 29% higher than that of the highest rated unit available, but the HSPF improvement was only 8%. With a 50%-oversized unit, the predicted HSPF improvement increased to 19%, while the SEER gain remained at 29%. An SEER of 20 appears to be the limit of present modulated reciprocating technology with conventional sizing and default cycling loss factors.

INTRODUCTION

The primary purpose of this work is to evaluate the performance improvement potential of a speed-modulated air-to-air heat pump *with high-efficiency heat exchangers and drives and current reciprocating compressor technology*. This near-term performance benchmark is obtained by a four-point parametric analysis of the most significant design and operating variables using a modulating heat pump design tool (Rice 1988a, 1991). The analysis further serves to demonstrate the use and capabilities of the system design program.

The modulating design tool is a major extension of an earlier single-speed air-to-air heat pump model (Fischer and Rice 1983). The modulating model features a number of improvements and additions to the single-speed version, such as four levels of modulating drive technology for compressors and fans, a range of variable-opening flow-control types, extended air-side heat exchanger correlations for modulating applications, and charge inventory prediction and balancing capability (Rice 1987, 1988a, 1991). Various versions of the two models have been validated against single-speed (Dabiri 1982; Fischer and Rice 1983, 1985; Damasceno et al. 1990; Spatz 1991), two-speed (Fagan et al. 1987), and variable-speed (Miller 1988a) heat pumps. The single-speed model has also been used with reported success in the simulation of variable-speed, engine-driven heat pumps (Fischer 1986; Monahan 1986).

APPROACH AND ASSUMPTIONS

Major Assumptions

The near-term benchmark analysis was conducted under assumptions made to facilitate comparisons with the highest SEER-rated variable-speed unit presently on the market—hereinafter referred to as the state-of-the-art (SOA) reference unit. The major assumptions were as follows:

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- a reciprocating compressor;
- an ECM-driven compressor, indoor blower, and outdoor fan;
- the same compressor speed turndown ratios in heating and cooling; and
- the same total air-side heat exchanger area per unit of design cooling capacity.

ECM Compressor Characterization

Reference Compressor The baseline reciprocating compressor used for the analysis was a present-generation modulating compressor using a three-phase, two-pole induction motor. This compressor was tested over a range of speeds and operating conditions with a variable-frequency sine-wave drive (by use of a motor-generator set). In this way, no direct or indirect inverter drive losses as reported by Miller (1988a) were introduced into the characterization of the baseline compressor performance.

ECM Conversion With this reference-compressor performance map, the modulating heat pump design tool (Rice 1988a, 1991) was used to replace the sine-wave-driven induction motor (SWDIM) with an ECM drive by use of a conversion routine built into the program. This conversion process was required to predict the performance of an ECM-driven reciprocating compressor because there were no publicly available data for such a combination at the time of the study. Performance data obtained from Zigler (1985) on a production two-pole, 2.75-hp (2.05-kW) SWDIM and from Young (1990) on a production four-pole, 3-hp (2.2-kW) ECM as functions of compressor speed and torque were used for this conversion. Also included in the motor models were correction factors for motor temperature effects and an approximate method to adjust for the performance effects of reduced-suction gas superheating with the more efficient ECM motor.

Speed Ranges A speed turndown ratio of 1-0.28 was selected for the ECM benchmark analysis so as to be comparable with that used by the SOA reference unit. The operating speed range selected for this turndown ratio was 5,400 to 1,500 rpm. This was judged to be the most efficient operating range for the selected compressor based on an analysis of the available calorimeter data.

ECM Indoor Blower and Outdoor Fan Characterization

Drive Efficiency and Nominal Speeds For both the indoor blower and outdoor fan modulating drives, an efficiency map obtained from Young (1990) for a 12-pole, 1/5-hp (0.15-kW) production ECM as a function of speed and torque was used. For the indoor blower, a typical nominal speed of 1,080 rpm (usually obtained from a six-pole induction motor) was assumed for compatibility with existing blower sizes. Similarly, an outdoor fan speed of 825 rpm (typically obtained from an eight-pole induction motor) was used.

Blower/Fan Efficiency An indoor blower efficiency of 45% was assumed to be achievable for typical indoor air-side pressure drop and flow requirements. Because blower/fan efficiency remains constant with changes in speed (from the fan laws), a single efficiency specification is sufficient. The outdoor fan efficiency also is assumed to remain constant over the range of fan-motor speeds but is allowed to change with fan *specific* speed, which varies inversely with the system pressure drop characteristic (AMCA 1973). This changing efficiency is accomplished through an algorithm built into the model (Fischer and Rice 1983) and reflects the more limited range of efficiency potential for lower pressure-drop configurations typical of outdoor units.

External Pressure Drop As specified by ARI Standard 210/240 (ARI 1989), the external (duct-only) pressure drop was set at 0.15 in. of water (37.4 Pa) at *nominal* indoor airflow conditions. Also included in the total air-side pressure drop were typical values for electric heater and filter pressure drop as a function of airflow (Fischer and Rice 1983).

Heat Exchanger Geometry, Sizing, and Augmentation

For convenience and consistency with a previous modulating system analysis by Rice and Fischer (1985), the heat exchanger geometric details (such as tube-and-fin spacing and tube sizes) of a first-generation commercially available unit were used for the benchmark heat pump. The total air-side surface area of the first-generation unit geometry was scaled up in size to be consistent with that of the SOA reference unit on a per-unit-of-nominal-capacity basis. This normalized sizing was maintained as a hardware constraint.

The air-side heat exchanger surfaces were assumed to be louvered, with pressure drop and heat transfer multipliers applied to baseline correlations as described by Fischer and Rice (1983). The baseline correlations, however, were updated (Rice 1988a, 1991) to reflect more accurate representations over a wide range of airflow rates (Gray and Webb 1986). The refrigerant-side heat exchanger surfaces were assumed to be internally augmented to provide an average increase in the heat transfer coefficient of 50%, with a corresponding 50% increase in the refrigerant-side pressure drop relative to the baseline correlations used by Fischer and Rice (1983). Because state-of-the-art heat pumps have various degrees of internal and external augmentation, these assumptions were made to approximate the upper limit of current heat exchanger technology.

Refrigerant Flow Control

With regard to refrigerant flow control, charge insensitivity was assumed, which requires that a sufficiently large accumulator or other charge storage reservoir be present in the refrigeration loop. The compressor inlet superheat was specified at a constant 10 F° (5.55 C°)[1] in the cooling mode and a constant 1 F° (0.55 C°) in the heating mode. The lower superheat value was used in heating to avoid a pinch point between the ambient temperature entering the evaporator and the superheated refrigerant temperature at the evaporator exit, which would have unnecessarily limited COP and capacity by 5% to 10%.

Condenser exit subcooling was controlled directly rather than by specifying a type of flow-control device. This approach allows the thermodynamically optimum flow control to be determined independently of the characteristics of a certain valve type.

Refrigerant Line and Reversing Valve Losses

Refrigerant line heat transfer losses were assumed to be zero for this analysis. Heat transfer, pressure drop, and refrigerant leakage losses in the reversing valve also were not included.

Dabiri (1982) has calculated that a typical discharge line loss could lower heating capacity by an average of 2.5% and system COP by about 2%. Cooling mode discharge line losses should be negligible because, in cooling, the heat loss is not a reduction in heat pump output. Liquid line losses were also shown by Dabiri (1982) to be generally negligible.

Estimates of the typical system overprediction incurred by neglecting reversing valve losses have been made by Krishnan (1986) and, more recently, in a survey paper by Damasceno et al. (1991a). Krishnan found system losses ranging from 4.0% to 5.5% in system capacity and 4.0% to 6.0% in EER for three valve brands in a typical design cooling condition. Damasceno computed system losses for three valves averaging 2.5% in capacity and 1.7% in EER in the cooling mode and 2.1% in capacity and 3.4% in COP in the heating mode.

Four-Point Design Analysis

Design Ambients and Speeds The approach taken to design a high-efficiency, variable-speed benchmark unit was to optimize the steady-state COP at four design-point ambients. The compressor speed and system capacity requirements at each of these ambients were assumed as follows:

Cooling Mode:

95°F (35°C)	—	Maximum speed, specified design capacity, acceptable sensible-to-total (S/T) capacity ratio;[2]
82°F (27.8°C)	—	Minimum speed, minimum capacity, acceptable S/T ratio;

Heating Mode:

47°F (8.3°C)	—	Minimum speed, minimum capacity, acceptable minimum supply temperature;
17°F (−8.3°C)	—	Maximum speed, maximum capacity.

For the usual sizing strategies, the compressor should be operating at or close to the assumed minimum or maximum speeds at the chosen design ambients. Standard ARI rating conditions (ARI 1989) were assumed at all ambients, with 80°F DB/67°F WB (26.7°C DB/19.4°C WB) indoor air in cooling and 70°F DB/60°F WB (21.1°C DB/15.6°C WB) in heating.

Configuration and Operation Optimization Strategy The four-point design optimization approach was further divided into a nominal design-point analysis and three off-design-point optimizations. The nominal design point is the 95°F (35°C) ambient, maximum compressor speed condition in the cooling mode with a specified design capacity requirement. The majority of the benchmark system component sizes and configurations were determined at this condition. The off-design analysis then was used to obtain an optimal bimodal operating strategy with speed and ambient temperature. A contour-data-generating version of the modulating heat pump design tool (named MODCON), containing a front-end for automated parametric analysis (Rice 1988a, 1991), was used to evaluate the optimum system requirements at each design point.

Relation to Seasonal Performance Analysis The maximized steady-state COPs and EERs obtained from these design points next were used with the determined control strategy to obtain compressor speed vs. ambient temperature performance maps in both the heating and cooling modes. These performance maps were used to predict seasonal and annual performance factors for the ECM benchmark heat pump in a representative DOE Region IV city—Columbus, Ohio. The seasonal results then were compared with those from a similar analysis conducted (based on available manufacturers' data) on three continuously modulated heat pumps.

[1]The notations for temperatures used herein are °F (°C) for temperature values and F° (C°) for temperature differences.

[2]Note that the S/T capacity ratio as used here is equivalent to the sensible heat ratio (SHR) or the sensible heat factor (SHF).

NOMINAL DESIGN-POINT ANALYSIS

Nominal Design-Point Assumptions

The following requirements were specified for the 95°F (35°C) design-point cooling condition:

- design capacity of 2½ tons (8.8 kW) cooling,
- compressor motor sized to operate at 136% of rated power at nominal speed,
- fan motors sized to operate at 75% of rated power at nominal speed,
- external pressure drop of 0.15 in. of water (37.4 Pa), and
- 10 F° (5.55 C°) compressor inlet superheat.

Nominal Design-Point Variables

The following design variables were optimized at the nominal design cooling condition:

- compressor displacement,
- nominal indoor and outdoor airflow,
- indoor fraction of total air-side heat exchanger area,
- number of coil rows and refrigerant circuits—indoor and outdoor, and
- condenser subcooling.

Automatic Motor Sizing Because compressor displacement and the nominal airflow values are design parameters *at the nominal condition*, the related compressor, indoor blower, and outdoor fan-motor sizes need to be adjusted appropriately to maintain constant drive efficiencies. The modulating design tool program was designed to allow the user the option of specifying a desired sizing criterion (as given above under "Nominal Design-Point Assumptions") for each motor and of having the program calculate the required motor size.

Nominal Design Methodology

A parametric evaluation was conducted (with two variables at a time) using MODCON to determine the optimum *nominal* system hardware configuration and operating conditions. The methodology adopted was to optimize the stronger design variables in pairs, starting with those having the greatest effect on capacity.

Compressor Displacement vs. Nominal Indoor Airflow Rate The compressor displacement and nominal indoor airflow were varied to find the maximum EER at a cooling capacity of 2½ tons (8.8 kW). The resultant contours of constant EER and capacity are shown in Figure 1. There, as denoted by the "x," a maximum capacity-constrained EER of 12.27 was found at a displacement of 1.59 in.^{3} (26.1 mL) with an indoor airflow rate of 925 cfm (437 L/s). Note that with a fixed total heat exchanger area, the unconstrained EER increases in the direction of smaller unit capacity.

Range of Parametric Evaluations Contour data sets also were generated for other appropriate variable pairs as follows:

- nominal outdoor airflow rate vs. compressor displacement,

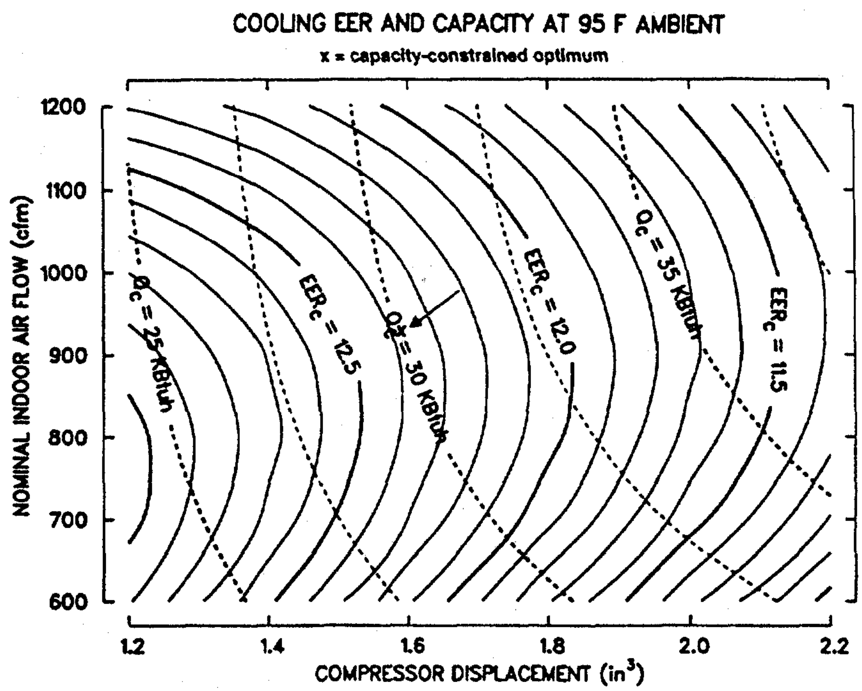

Figure 1 *95°F cooling mode—EER and capacity (Q_C) contours vs. compressor displacement and nominal indoor airflow rate.*

- nominal indoor vs. outdoor airflow rate,
- nominal indoor airflow rate vs. indoor area fraction,
- nominal outdoor airflow rate vs. condenser subcooling,
- nominal indoor airflow rate vs. number of indoor rows,
- nominal outdoor airflow rate vs. number of outdoor rows,
- number of indoor vs. outdoor refrigerant circuits.

All pairs except for the last three were considered interdependent variables; the last three pairs were found to be weakly interacting.

Interdependent Variables Contour data sets containing the most interdependent design variables—compressor displacement, nominal indoor and outdoor airflow, condenser subcooling, and indoor area fraction—were manually iterated from two to at most three times, successively updating the optimum values found for each of these five variables.

Weakly Interacting Variables For such variables as the integer number of coil rows, it was possible to choose the optimum values from one contour data set. The number of indoor rows was limited to four for indoor-cabinet-size considerations, although a small increase in COP was predicted with fewer rows. For the outdoor coil, a clear peak in COP with three rows was evident, with a 9% drop in COP predicted for a one-row coil.

We set the number of refrigerant circuits at values large enough to minimize COP penalties at the maximum refrigerant flow conditions, while keeping in perspective the need to maintain sufficient refrigerant velocities under low-speed operation to avoid significant refrigerant-side heat transfer degradation. Four circuits in the indoor coil and three in the outdoor coil were found to be a good compromise for the enhanced-tube geometry.

Optimum Design Cooling Performance An acceptable *nominal* S/T capacity ratio of 0.76 was obtained at the total-capacity-constrained point of maximum EER with an evaporator exit saturation temperature of 52.5°F (11.4°C) and a supply air temperature of 57°F (13.9°C) with 98.5% RH.

Optimum Hx Area Fraction Figure 2 shows a contour plot of EER and cooling capacity as a function of nominal indoor airflow and indoor coil area fraction. The indoor area fraction (which is defined in MODCON as the ratio of the frontal area times the number of rows times the fin pitch for the indoor coil divided by the sum of the same product for both coils) was found to be optimum at a value of 0.45, as denoted by the "*x*." Because of differences in coil tube-and-fin spacings between the indoor and outdoor coils, the related, geometry-independent *air-side surface area fraction* for the indoor coil is 0.39.

Relation to Off-Design-Point Analysis Once the major *nominal sizing and configuration* parameters were determined, the analysis shifted to the evaluation of the best operating conditions at low-speed cooling and at both low- and high-speed heating conditions. At this stage, it was not known whether or not the operational requirements for best performance at any of the remaining design points might exceed the nominal values selected so far, especially with regard to high-speed heating operation. Had this been the case, the affected components would

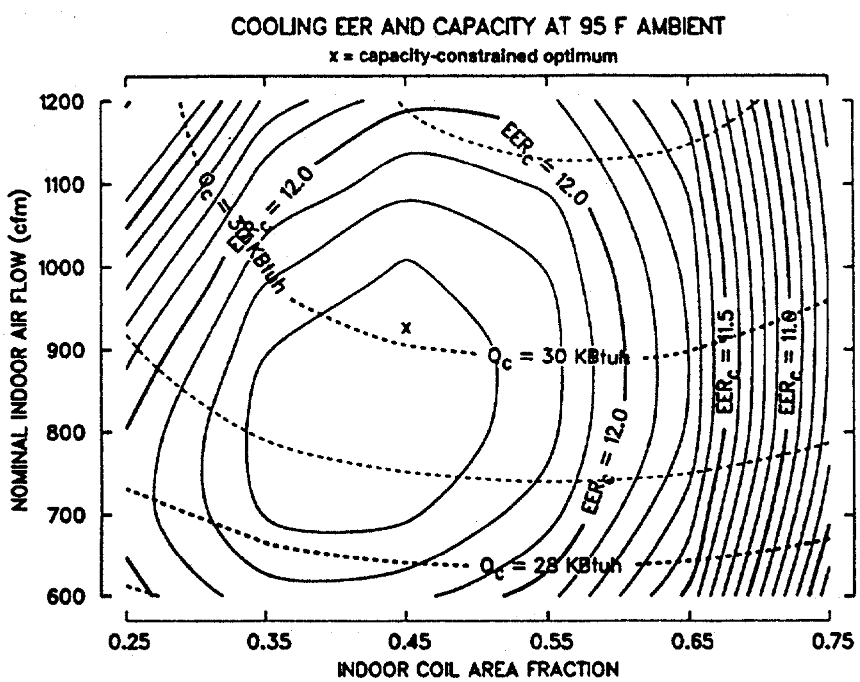

Figure 2 *95°F cooling mode—EER and capacity (Q_C) contours vs. indoor-coil area-fraction and nominal indoor airflow rate.*

have had to be resized and the preceding analysis repeated.

OFF-NOMINAL-DESIGN ANALYSIS

Off-nominal-design performance optimizations were conducted at 82°F (27.8°C) ambient, low-speed cooling conditions and at 47°F and 17°F (8.3°C and −8.3°C) ambient, low- and high-speed heating operation, respectively.

Fixed Parameters

Based on the nominal cooling design analysis, the following system parameters were fixed for the off-design analysis:

- compressor displacement,
- compressor motor size,
- *nominal* indoor and outdoor airflow,
- indoor blower and outdoor fan-motor sizes,
- indoor duct size,
- indoor fraction of total area, and
- number of coil rows and refrigerant circuits—indoor and outdoor.

Drive Efficiency Calculations Because the compressor displacement, the compressor and fan-motor sizes, and the nominal airflows have been fixed, the nominal operation points on the drive efficiency maps in MODCON have been established. Therefore, as compressor and fan speeds are modulated (by changing the frequency of the power supplied to the motors) and as the motor torque requirements change, the drive efficiencies are properly adjusted using the built-in efficiency vs. speed and torque characteristics.

The required indoor duct size was determined from the required pressure drop of 0.15 in. of water (37.4 Pa) at the optimum *nominal* airflow of 925 cfm (437 L/s). At off-design conditions, the use of the fixed duct size allows the duct pressure drop to reduce appropriately at off-nominal airflow conditions. This, in turn, provides the proper torque values for the indoor fan efficiency evaluation.

Operational Design Variables With the nominal parameters fixed, the optimization problem shifts to a more narrowly defined operational question. The operational design variables considered were the following:

- compressor inlet superheat,
- condenser subcooling, and
- indoor and outdoor fan frequency ratio.

For the cooling mode, the compressor inlet superheat was held constant at 10 F° (5.55 C°), while in the heating mode a constant value of 1 F° (0.55 C°) was used. This latter value was used because (as noted earlier) a pinch point was observed in heating when 10 F° superheat was required. This pinch point occurred because the evaporator saturation temperature is much closer to the evaporator inlet air temperature (10 F° or less) in the heating mode than in the cooling mode, where the entering air temperature is 25 F° to 30 F° higher.

Adjustment of condenser subcooling at different ambients and compressor speeds provides for optimum refrigerant flow control and thermodynamic system balance. Through direct control of condenser subcooling, the cycle optimum flow-control needs can be identified without imposing the constraints of a specific flow-control type.

The indoor and outdoor fan frequency ratios are defined such that at a frequency ratio of 1.0 the nominal airflow is obtained. Because ECM motors operate at synchronous speed (with no motor slip), the fan speed ratio is equivalent to the fan frequency ratio. Further, from the fan laws, for a fixed system pressure drop characteristic, the fan speed ratio is equivalent to the airflow ratio. Indoor airflow control is an effective way to control S/T cooling ratios and supply air temperature in the cooling and heating modes. Outdoor airflow control offers both efficiency improvement and noise-reduction advantages.

Low-Speed Cooling, 82°F (27.8°C) Ambient

Contour data sets were generated for low-speed cooling for the operational-variable pairs of indoor vs. outdoor fan frequency ratio and condenser subcooling vs. the outdoor fan frequency ratio. Maximum EER values were obtained at two levels of S/T ratio—one level of 0.71, which was the same as that provided by the SOA reference unit, and a higher level deemed to be more representative of the average S/T that would be required in Columbus, Ohio, at milder ambients. This use of two levels was based on consideration of the average S/T ratios calculated as a function of ambient temperature by a binned seasonal performance model (Rice et al. 1985).

Consideration of Relaxed S/T Constraint The rationale for considering the higher S/T ratio case is twofold. First, an S/T ratio of near 0.70 historically has been the design point for single-speed heat pumps. Because these units cycle at around 50% load factor at the 82°F (27.8°C) ambient, the *effective average* S/T ratio delivered by such units is certainly higher, perhaps around 0.80. Second, a variable-speed unit has the capability of adjusting the S/T cooling capacity ratio by raising the compressor speed or by lowering the indoor airflow when provided with an RH-discriminating signal from a humidistat. Such devices are presently offered as optional equipment on continuously variable and two-speed heat pumps. With such a device as *standard* equipment, a variable-speed heat pump could perhaps be designed to meet a higher average S/T ratio and thereby a higher baseline efficiency, with the humidistat control serving to lower the S/T ratio as needed under peak humidity conditions and in more humid climates.

Performance vs. Compressor and Indoor Blower Speeds Once the optimum airflows and condenser subcooling values were found, contour data sets were generated for 82°F (27.8°C) system performance as a function of compressor and indoor blower frequency ratios. Contour plots of cooling EER, S/T capacity ratios, and (total) capacity were generated from these data sets (Figures 3 through 5, respectively) to show a broad picture of how mild-ambient cooling performance is affected by these control variables.

In Figure 3, the cooling EER is shown to be a strong function of compressor speed and a rather weak function of indoor blower speed. In Figure 4, the S/T ratio is

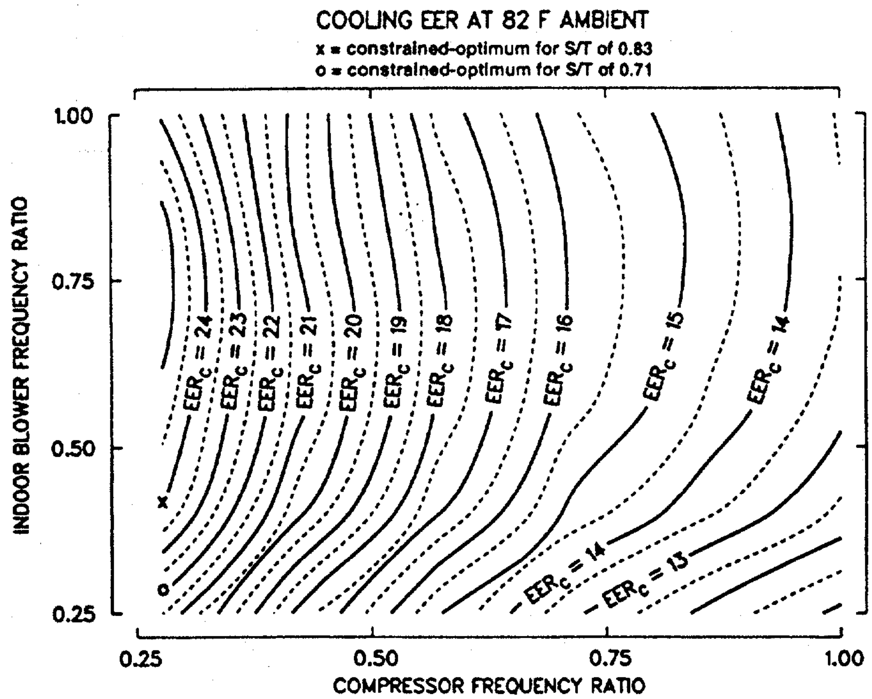

Figure 3 *82°F cooling mode—EER contours vs. compressor and indoor blower frequency ratios.*

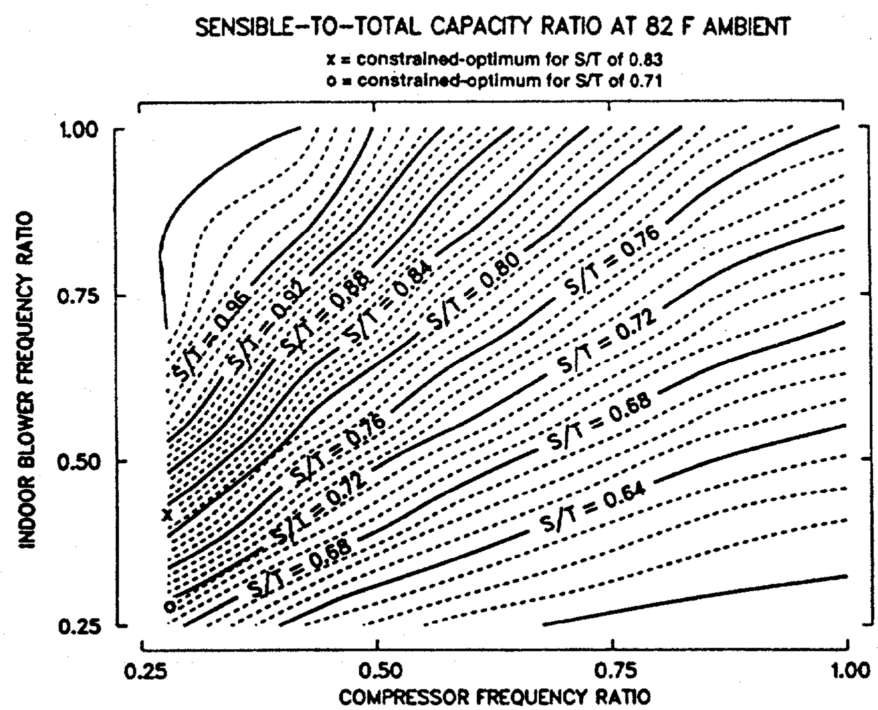

Figure 4 *82°F cooling mode—S/T capacity ratio contours vs. compressor and indoor blower frequency ratios.*

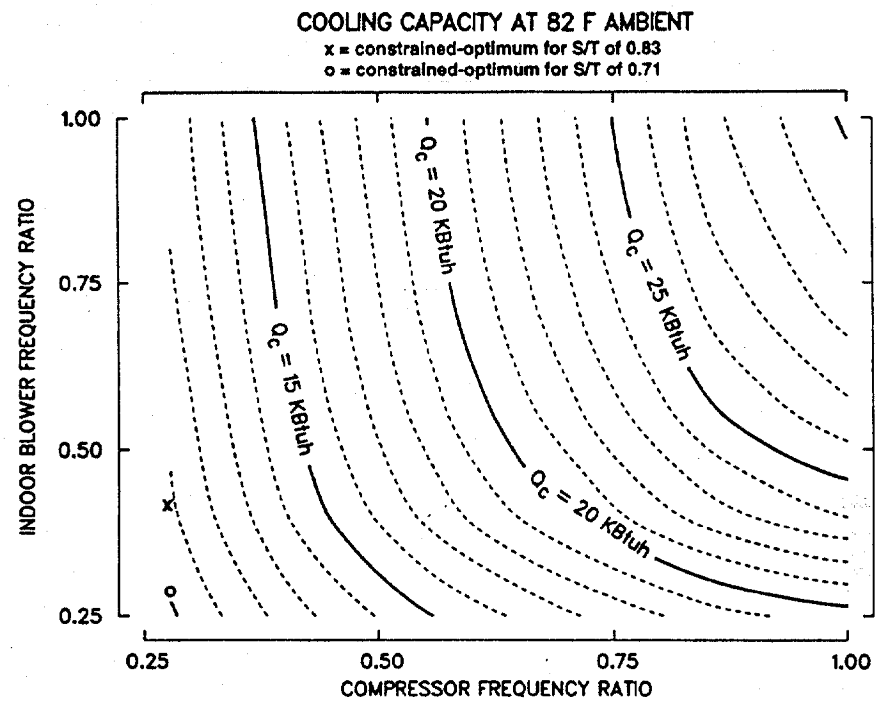

Figure 5 *82°F cooling mode—capacity* (Q_C) *contours vs. compressor and indoor blower frequency ratios.*

shown to be a stronger function of airflow rate with a moderate dependence on compressor speed. These figures suggest that both variables could be used to provide a wide range of S/T control. The effect on EER of relaxing the S/T ratio design requirement from 0.71, as denoted by the "o," to 0.83, as denoted by the "x," is evident in Figures 3 and 4, where the EER increases from 21.8 to 23.9. The total cooling capacity (Figure 5) shows that a wide range of capacity values is possible at 82°F (27.8°C), as compared with single-speed heat pump performance given by compressor and indoor frequency ratios of 1.0.

Low-Speed Heating, 47°F (8.3°C) Ambient

Performance vs. Compressor and Indoor Blower Speeds A similar three-operational-variable optimization was performed for low-speed heating. Figures 6 and 7 provide contour plots of heating COP and indoor supply air temperature. The mild-ambient heating COP is shown to be a slightly stronger function of airflow rate than is the cooling COP.

Indoor Supply Air Temperature Constraint A lower limit on indoor supply air temperature of 84.5°F (29.2°C), the same as for the SOA benchmark, was imposed on the ECM benchmark. From Figures 6 and 7, it is evident that this supply temperature constraint, as denoted by the "x," is just slightly above the optimum temperature from an efficiency-only perspective. Further analysis could be conducted based on the contour mappings to determine what loss in efficiency would result if warmer supply temperatures were required. Control strategy options for maintaining warmer supply temperatures in variable-speed heat pumps have been discussed by Sulfstede (1990).

High-Speed Heating, 17°F (−8.3°C) Ambient

The optimum COP point found at the 17°F (−8.3°C) high-speed heating condition from the three-variable optimization gave a supply air temperature of about 87°F (30.6°C) (before the addition of any required resistance heat), a value also close to that for the SOA reference unit. No plots of performance as a function of compressor speed were generated at 17°F (−8.3°C) because this ambient would almost always be below the balance point, and only the highest-speed operation is expected.

STEADY-STATE COMPARATIVE ANALYSIS

Efficiency and Capacity Trends

The resultant steady-state COPs, EERs, and capacity values for the four-point-optimized benchmark case were tabulated and compared to three commercially produced, variable-speed heat pumps (the SOA reference unit, an SOA alternative unit, and a first-generation unit) and to the laboratory-modified first-generation unit tested by Miller (1987). Nominal capacities for the comparative units ranged from 2.5 to 3 tons (8.8 to 10.6 kW).

Steady-State COP, EER, and Relative Capacity vs. Ambient In Figures 8 and 9, comparative data on heating COP, cooling EER, and fraction of nominal *cooling* capacity are plotted vs. ambient temperature. The

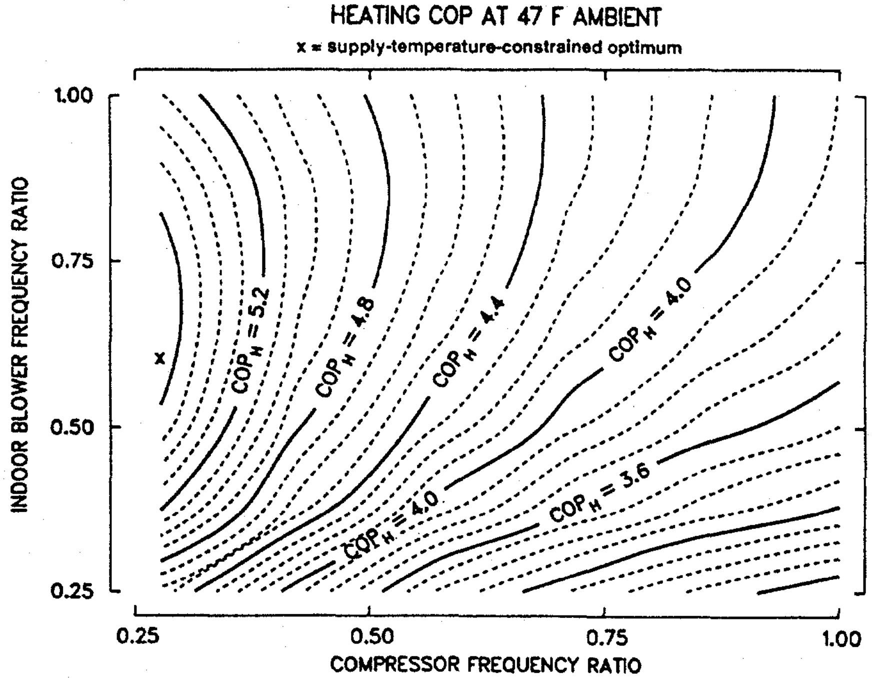

Figure 6 *47°F heating mode—COP contours vs. compressor and indoor blower frequency ratios.*

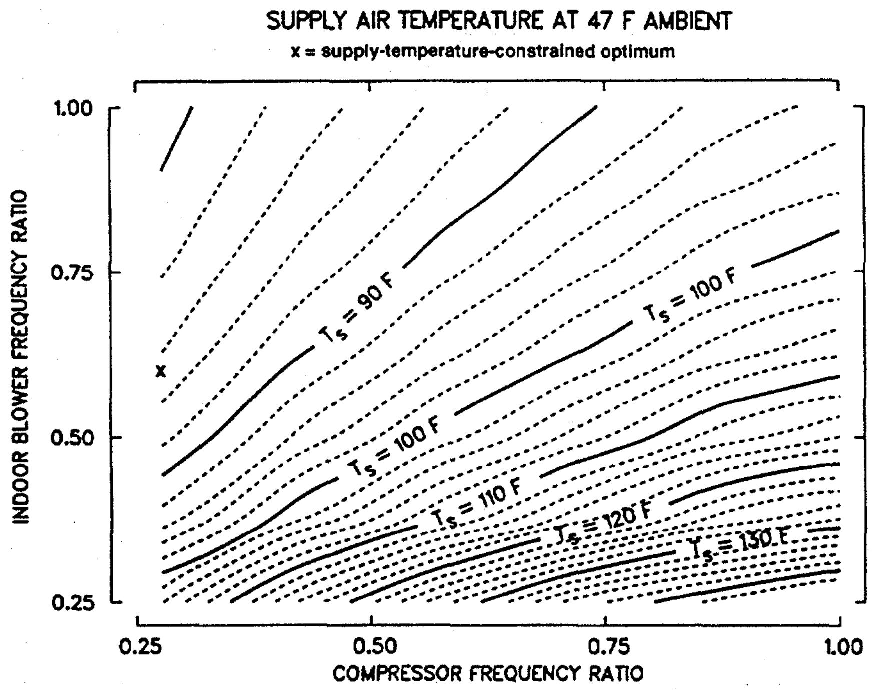

Figure 7 *47°F heating mode—supply air temperature* (T_S) *contours vs. compressor and indoor blower frequency ratios.*

data for each ambient temperature were taken at the most likely application speeds. The plots are intended primarily for comparative purposes at the ambients for which data are shown, as the lines connecting the points only coarsely represent the operating COP and capacity as a function of ambient. (More accurate data of this type would be obtained from the binned performance tabulations from a seasonal performance analysis for a specific house and climate.) However, from these simplistic plots, the following observations can be made:

- The SOA reference unit has the highest steady-state-efficiency values of all the commercially produced, variable-speed equipment considered.
- The heating COP of the ECM benchmark is significantly higher than that of the SOA refer-

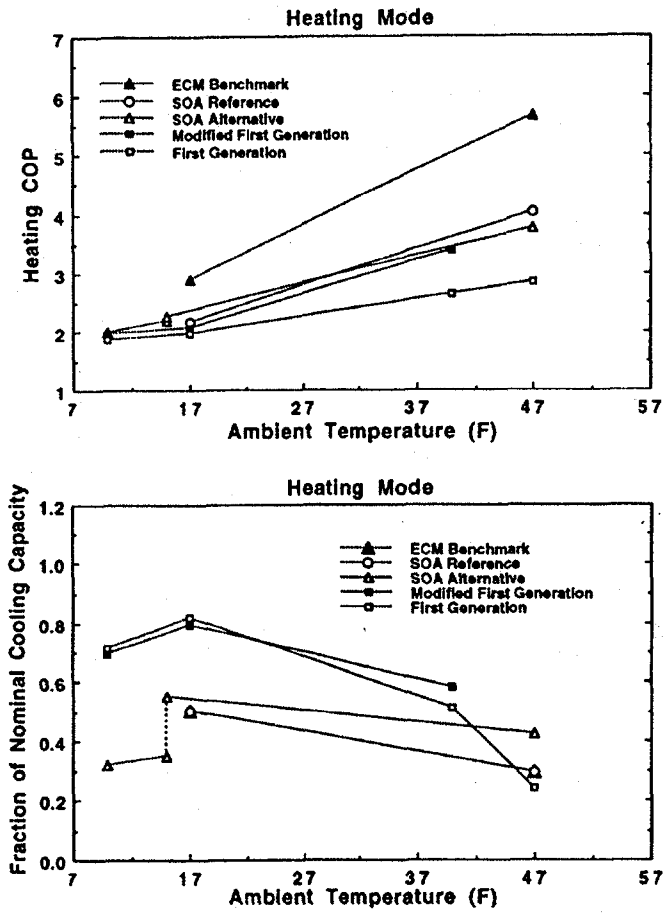

Figure 8 *Comparison of heating COP and fraction of nominal capacity vs. ambient temperature between four existing modulating heat pumps and the ECM benchmark.*

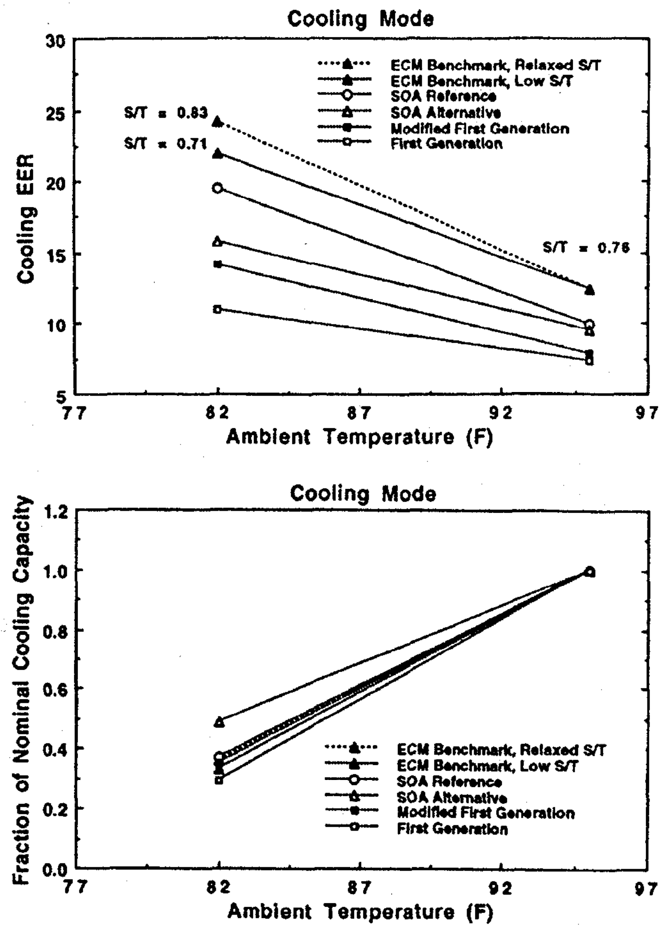

Figure 9 *Comparison of cooling EER and fraction of nominal capacity vs. ambient temperature between four existing modulating heat pumps and the ECM benchmark.*

ence at both extreme and mild ambients, although both have almost identical normalized[3] heating capacity levels and trends. The similarity in normalized capacities continues in the cooling mode, where the predicted EER advantage is smaller, especially if the same S/T ratio (of 0.71 at the 82°F [27.8°C] condition) as the SOA reference is maintained.

- The normalized heating capacity of the SOA alternative unit is about 10% higher than the capacities of the SOA reference and ECM benchmark units at ambients above 15°F (−9.4°C). The SOA alternative unit does not run at the highest speed below 15°F (the basis for the dotted lines in Figure 8) and has a turndown ratio of 1-0.36, as compared to the 1-0.28 ratio used in the ECM benchmark and SOA reference units.
- The first-generation units have a significantly higher heating capacity at the lower ambients. This is because of 50% overspeed operation in heating relative to the maximum cooling speed (turndown ratios of 1.52-0.21 in heating and 1.0-0.21 in cooling).

Steady-State COP and EER at Rating Conditions

In Table 1, the steady-state COP and EER comparisons between the ECM benchmark and the four existing modulating heat pumps are given at the conventional four-point rating conditions. The speed ratios and relative heat exchanger area used by the various units are provided as well. The ECM benchmark analysis predicts possible steady-state heating COPs 33% to 41% higher than that of the SOA reference unit. Cooling EERs about 24% higher than that of the SOA reference unit are predicted at the nominal cooling design point of 95°F (35°C), with about a 13% increase predicted at 82°F (27.8°C) when comparable S/T ratios are maintained. When the S/T ratio at 82°F is relaxed from 0.71 to 0.83, an EER gain nearly as large as at the 95°F condition is predicted.

Component Efficiencies and Sizes

The associated component efficiencies calculated by the MODCON program are given in Table 2. Both the

[3]All capacity values were normalized by the nominal *cooling* capacity of the individual heat pumps.

TABLE 1
Steady-State COP and EER Comparisons[a]
Between the ECM Benchmark and Existing Modulating Heat Pumps

Heat Pump	Speed Range	Heating COP		Cooling EER	
		17°F	47°F	82°F	95°F
ECM Benchmark[b,c]	1 – 0.28 c,h[d]	2.89	5.69	22.0 (24.2)[e]	12.4
[Percentage Increase from SOA Reference]	- - -	[+33.2%]	[+40.8%]	[+12.8%] [+24.1%][e]	[+24.7%]
SOA Reference[c]	1 – 0.28 c,h	2.17	4.04	19.5	9.94
SOA Alternative[f]	1 – 0.36 c,h	2.27	3.77	15.9	9.53
Modified First-Generation[b]	1 – 0.21 c 1.52 – 0.21 h	2.07	3.8[g]	14.1	7.85
First-Generation[b]	1 – 0.21 c 1.52 – 0.21 h	1.97	2.83	11.0	7.37
Speed Setting	- - -	Max	Min	Min	Max

[a]Indoor conditions of 70°F DB/60°F WB heating and 80°F DB/67°F WB cooling.
[b]Units have the same reciprocating compressor (or nearly equivalent in the ECM case) but different variable-speed drives (and motor for the ECM case).
[c]Total Hx area on these units was 26% greater than on the first-generation units.
[d]c = cooling, h = heating.
[e]If S/T constraint is relaxed from 0.71 to 0.83.
[f]Relative total area not available.
[g]Extrapolation from 40°F.

Conversion Table from IP to SI Units

To Convert	Multiply by	To Obtain
cfm	0.472	L/s
°F	(°F – 32) × 5/9	°C
F° (R)	5/9	C° (K)
ft^2	0.0929	m^2
horsepower	0.746	kW
$in.^3$ (volume)	16.4	mL
in. H_2O	249	Pa
KBtuh	0.293	kW
lbm	0.454	kg
ton	3.52	kW

modulating drive efficiencies (combined motor and inverter), η_d, and the overall compressor and pump/fan efficiencies, η_o, are provided. The compressor overall isentropic efficiency, η_o, is shown to stay in the upper 50s except for the lower ambient heating conditions. The blower and fan drive efficiencies are seen to drop much more than the compressor efficiency as speed is reduced. This is because the compressor provides more of a constant torque load for the motor, while the fan load drops with the square of the speed change ratio (Rice 1988b). Even with combined efficiencies decreasing to between 20% and 30%, the fan powers are reduced from nominal values of 110 to 150 watts to values of 20 to 35 watts, respectively. This large reduction results because ideal fan power drops with the cube of the speed ratio, which is much faster than the offsetting drive efficiency decreases.

The nominal compressor motor size required (per unit of nominal cooling capacity) for the ECM benchmark is about one-third less than that for the SOA reference unit. The motor size calculated for the indoor blower is less than half the size used in the SOA reference unit, while the outdoor motor is of comparable size. These calculated size requirements are based on the nominal-condition sizing assumptions noted earlier in this paper.

STEADY-STATE PERFORMANCE MAPPING OVER AMBIENT AND COMPRESSOR SPEED RANGES

To evaluate most accurately the seasonal performance of a modulating heat pump, the performance of the unit must be mapped over the full range of ambient temperature and compressor speeds. This requirement was completed with the aid of the MODCON program.

Optimal Control Variables as a Function of Compressor Speed

Control Strategy With the optimum operating conditions determined at the four design points, a control strategy was needed to tie the operational values of indoor and outdoor airflow ratio and condenser subcooling to an independent control variable. The compressor speed ratio was chosen for this purpose because the operational variables are more strongly a function of compressor speed than of ambient temperature. (A more refined

TABLE 2
Compressor/Fan Drive and Overall Efficiencies, η_d and η_o, Predicted for the ECM Benchmark

	Efficiencies (%)					
Ambient	Indoor Blower		Outdoor Fan		Compressor	
	η_d	η_o	η_d	η_o	η_d	η_o
Heating Mode						
17°F	78.4	35.3	72.3	31.1	89.4	47.5
47°F	63.9	28.8	57.7	25.0	78.0	57.5
Cooling Mode						
82°F	49.7	22.4	69.1	28.6	78.2	59.7
95°F	81.4	36.6	78.8	30.8	89.7	56.2

control algorithm could include both compressor speed and ambient temperature as independent variables but would require operational optimizations at more ambient temperatures and/or speed levels in both the heating and cooling modes.) The MODCON program presently is designed to accept single-variable control functions as a function of ambient temperature or compressor speed ratio. The latter option was used for this analysis, where two-point linear functions were defined in both the heating and cooling modes.

Control Values Table 3 lists the absolute and relative control values used for the speed/ambient mapping. The optimum control values at the low-speed cooling condition are given for S/T values of 0.71 and 0.83, although for the rest of the mapping and seasonal analyses only the higher S/T case was considered (where the S/T capacity ratio ranges from 0.83 at mild-ambient conditions to 0.76 at the design cooling condition).

The optimum condenser subcooling is shown in Table 3 to range from highs of 15 F° to 24 F° (8.3 C° to 13.3 C°) at the high-speed conditions in cooling and heating, respectively, to lows of 5 F° to 10 F° (2.8 C° to 5.6 C°) at the low speeds. The indoor and outdoor airflow ratios exhibit a much wider range of required airflow in the cooling mode than in the heating, with the outdoor airflow in heating ranging only from 45% to 67% of the nominal cooling values.

Generation of Speed vs. Ambient Performance Data

Performance Mapping With the control strategy defined for the ECM benchmark case, the MODCON program was run in the heating and cooling modes over grids of ambient temperature (from 7°F to 57°F [−13.9°C to 13.9°C] ambients in heating and from 67°F to 97°F [19.4°C to 36.1°C] ambients in cooling) and compressor speed (from the 1,500 rpm minimum to the 5,400 rpm maximum [nominal] value).

From these computer runs, contour data sets of selected dependent variables[4] were generated for both the heating and cooling modes. Contour plots of heating COP, capacity, and supply air temperature are shown in Figures 10 through 12, respectively, where an "x" and an "o" reference the location of the high- and low-speed steady-state rating points, respectively, at 17°F and 47°F

[4]More than 100 dependent parameters can be user-selected for data set output. These data sets can be processed later to analyze heat pump performance in greater detail, including component operating conditions and efficiencies.

TABLE 3
Optimal Control Values for the ECM Benchmark

	Heating			Cooling		
	17°F	47°F		82°F		95°F
Control Variables			S/T =	0.71	0.83	
Compressor Speed (rpm)	5400	1500		1500	1500	5400
Compressor Speed Ratio	1	0.28		0.28	0.28	1
Condenser Subcooling (F°)	24	10		5	5	15
Indoor Airflow (cfm)	815	555		260	390	925
Indoor Blower Airflow Ratio	0.88	0.6		0.28	0.42	1
Outdoor Airflow (cfm)	1675	1125		1400	1600	2500
Outdoor Fan Airflow Ratio	0.67	0.45		0.56	0.64	1

(−8.3°C and 8.3°C). Analogous plots for cooling EER, capacity, and S/T ratio also were developed but are not shown in this paper.

Figures 10 through 12 (and the analogous cooling plots) represent a full mapping of the expected performance of the ECM benchmark heat pump. Load lines for a specific house and climate can be plotted on the capacity contours as functions of ambient temperature to show the operating lines for a given application. These particular operating lines then can be overlaid on the remaining generalized plots (as done by Rice [1988b]) to define the specific operating COP, supply air temperature, or S/T ratios for a given house and climate. The seasonal analysis that follows is a computerized implementation of this process whereby the appropriate operating speed and COP are determined for each ambient temperature bin.

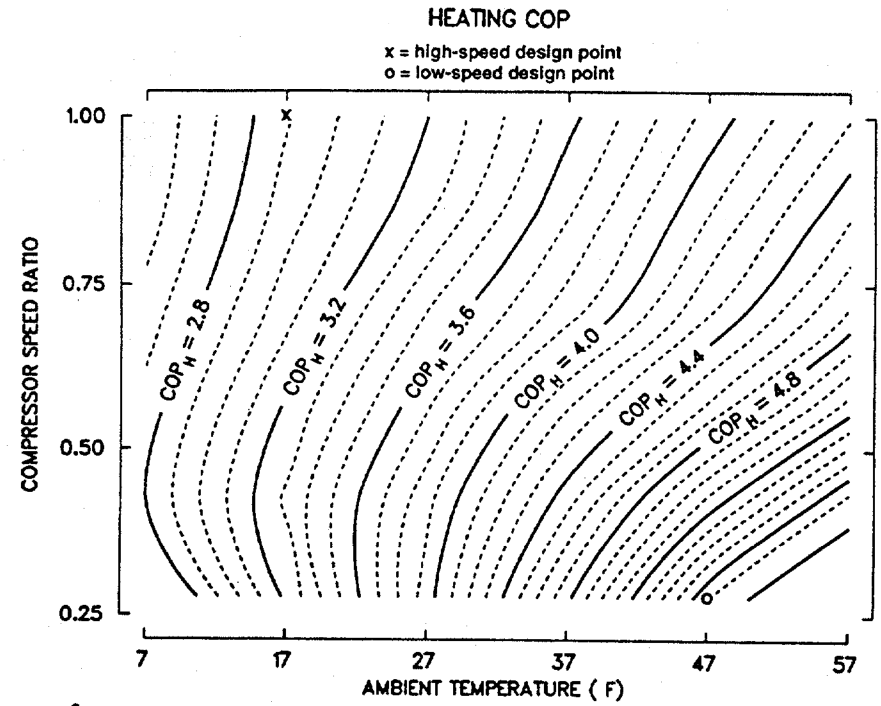

Figure 10 Heating performance mapping of ECM benchmark—COP vs. ambient temperature and compressor speed ratio.

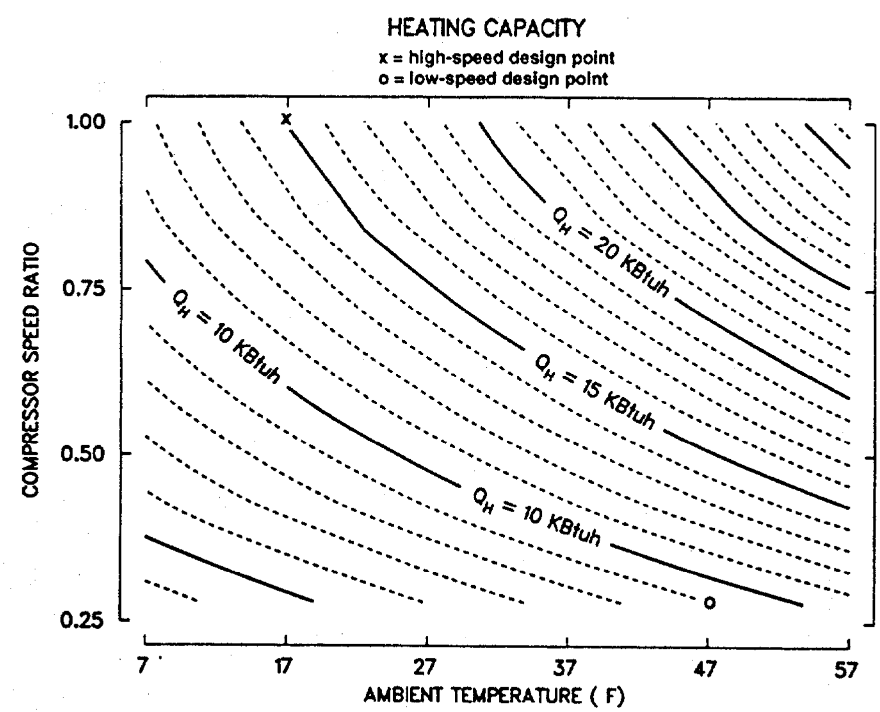

Figure 11 Heating performance mapping of ECM benchmark—capacity (Q_H) vs. ambient temperature and compressor speed ratio.

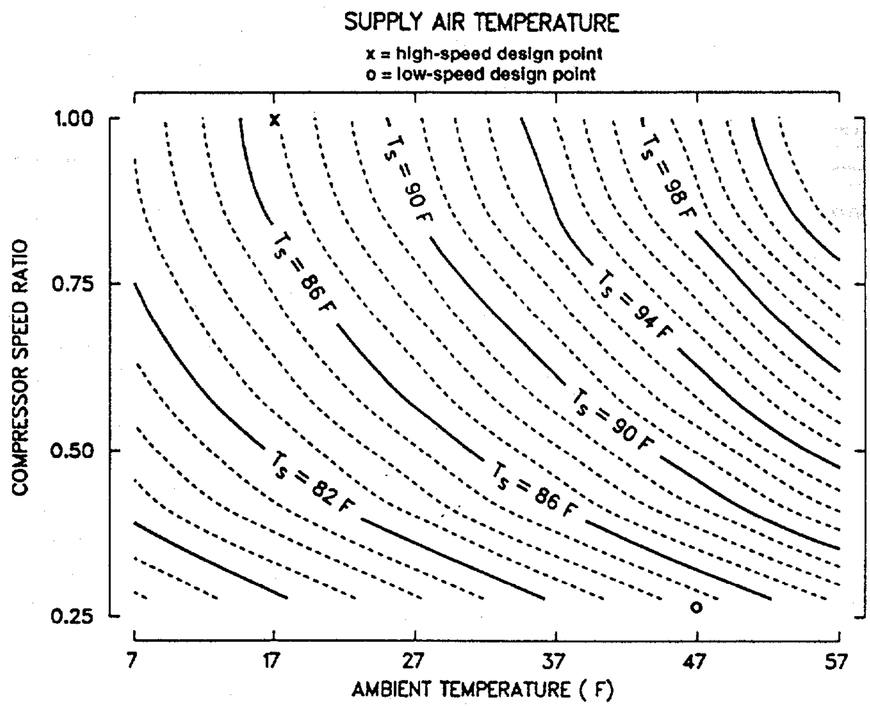

Figure 12 *Heating performance mapping of ECM benchmark—supply air temperature* (T_S) *vs. ambient temperature and compressor speed ratio.*

Optimal Refrigerant Charge Levels for a Modulating Heat Pump

Figures 13 and 14 represent the calculated refrigerant charge required to obtain the optimal refrigerant operating conditions. The Hughmark refrigerant inventory method was employed as discussed by Rice (1987) and recently was used with good agreement by Damasceno et al. (1991b). This method is especially appropriate for use in a modulating heat pump analysis because it includes the effects of varying mass flow rate.

The estimated required active charge of R-22 is shown to have lows of about 9.5 to 9.6 lbm (4.3 kg) for intermediate-speed, mild-ambient heating and cooling, while increasing slowly to between 9.8 and 10 lbm (4.4 and 4.5 kg) at the lowest and highest speeds in both modes. Only at the extreme ambient conditions in the heating mode do the requirements increase to more than 11 lbm (5 kg). The heating mode shows the widest variation in required charge; the rapidly increasing charge requirements below 17°F (−8.3°C) suggest that the condenser subcooling perhaps should be reduced at these ambients to minimize total charge requirements.

Over most of the operating range, however, the charge variation is about 5% and is small enough to be handled by a suction line accumulator or perhaps by a compressor with some excess storage capacity. Miller (1987) has observed similar[5] low variations in required charge in an experimentally optimized variable-speed unit. This variation would have been larger because of the effects of heat exchanger unloading on heat exchanger saturation temperatures (Rice 1987) but for the moderating effect of the drop in optimum condenser subcooling level with speed. These results suggest that a requirement of a constant active refrigerant charge with no storage capability would cause only a small performance compromise in an optimized modulating system, provided that the flow-control device could maintain the required subcooling levels.

[5]The trends are similar after making allowances for the wider speed ranges of the modified first-generation unit used in Miller's tests.

SEASONAL PERFORMANCE ANALYSIS

Available Steady-State Data for Seasonal Analysis

The steady-state performance data generated by MODCON were used as input to an annual performance factor (APF) and residential loads calculation program (an APF/Loads model) developed by Rice et al. (1985). Developed for seasonal analysis of single- and variable-speed heat pumps, this program uses performance data for as many speeds and ambients as are available. To best compare the seasonal results for the ECM benchmark to those for existing heat pumps, similar speed vs. ambient representations for these units were required. However, for the commercially sold modulating heat pumps, varying amounts of steady-state performance data were available. The most complete data sets available were for the first-generation unit, for which performance data were provided for four speeds in the cooling mode and six speeds in the heating mode over a range of ambients.

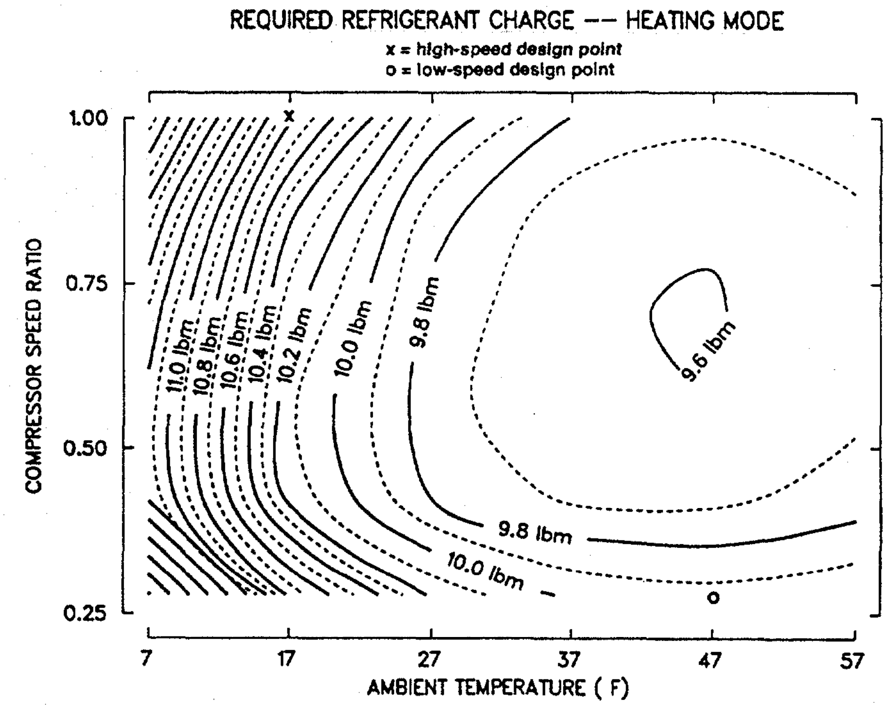

Figure 13 Heating performance mapping of ECM benchmark—required refrigerant charge vs. ambient temperature and compressor speed ratio.

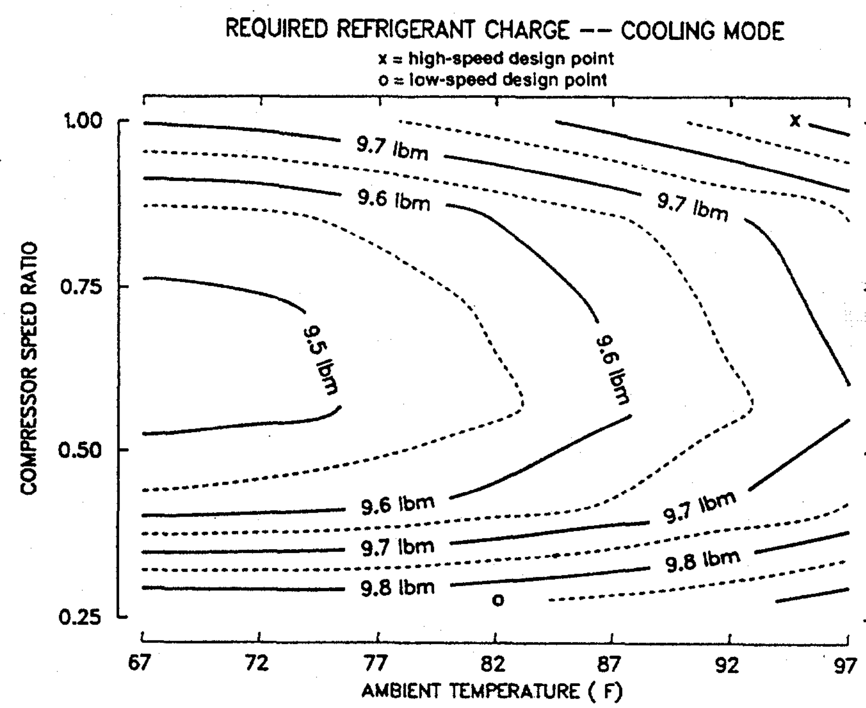

Figure 14 Cooling performance mapping of ECM benchmark—required refrigerant charge vs. ambient temperature and compressor speed ratio.

For the SOA alternative unit, data at three speeds—the low, intermediate, and high speeds of the DOE variable-speed test procedure described by Domanski (1988)—were provided over a range of ambients. However, in the heating mode, between 17°F and 47°F (−8.3°C and 8.3°C), the data include the effects of the integrated frost/defrost (F/D) cycle per the DOE test procedure. (This does not affect the steady-state comparisons made in Table 1, as the F/D effects do not extend to these ambients.) For the SOA reference unit, only performance data at the minimum and maximum speeds were available; and for the heating mode, similar F/D loss effects had been included.

Dynamic Loss Assumptions

The dynamic loss factors for the heat pumps to be compared on a seasonal basis were selected to include reasonable levels of cycling and F/D loss but without giving undue advantage to one unit or another in the process.

F/D Loss Correlations For the ECM benchmark and first-generation heat pumps, a set of normalized F/D losses for the first-generation coil configuration with demand defrost were used. These were measured and were added to the APF/Loads program by Miller (1988b) as functions of ambient temperature and relative humidity.

Integrated Steady-State Heating Data For the SOA alternative and the SOA reference heat pumps, because the primary F/D losses were approximately included in the provided performance data (per the DOE F/D test and interpolation assumptions between 17°F and 45°F [−8.3°C and 7.2°C]), only defrost tempering losses were added by use of the APF/Loads model. The defrost tempering losses were added in a manner consistent with the treatment used for the ECM benchmark and first-generation cases.

Cycling Loss Factors For all heat pumps considered, cycling loss degradation (C_D) factors of 0.25 in heating and cooling were assumed. The use of the upper-limit default C_D values of the DOE test procedure eliminated any potential for inflation of the HSPF and SEER numbers for the benchmark case from low loss-factor assumptions. Therefore, all seasonal predictions were made with standard cycling loss factors.

Bin Analysis Assumptions The seasonal analyses were performed for 5 F° (2.8 C°) temperature bins, with separate day and night load profiles averaged monthly over the heating and cooling seasons. Temperature bin data were obtained from U.S. Air Force engineering weather data (1978). An 1,800-ft^2 (167.2-m^2) house with HUD minimum insulation levels was assumed (Rice et al. 1985).

Adjustment of Results for Different Levels of Speed Mapping

A complicating factor in the seasonal analysis is the differing amounts of speed data available for the various heat pumps. Because the performance data are nonlinear with compressor speed, seasonal performance factors will differ for the same heat pump, depending on the number of speed data sets available (Domanski 1988). The way we adjusted for this effect was to run the ECM benchmark case for six, three, and two speeds and to use the resulting performance factor ratios of heating and cooling to correct the seasonal and annual numbers for the commercial units to the six-speed reference cases.

Seasonal Performance Results

Performance Relative to the DOE/ARI Rating Procedure The APF/Loads model was run for the various heat pumps to evaluate the HSPF, SEER, and APF values. It should be emphasized that the APF/Loads model does not evaluate seasonal performance according to the DOE/ARI rating procedure. This model was developed as a more rigorous alternative to that used in the rating procedure and is a more detailed and realistic bin analysis based on calculated day/night loads for a specified house. In the rating procedure (Domanski 1988), the slope of the heating load line depends on the nominal heating capacity of the unit at 47°F (8.3°C). Also, the energy use for defrost tempering heat is accounted for in all frosting temperature bins, whereas the rating procedure omits that portion of tempering heat that occurs above the high-speed balance-point temperature. As such, the HSPF and SEER values obtained with the APF/Loads model differ in various ways (especially in the heating mode) from those defined by DOE/ARI, and comparisons between the two approaches should be made with due caution.

Efforts were made, however, to make the more rigorous seasonal analysis as comparable as possible to the DOE/ARI procedure. First, a representative DOE Region IV city—Columbus, Ohio—was chosen. Second, the heat pump unit size relative to the calculated cooling load was selected to be equivalent to the DOE procedures, with the nominal unit size scaled in capacity to give 110% of the cooling load at a 95°F (35°C) ambient. Although these refinements improved agreement between the two approaches in the cooling mode, the APF/Loads model gave about a 40% higher heating load than the minimum design heating requirement of the rating procedure (Domanski 1988) at the 5°F (2.8°C) design temperature for Region IV.

For purposes of comparison, the DOE/ARI rating procedure calculation was performed for the ECM benchmark unit for Region IV with the minimum design heating requirement (DHR) commonly used for rating purposes. With the minimum DHR approach, a heating balance point of 17.6°F (−8.0°C) was obtained, compared to a 31°F (−0.6°C) average balance point in the APF/Loads model. For the units considered here, which have no relative overspeed capability in the heating mode, the 31°F balance point is a more realistic condition.

Performance Relative to First-Generation Unit In Table 4, seasonal performance results from the APF/Loads model are compared with those for the first-generation, the SOA reference, and the ECM benchmark modulating heat pumps. The first-generation unit is taken as the baseline in Table 4. Even though the first-generation unit has a wider modulation range in both heating and

TABLE 4
Seasonal Performance Factor Comparisons
Relative to a Nominally Sized First-Generation Modulating Heat Pump
1,800 ft² House—Nominal Unit Sizing
DOE Region IV—Columbus, Ohio

Heat Pump[a]	First-Generation[b]	SOA Reference[c,d]	ECM Benchmark[b]
Speed Range	1.52 – 0.21 heating 1 – 0.21 cooling	1 – 0.28 heating 1 – 0.28 cooling	1 – 0.28 heating 1 – 0.28 cooling
HSPF	7.04	7.72 (+9.66%)	8.31 (+18.0%)
SEER	11.20	16.0 (+42.9%)	20.6 (+83.9%)
APF	2.29	2.62 (+14.4%)	2.89 (+26.2%)

[a]All heat pumps have reciprocating compressors.
[b]C_D = 0.25, normalized modulating frost/defrost (F/D) losses (Miller 1988c).
[c]C_D = 0.25, manufacturer's F/D losses, defrost tempering heat added.
[d]Seasonal performance numbers are based on steady-state data at two speeds, adjusted to be comparable with results for six-speed benchmark predictions.

cooling, the poor performance of the inverter-driven induction motors, as measured by Miller (1988c) and as compared by Rice (1988b) to ECM drives, results in markedly poorer seasonal performance. This is especially evident in the cooling mode, where no significant offsetting effects of the higher-speed range occur; while in the heating mode, lower resistance heat requirements from the wider speed ratio mask to some extent the poorer steady-state performance of the first-generation unit.

Performance Relative to the SOA Reference Unit The two SOA modulating heat pumps on the market are compared with the ECM benchmark in Table 5, with the SOA reference unit as the point of reference. For comparison, the quoted DOE/ARI-rated HSPF and SEER rating values for Region IV are also provided for the SOA units, as are the computed DOE/ARI ratings for the ECM benchmark.

In the heating mode, the SOA alternative unit slightly outperforms the SOA reference unit, in rough agreement with the general trend of the DOE/ARI HSPF ratings, although the absolute values and the size of the predicted advantage are reduced. This occurs even though the SOA alternative unit has a smaller speed range because the relative capacity of the unit is higher in the heating mode than that of the SOA reference unit above 15°F (−9.4°C), as is shown in Figure 8. In cooling, the SOA reference unit shows a predicted performance advantage of only 2% over the SOA alternative unit, while the rating numbers have a 14% expected difference. The annual performance numbers for the two commercially available units are nearly identical because of the 75-to-25 split of heating-to-cooling load and the lower average performance level in heating than in cooling. (It should be noted that the SOA alternative unit also provides integrated domestic hot water capability, the added benefits of which are not considered in this paper.)

In Table 5, the ECM benchmark shows more than three times as much seasonal improvement potential in cooling than in heating. This is a reversal of the steady-state performance results from Table 1, where the steady-state heating performance gains are about half again as much as for cooling. While the steady-state heating COP gains for the ECM benchmark are about 35%, the calculated HSPF increase is not quite 8%. This is in sharp contrast to the calculated DOE/ARI HSPF rating for the ECM benchmark, which increases by 31% from the rating for the SOA reference—a gain close to the predicted steady-state COP gains.

The dominant reason for this small HSPF increase is that in the APF/Loads model about 35% of the heating input is for resistance heat backup, as compared to 14% for the DOE/ARI rating procedure. This energy use at a COP of 1 is a major dampening factor that prevents the HSPF increase from approaching the steady-state improvement level. The annual performance factor increase

TABLE 5
Seasonal Performance Factor Comparisons
Relative to a Nominally Sized
SOA Reference Modulating Heat Pump
1,800 ft² House—Nominal Unit Sizing
DOE Region IV—Columbus, Ohio

Heat Pump[a]	SOA Alternative[b,c]	SOA Reference[b,c]	ECM Benchmark[d]
Speed Range	1 – 0.36	1 – 0.28	1 – 0.28
HSPF	7.85 (+1.7%) $9.05_{DOE/ARI}$	7.72 $8.70_{DOE/ARI}$	8.31 (+7.64%) $11.4_{DOE/ARI}$
SEER	15.7 (−1.9%) $14.1_{DOE/ARI}$	16.0 $16.4_{DOE/ARI}$	20.6 (+28.8%) $19.9_{DOE/ARI}$
APF	2.65 (+1.1%)	2.62	2.89 (+10.3%)

[a]All heat pumps have reciprocating compressors.
[b]Seasonal performance numbers are based on steady-state data at two to three speeds, adjusted to be comparable with results for six-speed benchmark predictions.
[c]C_D = 0.25, manufacturer's frost/defrost (F/D) losses, defrost tempering heat added.
[d]C_D = 0.25, normalized modulating F/D losses (Miller 1988c).

is similarly limited to just over 10% due to the dominance of the heating requirements for the location selected. This dampening effect on HSPF is only weakly felt in the DOE/ARI HSPF rating procedure because the minimum DHR approach significantly underrepresents the typical heating load requirements.

Cooling Performance Limits In the cooling mode, the ECM benchmark unit has a predicted SEER of 20.6, as compared to a 16.0 value for the SOA reference unit. This seasonal performance advantage of 28.8% is more in line with the predicted steady-state increases of 24% to 25% at the 82°F and 95°F conditions (the former with a S/T ratio of 0.83). For a reciprocating compressor system with a relaxed S/T requirement at mild ambients, an SEER of 20 appears to be a plausible upper limit of performance for a Region IV climate when the unit is sized per the DOE/ARI rating procedure (Domanski 1988).

Heating Performance Improvement Potential A modulating heat pump designed to have overspeed capability in the heating mode (relative to the nominal design speed in cooling) would go a long way toward realizing the predicted steady-state potential in heating performance. Oversizing of the heat pump for heating-dominated climates would be another method to more closely approach the potential of the steady-state performance numbers.

Potential of unit oversizing Unit oversizing was briefly investigated, and results for a 50% oversizing for the Region IV climate are shown in Tables 6 and 7. Table 6 tabulates the percentage increases in seasonal and annual performance. The effect of 50% unit oversizing is shown to increase the HSPF and APF values by 10% to 12% for the SOA units and by about twice that (22% to 24%) for the ECM benchmark. Because oversizing results in more operating hours at the slower speeds, the units with the highest low-speed efficiencies should derive the most benefit. The SEER also increases by about 14% for the two units with higher low-speed cooling efficiencies. A previous investigation of oversizing effects by Miller (1988b) on the modified first-generation unit showed only marginal benefits; however, this unit is shown in Table 1 to have poorer low-speed efficiency than the SOA units. A more recent analysis by Sulfstede (1990) on the HSPF benefits of oversizing the SOA reference unit in a Region V location showed benefits closer to those predicted here.

TABLE 6
Effect of 50% Oversizing on the Seasonal Performance Factors of SOA and ECM Benchmark Modulating Heat Pumps
1,800 ft² House—150% Nominal Unit Sizing
DOE Region IV—Columbus, Ohio

Heat Pump	SOA Alternative	SOA Reference	ECM Benchmark
HSPF Change (%)	+11.5	+12.2	+23.9
SEER Change (%)	+1.9	+13.8	+14.1
APF Change (%)	+9.8	+12.2	+22.1

Table 7 shows the seasonal performance factors (and the increases relative to the SOA reference case) that are possible through moderate oversizing with efficient modulating heat pumps. House-loads-based HSPF values higher than 10 and SEER values of 23.5 are predicted to be possible for Region IV. (Some modification of the DOE/ARI rating procedure [Domanski 1988] might be considered to provide full credit for such oversizing, as the present approach increases the load lines in proportion to the unit capacity rather than to the house load requirement.) Such an oversizing strategy would appear to be an effective way of achieving additional performance gains by displacing backup heat requirements while reducing the average loading on the heat exchangers. This strategy should be especially beneficial when the load requirements are dominant at intermediate speeds with a nominally sized unit.

However, care should be taken that the performance gains achieved will offset the added cost of the oversized unit over a payback period acceptable to the customer. No such economic analysis was undertaken in this study. For the selected house in Columbus, Ohio, however, unit oversizing by 50% resulted in a savings of 1,550 kWh/year or about $110/year at $0.07/kWh. Another oversizing consideration is that if the low-speed balance

TABLE 7
Seasonal Performance Factor Comparisons Relative to a 50%-Oversized SOA Reference Modulating Heat Pump
1,800 ft² House—150% Nominal Unit Sizing
DOE Region IV—Columbus, Ohio

Heat Pump	SOA Alternative	SOA Reference	ECM Benchmark
Speed Range	1 – 0.36	1 – 0.28	1 – 0.28
HSPF	8.75 (+1.0%)	8.66	10.3 (+18.9%)
SEER	16.0 (−12.1%)	18.2	23.5 (+29.1%)
APF	2.91 (−1.0%)	2.94	3.53 (+20.1%)

point in cooling is raised too high, dehumidification capability at mild ambients may suffer due to cycling effects. Effective use of a humidistat with fan speed control would minimize this potential problem.

Potential of overspeed operation in heating An alternative way of increasing heating seasonal performance with modulating heat pumps is to design the compressor drive to be oversped only (or over a somewhat wider range) in the heating mode (Rice 1988a, b). Increasing the overspeed heating capability relative to design cooling speed would obtain only part of the performance gain of unit oversizing (because of the absence of additional heat exchanger unloading benefits), but the cost should be much less. One manufacturer has recently introduced a heat pump with 10% more overspeeding capability in the heating mode than in the cooling mode. The use of a modulating ECM-driven scroll compressor would be another way to maintain a higher heating capacity at low ambients with or without an expanded speed range because of the reduced dropoff in volumetric efficiency at higher pressure ratio conditions (Rice 1988a).

The individual and combined merits of overspeed operation in heating with reciprocating and scroll compressors and of unit oversizing should be further investigated for modulating heat pumps, since at current seasonal performance levels a unit increase in HSPF has the potential for more energy savings than a unit increase in SEER. Even if the SEER levels were reduced to achieve higher HSPF values, the net energy use in moderate to northern U.S. locations would decrease.

Potential relative to DOE rating procedure Finally, the significant energy-saving advantages of oversizing a modulating unit relative to the design cooling condition (or alternatively of providing overspeeding in heating) will not be as apparent using DOE minimum DHRs as when a more realistic house-characteristics-based loads approach is used. Calculations of energy use from the DOE/ARI HSPF ratings at minimum DHRs will underestimate the possible heating-mode energy savings of oversizing and/or overspeeding modulating heat pumps.

CONCLUSIONS AND RECOMMENDATIONS

The purpose of this near-term benchmark analysis was twofold. One purpose was to evaluate the potential performance improvement predicted by a modulating heat pump model *with high-efficiency heat exchangers and drives and current reciprocating compressor technology* relative to an SOA modulating heat pump. The second was to demonstrate a methodology using a modulating heat pump design tool for such a system design analysis.

With regard to the first purpose, a potential increase in steady-state cooling performance ranging from 13% to 25% was found. A 29% increase in SEER may be possible with SEERs exceeding 20 if the mild-ambient design-S/T-ratio requirements are relaxed. Steady-state heating performance improvements of 33% to 41% also were predicted. The resultant HSPF gains, however, were less than 8% for DOE Region IV. HSPF gains of close to 20% are shown to be possible if the heat pump units are oversized 50% relative to the required design cooling load. The oversizing of modulating units and other alternative approaches to improve heating seasonal performance—such as a wider compressor overspeed range in heating and/or a modulating scroll compressor—should be further investigated.

With regard to the second purpose, our experience with this analysis suggests that a reasonably optimized modulating system can be obtained by the four-point design approach employed here. Comparing this design process with a black-box optimization approach conducted in an earlier assessment of variable-speed potential (Rice and Fischer 1985), we find the present approach intuitively superior in maintaining engineering control of the design process and by providing a visual (and tabular) mapping of the design objectives and constraints about the vicinity of the optima.

The modulating design tool used for this analysis will be made available to the HVAC community. The program is also capable of using R134a as a refrigerant, and other pure refrigerant alternatives can be added with minimal effort as their thermodynamic and thermophysical properties become available (Spatz 1991). Given compressor performance maps for these candidate alternative refrigerants, the program could be used to determine their comparative performance in optimally configured and controlled single- or variable-speed air-to-air heat pump systems.

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DISCUSSION

Carl T. Sgamboti, Senior Research Engineer, United Technologies Research Center, East Hartford, CT: Can the current version of the heat pump code perform analyses of multiple indoor units in simultaneous heating and cooling? Also, does Oak Ridge National Laboratory plan to add transient analysis capability to predict heat pump response and track refrigerant location?

C.K. Rice: The ORNL modulating model has no special provisions to handle more than one indoor coil at a time, nor is it capable of tracking simultaneous heating and cooling. However, such combined operation could probably be handled by an appropriate executive routine that defines various heat pump configurations appropriate to each possible operation combination and that calls the heat pump model as a subroutine for each configuration. The source code is available for those interested in such modified uses of the program.

There are no plans to add transient analysis capability to the heat pump model. The model does include refrigerant charge inventory estimation and balancing capability, a wide range of charge estimation methods, and a suction line accumulator model. One result of this capability is that the required steady-state on- and off-cycle charge distributions in each component are provided.