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EFFICIENCY CHARACTERISTICS OF SPEED-MODULATED DRIVES AT PREDICTED TORQUE CONDITIONS FOR AIR-TO-AIR HEAT PUMPS

C.K. Rice, Ph.D.
ASHRAE Member

ABSTRACT

Examples of system (motor plus inverter drive) efficiencies of two broad types of adjustable speed control are compared for predicted compressor and indoor blower load profiles. The two classifications are inverter-driven induction motors (IDIMs) and permanent-magnet electronically commutated motors (PM-ECMs). Reference sine-wave-driven induction motor (SWDIM) efficiencies are also given.

Available bench data in the ASHRAE literature (Lloyd 1982) on late 70's IDIM compressor drives are compared to recent bench and application data obtained on IDIM and PM-ECM drives. The drive efficiencies are compared over a common set of predicted operating torque requirements for heating and cooling conditions. A modulating heat pump model was used to develop predicted reciprocating compressor torque / drive-frequency mappings and the expected operating torque ranges. The variation in modulating compressor torque requirements is analyzed and the major determining factors examined. Ways to adjust the given torque relationships for different compressor types and sizing strategies are also discussed.

Similar performance comparisons are made for modulating indoor blower drives. Modulating blower performance data on an early '80s generation modulating heat pump with an IDIM drive (and SWDIM reference drive) were obtained from Miller (1988). These data are compared to bench data on recent IDIM and PM-ECM drives under similar torque conditions.

In both compressor and blower applications, the combined system efficiency of the PM-ECM drives is shown to be nearly equal to or higher than that of the reference SWDIM cases and significantly better than IDIMs available in the late '70s. When compared to available data on more recent IDIMs, the PM-ECM efficiency advantage over IDIM compressors has been reduced 40 to 50% between half and nominal speed (3600 rpm) but still remains 14 to 9% higher, respectively. For blowers, the IDIM improvement has not been as great.

PM-ECM-driven blowers appear to be very competitive at present with IDIM blowers on the combined basis of efficiency, cost, and reliability. For the compressor PM-ECM drives, if progress continues on cost and reliability issues with minimal loss in efficiency, these drives will likely be a leading candidate for future-generation (5 to 10 year range) modulating heat pumps.

C. Keith Rice is a research staff member in the Building Equipment Research Program, Efficiency and Renewables Research Section, Energy Division, Oak Ridge National Laboratory, Oak Ridge, Tennessee.

INTRODUCTION

A key issue in continuously modulating heat pump development is the performance of the speed-modulated drives over the required operating conditions in heating and cooling modes. For rotating equipment, these application requirements can be expressed by torque vs drive frequency (or frequency ratio) relationships. A proper comparison of the potential performance of various modulating drives is ensured if evaluation is made over the entire expected operating range rather than at one or two design conditions.

An appropriate set of expected steady-state operating conditions were obtained based on previous work by the author (Rice and Fischer 1985) on an initial assessment of continuously variable-capacity modulation benefits. In this work, an air-to-air heat pump with a reciprocating compressor and fixed total heat exchanger surface was optimized for best annual performance in a moderate climate (60% heating load, 40% cooling load). This was done analytically using a modified version of a steady-state heat pump design model (Fischer and Rice 1983) in conjunction with a numerical optimization program. The resulting optimal heat pump configuration and optimal compressor and indoor blower speed relationships, with one modification, were used to determine a set of appropriate steady-state operating conditions to be imposed on the variable speed drives under comparison.

One purpose of this paper is to present these predicted pressure ratios, operating conditions, and envelopes of torque vs drive frequency ratio for a reasonably optimal modulating heat pump. The factors which determine the obtained frequency/torque characteristics are discussed, as are possible ways to extend the results to other compressor types and frequency or speed ranges. From this evaluation, a broader understanding is obtained of the types of compressor and blower loadings to be expected in modulating heat pump applications and of any resultant limitations to drive performance.[1]

A second purpose of this analysis is to provide information on speed-modulated efficiency characteristics of compressor and blower drives at various technology levels under modulating heat pump conditions. Jeter et al (1987) have noted the lack of data in the literature appropriate for residential variable-speed system analysis. Because of this, they assumed constant drive efficiencies in their analysis as did Miller and Jaster (1980) and Muir and Griffith (1980). Another recent variable-speed model by Krakow and Lin (1987) omitted drive efficiency considerations from the analysis and considered only the thermodynamic performance.

Efficiency data are somewhat more available on industrial duty IDIM drives and motors for general applications (e.g., Mohan 1981, Andreas 1982, Kirschen 1985). More recently, some performance comparisons between PM-ECM and IDIM drives for 5 and 10 hp (3.75 and 7.5 kW) industrial applications have been given by Wallace (1987) and Cathey (1987), respectively. While these data and discussions are useful in observing general drive performance trends over ranges of speed and torque, it is difficult to determine how transferable these efficiencies are (either in absolute or relative terms) to residential heat pump applications.

Modulating drive efficiency data more appropriate to residential heat pump application were obtained from five sources for the determined operating torque ranges.

1. Compressor speed-drive performance for a late '70s industrial pulse-width-modulated (PWM) IDIM drive with a standard compressor motor as well as reference SWDIM data were given in bench test results reported by Lloyd (1982).
2. Bench data on residential state-of-the-art (SOA) compressor and blower drive systems were also recently provided by Lloyd (1987) for 6-step production IDIM's and their reference SWDIM's.
3. PM-ECM modulating-drive efficiency for compressor and indoor blower systems were obtained from prototype bench test data provided by Harms (1985), the characteristics of which have been discussed by Erdman (1984) and Harms (1986).
4. Another set of 6-step IDIM and reference SWDIM data for variable-speed indoor blower performance was obtained from Miller (1988) based on tests of the as-installed performance in the air handler of a 3-ton, split system, modulating heat pump of early '80s design.

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[1]All torque discussions refer to average (over one revolution) steady-state values. No considerations have been given here to transient torque conditions such as startup (pulldown), cycling, or defrosting operation. Maximum steady-state operating conditions (ARI 1984) are included, however.

5. Finally, reference SWDIM data from Zigler (1987) were used in conjunction with hermetic compressor tests by Miller (1988) (on the same early '80s modulating heat pump) to estimate the application efficiency of the original residential 6-step IDIM drive system and that with an industrial PWM IDIM drive.

The SWDIM modulating drive data obtained from motor-alternator drives are included only as laboratory reference points for the efficiency levels of the tested induction motors used in the IDIM's and as a means to quantify and compare the inverter drive losses. SWDIM drives are not economically viable for heat pump applications considered here.

A third purpose of this analysis is to determine the most appropriate high-efficiency speed control drive to be used in advanced modulating system assessments. Issues considered are:

— the ability of the PM-ECM drive to perform efficiently and operate acceptably over the required torque vs frequency ratio ranges,

— the degree of efficiency improvement over that reported for the alternative, more accepted, IDIM speed drives, and

— the likelihood of near-term resolution of remaining cost, reliability, and manufacturing questions without significant degradation of performance advantage.

Where appropriate in the following discussion, drive speed and drive frequency ratios have been used interchangeably. *For synchronous motors* such as PM-ECM drives, *speed and frequency ratios* (turndowns) or ranges *are equivalent,* while *for induction motors, they differ by the rotor slip* (which increases with torque). All results are presented in terms of drive frequency (and/or drive frequency ratios) rather than speed because this was the more commonly available controlled variable for the modulating-drive tests.

MODULATING COMPRESSOR OPERATING ENVELOPES

Modulating Heat Pump Model

Speed Ranges and System Control. In the capacity modulation assessment of Rice and Fischer (1985), a reciprocating compressor was assumed to be speed-modulated over ranges[2] of

– 25% to 150% of nominal speed in heating mode, and

– 25% to 100% of nominal speed in cooling mode.

The indoor blower was allowed a range of

– 40% to 100% of nominal speed in either mode.

The outdoor fan was not modulated.

For the compressor, nominal speed was set at 3600 rpm. The compressor was sized, so when operating at nominal speed, to meet the design day cooling load in an 1800 ft^2 house in the selected moderate climate of Nashville, TN (60% heating load, 40% cooling load).

The modulating system was controlled to give minimal compressor inlet superheat and optimal condenser subcooling. For present purposes, one modification was made to the reported optimal configuration. The optimal indoor blower flow rate in heating mode was reduced uniformly by 25% to obtain more acceptable supply air temperatures.

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Revised Compressor Model. The original reciprocating compressor model (Rice and Fischer 1985) was replaced by an improved version based on more extensive calorimeter testing of a hermetic reciprocating compressor using a three-phase, 2-pole, induction motor over a sine-wave frequency range of 15 to 75 Hz.[3] This was supplemented by 15 Hz application compressor data taken by Miller (1988) on the same model compressor. Operating conditions appropriate to each frequency ranged from −5 to 55 F (−20.6 to 12.8°C) evaporating temperatures and 90 to 130 F (32.2 to 54.4°C) condensing temperatures.

Data to simulate the specific modulating induction motor performance were obtained from Zigler (1987) who provided motor efficiency and slip information (at typical motor operating temperatures) for 15 to 90 Hz operation over a wide range of torque. These data were used to iteratively evaluate compressor torque from the compressor calorimeter data and also to convert from induction to synchronous motor simulation. With the revised model, the optimally-controlled modulating heat pump (Rice and Fischer 1985) was simulated over the assumed ranges of frequency ratio for heating and cooling mode operation.

Predicted Operating Conditions

Pressure Ratios. The resultant range of pressure ratios seen across the compressor are shown in Figure 1 for the optimally-controlled design (Rice and Fischer 1985) over heating and cooling ambient and frequency ratio ranges. Here contour plots are used to show lines of constant pressure ratio as a function of compressor frequency ratio and ambient temperature. Indoor conditions were fixed at 70 F (21.1°C) in heating mode and 78 F (25.6°C) in the cooling mode. Shown are the entire range of conditions for all frequency ratio and ambient temperature combinations as well as the conditions given by the optimal operating frequency line for the selected house and climate. The optimal line does not reach the nominal design frequency ratio of 1.0 in cooling mode (denoted by the "x" in Figure 1) because it is based on meeting the average load which was about 7% less than the design day load.

Along the optimal operating lines, the maximum pressure ratios in cooling mode are seen to be smaller than in heating mode. At lowest frequency operation, the pressure ratios are less than 2.5 in heating mode and less than 2 in cooling. At the ambients of maximum heat pump yearly output (house load times operating hours) of 34 F (1.1°C) heating and 84 F (28.9°C) (Rice and Fischer 1985), the pressure ratios are 3.1 and 2.0, respectively.

This last observation suggests that to maximize compressor performance on an annual basis, for a modulating heat pump application in a balanced climate, compressors should be designed for optimal efficiency at pressure ratios near 2.5. For cooling only applications, a design pressure ratio of 2.0 would be more optimal for best SEER.

Evaporating and Condensing Temperatures. In Figure 2, the range of refrigerant-22 evaporating and condensing (saturation) temperatures are shown for the same ranges of ambients and compressor frequency ratio. Along the modulating operating line in heating mode, as the compressor speed is decreased, the evaporating temperature ranges from −5 to 45 F (−20.6 to 7.2°C) while the condensing temperature remains about 95 F (35°C). As speed is decreased along the cooling operating line, the evaporating temperature ranges from 42 to 56 F (6.1 to 13.3°C) and the condensing from 125 to 90 F (51.7 to 32.2°C).

Both Figures 1 and 2 show the predicted operating condition effects of speed reduction commonly referred to as heat exchanger unloading. These operating conditions are next used to determine the resultant compressor torque variations for a modulating heat pump application.

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[3]Personal communication with a compressor manufacturer, 1986.

PREDICTED MODULATING COMPRESSOR TORQUE CHARACTERISTICS

Factors in Determining Compressor Torque

Compressor torque is given by

$$torque = shaft\ power\ /\ motor\ speed$$

$$\propto (\dot{W}_{cm} \bullet \eta_{inv} \bullet \eta_{mot})\ /\ S \tag{1}$$

where

$\dot{W}_{cm}$ = compressor input power,

η_{inv} = inverter efficiency,

η_{mot} = motor efficiency, and

S = motor operating speed.

Defining compressor shaft isentropic efficiency η_{is} as

$$\eta_{is} = \frac{ideal\ shaft\ power}{actual\ shaft\ power} = \frac{\dot{m}_r \bullet \Delta h_{is}}{\dot{W}_{cm} \bullet \eta_{inv} \bullet \eta_{mot}} \tag{2}$$

Equation 1 can be rewritten as

$$torque \propto (\dot{m}_r \bullet \Delta h_{is}\ /\ \eta_{is})\ /\ S \tag{3}$$

where

$\dot{m}_r$ = refrigerant mass flow rate, and

Δh_{is} = enthalpy change for an isentropic compression from shell inlet conditions to shell outlet pressure.

The shaft isentropic efficiency in Equation 3 includes the effects of compression efficiency, mechanical efficiency, and internal suction gas superheating. While motor efficiency cannot, by definition, affect torque directly, it does have an indirect influence through the motor cooling done by internal suction gas superheating.

The refrigerant mass flow rate in Equation 3 can be expanded to give

$$\dot{m}_r = S \bullet D \bullet \rho \bullet \eta_{vol} \tag{4}$$

where

D = compressor displacement,

ρ = refrigerant density at compressor shell inlet conditions, and

η_{vol} = compressor volumetric efficiency, defined by Equation 4 as the actual compressor mass flow rate divided by the ideal flow rate.

Combining Equations 3 and 4 gives

$$torque \propto D \bullet \rho \bullet \eta_{vol} \bullet \Delta h_{is}/\eta_{is} \tag{5}$$

where the *direct* motor speed dependence has canceled out.

For speed modulation, the displacement D in Equation 5 is a constant.[4] Thus, compressor torque can be expressed in terms of the four remaining factors in Equation 5, each of which is dependent in a different way on the compressor operating conditions.

Rather than look at each individual term separately, the dependence of two types of logical pairings of the terms will be examined. The first pairing relates to the two components of loading the compressor places on the motor—*the mass pumping and the work per unit mass requirements.* The product of ρ and η_{vol} is proportional to the mass pumping (per unit speed) component of the torque while the Δh_{is} / η_{is} term relates to the work per unit mass component.

The second pairing separates the torques into *an R-22 operating conditions component and a compressor efficiency effects component.* For a given type of compressor,[5] the R-22 conditions component can be viewed as a primary effects torque based on heat exchanger conditions with R-22 properties. This is multiplied by the secondary compressor efficiency effects to obtain the actual torque. The product of ρ and Δh_{is} gives the R-22 conditions component and the ratio of η_{vol} to η_{is} gives the compressor effects multiplier.

The following section describes how each of these torque components are affected by modulating heat pump operation over the range of heating and cooling ambients.

Torque Component Variations and Net Requirements with Speed Modulation

In Figures 3–6, the variation in the component values for both types of logical groupings are shown as contour lines over the range of compressor frequency ratios and ambient temperatures. The optimal operating line for the selected house and climate is also shown superimposed on each contour plot. Heating and cooling mode plots are shown side-by-side in each figure.

All the component values in Figures 3–6 have been normalized to 1.0 at the selected nominal cooling design condition.[6] The subscript "n" denotes normalized values.

Mass Pumping Component (per unit speed). In Figure 3, the normalized mass pumping requirements of compressor loading, $M_n = \rho_n \; \eta_{vol,n}$, is shown. The pumping requirements on the compressor are seen to be most strongly dependent on ambient in the heating mode (increasing with ambient) and on compressor speed in the cooling mode (decreasing with speed increase). This is because the generally dominant factor in M_n is the refrigerant density which tracks directly with evaporating temperature (as given in Figure 2). Only at the higher pressure ratio (lower speed conditions) do strong dropoffs in the volumetric efficiency cause the trends of pumping requirement to shift noticeably lower.

Work Per Unit Mass Component. Figure 4 contains contours of the second component of compressor loading given by $E_n = \Delta h_{is,n} / \eta_{is,n}$. Here the dominant factor is generally the ideal work per unit mass requirements, $\Delta h_{is,n}$, which tracks along with pressure ratio (as given in Figure 1). As in the case of the mass pumping component (only at the lower speed, i.e., higher pressure ratio conditions) do strong dropoffs in shaft isentropic efficiency cause the work requirements to shift markedly higher in heating mode and to stop falling in cooling mode.

R-22 Conditions Component. The primary effect torque contours for a reciprocating compressor are shown in Figure 5. Here the direct effects of the refrigerant conditions on torque are given by $T_R = \rho_n \bullet \Delta h_{is,n}$.

In the heating mode, for fixed ambients, the contour lines are near vertical for the upper two-thirds of the figure. This indicates that the refrigerant density and the ideal work changes with speed have balanced out each other leaving a primary torque independent of speed. Here the torque decreases nearly uniformly with

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[4]Note that for variable displacement (or stroke) modulation, the torque is directly proportional to stroke and thereby has different torque vs modulation characteristics than those to be discussed here.

[5]The volumetric efficiency trends of each compressor type can significantly affect the heat exchanger loading and thereby the operating conditions seen over the range of speeds and ambients.

[6]The nominal design condition was selected to be cooling mode operation (at a frequency ratio of 1.0 (3600 rpm) with 95 F (35°C) ambient, 78 F (25.6°C) indoor conditions. From Figure 2, nominal design condition operation gave compressor inlet/exit saturation temperatures of 37.2/126 F (2.9/52.2°C).

decreasing ambient indicating that refrigerant density effects are dominant with ambient. At the milder heating ambients, however, the contours shift slope so that the torque requirement reduces at the lower speeds (even though the evaporating temperatures still increase). This indicates that the reducing pressure ratios (and thereby reduced ideal work) resulting from heat exchanger unloading have overcome the increasing refrigerant densities. Here the primary torque (as a function of speed) has shifted from refrigerant density and pressure ratio offsetting each other to a pressure ratio dominant situation.

In the cooling mode, as speed is reduced at a fixed ambient, the primary torque decreases. This indicates that pressure ratio effects are dominant over refrigerant density changes. Torque reductions are even greater as the ambient is reduced at fixed speeds. This is because evaporator temperature drops slightly with decreasing cooling ambients and both factors of T_R move in the same direction.

Compressor Effects Component. This torque component can be thought of as a *secondary* compressor effects multiplier to the *primary* R-22 conditions torque. The compressor torque multiplier T_{cm} is given by $\eta_{vol,n} \ / \ \eta_{is,n}$, the ratio of normalized volumetric to shaft isentropic efficiencies. This component, as shown in Figure 6, represents the secondary effect of operating conditions through their influence on the compressor performance. A value of T_{cm} greater than 1.0 indicates that the compressor pumping efficiency (requirement for more torque) is greater than the work efficiency (which reduces torque needs), relative to the ratio at the nominal design cooling condition.

From Figure 6, in the heating mode, the compressor effects component is rather complex, ranging from a maximum of 1.13 at a low ambient, intermediate speed (frequency ratio of 0.75) to a low of 0.82 at the low ambient, high speed condition. Another local minimum occurs at a frequency ratio of 0.50 over all heating ambients. The local peaks and valleys at frequency ratios of 0.75 and 0.50 are the results of peaks in volumetric and shaft isentropic efficiency, respectively. The volumetric efficiency peak shown at 75% of nominal speed may be exaggerated at the lowest ambients because of extrapolations beyond the limits of available data at that speed. However, the general trend there is supported by the data as is the increasing isentropic efficiency trend at 50% of nominal speed. Above nominal speed, the compressor multiplier is seen to reduce the required torque due to the worsening volumetric efficiencies, relative to the isentropic.

In cooling mode, the compressor multiplier increases at lower speed, lower ambient conditions. Here the volumetric efficiency has increased while the shaft isentropic drops rather sharply from a peak at 50% of nominal speed. At other cooling conditions, the compressor multiplier is near unity, dropping slowly below one only with operation at 50 to 75% of nominal speed at higher ambients.

However, in both modes, the compressor multiplier is seen to be a secondary effect compared to the primary effects of the R-22 conditions determined by heat exchanger loading and the refrigerant properties.

Net Torque Requirements. The product of either the mass pumping and work requirements or the primary R-22 conditions and the secondary compressor efficiency effects is the net torque required of the compressor motor. The resultant normalized torque contours T_n are shown in Figure 7 for heating and cooling operation. The general trends of the R-22 condition effects (Figure 5) discussed earlier are seen to hold although the local peaks and valleys of the compressor efficiency variations (Figure 6) complicate the final curves at intermediate to low speeds, especially in heating mode.

The compressor torque is thus seen to be a somewhat involved interplay between various factors *primarily* determined by refrigerant operating conditions and properties (the factors always opposing each other to varying degrees) and *secondarily* influenced by compressor efficiency parameters. The primary torque directions are determined by whether refrigerant density changes or pressure ratio changes have the dominant effect.

Compressor Torque vs Frequency Ratio Mappings

The torque information in Figure 7 has been replotted in Figures 8 and 9 in the form of more conventional frequency ratio vs torque (equivalent to speed vs torque) plots. The torque curves are shown for a family of ambient temperatures while indoor temperature is held constant. The optimal operating lines for the selected house and climate are again superimposed on the full operating range. Curves have been added for maximum design operating conditions in heating and cooling mode as specified by ARI Standard 210/240 (1984). Also shown in the figures are reference evaporator/condenser conditions for heat pump compressors given by ARI 520-78 (1978) and other operating conditions at selected points of interest.

Heating Mode Conditions. For normal operating conditions in Figure 8, the torque variation *along the operating line* is ±20% about an average value of 60% of nominal design cooling torque. This is a little more than half of the normal operating torque range of a single-speed heat pump in the heating mode (as seen from the torque variation at nominal speed). Note also that the maximum heating design condition yields torques 17% larger than nominal design cooling torque.

In general, compressor torque requirements are seen to remain nearly constant with overspeeding (operation above nominal speed). Along the heating operating line, overspeed operation does not require torques larger than 60% of nominal design torque. However, overspeed operation at the maximum heating design condition, if necessary, would require motor oversizing for either IDIM or PM-ECM drives as the torque requirement quickly becomes greater than 1.2 times nominal torque (not shown on Figure 8). Oversizing would be needed in such a case because motor torque capability is reduced in the overspeed (constant power) region (Mohan 1981). Motor voltage cannot be increased over line voltage, when overspeeding, to maintain a constant strength (constant volts/hertz) motor as can be done at frequencies below 60 Hz.

Cooling Mode Conditions. Under normal operating conditions in Figure 9, the torque variation along the operating line ranges from 60 to 100% of nominal design torque. This is about twice the normal operating torque range of a single-speed heat pump in the cooling mode.

The maximum cooling design condition at nominal speed results in compressor torque requirements 20% above nominal torque. Should the compressor be required to run at 50% overspeed in cooling mode, the overspeed torque capability will have to be twice as much as if limited to overspeed operation only in the heating mode at lower ambients.

Summary. For modulating heat pump operation with a reciprocating compressor, the *overall* range of expected torques are nearly the same as for single-speed heat pumps although the distributions with ambients are different. On an annual performance basis, the results of Figures 7 and 8 suggest that *speed-modulated motors should be optimized for best efficiency at 60 to 70% of nominal design cooling torque.*

Discussion As a further point of reference, *the nominal design torque* used here *corresponds* most closely *to the rating point torque*—seen in Figure 9 to approximately equal to the torque at the standard compressor rating conditions (ARI 520/78) of 45 F/130 F (7.2°C/54.4°C). The torque which gives the *rated horsepower,* often termed *full load torque, is* typically *two-thirds to four-fifths of the rating point torque* (alternatively the rating point torque is 25 to 50% higher[7] than the full load torque). Therefore, in terms of this more common reference point, the operating torque ranges of Figures 8 and 9 suggest that *speed modulated motors should be optimized* for best performance at or slightly below (0 to 20% below) the *full load* (rated horsepower) *torque,* depending on the specific motor design.[8]

MODULATING FAN TORQUE CHARACTERISTICS

For the indoor blower, the torque varies approximately with the square of the speed, i.e.,

$$\begin{aligned} \text{torque} &= \text{shaft power} \,/\, \text{blower speed} \\ &= (Q \cdot \Delta P) \,/\, (S \cdot \eta_f) \qquad (4) \\ &\propto C \cdot S^2 \end{aligned}$$

[7]An intermediate value of 36% higher is assumed here for rated horsepower determinations of a later section.

[8]A compressor torque profile is often assumed to be "constant torque" (usually the torque at rated horsepower) up to nominal speed and "constant power" for higher speeds (i.e., torque reduces with the inverse of the speed ratio). For the heating mode, if 73.5% of the nominal design cooling torque (1/1.36) is taken to be the full load torque, then the torque at 50% overspeed from Figure 8 is .53/.735 or 0.72 of that as compared with the inverse speed ratio of 1/1.5 or 0.67). Thus there is a reasonably close match between the simplified assumption and the results shown here. However, modulating motor performance up to nominal speed would be better indicated by using torque profiles slightly larger than "constant torque" in the cooling mode and slightly smaller in the heating mode as shown in Figures 8 and 9.

where

Q = volumetric air flow rate $\propto S$,

ΔP = system pressure drop $\propto C \bullet Q^2$,

C = system characteristic constant,

η_f = blower efficiency, which according to the general fan laws is independent of blower speed [McQuiston 1977], and

S = blower speed.

At fixed speed, the system characteristic constant C can change considerably, with transition from dry coil to wet coil (a possible doubling or more of the coil pressure drop) and in cases of dirty filters and closed off branch ducts. However, examination of blower (as opposed to axial propeller fans) ΔP *vs* Q curves shows that at constant blower speed, as system pressure drop increases, the blower shaft horsepower generally decreases;[9] therefore the torque requirement will decrease [Torin Corporation 1980]. *Design* torque will then be seen at the minimum design system characteristic, i.e., the maximum design flow rate at maximum speed. Therefore a square law variation in torque with blower speed (or frequency for PM-ECM drives) was assumed with 100% design torque occurring at maximum speed (frequency). A more restricted indoor air flow system will result in operation at less than design torque.

MODULATING DRIVE EFFICIENCIES AND COMPARISONS

Compressor Drives

With the expected operating torque characteristics established, the modulating-drive efficiencies of the compressor drive alternatives were evaluated at the appropriate torque/frequency ratio combinations.

Prototype PM-ECM Drive. In Figure 10, the torque/frequency curves representing the normal compressor operating lines of Figures 8 and 9 are shown together for heating and cooling mode operation. The points of nominal design cooling condition and maximum design heating (H) and cooling (C) conditions at nominal speed are also shown. These operating curves are superimposed on a contour plot of *system* (combined motor and speed control) efficiency of a compressor PM-ECM modulating drive as a function of frequency ratio and fraction of nominal design torque.

The efficiency contours were generated from data taken from a bench test of a prototype, 4-pole, surface magnet, PM-ECM with a pulse-width-modulated (PWM), IGT-based control. The motor is appropriate for a 2 to 3 ton (6.8 to 10.2 kW output) application (depending on the choice of nominal speed). Windage and friction losses from the bench test were included in the data shown.[10] The diagonal dotted line in the upper right-hand corner of Figure 10 is the operation limit line (into the constant power region), which indicates that the highest torques can only be obtained at the lower drive frequencies.

For a nominal speed of 3600 rpm, the prototype PM-ECM motor provided 2.1 hp at full load torque (assumed to be 1/1.36 times nominal design cooling torque). Because the specific application considered here required 3 hp at full load, we have assumed for present purposes that the motor could be scaled up in size from 2.1 to 3 hp with minimal effect on the efficiency contours. In this way, the PM-ECM efficiency can be more properly evaluated under a realistic sizing strategy.

[9]This occurs because of increasing blower efficiency at higher pressure drops.

[10]Excluding the bench windage and friction losses, the PM-ECM drive performance would be higher by 3.4 to 0.4 percentage points at the lowest speed as torque increases from lowest to highest. At the highest speed, the numbers would be 7 to 1.4 points higher from lowest to highest torque, respectively.

From the superposition of torque requirements on efficiency contours, it can be seen that the matching of motor performance to compressor loading, while not ideal, gives good performance overall.[11] The highest efficiencies are at the higher frequency operation points in heating mode, followed by the nominal to intermediate frequencies in either mode. As to be expected, low speed performance is reduced because of the increasing effect of fixed motor and inverter losses, but the reduction is only by about 10% in either mode.

Without the bench test windage and friction losses, the PM-ECM efficiency would be about 1 percentage point higher *along the operating line* in the cooling mode and 1 to 2.4 points higher from low to high speed operation in the heating mode. However, it should be noted that the bench windage and friction losses are likely to be lower than those in a hermetic compressor, refrigerant environment.

Late '70s Industrial IDIM Drives. A similar superposition of operating torque was conducted on IDIM and reference SWDIM efficiency data reported by Lloyd (1982). Here the corresponding performance values were obtained for an IDIM consisting of a 2-pole, 4 hp (3 kW), standard production three-phase compressor motor with a PWM power-transistor inverter drive. The motor size was scaled to 3 hp (2.24 kW), again to properly apply the predicted torque range. A constant volts/hertz ratio was maintained in all tests for near maximum motor performance. The Lloyd data were limited to 30 and 60 Hz operation. It was understood[12] that bench windage and friction losses were included in these data.

State-of-the-Art (SOA) Production Residential IDIM Drives. Compressor drive system bench efficiency data for a SOA production residential IDIM drive were obtained from Lloyd (1987) as were estimates of the corresponding reference SWDIM performance. The SOA IDIM drive uses a 2-pole, 3 hp (2.24 kW), three-phase motor designed for variable speed operation;[13] therefore, low rotor resistance and high efficiency stator design features were included. The inverter is of the 6-step voltage-source (VSI) design using modern power device technology. Bench windage and friction losses are understood to be included in the data as well.

First Generation Production Compressor. The preceeding three sets of data were based on dynamometer bench tests at 77 F (25°C) motor temperature. Data were also available from Miller (1988) on the relative installed performance of a first generation modulating reciprocating compressor alternately driven by variable-frequency sine wave and two types of IDIM designs. The sine-wave to IDIM comparison tests were each conducted under the same modulating conditions (with constant volts per hertz control) for heating and cooling mode operation. All tests used the same compressor motor. The original inverter was a 1980 production residential 6-step (or VSI) type which was later replaced by a comparably-sized PWM industrial inverter of early '80s design.

Sine-wave efficiency information was obtained from the manufacturer (Zigler 1987) for the specific 3 hp (2.24 kW), 2-pole, 3-phase induction compressor motor used in this test over a range of frequencies (with constant volts per hertz) from 15 to 90 Hz. The motor was sized for maximum cooling mode operation of 60 Hz. The motor efficiency information was generated from simulations[14] using the known motor characteristics and assumed stator and rotor operating temperatures of 176 F (80°C) and 185 F (85°C), respectively. No friction or windage effects were included in these efficiencies. The provided efficiency information was used to obtain a SWDIM reference curve by applying the torque profiles shown in Figure 10 for heating and cooling modes.

System drive efficiency curves for the two IDIM designs tested by Miller (1988) were then generated by applying his calculated compressor isentropic efficiency loss percentages to the common reference SWDIM curve. Adjustments were made to the losses based on Miller's data at various ambients to approximate as closely as possible the operating line ambients used with the other drive systems.

[11]Should overspeed operation be required of this motor in the cooling mode, the operating lines would have to be shifted to the left by at least 20% of nominal torque to allow overspeed operation at the maximum design condition. This would reduce efficiencies by 1 to 2 percentage points at the higher heating speeds and increase efficiency by about 1 point at the lower speeds.

[12]Personal communication with J. Kingman, Emerson Electric, October 1987.

[13]The inverter losses were backed out directly while the waveform motor losses were estimated from theory.

[14]Motor model predictions were also made at the standard 77 F (25°C) motor temperature for 60 Hz operation and were found to compare well to standard measured bench data over the range of load torques.

The main differences between the first generation drive efficiency estimates and the other cases are threefold. First, the Zigler reference SWDIM data is for a more typical motor operating temperature than the other reference SWDIMs. Second, friction and windage effects are not included in the Zigler data.[15] Third, the system IDIM efficiencies are results derived from the SWDIM reference curve based on actual compressor operating data not bench tests. Therefore the effects of torque variations per revolution on motor performance are included as are any additional losses in compressor efficiency from increased suction gas superheating because of the increased motor losses.

Consideration should be given to these differences in any absolute comparisons made between the derived application results and the other bench data cases. Even so, the comparisons should serve to provide useful indications of drive performance at various levels of technology development.

Comparative Results. The comparative efficiency results for the various types and development levels of modulating drives are shown in Figure 11. The results are shown as a function of frequency ratio and alternatively compressor drive frequency appropriate to either the 2-pole SWDIM and IDIM cases or the 4-pole PM-ECM motor. As noted earlier, the efficiencies were all evaluated at appropriate torque profiles for heating and cooling operation. The given SWDIM curves provide useful reference points between the ideal (or nearly ideal) modulating performance of a given induction motor, that when inverter-driven, and that of an inverter-driven PM-ECM.

From the comparisons in Figure 11, *the system efficiency of the prototype PM-ECM is seen to be nearly equal to or higher than the* average of the three available SWDIM efficiency curves, even though the ECM performance includes inverter losses.[16] Compared to the SOA production SWDIM (drive B), the ECM drive is almost identical in heating and cooling mode except for losing slightly at a frequency ratio of 0.5 in heating mode. However, at frequency ratios below 0.5, drive B and D SWDIM curves would be expected to fall off sharply as in SWDIM curve C (because of fixed slip losses), again giving the PM-ECM system an advantage.

This comparable performance for the PM-ECM system is possible because permanent magnet motors have inherent efficiency advantages over induction motors. Two major induction motor losses, rotor slip and stator magnetization losses [Richter 1984], are eliminated by use of the permanent magnet synchronous design. Furthermore, the inverter-driven PM motor is much less susceptible to harmonic losses than are IDIM's and therefore the PM-ECM system efficiency holds up better at lower speeds. *The PM-ECM motor efficiency advantages essentially offset the ECM inverter efficiency loss.*

Comparison with the available IDIM data in Figure 11 *shows* a *substantial efficiency advantage for the PM-ECM,* especially when compared to the late '70s and early '80s drives. Although the improved SOA IDIM (curve B) has reduced the PM-ECM efficiency advantage by 40 to 50% between half and nominal speed compared to the late '70s industrial IDIM (curve D), the difference still remains 11 to 14% and 9 to 10%, at half and nominal speeds in heating and cooling modes, respectively. No conclusions can be made regarding the relative performance of the PWM versus 6-step drives, as the available data represent different technology levels.

These and additional quantitative comparisons of the curves in Figure 11 are given in Tables 1 and 2. In Table 1, the prototype PM-ECM system efficiency gains are shown relative to each of the IDIM systems. The PM-ECM performance at its lowest frequency ratio is seen to be nearly double that of the derived application efficiencies of the older IDIM systems (for which comparable data are available). It remains to be seen how much this gap has been narrowed with SOA IDIMs. (Such data are needed to determine the point at which drive losses outweigh the heat exchanger unloading gains of the lower speeds).

In Table 2, comparisons of all the drive systems are made to their most appropriate reference SWDIM efficiencies. For the IDIM cases, these comparisons show the net drive losses from the inverter and waveform inefficiencies. These losses are contrasted with the prototype PM-ECM's much closer tracking of the performance of the SOA SWDIM.

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[15]The effects of a higher operating temperature and the omission of windage and friction losses should somewhat offset each other.

[16]This implies that the peak power demand of a PM-ECM system should be less than that of a single-speed heat pump with a *single-phase* motor. Also, peak demand would be further reduced by variable-speed operation at just the output needed rather than cycling with an oversized single-speed unit.

IDIM Loss Breakdown Discussion. The losses in the IDIM drives compared to sine wave performance can be categorized into two parts. The first are the *losses due to energy dissipation in the inverter drive* itself. Lloyd's (1982) data indicated that, for the late '70s PWM IDIM, the inverter-only losses were around 75 to 80% of the total losses shown in Table 2 at 60 Hz and 35 to 43% at 30 Hz.[17] Miller [1988] measured inverter-only losses on an early '80s PWM IDIM ranging from just below 50% of the total loss at 60 Hz and above, 30 to 32% at 30 Hz, and 21 to 25% at 15 Hz. In Miller's case, total losses were derived from operating hermetic compressor tests instead of a bench test.

The second type of drive losses are the *additional motor losses* generated by the harmonic content of the inverter-supplied waveforms. For the late '70s inverter shown by Lloyd (1982), the magnitude of the harmonic losses at 30 Hz was attributed to a simplified inverter design. He noted that the PWM-IDIM tested did not optimize the pulse sequence with frequency but only varied the on-time per pulse to maintain constant volts per hertz operation. The early '80s inverter tested by Miller (1988) was purported to diminish the harmonic effects by incorporating a microprocessor to optimize the pulsing sequence. However harmonic effects (100%–inverter losses) were larger in Miller's case at 30 Hz operation—68 to 70% compared to 57 to 65% shown by Lloyd.

Because the harmonic loss would be expected to be less for the PWM tested by Miller, this suggests that, in a hermetic compressor application, an indirect source of additional loss occurs. This is most likely the loss resulting from the *additional suction gas superheating* caused by the motor heating from the waveform harmonics. The reduced refrigerant flow rate available for motor cooling at low speeds would also tend to aggravate the amount of additional superheating. Because PM-ECMs do not suffer similarly from harmonic heating effects, the bench tests differences between IDIMs and PM-ECMs may underestimate the PM-ECM performance advantage in a hermetic compressor application. Alternatively, comparisons of bench tested PM-ECM performance to the derived IDIM application efficiency data may provide a more representative comparison of application performance differences than bench to bench data comparisons.

Indoor Blower Drives

In Figure 12, the blower nominal design torque load is shown as a square law function of drive frequency.

Prototype PM-ECM Blower Drive. The blower load in Figure 12 is superimposed on a contour plot of system (combined inverter and motor) efficiency for a prototype 12-pole, 1/3 hp (0.25 kW) indoor blower PM-ECM. The inverter control was a 1/3 hp (0.25 kW), FET-based PWM design. These prototype performance data were also obtained from Harms (1985) as a complement to the PM-ECM compressor data. PM-ECM system efficiencies were read from the blower load curve in Figure 12 for a frequency ratio range of 1/3 to nominal for comparison to other tested modulation cases. Extrapolations of the efficiency data were required for the lowest speed/torque points. Nominal speed for the PM-ECM drive was selected here to be 1400 rpm at 140 Hz operation.

Early '80s IDIM Blower Drive. Combined blower and blower motor (including inverter) efficiencies were obtained from tests by Miller (1988) on installed performance in a split-system indoor air handler using an early '80s model 6-step IDIM drive and a reference SWDIM drive. The nominal sine wave efficiency of the three-phase, 1/3 hp (0.25 kW) air handler motor used in Miller's test was obtained from the manufacturer as 75%. From this information and the knowledge that the blower-only efficiency should remain essentially constant with motor speed [McQuiston 1977], a constant blower-only efficiency of 28.5% was computed and assumed for all speeds. This value was applied to Miller's air handler data to convert to modulating drive-only efficiencies (motor and inverter only).

For the reference sine-wave (SWDIM) drive tests, the applied volts/hertz ratio was adjusted in-situ (Miller 1988) at each tested operating frequency for minimum current and thereby maximum efficiency. Thus the measured SWDIM efficiencies should represent ideally driven induction motor performance.

For the 6-step IDIM tests, the voltage supplied to the motor was controlled directly by the 6-step voltage-source-inverter (VSI). The VSI did, to our understanding, also reduce the voltage at lower frequencies in a manner intended to maximize motor performance at reduced torques (Mohan 1981, p. 4–6, [18]).

[17]At the appropriate torque conditions in heating and cooling modes.

[18]Personal communication with Ken Cooper, York International, York, PA, May 1982.

SOA IDIM Blower Drive. Comparable blower drive system efficiency data for a SOA production IDIM drive and reference SWDIM drive were also obtained from Lloyd (1987). The SOA blower IDIM drive uses a 6-pole, 1/3 hp (0.25 kW) three-phase motor with a nominal speed of 1080 rpm at 60 Hz frequency. The inverter is of 6-step VSI design. The efficiency data was for a square law blower load with full load torque at maximum speed.

Efficiency Comparisons. The three sets of efficiency data are compared in Figure 13. The PM-ECM blower drives were found to equal the performance of the sine-wave-driven cases at the highest frequencies and to exceed the reference values of the early '80s and SOA SWDIMs by 13% and 50% (7 and 20 percentage points) respectively, at the lowest frequency ratio. The reason why the SOA SWDIM motor efficiency is exceeded by Miller's early '80s design is not known. However, *in a comparison of the total inverter-drive losses* (referenced to the respective SWDIMs), *the SOA SWDIM drive has smaller losses* of 7.4% at maximum frequency and 52.5% at lowest frequency as compared to 18% and 77% for the early '80s inverter design. The SOA IDIM has thus reduced the 60 Hz losses by 60% while reducing the 20 Hz losses by only 32%.

Comparison of the PM-ECM system efficiencies *to the 6-step IDIM systems* show efficiency gains of 216 and 344% (46.5 and 41 percentage points) at the lowest frequency ratio and 8 and 21% (5.5 and 13 percentage points) at the highest frequency ratio.

Thus the efficiencies of the two IDIM drives are seen to be strongly degraded at frequency ratios below about 2/3 of nominal. Again, this must be accounted for by inverter and harmonic (waveform) losses. Additional data (not shown) provided by Lloyd (1987) and further measurements shown by Miller (1988) indicate that *most of this loss* (70 to 90% in Lloyd's data and 90 to nearly 100% in Miller's data) *is due to direct inverter losses.* This is due to direct inverter losses. This is in contrast to the compressor case where more to most of the losses at lower frequency ratios were due to the motor harmonics. Thus further improvements in the lower-power blower inverter components or new design approaches to minimize inverter energy dissipation may reduce these losses.

Blower and Compressor Drive Comparison Summary—Power Reductions

Although the blower system IDIM efficiencies at the lowest frequency ratio operation are bettered by the PM-ECM drive by factors of 3 or more, the improvements in performance at intermediate to maximum speeds are equally important in terms of reduced absolute blower power use. This is because, for a blower load, the blower shaft power is proportional to the cube of the blower speed. At one-third speed, the ideal power is reduced to 1/27th of the nominal speed value. Because of this, a further two-thirds decrease in power requirement, because of the PM-ECM efficiency increase, is approximately the same as the power reduction at higher speeds. The power reductions relative to SOA IDIM power *at nominal speed* are shown in Table 3 for a range of blower frequency ratios. Power reductions of the PM-ECM drive over the SOA IDIM drive are seen to peak at almost 12% at a frequency ratio of about two-thirds of nominal. Elsewhere, reductions stay fairly constant around 10% except for a dropoff at maximum speed.

For a compressor load, the shaft power requirements only decrease approximately linearly with speed (power = torque x speed) because torque requirements stay relatively constant (compared to a blower load). As shown earlier in Figures 8 and 9, while compressor torque may either decrease (in heating) or increase (in cooling) with speed, in either case the change is much less than the speed change ratio (torques of 73% down to 48% of nominal in heating mode and 60% up to 100% of nominal in cooling).

However, offsetting the higher required shaft power at lower speeds, the percent drive efficiency reductions at lower speeds for the SOA IDIM case (at least down to 30 Hz) are not nearly as great in Figure 11 as Figure 13. The net compressor power reductions are given in Table 4. Power reductions are seen to be lowest[19] at a 0.5 frequency ratio and increase slowly in heating mode to a maximum of just over 5% at 25 percent overspeed. In cooling mode, savings are greater than 5% at half speed, increasing to 9% at nominal speed.

[19]Power reductions at quarter speed (although SOA IDIM data are not available) will be higher than at half speed if the SOA IDIM system efficiency falls off like the earlier model IDIMs (curves C/C' in Figure 11.

In summary, *for indoor blower applications,* the PM-ECM drive shows fairly uniform power reductions, relative to a SOA IDIM, of 10 to 12% at all speeds except the highest. *For the compressor drives,* the power reductions are generally larger at higher speeds ranging from 3.5 to over 5% in heating and 5.4 to 9% in cooling mode.

DISCUSSION

IDIM Drives

The IDIM results shown in Figures 11 and 13 are in general agreement with the overall drive-efficiency trends noted by Mohan [1981, section 4.7] for PWM and VSI (6-step) drives at full and reduced speeds with "constant torque" (compressor) and blower load applications, respectively. Although the drive efficiencies of the production IDIM's shown in Figures 11 and 13 are somewhat below that of the prototype PM-ECMs, there are other important considerations favorable to the IDIM drives at the present time. They include the following:

- The reliability of three-phase induction motors have been proven over time and represent an improvement over the presently used single-phase induction motors and proposed ECM drives.
- The cost of the IDIM drives for compressors are lower than PM-ECM drives. This is mainly because of the increased cost from the permanent magnet material which is not offset by the reduced steel content of the rotors. Inverter costs should be about the same for both designs because of offsetting design requirements.
- Even though the IDIM drive efficiencies are lower, they still will provide a large portion of the modulating *system* gains from heat exchanger unloading. The efficiency advantages of the PM-ECM must be justified on the basis of the remaining marginal gain.

PM-ECM Drives

Based on the information available to the author and presented here, the prototype PM-ECM drives offer a clear efficiency advantage to production IDIM drives over the expected range of compressor and blower loads for modulating heat pumps. The degree to which production versions of the PM-ECM drives achieve these efficiencies will depend on the approaches taken to deal with the reliability, cost, and production factors.

Reliability issues relate primarily to two new possible failure modes that do not exist with induction motors—magnet retention and demagnetization.

Present ECM prototype motors typically have permanent magnets mounted circumferentially with an adhesive on the surface of an iron rotor core. Various approaches are being tried to prevent the magnet adhesive or pieces of the brittle magnet material from dislodging at higher speeds. (This problem is aggravated if alternative compressors with higher maximum speeds are considered.) The hermetic compressor application is more difficult than an open application because the refrigerant gas cooling the motor tends to dissolve the adhesive bonds. Any bonding or wrapping approaches found successful in prototypes must also prove practical for the production line.

Other magnet retention possibilities include embedded (or buried) magnets within the rotor and various forms of metal retaining bands or wire meshes. The band approaches add to the costs and incur varying degrees of efficiency loss due to eddy currents. The embedded magnet approach is less costly and offers excellent demagnetization protection but with some loss in efficiency.

The demagnetization problem with surface magnets is greatest at low temperatures, especially with the ferrite magnets considered here. The compressor application would be the most susceptible at cold ambients in the heating mode. Under such conditions, a failure of the power transistor circuit in the inverter to prevent high currents from passing to the motor could make the motor useless.

Another issue has been raised by Mohan and Ramsey [1986] on the the technical feasibility of PM-ECMs for heat pump application. They have suggested that the use of PM-ECMs "may impose a severe penalty in terms of oversizing the compressor motor and inverter..." to allow operation at 50% above rated speed.

Discussions with a PM-ECM manufacturer[20] indicate that present PM-ECM designs (as opposed to more conventional synchronous motors) do not require proportional motor or inverter oversizing to operate up to 50% above rated speed at reduced torque such as that shown in Figures 8 and 10 for heating mode conditions. In such cases, the motor size would instead be dependent on the torque requirements at the rated condition and would not change. The inverter current requirement would only be increased a portion of the overspeed percentage. This is because the no-load speeds of the PM-ECMs are already over 50% over nominal 60 Hz speed. Therefore, this oversizing penalty appears to be somewhat overstated.

There are also differing technical opinions on how soon production ECMs in compressor applications will approach the system prototype efficiencies shown here. However, it seems highly probable to the author that improvements in the PM motor and inverter technology over the next five years will offset most of any effficiency penalties taken to ensure adequate motor reliability. Therefore in the time frame of advanced modulating system considerations (5 to 10 years), the PM-ECM efficiency values given here would be suggested as appropriate values for use by researchers and designers of advanced modulating systems.

The higher efficiency capabilities of the PM-ECM, future magnet cost reductions, and sufficient reliability experience will likely make PM-ECMs a leading candidate for compressors of future generation modulating heat pumps. The main point of contention at present seems to be how the PM-ECM systems would be phased into modulating compressor applications. A possible scenario might be that PM-ECM compressor drives could be adopted in variable speed systems as compressor manufacturers retooled for scroll or rotary compressor technology—provided cost and reliability issues were resolved. However, some U.S. manufacturers are presently developing reciprocating compressor modulating systems using PM-ECM technology.

For indoor blower drives, present PM-ECM drive costs are in the same range as IDIM drives. The reliability issues of magnet retention and demagnetization are less of a concern here because of the lower motor speeds, the higher indoor temperatures, and the more favorable operating environment. The prototype efficiencies shown in Figure 13 have been closely matched in production. Thus more operating experience appears to be the only element needed for wider market acceptance in modulating blower applications.

Generalization of Modulating Compressor Operating Conditions and Torque Characteristics

The operating conditions and torque results shown in Figures 1–9 were developed for a reciprocating compressor with a specific modulation strategy. However, with appropriate interpretation and minimal adjustment, these results can be generally applied to other *reciprocating* compressors, speed ranges, and sizing strategies. For compressor with significantly different volumetric efficiency trends, such as scroll or rotary, a more limited but still useful generalized application can be made along the operating lines.[21]

In generalizing the results of Figures 1–9, the key assumption is that the obtained operating conditions (i.e., evaporating and condensing temperatures, pressure ratios) will remain unchanged with appropriate speed scalings, at least along the operating line. This is equivalent to assuming a fixed modulating heat pump design (i.e., fixed compressor to indoor blower speed ratios, specific condenser subcooling variations with ambient, and specific heat exchanger sizes). Effects of different relative heat exchanger sizings, sub-optimal refrigerant control, or different blower speed control strategies would change the torque profile to some degree. The operating conditions given here can be assumed to be a close approximation (especially along the operating line) to those expected in an optimally-controlled modulating heat pump with similar frequency ratio ranges, provided the nominal speeds are scaled properly.

For modulating systems with similar compressor volumetrics, the controur plots relating directly to operating conditions (Figures 1, 2, and 5) can be assumed to apply exactly. Those plots which include compressor efficiency effects (Figures 3, 4, 6 and 7) would hold only approximately. However, using the fixed

[20]Personal correspondence with R. V. Zigler, General Electric Motor Technology Department, November 1986.

[21]These generalizations can be used to see approximately how the torque will change with speed, without recourse to a completely new modulating system analysis.

operating conditions of Figure 2, a new plot of compressor efficiency effects (Figure 6) could be obtained for each different compressor to adjust the basic R-22 conditions torque plot (Figure 5). In this way, a new torque curve could be obtained (with minimal recalculation) which included the specific compressor efficiency characteristics as a function of ambient conditions and frequency ratio.

A similar argument can be made to generalize the torque curves and operating conditions *for other sizing or speed ranges* with comparable compressor volumetrics. (Note that in Figures 1–9, the given frequency ratios only apply "exactly" for a nominal speed of 3600 rpm.) The compressor would simply be assumed to be scaled in size to give the same capacity and thereby heat exchanger loading at the corresponding new nominal speed. Only the new compressor efficiency effects (Figure 6) at the translated speed ratios would be needed to obtain a new set of torque predictions. This generalization would apply to speed modulated compressors, but not to stroke or other means of displacement modulation. This is because, as shown in Equation 5, with speed modulation, torque is only indirectly related to speed through the compressor efficiencies.

For modulating systems with different compressor volumetrics, such as scroll or rotary, the generalizations must be narrowed to the operating lines only (rather than over the entire frequency ratio and ambient region). This is because compressors with strongly different volumetric efficiency trends will significantly change the heat exchanger loadings as a function of speed or ambient. In this case, only along the operating lines can a generalization of operating conditions be applied. This is because the operating lines correspond to fixed building load lines and thus fixed capacity requirements.

While the different compressor type can be assumed to operate over the same operating lines, the corresponding frequency ratio will be different from the baseline reciprocating case (to give the same capacity at a fixed ambient) to reflect the difference in volumetric efficiency trend. For example, the operating line over the heating frequency ratios of 0.25 to 1.5 for a reciprocating compressor might be traversed by a frequency ratio of only 0.25 to 1 for a scroll compressor because its volumetric efficiency falls off less with increased pressure ratio. This translation of speed range will not necessarily be linear. Therefore the intermediate speed values must also be evaluated to obtain speed vs torque operating curves like those given in Figure 10.

To allow such a translation to be made, the normalized volumetric efficiencies of the base reciprocating compressor are given in Table 5 at selected speeds along the operating line. The frequency ratios, when multiplied by these normalized efficiencies, give the generalized volumetric pumping ratio. With similar volumetric efficiency numbers for the scroll or rotary compressor, the generalized volumetric pumping ratio can be used in reverse to obtain the equivalent frequency ratio of the different compressor.

Once the new (non-linear) frequency scale has been obtained, the operating lines can be adjusted to the new compressor by the same procedure as if it were just a different reciprocating compressor (i.e., the compressor effects multiplier) but only along the operating line points. For cases where the new compressor has a narrower modulating range than the base curves, only the portion that coincides with the base operating line can be evaluated accurately.

From these generalizations, it can be argued that the torque ranges to be expected from scroll and rotary compressors should not be any wider than those shown herein. The main differences to be expected is that similar torque ranges will be seen over narrower speed ranges.

CONCLUSIONS

1. Sets of of modulating drive performance characteristics for a range of technology levels for compressors and blowers have been presented which are suitable for advanced variable-speed system analysis.
2. For both compressor and indoor blower modulating applications, prototype PM-ECM drive system efficiencies (motor plus inverter drive) are shown to be nearly equal to or higher than those of sine-wave-driven three-phase induction motors and significantly better than IDIMs available in the late '70s and early '80s.
3. When compared to available data on more recent IDIMs, the PM-ECM efficiency advantage over IDIM compressors has been reduced 40 to 50% between half and nominal speed (3600 rpm), but still remains 14 to 9% higher, respectively. For blowers, the IDIM improvement has not been as large.

4. The bench measured efficiency advantages of PM-ECM drives are likely to be even larger in hermetic applications because of the compounding effect of increased suction gas superheating with IDIM drives.

5. Compared to SOA IDIM drives, the power reductions possible from the use of PM-ECM blower and compressor drives are around 10% and 5% of the respective nominal powers. These reductions are realized about equally over the full range of frequency for the blower while increasing with speed for the compressor. On a system energy basis, the compressor savings will be about 5 times the blower savings.

6. PM-ECM-driven *blowers* appear to be very competitive at present with IDIM blowers on the combined basis of efficiency, cost, and reliability. For the *compressor* PM-ECM drives, if progress continues on cost and reliability issues with minimal loss in efficiency, these drives will likely be a leading candidate for future-generation (5 to 10 year range) modulating heat pumps.

7. To maximize compressor performance on an annual basis for modulating heat pump application in a balanced climate, compressors should be designed for optimal efficiency at pressure ratios near 2.5. For cooling-only applications, a pressure ratio of 2.0 would be more optimal for best SEER. Furthermore, the compressor conditions given along the operating lines in Figures 1 and 2 should be representative, with proper speed scaling, of expected modulating conditions for various types of compressors with similar speed ranges.

8. The effective operating torque range for a modulating heat pump (with a reciprocating compressor) compared to a single-speed heat pump is about half as wide in the heating mode and twice as wide in the cooling mode. Over all heat pump ambients, however, both types of units cover nearly the same torque range. The modulating torque profiles with ambient suggest that, for a moderate climate, speed-modulated reciprocating compressor motors should be optimized for best performance at or slightly below full load (rated horsepower) torque.

9. A motor designed for 50 percent *cooling* overspeed operation will require twice as much overspeed torque capability than if limited to the same overspeed operation only in the heating mode at low ambients.

10. For speed modulated systems, the compressor torque / drive frequency mappings given in Figures 8 and 9, with suitable interpretation and minimal adjustment, should be applicable to other types of speed ranges and sizing strategies for reciprocating compressors. Compressor types with significantly different volumetric efficiency characteristics, such as scroll and rotary, will have somewhat altered torque profiles versus speed than shown here (although the torque ranges are expected to be no greater). Appropriate profiles for these compressors should be evaluated as the necessary modulating compressor data are available.

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TABLE 1
Efficiency comparisons of prototype PM-ECM to various IDIM compressor drives—2 to 4 hp (1.5 to 3 kW) motor sizes.

Frequency Ratio	PM-ECM Efficiency Gain Relative to IDIM Systems (bench data) B	D	(application data) C	C'
Heating Mode				
0.25	>>11.3*		91.9	97.6
0.5	11.3	22.3	50.9	34.8
0.75	8.9		31.9	24.6
1.0	8.6	15.2	18.4	20.0
1.25	8.1			17.9
1.5	5.9			19.3
Cooling Mode				
0.25	>>13.8*		84.9	97.1
0.5	13.8	24.2	36.2	35.5
0.75	10.0		24.9	24.9
1.0	9.9	20.2	26.8	26.8

B—SOA residential production IDIM (6-step).
C—1980 residential production IDIM (6-step).
C'—Early '80s industrial drive (PWM) with motor from "C".
D—Late '70s industrial drive (PWM) with standard motor.
*Estimated

TABLE 2
Comparison of net IDIM losses and PM-ECM efficiency relative to reference SWDIMS for compressor drives—2 to 4 hp (1.5 to 3 kW) motor sizes.

Frequency Ratio	PM-ECM to B SWDIM	IDIM Efficiency Compared to Reference SWDIMs B	C	C'	D
Heating Mode					
0.25	<−3.3*	>−13.0*	−36.1	−37.0	
0.5	−3.3	−13.0	−30.0	−21.6	−16.6
0.75	−1.1	−9.2	−20.6	−16.0	
1.0	0.0	−7.9	−12.1	−13.2	−11.3
1.25	+0.4	−7.1		−12.2	
1.5	+0.1	−5.4		−12.1	
Cooling Mode					
0.25	⩾ 0.0*	>−12.8*	−33.4	−37.5	
0.5	−0.8	−12.8	−19.4	−19.1	−15.8
0.75	+0.7	−8.5	−14.0	−14.0	
1.0	+1.0	−8.1	−15.9	−15.9	−12.6

B—SOA residential production IDIM (6-step).
C—1980 residential production IDIM (6-step).
C'—Early '80s industrial drive (PWM) with motor from "C".
D—Late '70s industrial drive (PWM) with standard motor.
*Estimated.

TABLE 3
Relative power reductions of PM-ECM blower compared to SOA IDIM drive.

Blower Frequency Ratio	Indoor Blower Power Reductions* (% of IDIM power at *nominal* speed)
1/3	10.6
1/2	10.1
2/3	11.6
3/4	11.4
5/6	11.0
1	7.4

*To obtain net energy savings, these reductions must be weighted by the fraction of operating hours at each frequency ratio.

TABLE 4
Power reductions of PM-ECM compressor drive compared to SOA IDIM drive.

Compressor Frequency Ratio	Compressor Power Reductions* (% of IDIM power at *nominal* speed) Heating Mode	Cooling Mode
0.25	?	?
0.50	3.5	5.4
0.75	4.4	6.7
1.0	4.7	9.0
1.25	5.1	
1.5	4.1	

*To obtain net energy savings, these reductions must be weighted by the fraction of operating hours at each frequency ratio.

TABLE 5
Normalized volumetric efficiencies of base reciprocating compressor along modulation operating lines.

Frequency Ratio	Ambient F	(°C)	Normalized* Volumetric Efficiency (relative to *nominal* cooling design point)
Heating Mode			
0.25	57.0	(13.9)	1.19
0.25	47.0	(8.33)	1.12
0.25	42.0	(5.56)	1.05
0.5	32.4	(0.22)	0.97
0.75	25.7	(−3.50)	0.95
1.0	21.0	(−6.11)	0.82
1.25	17.3	(−8.17)	0.77
1.5	14.0	(−10.0)	0.73
1.5	7.0	(−13.9)	0.69
Cooling Mode			
0.25	77.0	(25.0)	1.17
0.25	81.8	(27.7)	1.19
0.5	89.8	(32.1)	1.14
0.75	96.4	(35.8)	1.03
0.775	97.0	(36.1)	1.02
1.0	95.0	(35.0)	1.0

*The reference volumetric efficiency at the nominal cooling design point was 70.3%.

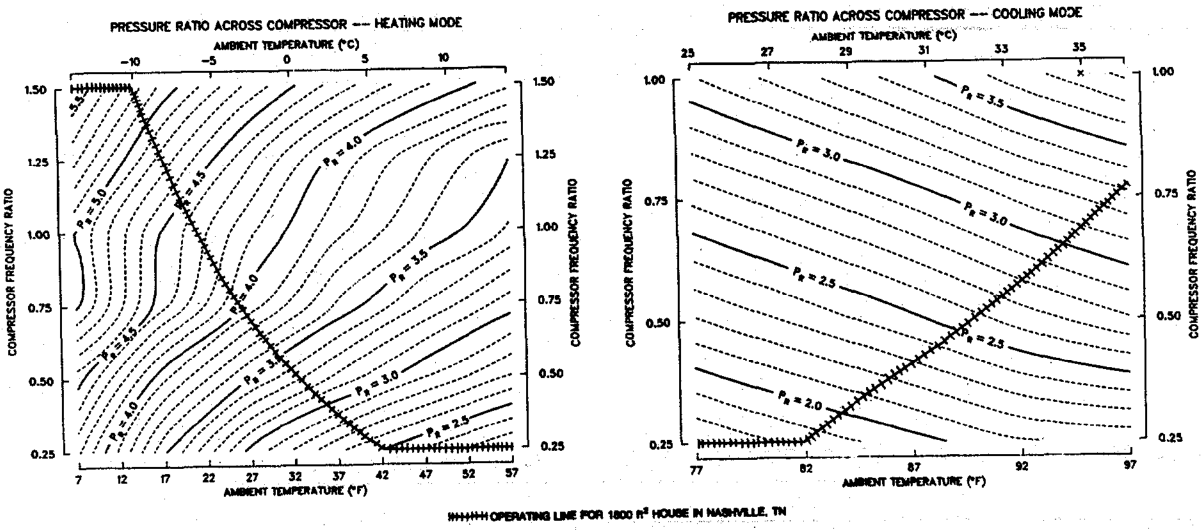

Figure 1 Contours of predicted compressor pressure ratios as a function of ambient temperature and compressor frequency ratio

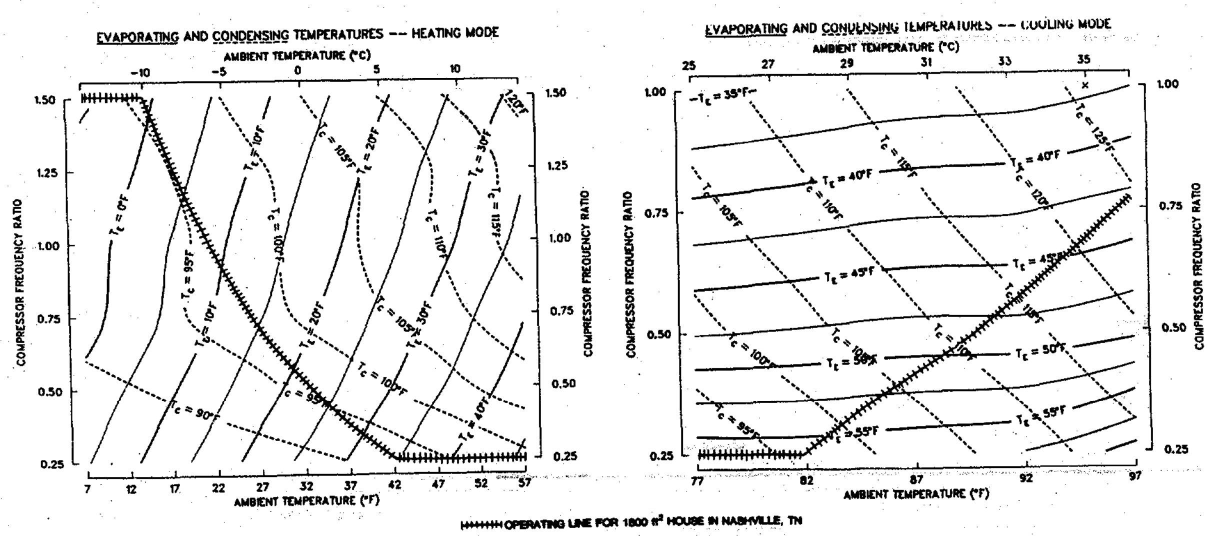

Figure 2 Contours of predicted refrigerant-22 saturation temperatures at compressor inlet and exit as a function of ambient temperature and compressor frequency ratio

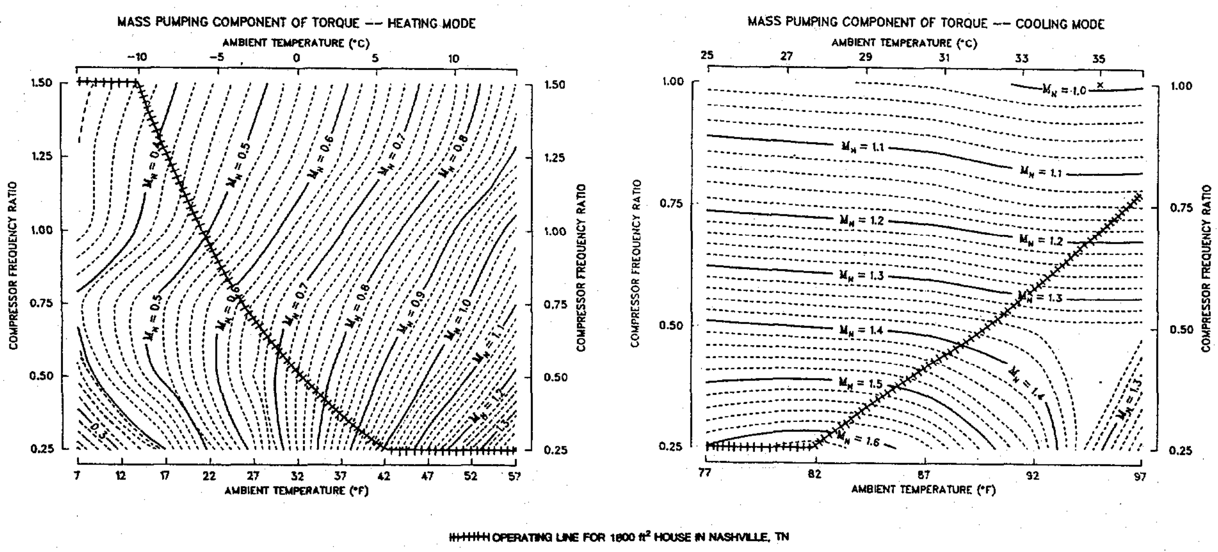

Figure 3 Contours of the <u>mass pumping component of torque</u>, $M_n = \rho_n \cdot \eta_{vol,n}$, as functions of ambient temperature and compressor frequency ratio

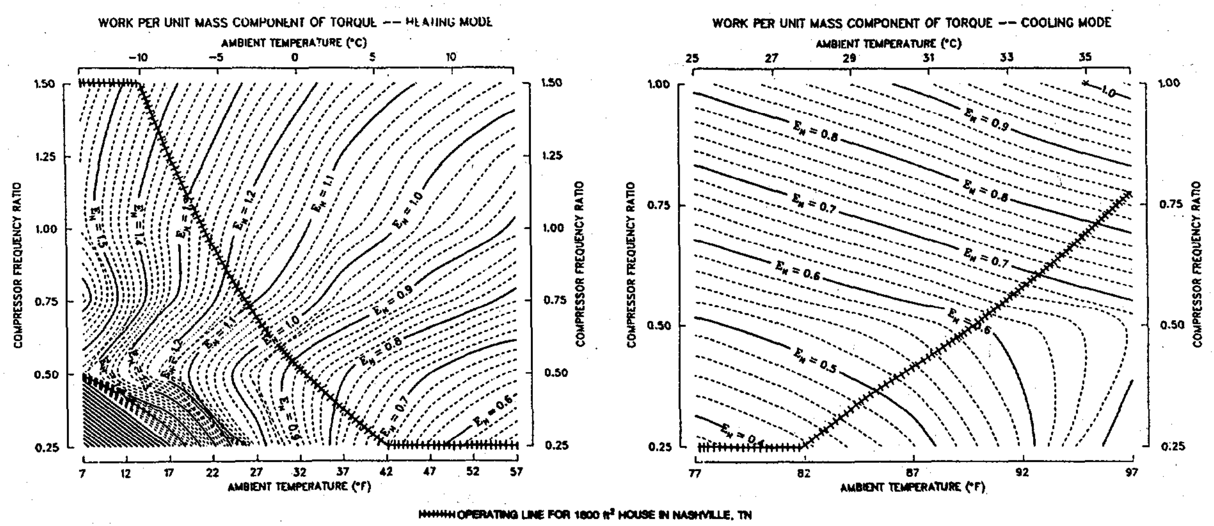

Figure 4 Contours of the <u>work-per-unit-mass component of torque</u>, $E_n = \Delta h_{is,n} / \eta_{is,n}$, as functions of ambient temperature and compressor frequency ratio

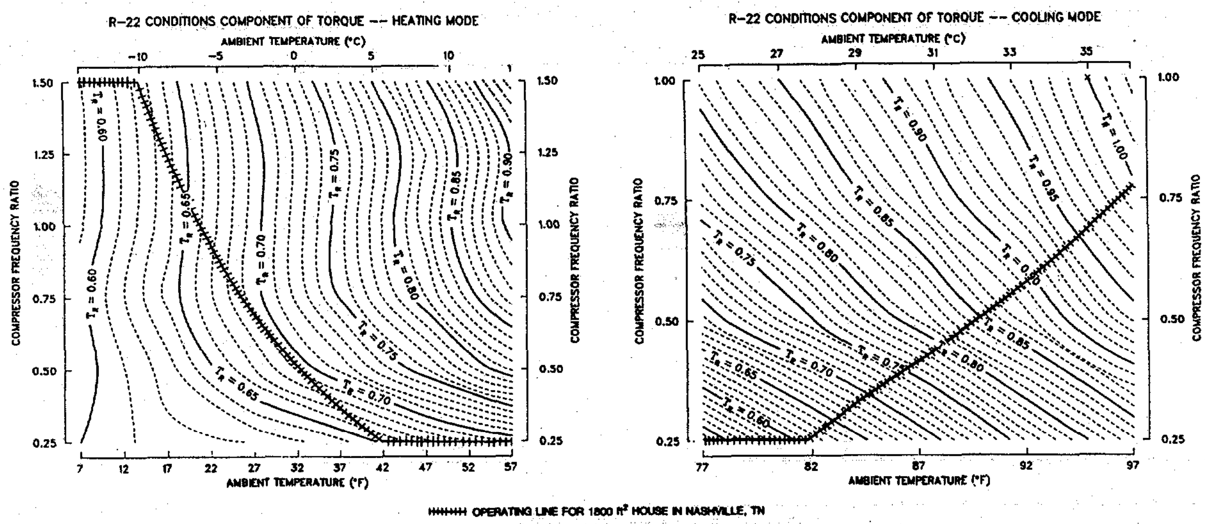

Figure 5 Contours of the R-22 operating conditions component of torque, $T_r = \rho_n \cdot \Delta h_{is,n}$, as functions of ambient temperature and compressor frequency ratio

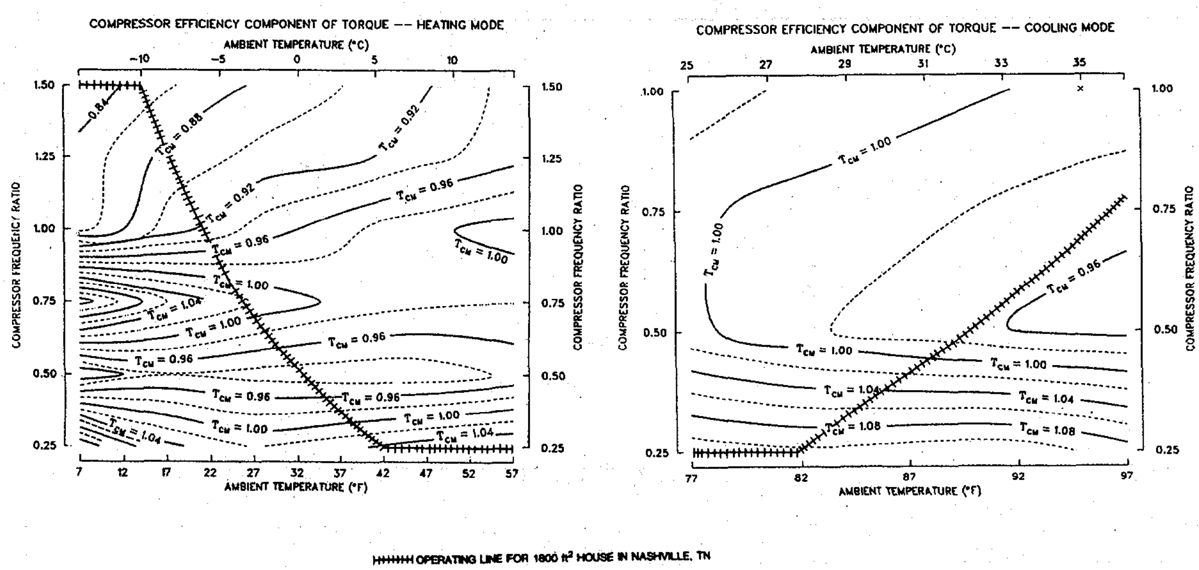

Figure 6 Contours of the compressor efficiency component of torque, $T_{cm} = \eta_{vol,n} / \eta_{is,n}$, as function of ambient temperature and compressor frequency ratio

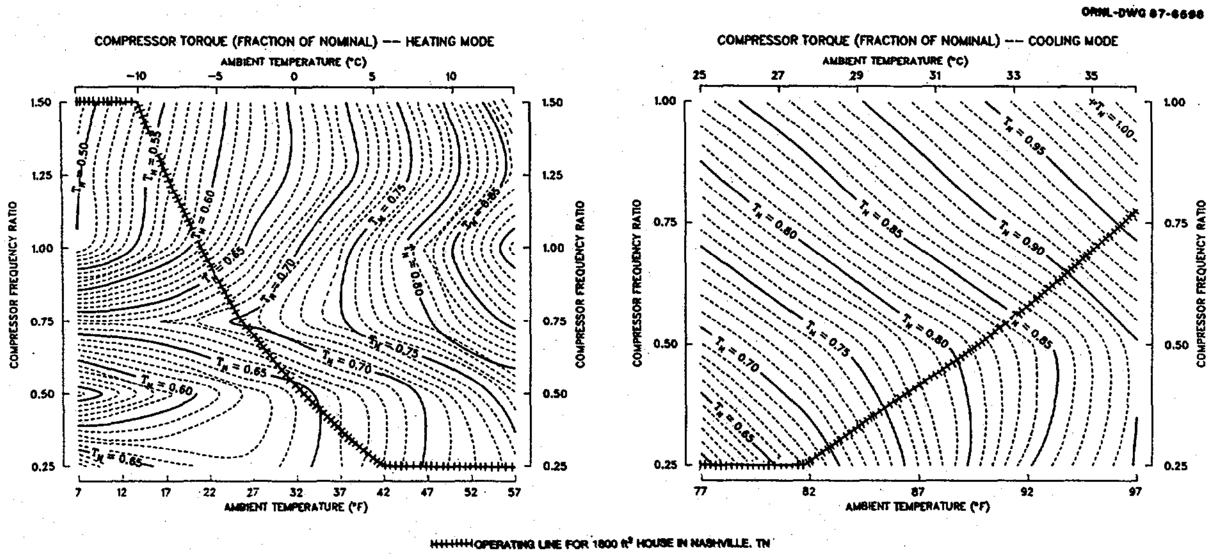

Figure 7 Contours of <u>compressor torque</u> (fraction of design cooling torque) as a function of ambient temperature and compressor frequency ratio

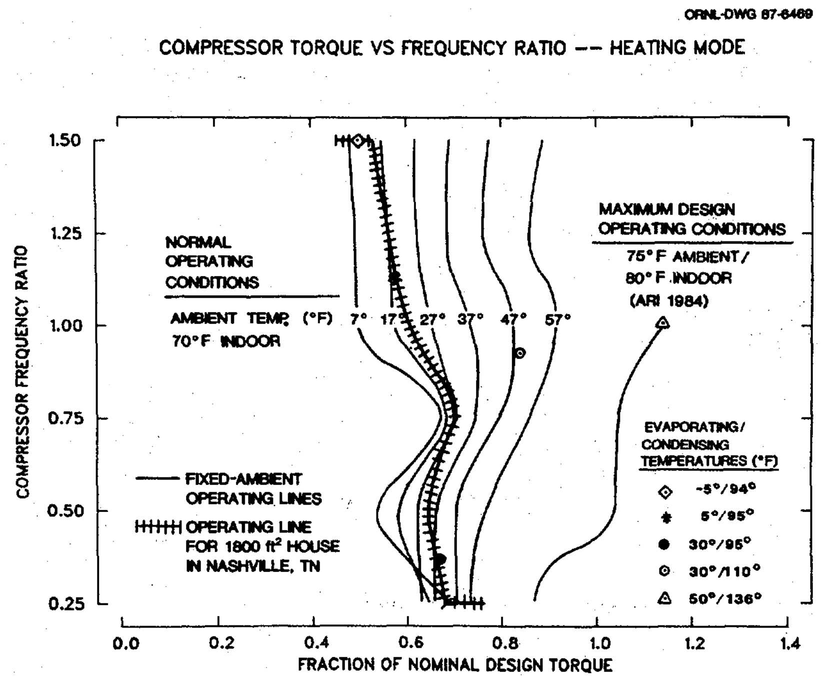

Figure 8 Reciprocating compressor <u>torque vs. frequency ratio characteristics</u> of a modulating heat pump for a range of ambient and fixed indoor temperatures--<u>heating mode</u>

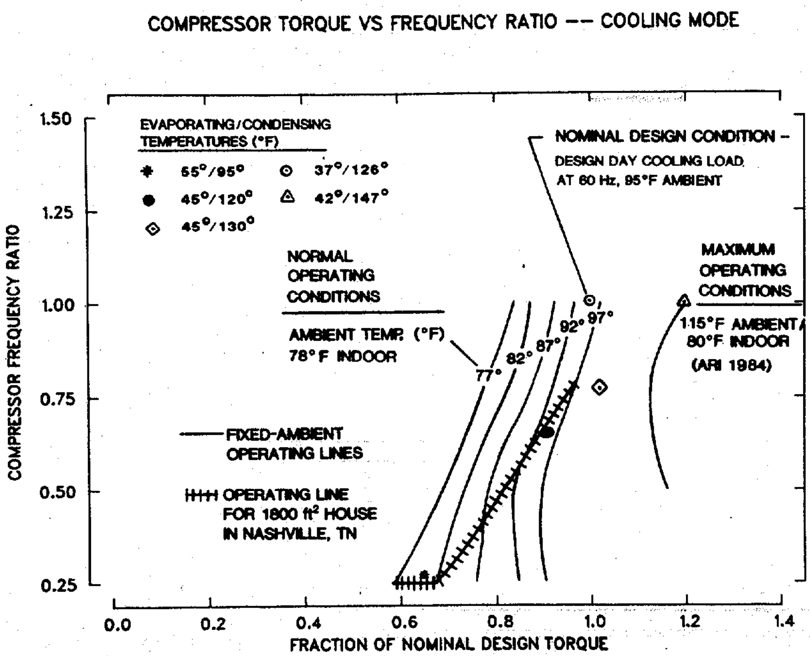

Figure 9 **Reciprocating compressor torque vs. frequency ratio characteristics of a modulating heat pump for a range of ambient and fixed indoor temperatures--cooling mode**

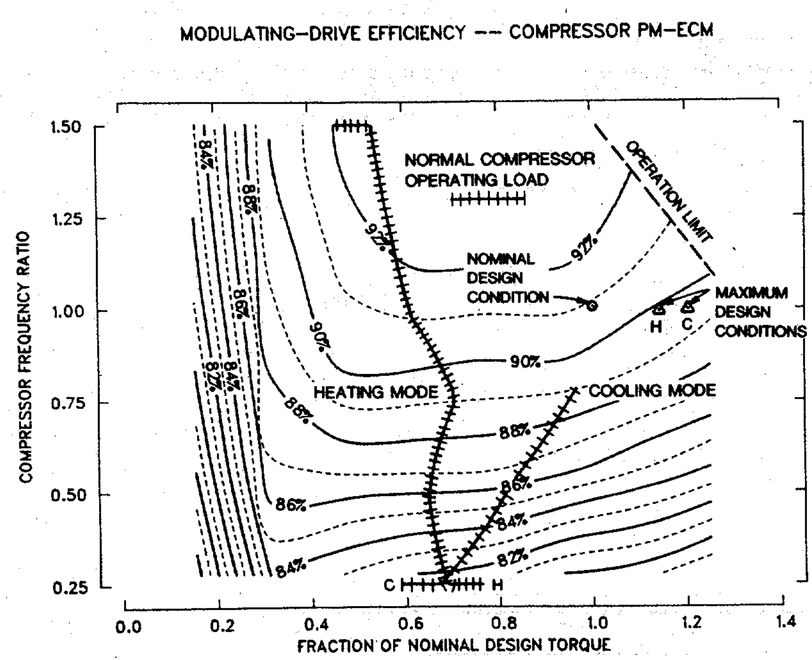

Figure 10 **Predicted modulating-compressor load characteristics superimposed on contours of modulating-drive efficiency for a prototype compressor PM-ECM [2 hp (1.5 kW) at 3600 rpm]**

COMPRESSOR MODULATING–DRIVE EFFICIENCIES
2 to 4 hp (1.5 to 3 kW) DRIVES

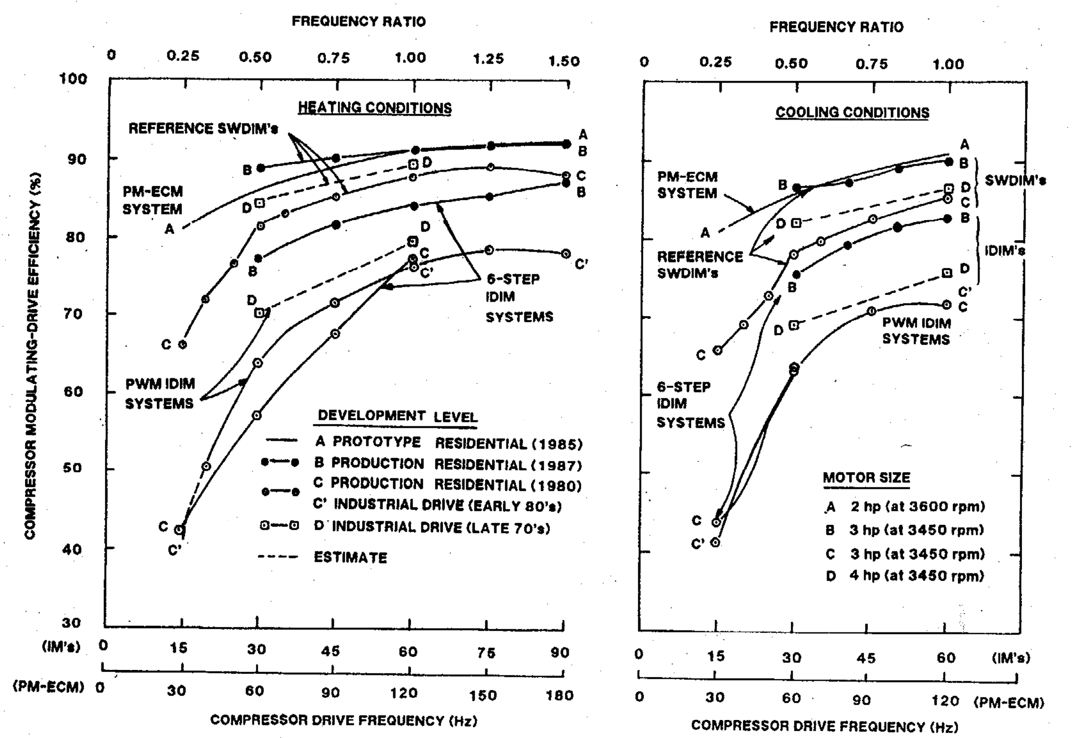

Figure 11 Comparison of compressor modulating-drive system efficiencies vs. frequency ratio / drive frequency for predicted compressor load conditions

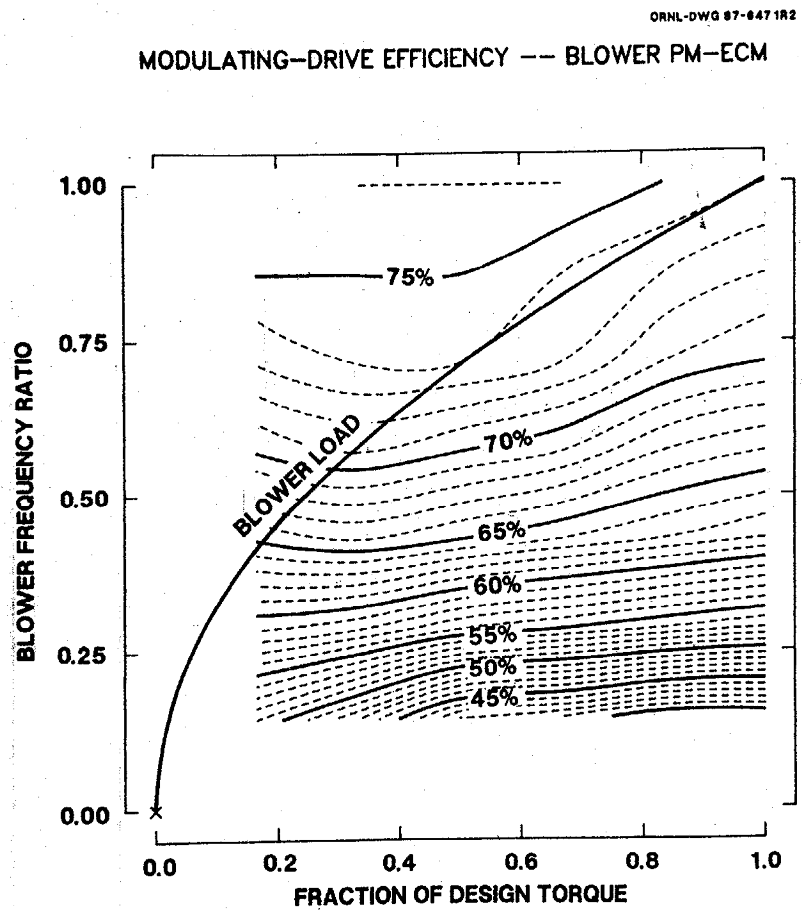

Figure 12 Modulating-blower load characteristics superimposed on contours of modulating-drive efficiency for a 1/3 hp (1/4 kW) blower PM-ECM

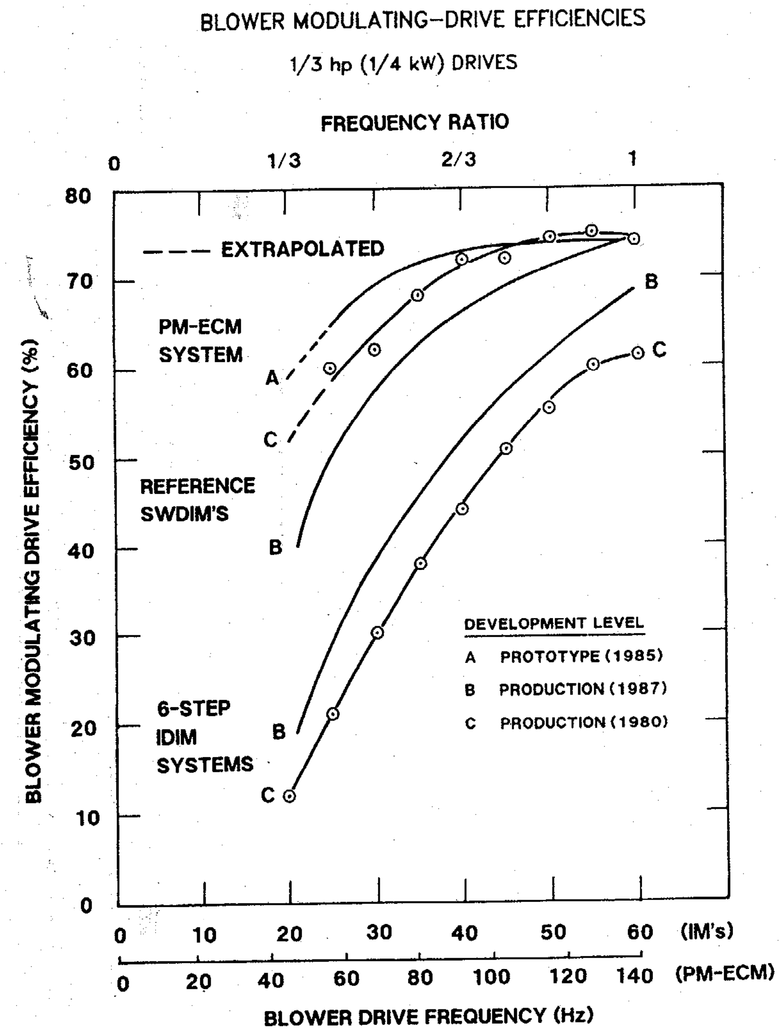

Figure 13 Comparison of blower modulating-drive system efficiencies vs. frequency ratio / drive frequency for typical blower load conditions

Discussion

C. K. Rice: Some supplemental tables are given below of the power reduction potential of PM-ECM drives relative to IDIM designs. These tables compare savings from IDIM power draw at the same frequency rather than from nominal IDIM power (as was done in Tables 3 and 4 of the paper).

The comparison to nominal power is most useful in observing absolute drive power reduction referenced to a single-(nominal)speed system. These nominal reductions can be used by drive manufacturers to determine the savings payback for an improved modulating drive of a given nominal size and power draw (provided the percent reductions are weighted for operating hour distributions with speed).

Comparisons to an IDIM drive at the same frequency show better the relative improvement from one modulating system to the next.

Power Reductions of Compressor PM-ECM Drive Compared to First Generation IDIM Drives

Compressor Frequency Ratio	Relative Power Reduction (from IDIM at same frequency) C (application data)	C' (application data)	D (bench data)
Heating Mode			
0.25	47.9	49.4	
0.5	33.7	25.8	18.2
0.75	24.2	19.7	
1.0	15.5	16.7	13.2
1.25		15.2	
1.5		16.2	
Cooling Mode			
0.25	45.9	49.3	
0.5	26.6	26.2	19.5
0.75	19.9	19.9	
1.0	21.2	21.2	16.8

Power Reductions of Compressor PM-ECM Drive Compared to SOA IDIM Drive

Compressor Frequency Ratio	Relative Power Reduction (from IDIM at same frequency) SOA IDIM (B) (bench data)
Heating Mode	
0.25	>10.2
0.5	10.2
0.75	8.2
1.0	7.9
1.25	7.5
1.5	5.6
Cooling Mode	
0.25	>12.1
0.5	12.1
0.75	9.1
1.0	9.0

Power Reductions of Blower PM-ECM Drive Compared to First Generation and SOA IDIMs

Blower Frequency Ratio	Relative Power Reduction (from IDIM at same frequency)	
	SOA (B)	1st Generation (C)
1/3	71.2	79.7
1/2	44.9	56.5
2/3	29.5	39.7
3/4	22.6	31.0
5/6	17.0	24.2
1	7.4	17.6

Average Reductions in Component Energy Use With PM-ECMs (Moderate Climate)

Component	Estimated Average Frequency Ratio	PM-ECM Energy Use Reductions From Base IDIMs	
		1st Generation	SOA
Compressor			
Heating	1/2	30%	10%
Cooling	1/3	35%	12%
Blower	1/2	57%	45%

- Of the maximum PM-ECM savings
  - Up to 2/3 of compressor savings may be obtainable with SOA IDIM
  - Less than 1/4 of blower savings are obtainable with SOA IDIM

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