LABORATORY EXAMINATION AND SEASONAL ANALYSES OF THE DYNAMIC LOSSES FOR A CONTINUOUSLY VARIABLE-SPEED HEAT PUMP

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ABSTRACT

An air-to-air split-system, continuously variable speed heat pump of nominal 2-1/2 ton (8.8 kW) cooling capacity was instrumented and tested in the laboratory. The system operating conditions, coefficient of performance, and system capacity were measured during frosting/defrosting tests and during both heating and cooling mode cycling tests. Frosting/defrosting tests were conducted for a range of compressor speeds, outdoor air temperatures, and relative humidities. Cycling tests were conducted for a range of compressor speeds and outdoor temperatures.

Dynamic loss algorithms were developed from the data and seasonal analysis simulations were made to estimate the potential energy savings due to compressor and indoor blower modulation. For Ft. Worth (Tex.), Knoxville (Tenn.), and Syracuse (N.Y.), analysis of energy cost savings due to modulation justify a cost premium of $400 for three-year payback.

INTRODUCTION

Continuously variable speed heat pumps (CVSHP) were first introduced in the American market in 1979 and in the Japanese market in 1983. The initial American marketing effort was not successful. However, in Japan, 1.1 million CVSHP units were sold during 1987. The increase in sales was due to the concern for energy efficiency rather than for cost effectiveness, and the concern emanated from the increased costs of imported fossil fuels. In the American market, the near doubling of the national average cost of electricity per kilowatt hour (Department of Energy 1987) for residential heating and cooling in the last five years has given justification for continued research of CVSHPs, which could be cost effective.

The CVSHP has many potential benefits. Compared with the single-speed heat pump (SSHP) the CVSHP offers improved comfort control, improved energy efficiency, and reduction in peak demand to the utility (Petersen 1987).

Previous work conducted in the laboratory demonstrated optimal control of refrigerant flow and airflow for a breadboard CVSHP (Miller 1987a). This previous work was continued in the present study by investigation of the frosting, defrosting, and cycling losses of the breadboard CVSHP while it operated under optimal refrigerant flow and airflow control settings. Previous studies by Tanaka and Yamanaka (1982) and by Miller (1985), dealing with cycling dynamics; by Kuwahara et al. (1986), dealing with defrosting efficiency; and by Glamm (1987), dealing with dynamic control, were gleaned for techniques to minimize the dynamic losses of the test CVSHP. A best dynamic control strategy for the test unit was developed in the laboratory, and dynamic loss algorithms were developed from the data. Seasonal analysis simulations were made to estimate the energy and cost savings directly attributable to the reduction of dynamic losses for SSHPs and the efficiency improvement due to CVSHP compressor and indoor blower modulation. The energy cost savings are estimated to determine the allowable cost premiums for three-year payback to the consumer due to the efficiency improvement.

LABORATORY FACILITY

The Breadboard Heat Pump

The test unit was a split-system residential air-to-air CVSHP that was commercially available in 1979. The original capillary tubes of the heat pump were replaced with fine metering hand valves having variable flow area for both heating and cooling mode throttling of refrigerant. Both indoor and outdoor heat exchangers were of tube-and-plate-fin construction. The indoor heat exchanger was an A-frame coil and had a distributor downstream of the variable area throttle to meter refrigerant between the two parallel heat exchanger circuits. The outdoor coil has 13 fins per inch; each fin was wavy or patterned with two complete waves per row. The coil also had three tube rows and was installed at roughly a 30° slant. The heat pump had a suction line accumulator.

The indoor blower motor and the compressor motor were three-phase, three-wire induction motors with Y-connected stator windings. The reciprocating compressor was driven by a high-efficiency pulse-width-modulated inverter, while the indoor blower was driven by a six-step inverter that was supplied by the manufacturer. For best COP, the outdoor fan was driven by a 220-V ac two-speed motor that was manually controlled.

THE TEST STAND

Testing was conducted in environmental chambers as previously described by Miller (1987b). The moisture content of conditioned air in the environmental chambers was measured using dew point hygrometers. A host computer and data acquisition system (DAS) was used to monitor all temperatures, pressures, powers, and flows.

EXPERIMENTAL PROCEDURE

Steady-State Testing

Results of steady-state testing (completed in previous work by Miller 1987a) were used as a base for settings of optimal refrigerant and airflow control. Steady-state coefficient of performance (COP) and capacity were used as a base for calculating dynamic losses for the breadboard CVSHP.

Frosting and Defrosting Tests

Tests were conducted to observe the effect of compressor speed, outdoor temperature, and relative humidity on the COP and capacity of the test unit. The refrigerant charge was held constant for all testing at 8.1 lb (3.7 kg). Testing was conducted at outdoor temperatures of 35, 25, and 17 F (1.7, -3.9, and -8.3°C). At 35 F (1.7°C) outdoor temperature, tests were conducted at outdoor relative humidities of 60, 70, 80, and 90%. For testing at 25 F and 17 F (-3.9°C and -8.3°C), the outdoor relative humidity was held constant at 80%. Control of both the CVSHP and data collection was performed automatically by the DAS. Prior to actual data collection, the heat pump was run through a preliminary frosting and defrosting cycle at each ambient air test condition. This was done to better simulate actual frosting and defrosting operation.

The defrost cycle was initiated automatically by a demand defrost control, which was based on an air-side total pressure drop of 0.51 in (0.13 kPa) of water through the outdoor coil. Signals from the manufacturer-supplied demand sensor and from a differential pressure cell, which measured air-pressure drop, were both used as input to the DAS for the start of defrosting. During defrosting, the effect of overspeeding the compressor and increasing the flow area through the active throttle was tested for improvements in defrost efficiency. Following the defrost cycle the heat pump was returned to heating-mode operation and the recovery time required to achieve 95% of steady-state capacity was recorded, as were the refrigerant temperatures and pressures.

Cycling Tests

Tests were conducted in heating and cooling modes at outdoor ambient temperatures of 40 F (4.4°C) and 82 F (27.8°C), respectively. At start up the compressor was energized at 60-, 30-, and 15-Hz drive frequencies to observe the effect of compressor speed on cycling efficiency. The cycling rate was set at 12-min-on and 48-min-off (i.e., 20% on-time for a thermostat with a maximum of 1.5 cycles per hour), a cycling rate suggested by the air-conditioning industry as more typical of actual cycling rate for variable speed residential units.

The cycling tests were run immediately following steady-state tests. At each compressor speed, the refrigerant charge was set to those amounts previously determined by Miller (1987a) that gave optimal refrigerant subcooling at the condenser exit and low superheat at the inlet to the compressor.

The refrigerant flow area of the active throttle in each cycling test was controlled in one of two different modes: the throttle was fixed to previously established steady-state optimal flow conditions and the throttle was fully open at the start of an on-cycle and reset to steady-state optimal flow conditions when a refrigerant level was observed developing in the accumulator. These two modes were used for normal mode cycling (i.e., compressor and fans cycle on and off simultaneously) and for tests in which off-cycle refrigerant migration was controlled and the indoor blower operation was extended into the off-cycle.

Control of the heat pump and data collection was performed automatically by the DAS and host computer system. Representative cycling operation of the heat pump was established by automatically cycling the heat pump through three on- and off-cycles prior to data collection. Data on three to four cycles of heat pump operation were then collected and filed on the host computer.

Data Collection Procedure

Dynamic loss data (i.e., frosting, defrosting, and cycling) were monitored at preset time intervals into the respective cycles. As time progressed into the respective cycle, the scan rate was automatically adjusted according to the severity of the transients. Upon completion of a test, data were recorded and stored by the host computer for further reduction and analysis.

A statistical error analysis was performed to determine the precision of individual data measurements within given time bins for frosting, defrosting, and cycling tests. The error analysis was based on a normal distribution and assumed a 95% probability that the mean value fell within two standard deviations of the true value.

FROSTING OF THE OUTDOOR COIL

The frosting efficiency of the breadboard test unit operating as an SSHP was previously reported by Miller (1987b). The previous frosting study of the tube-and-plate-fin outdoor coil, having 13 fins per inch, revealed only a slight effect on the COP and capacity of the test unit operating with capillary tube flow control.

CVSHP Frosting Traits at a Compressor Drive Frequency of 60 Hz

The COP and capacity of the CVSHP are displayed in Figures 1 and 2 for tests conducted at an outdoor ambient of 35 F (4.4°C) with relative humidities of 60, 70, 80, and 90%. Both the COP and capacity degraded roughly 4% over the respective time period of each test. Increasing the moisture content of the outdoor air caused an increased rate of COP and capacity degradation, as seen in Figures 1 and 2.

Defrost was initiated by a demand defrost control sensing the air-side total pressure drop through the outdoor coil. The plots of pressure drop as a function of time (Figure 3) give an indication of the rate of frost accumulation on the outdoor heat exchanger for the respective outdoor humidity conditions. Testing at 60-Hz drive frequency, 35 F (4.4°C) outdoor temperature, and 90% relative humidity required 79.8 min for defrost initiation. At 70% relative humidity the time to defrost increased to 185 min and at 60% relative humidity defrosting was initiated after 14.2 hours.

Frosting as Affected by Outdoor Temperature and Compressor Modulation

The COP and capacity plots in Figures 4 and 5, respectively, were for tests conducted at 35 F (1.7°C), 80% relative humidity, and at discrete compressor speed. The tests revealed only marginal drops during the test despite the accumulation of frost on the outdoor heat exchanger. The density of refrigerant entering the compressor remained fairly constant for all tests and, as a result the evaporator capacity remained approximately constant. The near constant density of refrigerant at compressor inlet caused the refrigerant mass flowrate and compressor power to be only slightly affected by frosting of the outdoor coil.

The rate of COP degradation due to frosting decreased as compressor speed decreased (Figure 4). The reduction in compressor speed unloaded the heat exchangers, which caused an increase in evaporator pressure and temperature, and therefore decreased the humidity gradient from ambient air to the tube wall of the outdoor coil. This in turn decreased the rate of frost accumulation and the rate of performance degradation.

Frosting tests conducted at 35, 25, and 17 F (1.7, -3.9, and -8.3°C) outdoor temperature and 80% relative humidity, were followed by an automatically initiated demand defrost based on outdoor coil air-side pressure drop. The outdoor coil was heavily frosted at the start of each defrost. Capacity plots in Figures 5, 6, and 7 indicate that reducing the outdoor temperature and/or compressor speed caused significant increase in the time to defrost initiation. The drop in temperature reduced the moisture content of the air, while reducing compressor speed increased outdoor coil temperature. Both trends decreased the frosting rate on the outdoor coil. No frosting was observed at the lower temperature of 17 F (-8.3°C) with the compressor driven at 30-Hz drive frequency.

CVSHP operation at 25 and 17 F (-3.9 and -8.3°C) outdoor temperature (capacity plotted in Figures 6 and 7) revealed no noticeably different frosting trends even with the compressor driven at 90-Hz speed. Reducing compressor speed caused near proportional reduction in capacity. Time-to-defrost initiation increased as compressor speed decreased from 90- to 45-Hz drive frequency.

DEFROST EFFICIENCY OF THE CVSHP

Defrosting data are listed in Table 1 for tests conducted as a function of compressor speed and of outdoor temperature with the relative humidity held constant at 80%. The compressor was operated at 60-Hz drive frequency during the frosting cycle, while during defrosting the compressor drive frequency was set at 60 Hz

or increased to 90 Hz. Defrosting data are listed in Table 2 for indication of the effect of compressor overspeeding and throttle control.

Overspeeding the Compressor

The defrost intervals (time between defrosts) for tests conducted at the given outdoor temperatures are roughly equal (Table 1). Overspeeding the compressor during defrost decreased the defrost time 8% at 35 F (1.7°C) and roughly 15% at 25 and 17 F (-3.9 and -8.3°C). However, defrost power (compressor and indoor blower power) was increased roughly 20% by increasing the compressor speed to 90 Hz. In turn, this increase in defrost power caused the COP, which was integrated over the frost and defrost cycles (including 5 kW of auxiliary heat), to decrease. The results in Table 1 indicate that overspeeding the compressor to 90 Hz during defrosting does not improve system efficiency.

Effect of Throttle Control

Additional defrost tests were conducted to observe the effect of increasing the flow area through the throttle, which was active during reverse cycle defrosting. Previous work by Miller (1987b) indicated that increasing the effective flow area through the throttle would improve defrost efficiency by increasing refrigerant flow during the first few minutes of defrosting.

Results of defrosting tests following frosting tests, both of which were conducted at 35 F (1.7°C) and 80% relative humidity, are listed in Table 2 as a function of the effective throttle control. The defrost intervals for these tests were similar, indicating similar frost loads. Roughly doubling the flow area of the throttle caused both the defrost interval to decrease 15% and the defrost power to decrease roughly 10% of measurements compared to tests with fixed throttle flow area (Table 2). This throttle control resulted in a 2% increase in COP,[a] integrated over the frost and defrost cycles.

The combined effect of increased throttle opening and compressor overspeeding during defrosting was also tested (Table 2). Doubling the throttle opening and overspeeding the compressor caused a 1% drop in integrated average frost/defrost COP as compared with defrosting at 60 Hz (Table 2).

The works of Kuwahara et al. (1986) and Itoh (1986) indicated improvement in defrost and system efficiency through compressor overspeeding and control of the throttle flow area during defrosting. Glamm (1987), however, indicated only slight improvement in average frost/defrost COP resulting from control of the throttle flow area. The results in Tables 1 and 2 indicate, for the given system, that the best average frost/defrost COP was achieved by increasing the flow area of the throttle active in defrost and by operating the compressor at 60 Hz drive frequency during defrosting. Further testing also revealed that regardless of the drive frequency of the compressor prior to the defrost cycle, the best average frost/defrost COP was observed with the compressor driven at 60 Hz during defrosting.

CYCLING PERFORMANCE OF THE BREADBOARD CVSHP

One of the major advantages of a CVSHP is its ability to follow the house heating and cooling load. The load matching characteristic of a CVSHP reduces the number of on/off cycles and improves seasonal cycling efficiency as compared to that of single-speed equipment. However, at compressor speeds less than 60 Hz, the time response of the heat pump is slower and, therefore, the cycling efficiency per cycle for a CVSHP can be less than that of an SSHP.

Effect of Compressor Speed on Part-Load COP

Heating Mode. Tests were conducted at an outdoor ambient temperature of 40 F (4.4°C). The cycling COP, normalized to steady-state COP value per respective compressor speed, is displayed in Figure 8 as a function of time into the on-cycle. At 60- and 30-Hz drive frequency, the system refrigerant charge was set at

[a] COP includes 5 kW of auxiliary heat during defrost period.

8 lb (3.6 kg), while at 15 Hz the charge was set at 12 lb (5.5 kg) for duplication of previous optimal steady-state refrigerant flow settings (Miller 1987a).

At 60-Hz drive frequency, the test unit reached steady-state COP in roughly 8 min. However, when the compressor was started at 15- or 30-Hz drive frequency, steady-state COP was not achieved during the 12-min on-cycle. Mulroy and Didion (1985) stated that at a point 15 s after start up, 38% of the heat pump charge would be in the accumulator. Mulroy and Didion concluded that the gradual return of liquid refrigerant to heat pump circulation caused the majority of the cycling loss. Therefore, at a compressor speed less than 60 Hz, the time response of the system would be more sluggish and lead to larger losses.

The slow response of the test unit at speeds less than 60 Hz caused an increase in the degradation coefficient (C_D) for normal mode cycling tests (i.e., compressor and fans cycle on and off simultaneously) (Table 3). Attempts were made to improve cycling efficiency by increasing the flow area of the throttle during start up. Results indicated that the larger flow area only increased the level of refrigerant in the accumulator after start up (visually observed through sight glasses on the accumulator) and caused a decrease in cycling efficiency.

Cooling mode. Tests were conducted at an outdoor ambient temperature of 82 F (27.8°C). At 60-Hz drive frequency, the system refrigerant charge was set at 9.5 lb (4.3 kg), at 30 Hz the charge was set at 8.6 lb (3.9 kg), and at 15 Hz the charge was set at 9.75 lb (4.4 kg) for duplication of previous optimal steady-state refrigerant flow settings (Miller 1987a).

Testing with the throttle flow area fixed yielded similar results as displayed for heating mode. However, by controlling the throttle during start up, the compressor was able to establish charge distribution more rapidly and to improve low compressor speed cycling efficiency as reflected in improvement in degradation coefficients (C_D), which are listed in Table 3 for normal mode cycling. In cooling mode the refrigerant charge was more evenly distributed between the heat exchangers prior to start up. The larger flow area through the throttle caused less restriction to flow and allowed the compressor to rapidly establish charge distribution. Cycling COP (normalized to steady-state COP) per compressor speed tested is displayed in Figure 9. This figure illustrates the improvement in low compressor speed cycling efficiency in cooling mode as compared to heating mode results (Figure 8). The CVSHP achieved steady-state cooling performance in roughly 12 min for testing at 30- and 15-Hz compressor drive frequency. At 60-Hz drive frequency, steady-state COP was reached in roughly 7 min (Figure 9).

CYCLING CONTROL STRATEGIES

Both heating and cooling mode test results indicated a decrease in cycling efficiency as compressor speed was reduced from 60- to 15-Hz drive frequency. For normal mode cycling the increase in C_D (Table 3) as compressor speed decreased revealed the loss in cycling efficiency. Cycling control strategies developed from previous work by Miller (1985) were tested on the breadboard CVSHP to observe any improvement in cycling efficiency at compressor speeds less than 60 Hz.

Heating and cooling mode tests were also conducted with the refrigerant isolated in the system high-side during the off-cycle (refrigerant migration control) and with the indoor blower continuing to operate for 2 min into the off-cycle (indoor blower delay). At start up the throttle was controlled for best cycling efficiency. The results of both heating and cooling mode tests are displayed in Figure 10 and are compared to normal mode[b] cycling test results.

Heating mode cycling test results (Figure 10) indicate a significant improvement in efficiency. The tests conducted at 30-Hz compressor drive frequency show the CVSHP to achieve near steady-state performance in roughly 10 min through the control off-cycle refrigerant migration. As seen in Table 3 the C_D decreased from 0.329 (for normal mode cycling) to 0.133, with combined control of off-cycle migration, 2 min of indoor blower delay, and throttle control.

Cooling mode cycling test results observed at 15-Hz compressor drive frequency were not as dramatic as those observed in heating mode. Off-cycle control of refrigerant migration did improve the cycling COP of the

[b] Compressor and fans cycle on and off simultaneously and throttle controlled at start up.

CVSHP as seen in Figure 10. The C_D (Table 3) decreased from 0.362 (for normal mode cycling at 15-Hz speed) to 0.102 (for cycling with off-cycle migration control, 2-min delay in indoor blower operation, and throttle control).

The reduction of C_D due to extending indoor blower operation into the off-cycle is illustrated in Figure 11. In heating mode with the compressor starting at 30-Hz drive frequency, the C_D was reduced 31.6% of normal mode C_D due to the combined effect of throttle control at start up and control of refrigerant migration during the off-cycle. Delaying the shutdown of the blower further reduced the C_D by an additional 27.9%. In cooling mode, the combined effect of throttle control at start up and off-cycle migration control reduced by 48.2% the C_D for normal mode cycling, and the delay in blower shutdown reduced it further by 23.6%.

SEASONAL PERFORMANCE ANALYSES

Seasonal analyses were conducted using a computer code developed by Rice et al. (1988). Frosting and defrosting algorithms were developed from the CVSHP test data as functions of outdoor ambient temperature, relative humidity, and drive frequency of the compressor. The algorithms were coupled to a quadratic interpolation routine for calculating frosting/defrosting energy consumption. Cycling, including off-cycle parasitics, was calculated based on the Department of Energy test procedure (1979), with the exception of cycling rate.

Simulated operation of the breadboard CVSHP was applied to a "standard house" for three cities in the United States. The standard house is a single-family detached residence, 1800 ft^2 (167 m^2) ranch style home that has a crawl space. The ceilings are insulated to R-19, the walls to R-11, and the floors to R-9. Further detail on the house can be found in the work of Rice et al. (1988). Heating and cooling loads were calculated for the cities investigated using weather data base obtained from the U.S. Air Force, Army, and Navy engineering weather data manual (1978).

Comparisons of seasonal performance simulations were made for the different heat pump configurations listed in Table 4. Steady-state COPs of each system were based on the compressor and indoor blower, both of which were powered by ideal induction motor drives (i.e., sine-wave driven). A conventional SSHP, having capillary tube flow control and a 90-min time/temperature defrost control (Miller 1987b), was compared with the optimally controlled CVSHP operating only at 60-Hz drive frequency (i.e., single-speed unit with optimal flow control and reduced dynamic losses). The optimally controlled breadboard CVSHP, having reduced dynamic losses, was then compared with the CVSHP operating only at 60-Hz drive frequency (i.e., optimal control single-speed) to quantify the benefit of modulation on a seasonal basis. The CVSHP modulation ratios used in the seasonal simulations were 3:1 (90- to 30-Hz compressor speed) in the heating mode and 4:1 (60- to 15-Hz compressor speed) in the cooling mode.

For purposes of this study the computer code was used to evaluate the following:

1. A breakdown of heat pump energy consumption on an annual basis,
2. The effect of climate on the magnitude of dynamic losses,
3. Energy cost and potential payback for improving SSHP and CVSHP seasonal efficiency, and
4. The effect of oversizing the heat pump.

Heat Pump Comparisons and Energy Consumption

Total annual energy use due to frosting/defrosting, cycling, and supplemental resistance heat was calculated for the heat pump systems (Table 4) operating in Fort Worth, Knoxville, and Syracuse. Energy use for each of the loss categories, along with steady-state and total annual operating energy use, are listed in Table 5.

SSHP reduced dynamic losses. Only 1% of the total annual energy consumption was attributable to frosting. Demand defrost control as compared to a 90-min time-temperature defrost control reduced the number of seasonal defrosts by roughly a factor of 3 in all three locations. Both demand defrost control and improvement in defrost efficiency[c] caused the frosting/defrosting energy consumptions for an optimally

[c] Throttle fully open during defrost to increase refrigerant circulation for improved defrost efficiency.

controlled SSHP to be roughly half that observed for an SSHP with fixed throttle and time-temperature defrost control.

The combined control of off-cycle refrigerant migration and delay in shutdown of the indoor blower substantially reduced cycling losses for the optimally controlled SSHP by roughly 50% for simulated operation in Fort Worth, Knoxville, and Syracuse (Table 5).

CVSHP modulation effect. The added feature of continuous modulation of the compressor and indoor blower motors further improved system efficiency significantly as seen by the reduction of energy consumption (Table 5). This improvement in system efficiency can be attributed to (1) reduced supplemental heat, (2) reduced cycling losses, (3) heat exchanger unloading, and (4) reduced frosting/defrosting losses as explained in the following paragraphs.

Reduced supplemental heat. Overspeeding the compressor reduced the balance point of the CVSHP as compared with the optimally controlled SSHP and reduced the amount of supplemental heat required to match load (i.e., balance points listed in Table 5). In the severe winter climate of Syracuse the CVSHP reduced supplemental heat usage by 43%. The compressor overspeeding capability would also reduce the peak power demand for the electric utility company.

Reduced cycling losses. For operation at ambient temperatures greater than the balance point, the load following capability of the CVSHP reduced cycling losses simply by reducing the number of cycles. Assuming that both configured heat pumps have reduced dynamic losses, the compressor and indoor blower speed modulation decreased cycling energy consumption roughly by 40% of that for the optimally controlled SSHP.

Heat exchanger unloading. Modulating the CVSHP at low loads reduced the condensing and raised the evaporating pressure (heat exchanger unloading) and thereby improved steady-state COP as previously observed by Miller (1987a). This heat exchanger unloading yielded the largest reduction in heat pump energy usage for Fort Worth and Knoxville. The steady-state energy consumption of the CVSHP was reduced by 19% compared with the optimally controlled SSHP in Fort Worth. In Knoxville, the steady-state usage was reduced by 17.5% but by only 2% in Syracuse. Probably the reduction in Syracuse was small because the majority of load hours occurred at nominal compressor speed (i.e., 60-Hz drive frequency) in this predominantly heating load climate.

Reduced frosting/demand defrosting losses. The frosting/demand defrosting efficiency of the CVSHP improved slightly over that of the optimally controlled SSHP. A maximum decrease in frost/demand defrost energy consumption of 30% was observed for the CVSHP in Knoxville as compared with the optimally controlled SSHP. The number of simulated defrosts in Knoxville for the CVSHP was 236 as compared with 304 defrosts for the SSHP. During low-speed modulation the evaporator temperature increased and, therefore, the defrost interval increased, resulting in the reduced number of seasonal defrosts. In Syracuse the number of defrosts were nearly equal, again probably because the majority of heating load hours occurred at nominal compressor speed (i.e., 60-Hz drive frequency).

Allowable Cost Premiums for Efficiency Improvements

Energy use due to frosting and defrosting, cycling, and supplemental heat was translated into direct costs to determine incremental paybacks for improving heat pump efficiency. Energy costs, based on electrical rates cited in the DOE publication *Typical Electric Bills* (1987), were used to calculate annual operating costs. The electric rates, which were based on winter and summer heating and cooling usage per 2500 kWh, were used to calculate cost data (Tables 6 and 7) from the energy consumption data (Table 5).

For Fort Worth and Knoxville, which have a heating-load-to-total-load ratio less than 0.6, the cost of consumed energy for supplemental heat was less than $15 for the SSHP and less than $6 for the CVSHP. In contrast, the CVSHP reduced supplemental heat cost from $200 to $113 per year for Syracuse, which has a heating-load-to-total-load ratio of 0.84.

SSHP Reduced Dynamic Losses. The work of Glamm (1987) supports the results of this study, which indicate defrost and cycling efficiency improvements through control of the throttle valve (i.e., an electronic expansion valve). The control of off-cycle refrigerant migration via control of the throttle valve and the delay in shut down of the indoor blower resulted in roughly a $40 per year energy cost savings for the SSHP comparisons in Table 6. For simulated SSHP comparisons, a demand defrost initiator vs a time-temperature initiator resulted in a per year savings of $14 in Fort Worth, of $25 in Knoxville, and of $50 in the severe climate of Syracuse.

The consumer could afford the three-year premiums listed in Table 7 for the improvement in SSHP. A demand defrost control and a controllable expansion device (i.e., electronic valve) would have to cost no more than about $300 to $400 for a three-year payback to the consumer.

CVSHP Modulation Effect. The continuous modulation capability of the CVSHP vs the optimally controlled SSHP reduced frosting/defrosting, cycling, and steady-state (i.e., heat exchanger unloading) energy consumptions. Cost of energy savings due to reduced supplemental heat usage was largest for Syracuse because of its predominant heating load and high utility rates. The improvement in cycling performance resulted in roughly a $15 annual savings for all cities investigated. Frosting/defrosting annual cost savings were $14 for Syracuse and only $6 and $2 for Knoxville and Fort Worth, respectively. However, for Fort Worth and Knoxville the major cost savings resulted from heat exchanger unloading. In Fort Worth $105 annually were saved and in Knoxville the savings were $67 because the CVSHP could operate at low compressor speeds with improved efficiency as compared to performance at 60-Hz speed. In Syracuse, the cost savings were only $13 per year due to heat exchanger unloading; however, as previously discussed the reduction of supplemental heat usage saved $87 per year as compared to supplemental heat usage in an SSHP.

Again the allowable premium for modulation capability for three-year paybacks are listed in Table 7. The consumer could afford roughly $300 to $400 for the added feature of continuous modulation of the compressor and indoor blower motors. These results indicate that for the test system the cost of the inverter drive or drives and smart controls must be no more than $400 for a three-year or less payback period.

The Effect of Sizing on Annual Performance for the CVSHP and the SSHP

All seasonal analyses discussed thus far were based on the 2.5-ton (8.8-kW) test heat pump. Previous work by Miller (1987b) reviewed the effect of heat pump sizing on the annual performance factor (APF). Similar work is again presented for sizing of the optimally controlled SSHP and the CVSHP, both having reduced dynamic losses.

To observe the effect of oversizing on APF, steady-state capacities and power consumptions (both heating and cooling) and frosting, defrosting, and cycling capacities and power consumptions measured for the test heat pump (operating with demand defrost) were scaled to yield various degrees of oversizing.[d] It was assumed that the frosting and defrosting time intervals would be independent of size (i.e., coil loading constant with scaling). Plots of APF as a function of degree of heat pump oversizing (termed CDF on abscissa of Figure 12) are illustrated for the optimally controlled SSHP and the CVSHP.

Hypothetical SSHP. The plots of APF in Figure 12 for the SSHP indicate that the 2.5-ton (8.8-kW) test unit yielded the best APF for both Fort Worth and Knoxville. In Syracuse the SSHP could be sized to 2 1/4 times the design cooling load. This adjustment would have improved the APF for the SSHP from 2.0 for the 2.5-ton (8.8-kW) test unit to 2.05, a 2.5% improvement due principally to the reduction of supplemental heat usage. Such oversizing probably would not be cost justifiable and would cause poor indoor humidity control during the summer due to frequent cycling of the hypothetically oversize SSHP.

Hypothetical CVSHP. Oversizing the CVSHP in Syracuse would improve the APF (Figure 12). An oversizing of 2 1/4 times the design cooling load would improve the APF from 2.3 for the test unit to 2.4, a 4% improvement. As compared to the SSHP, the CVSHP would better control temperature and humidity even with the system oversized because of the more continuous operation of the compressor and indoor blower.

For Fort Worth and Knoxville, the CVSHP could be oversized to 1.5 times the design cooling load; the test unit was roughly this size for Knoxville. For Fort Worth, the APF improvement was only 1.5% compared with the breadboard CVSHP. Oversizing in these two cities would help supplemental heat usage, which is most beneficial to the utility at peak power demand, while still minimizing the cycling losses through the near continuous operation of the compressor.

[d] Ratio of the unit cooling capacity to the design cooling load calculated at a design temperature that is 97.5% of the time greater than observed daily temperatures.

DISCUSSION

The seasonal analysis results depicted in Figure 12 for simulated operation in Syracuse showed that a CVSHP improved the APF by 15% compared with the optimally controlled SSHP. The test breadboard CVSHP operating in Knoxville and Fort Worth would improve the APF roughly 23% and 36%, respectively, when compared with the SSHP. The major improvement in seasonal efficiency in the moderate climates resulted from the ability to unload the heat exchangers during low speed operation. In the predominantly heating load climate of Syracuse the major improvement resulted from the reduction of supplemental heat usage. The CVSHP reduced both cycling losses and frosting/defrosting losses roughly 50% of those observed for the optimally controlled SSHP.

CVSHPs offer other benefits that are not easily quantifiable. In the heating season the CVSHP can reduce utility peak demand by overspeeding the compressor. The overspeed feature would also improve comfort by increasing the supply air temperature as compared to an SSHP (Miller 1987a). In the cooling mode the CVSHP's near constant operation would help improve humidity control as compared to an SSHP. Also the fewer number of on/off and defrost cycles would improve compressor reliability. Finally, the CVSHP could be used for zone control by adjusting capacity appropriately for the zoned load. Rice (1984) predicted that a 25% reduction in load through zoning would increase the allowable first cost by 69% for the three-year payback of a CVSHP.

CONCLUSIONS

- Modulation of compressor and indoor blower speed resulted in reduced annual energy use of about 15-20% compared with optimally controlled SSHPs.
- For the configured breadboard CVSHP, the cost to the consumer for inverters and smart control can be roughly $400 for a three-year payback.
- In the moderate climates of Fort Worth and Knoxville, the heat exchanger unloading of the CVSHP caused the largest reduction in energy usage when compared with an optimally controlled SSHP.
- In the predominantly heating load climate of Syracuse, the major improvement in CVSHP vs SSHP performance resulted from reduction of supplemental heat usage.
- Both frosting/defrosting and cycling losses were reduced by roughly 50% of those for an optimally controlled SSHP.
- The increased cost of demand defrost hardware and a controllable expansion device (i.e., electronic valve) is cost justifiable for an SSHP.
- For the locations studied, oversizing a CVSHP above the design cooling load was beneficial. In the northernmost climates, oversizing to 2.25 times the design cooling load would improve APF 4% of SSHP APF and would still provide adequate cooling mode comfort.

ACKNOWLEDGEMENT

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TABLE 1
Effect of Overspeeding the Compressor above 60-Hz Drive Frequency on Reverse Cycle Defrosting Efficiency

Outdoor Ambient Temperature	35 F (1.7°C)		25 F (-3.9°C)		17 F (-8.3°C)	
Compressor Speed During Frost/Defrost (Hz)	60/60	60/90	60/60	60/90	90/60	90/90
Defrost Interval (min)	105.5	99.1	146.1	152.6	128.0	106.7
Defrost Time (min)	5.7	5.3	6.0	4.8	5.7	4.8
Defrost Power[a] (W-H)	210.3	280.6	231.6	263.1	223.9	279.2
Frost/Defrost COP[b]	2.22	2.18	2.03	2.06	1.76	1.73

[a] Measured power consumption of compressor and indoor blower.

[b] Integrated average COP includes 5 kW of auxiliary heat during defrost period.

TABLE 2
Defrost Efficiency as Affected by Flow Area of the Indoor Throttle and Compressor Overspeeding

Laboratory Data	Compressor Frost/Defrost Speed (Hz)			
	60/60		60/90	
Effective Throttle Area (in^2)	0.06	0.10	0.06	0.10
Defrost Interval (min)	105.50	104.80	99.10	102.40
Defrost Time (min)	5.70	4.80	5.30	4.50
Defrost Power[a] (W-H)	210.30	190.00	280.60	244.20
Frost/Defrost COP[b]	2.22	2.27	2.18	2.20

[a] Measured power consumption of compressor and indoor blower.

[b] Integrated average COP inculdes 5 kW of auxiliary heat during defrost period.

TABLE 3
Degradation Coefficients (C_D) Observed for Various Compressor Speeds and Cycling Strategies

Mode of Operation	Compressor Speed (Hz)	C_D for Various Cycling Control Strategies				
		Normal		Overspeed 20 s[a]	Refrigerant Control, Blower Delay (off-cycle)	
Expansion Valve Setting		Fixed	Variable	Fixed	Fixed	Variable
Heating	60	0.225			0.113	
	30	0.329	0.391	0.295	0.215	0.133
	15	0.405				
Cooling	60	0.169			0.060	
	30	0.365	0.237			
	15	0.362	0.217	0.209	0.162	0.102

[a]At start up the compressor is driven at 60 Hz for 20 s.

TABLE 4
Heat Pump Systems used in APF Comparisons, Compresor and Fans Sine-Wave Driven

Heat Pump System	Refrigerant Flow Control	Indoor Airflow	Cycling C_D		Defrost Control	COP[a] Heat	EER[b] Cool
			Heat	Cool			
Conventional Single-Speed	Capillary	Constant	0.25	0.18	90-min Time & Temperature	3.0	8.70
Optimal Control Single-Speed	Variable Area	Constant	0.11	0.06	Demand	3.1	9.66
CVSHP	Variable Area	Variable	0.13	0.10	Demand	3.1	9.66

[a]Heating COP measured at 47 F (8.3°C) outdoor air temperature (60-Hz compressor drive frequency).

[b]Cooling EER measured at ARI rating point of 80 F (27°C) DB/67 F (19°C) WB indoor and 82 F (28°C) outdoor temperature (60-Hz compressor drive frequency).

TABLE 5
Simulated Annual Energy Use Breakdown for Single-Speed Heat Pump and Continuously Variable Speed Heat Pumps

City and State	Heat Pump Configuration	Heating Balance Point [F (°C)]	Annual Back-Up Heat[a] (kWh)	Dynamic Loss Energy		Steady-State Energy (kWh)	Total Annual Energy[d] (kWh)	Yearly Defrosts
				Fr/Def[b] (kWh)	Cycling[c] (kWh)			
Ft. Worth, TX	Conventional Single-Speed	21.0 (-6.1)	94.5	437.3	1257.7	9032.4	10822.0	519
	Optimal Control Single-Speed	21.0 (-6.1)	91.6	140.7	549.0	8332.6	9114.0	139
	CVSHP	17.5 (-8.1)	30.3	97.2	293.0	6742.3	7162.8	105
Knoxville, TN	Conventional Single-Speed	22.0 (-5.6)	234.7	784.1	1278.5	7502.2	9799.4	911
	Optimal Control Single-Speed	22.0 (-5.6)	244.9	338.0	594.7	6815.0	7992.6	304
	CVSHP	18.0 (-7.8)	94.1	225.7	345.0	5618.3	6283.2	236
Syracuse, NY	Conventional Single-Speed	27.0 (-2.8)	3073.2	1994.7	1156.3	10232.3	16456.5	2008
	Optimal Control Single-Speed	27.0 (-2.8)	3178.1	1192.8	545.4	9555.2	14471.6	855
	CVSHP	21.0 (-6.1)	1806.3	950.8	321.0	9344.0	12422.2	847

[a] Auxiliary heat required to satisfy house heating load when heat pump operates below balance point. Does not include auxiliary heat (5 kW) attributed to defrost.

[b] Frosting/defrosting includes recovery energy following defrost and 5 kW of auxiliary heat during reverse-cycle defrost.

[c] Cycling includes off-cycle parasitics.

[d] Heat pump yearly energy consumption including back-up heat.

TABLE 6
Cost Breakdowns of Simulated Annual Energy Consumption for Single-Speed and Continuously Variable Speed Heat Pumps

City & State	Heat Pump Configuration	Ratio of Heating Load to Cooling Load	Annual Backup Heat[a] Cost ($)	Annual Cost of Dynamic Energy Losses ($)		Steady-State Energy Cost ($)	Total Operating Energy Cost[b] ($)
				Frosting/Defrosting	Cycling		
Ft. Worth, TX	Existing Single-Speed	0.32	4.62	21.39	76.98	573.30	676.29
	Optimally Controlled Single-Speed		4.46	6.69	33.05	529.15	573.35
	CVSHP		1.40	4.90	17.14	423.96	447.40
Knoxville, TN	Existing Single-Speed	0.58	12.97	43.24	70.79	413.42	540.42
	Optimally Controlled Single-Speed		13.67	18.51	32.62	375.98	440.78
	CVSHP		5.20	12.47	19.06	309.77	346.50
Syracuse, NY	Existing Single-Speed	0.84	193.89	125.46	72.58	644.90	1036.83
	Optimally Controlled Single-Speed		200.59	74.77	34.65	601.76	911.77
	CVSHP		113.48	60.26	20.35	588.56	782.65

[a] Auxiliary heat required to satisfy house heating load when heat pump operates below balance point. Does not include auxiliary heat (5 kW) attributed to defrost.

[b] Costs based on winter and summer electric rates for space heating taken from DOE publication DOE/EIA-0040(87).

TABLE 7
Allowable Cost Premium for Three-Year Payback in $ for Improvement in Heat Pump Efficiency

	Single-Speed[a]	Continuous Modulation[b]
City & State	Optimally Controlled vs. Existing	CVSHP vs. Single-Speed Optimally Controlled
Ft. Worth, TX	308.82	377.85
Knoxville, TN	298.92	282.84
Syracuse, NY	375.18	387.36

[a] Cost premium for single-speed improvement based on reduction of dynamic losses and reduction of steady-state energy consumption.

[b] Cost premium based on the effect of continuous modulation for the identical system with optimal flow control and optimal dynamic control.

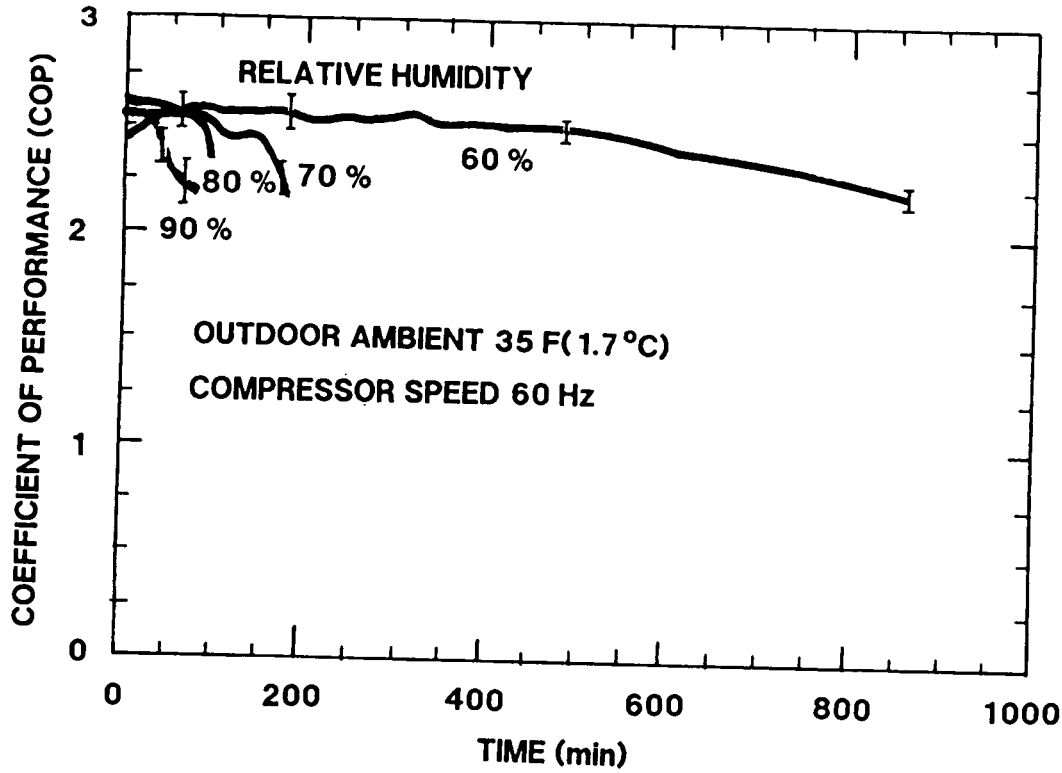

Figure 1 The effect of outdoor relative humidity on average COP observed during frosting of the CVSHP's outdoor coil

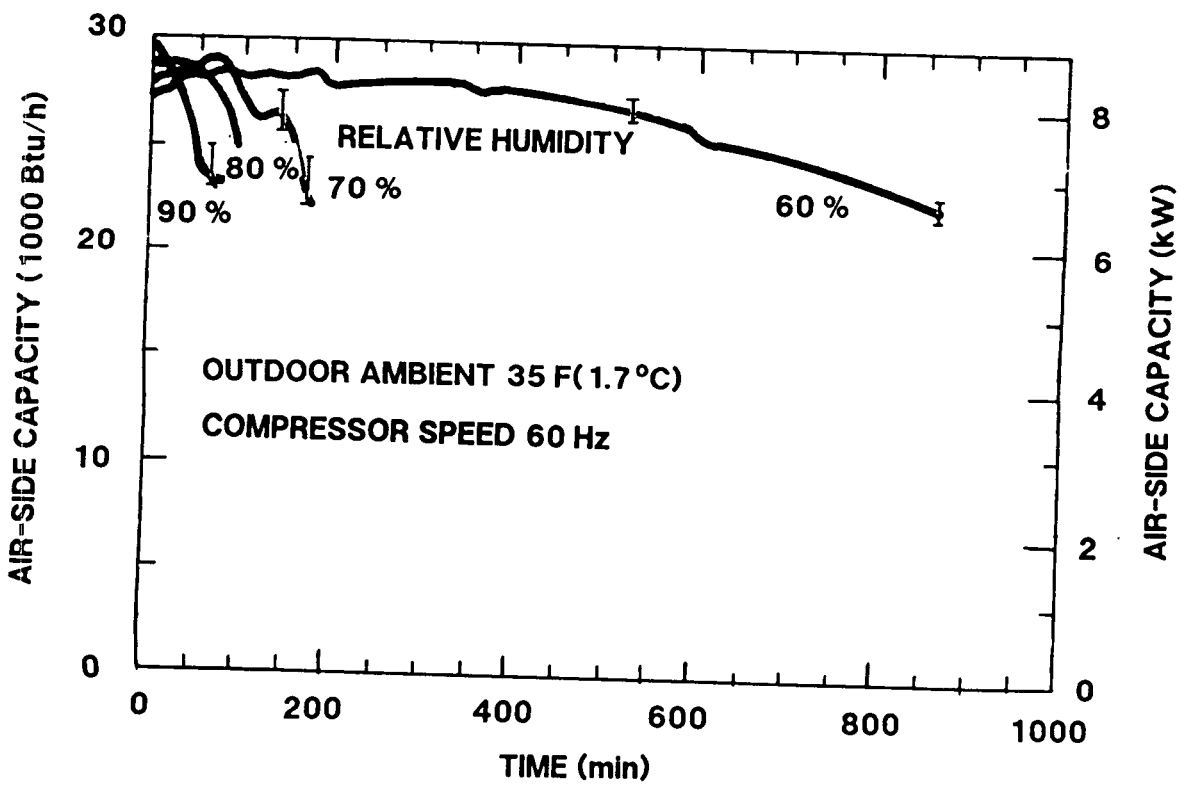

Figure 2 Air-side capacity of the CVSHP measured during frosting of the outdoor coil

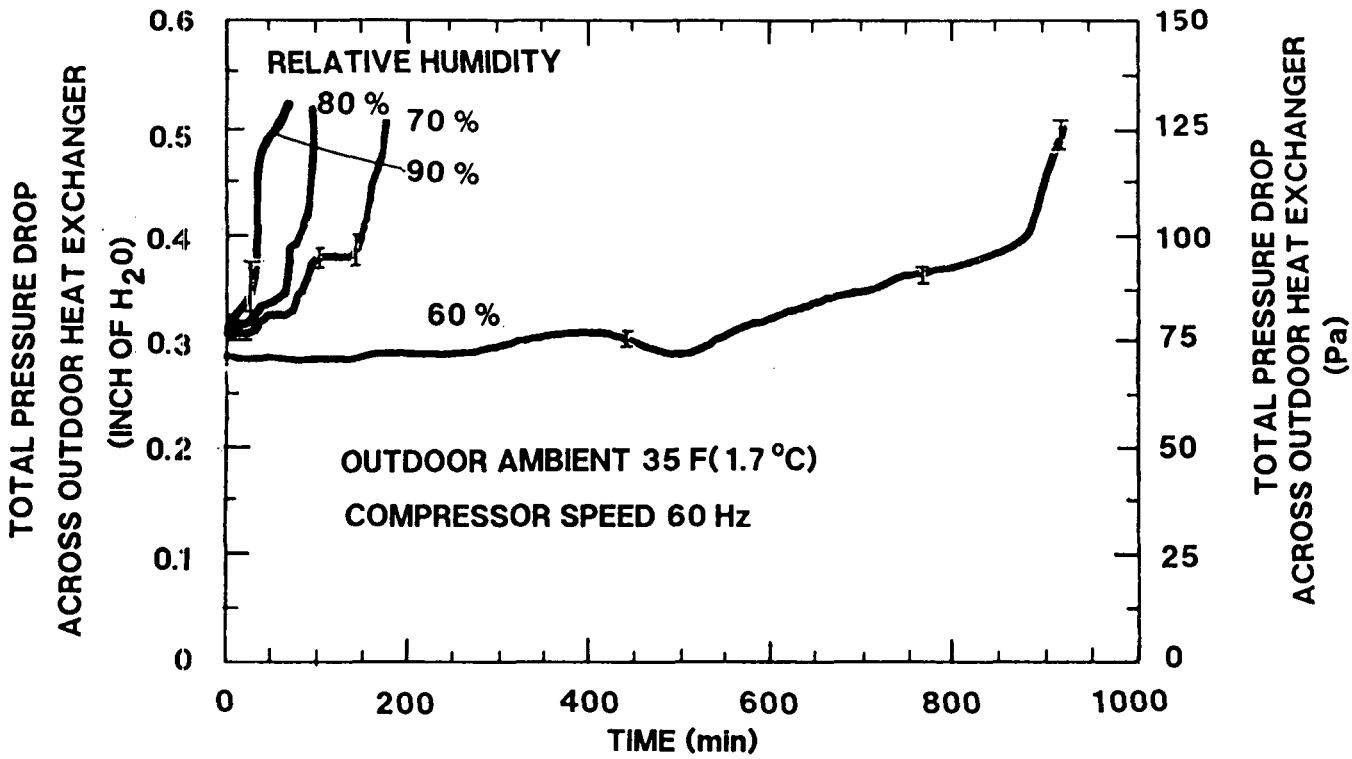

Figure 3 Total air pressure drop across the outdoor coil measured while frost accumulated on the coil

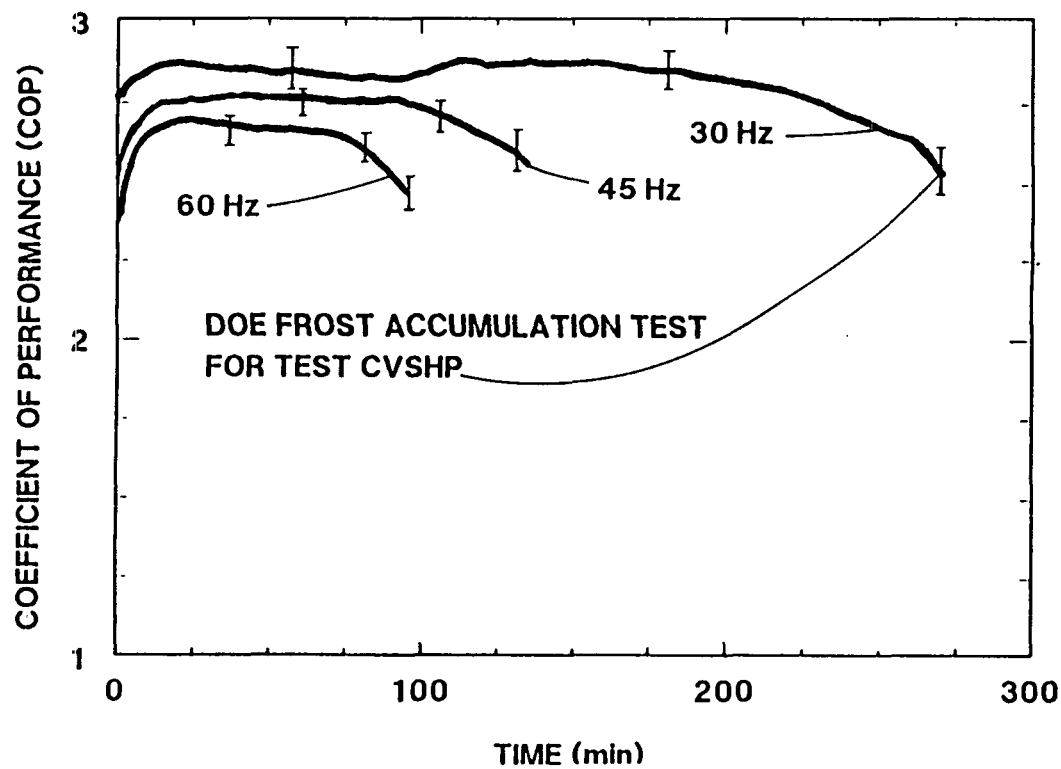

Figure 4 COP measured at 35 F (1.7°C) outdoor temperature, 80% outdoor relative humidity and at discrete compressor speeds

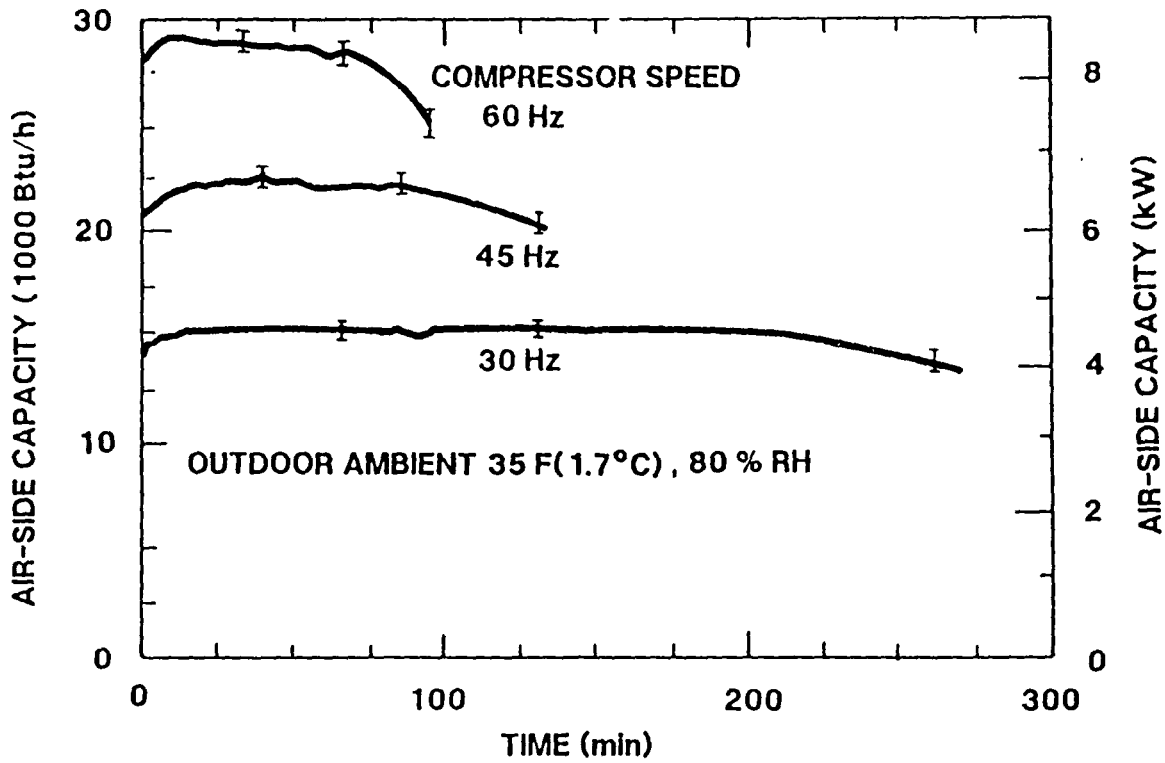

Figure 5 Air-side capacity measured at 35 F (1.7°C) outdoor temperature and at discrete compressor speeds

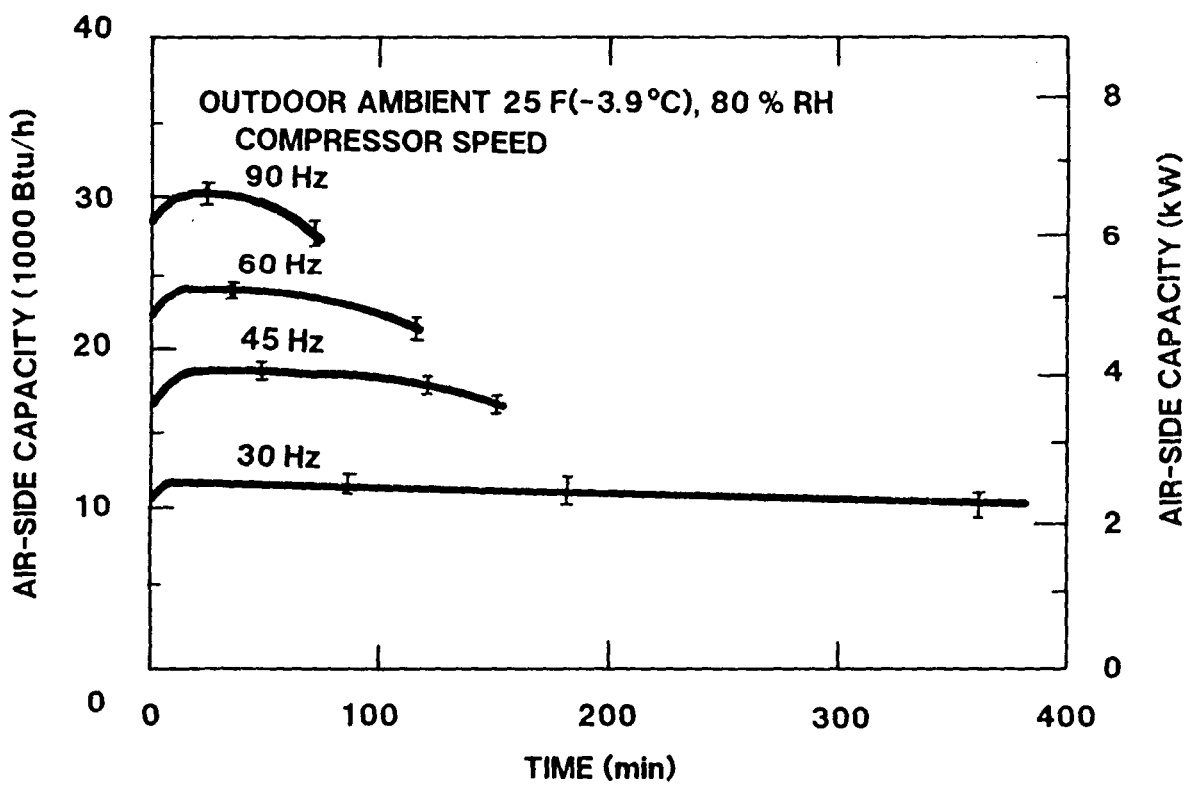

Figure 6 Air-side capacity measured at 25 F (−3.9°C) outdoor temperature and at discrete compressor speeds

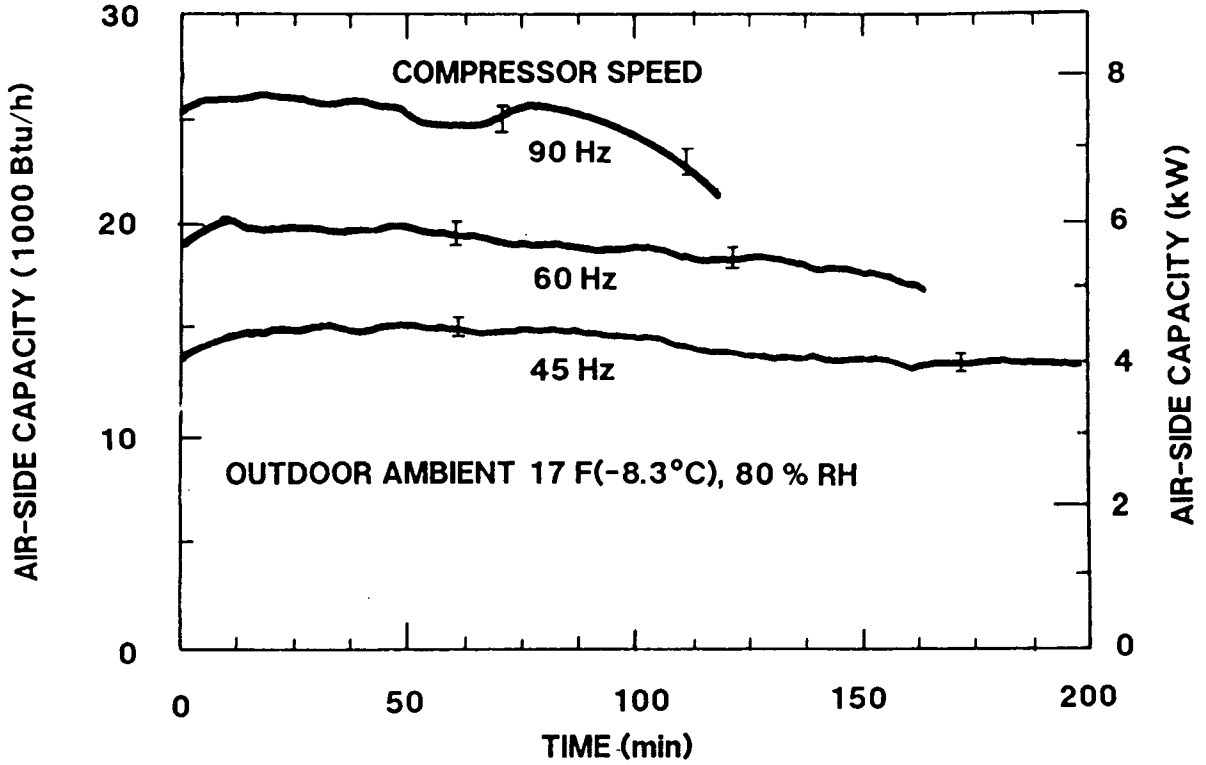

Figure 7 Air-side capacity measured at 17 F (−8.3°C) outdoor temperature and at discrete compressor speeds

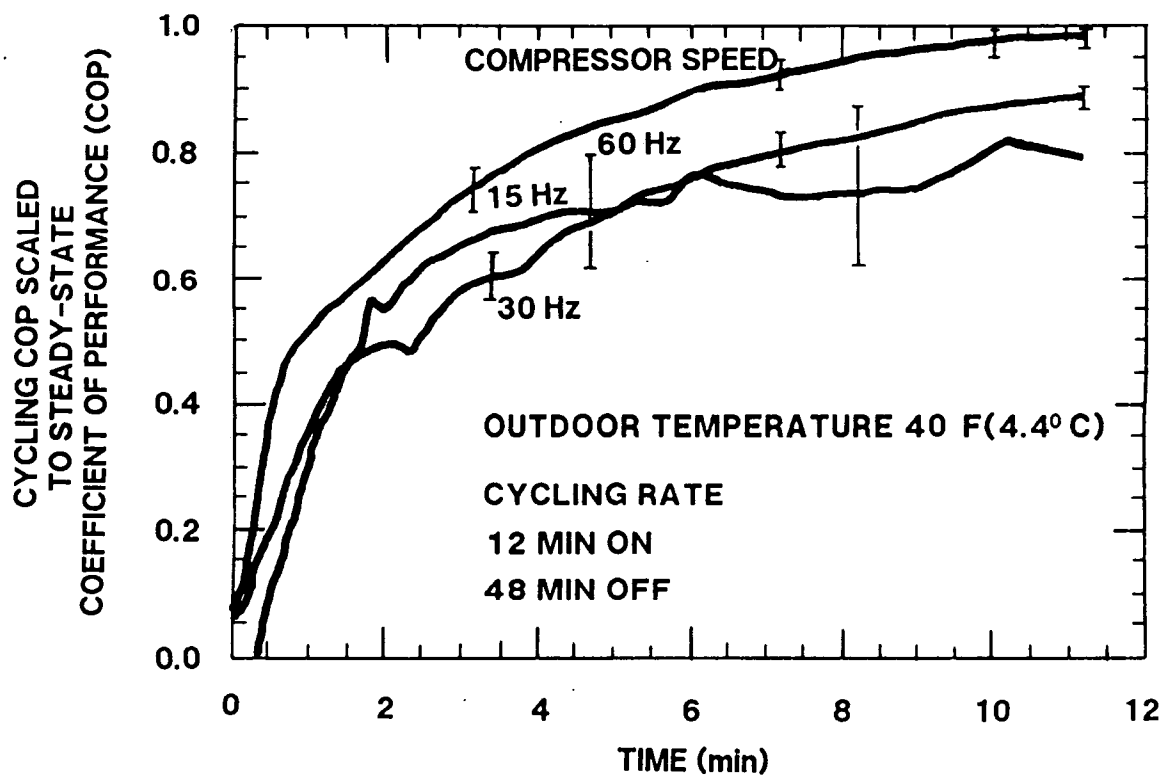

Figure 8 Heating-mode part-load COP scaled to steady-state COP at respective compressor speed

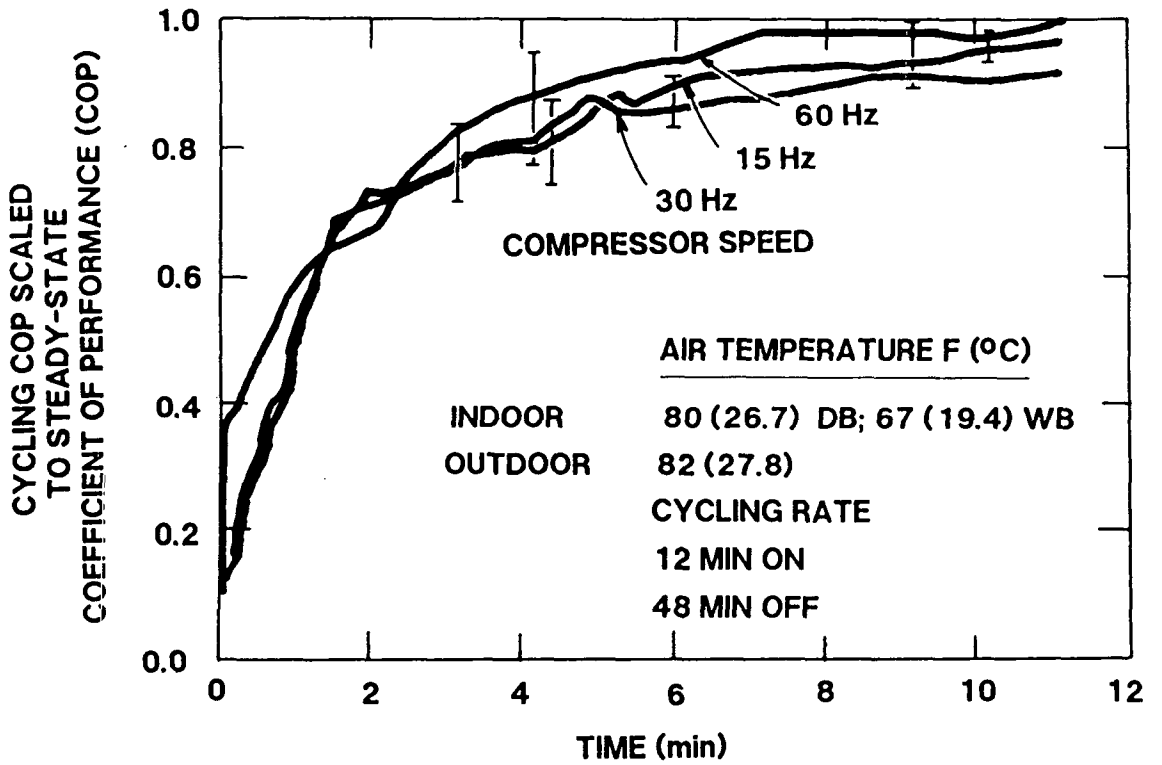

Figure 9 Cooling-mode part-load COP scaled to steady-state COP at respective compressor speed

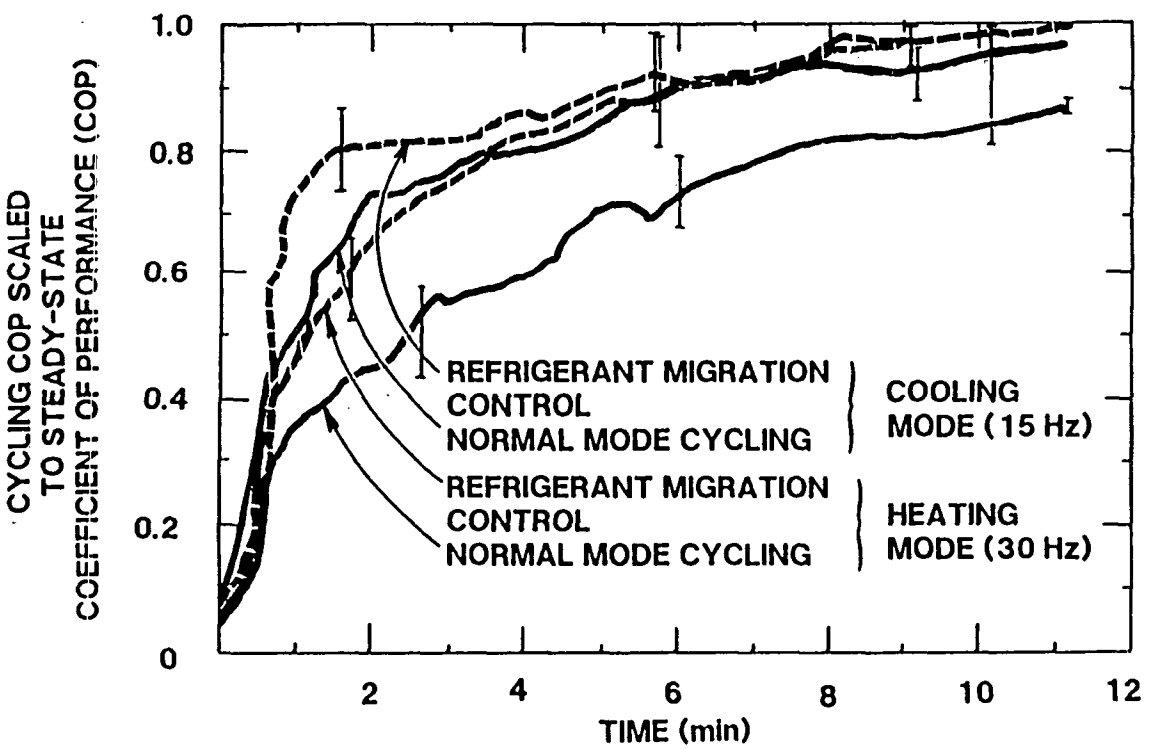

Figure 10 Improvement in both heating and cooling mode part-load COP due to cycling control strategy

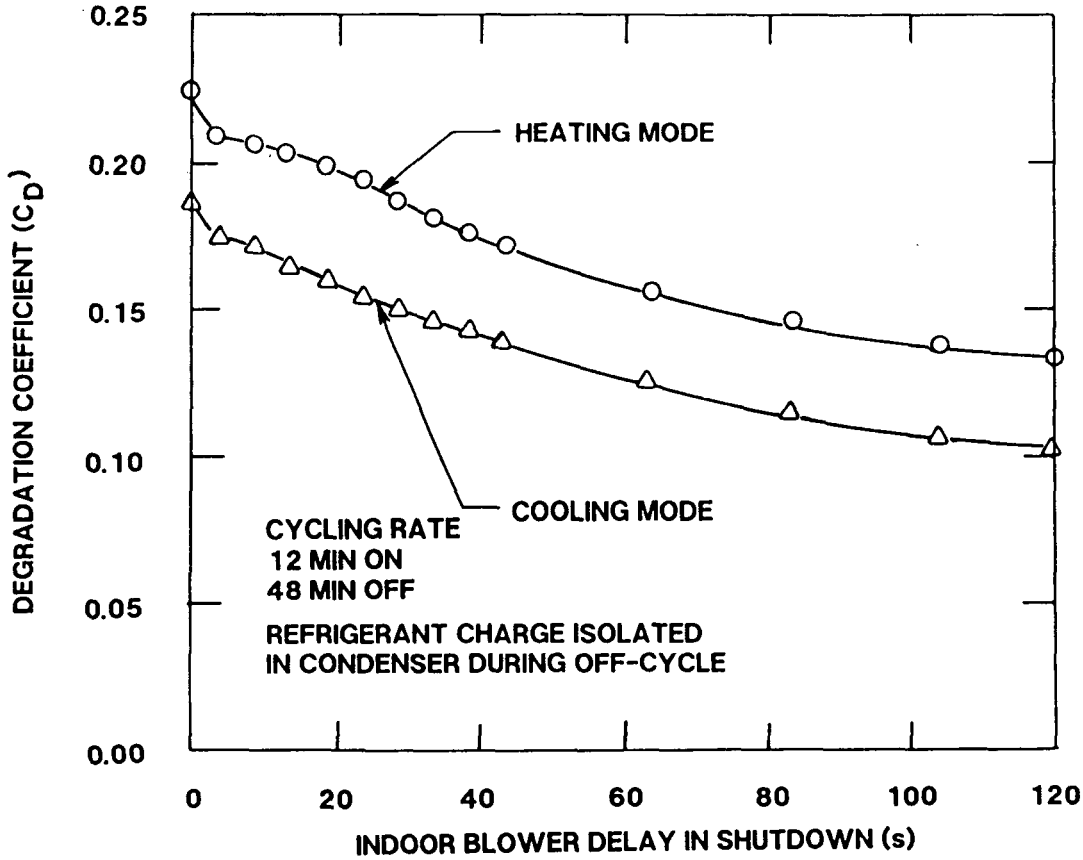

Figure 11 The effect of indoor blower delay on C_D for cycling tests controlling off-cycle refrigerant migration

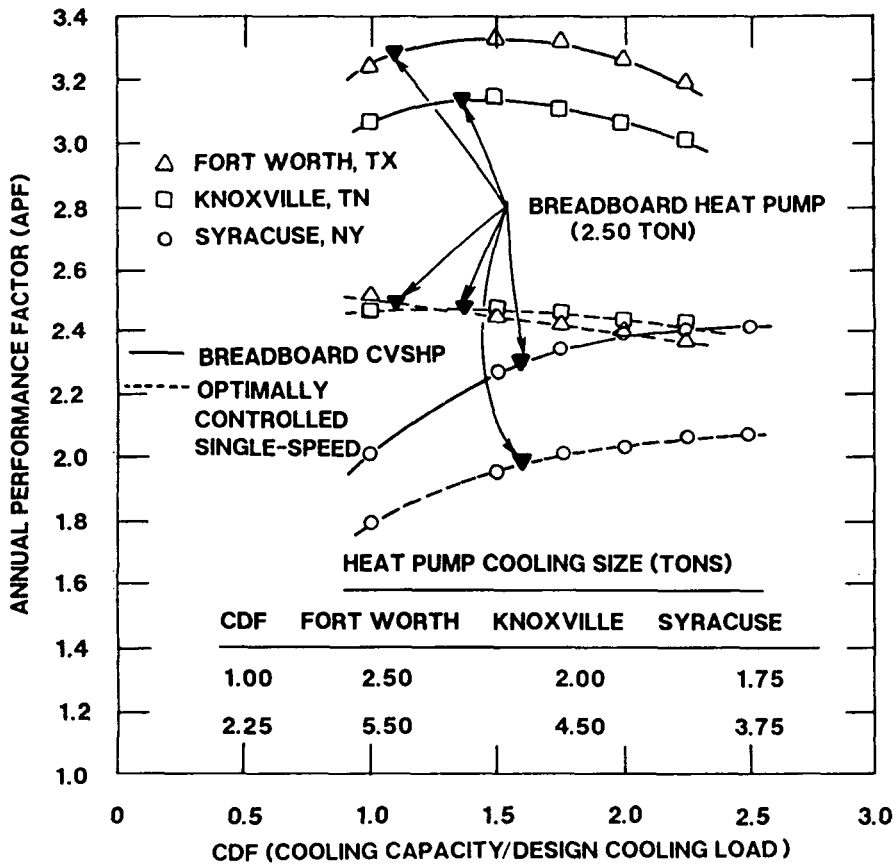

Figure 12 The annual performance factor for SSHP and CVSHP as function of oversizing ratio. Compressor and fan motors were sine-wave driven.

DISCUSSION

H. Akbari, Staff Scientist, LBL, Berkeley, CA: In your simulation you have not included the energy consumption by the controller itself. Knowing that the variable speed controller increases the energy use of the drive by 3% to 5%, how would they affect your simulated SEER? How would it affect the cost-effectiveness analysis?

W.A. Miller: The steady-state COPs of the optimally controlled, continuously variable speed heat pump (CVSHP) were based on the compressor and indoor blower motors both being driven by separate ideal induction motor drives (i.e., sine-wave driven). Seasonal simulations did not include the effect of inverter drive energy use. For the "tested system," the SEER for simulated operation in Knoxville (i.e., DOE Region IV) would degrade from 13.5 to 11.5, a 15% drop due to the effect of the "first generation" inverter drives.

The "first generation" inverter drives of the tested CVSHP were characterized and presented in work by Miller (1988). Rice (1988) showed that an advanced prototype, variable-speed motor and inverter drive [i.e., permanent magnet, electronically commutated motor (PMECM)] would have a combined efficiency even higher than that of an induction motor with ideal induction motor drive. In fact, state-of-the-art CVSHPs currently being introduced to the U.S. market have manufacturer-rated Region IV SEERs of about 15. This rating includes the energy consumption of PMECM drives for the compressor, indoor blower, and outdoor fan. Thus, for this paper, seasonal simulations were conducted based on performance for ideal induction motors drives that would better simulate state-of-the-art CVSHPs. Therefore, in terms of actual equipment, the cost premiums are probably conservative.