STEADY-STATE REFRIGERANT FLOW AND AIRFLOW CONTROL EXPERIMENTS FOR A CONTINUOUSLY VARIABLE SPEED AIR-TO-AIR HEAT PUMP

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ABSTRACT

A continuously variable speed, air-to-air split-system residential heat pump of nominal 2 3/4-ton (9.7-kW) capacity was instrumented and tested in the laboratory. The coefficient of performance (COP), capacity of the system, and component efficiencies were measured during steady-state heating- and cooling-mode operation. The indoor blower speed and the refrigerant subcooling at the condenser exit were varied at discrete compressor speeds to determine the best COP as constrained by residential comfort requirements.

Heating- and cooling-mode test results showed a 10% improvement in COP for operation with a variable area throttle as compared with a capillary tube throttle. Improved control of refrigerant flow and reduction of the indoor blower speed improved humidity control. As compared with results for capillary tube flow control, the sensible heat ratio was reduced from 0.95 to 0.77 at 15-Hz compressor speed.

INTRODUCTION

The capacity modulation of large commercial refrigeration systems is a well-established practice. However, the recent development of reliable, low-cost inverter drives has stimulated manufacturers to focus their attention on residential-sized heat pumps. An inverter drive allows variable-speed operation of the compressor for modulation of capacity output to follow the house load requirements. This results in improvements in seasonal energy efficiency as compared with single-speed heat pump systems through the reduction of dynamic losses, auxiliary heat requirements, and heat exchanger unloading.

Much work has been done to improve heat pump efficiency through inverter control of the compressor and indoor blower (Iida et al. 1982). A comparative study of six methods to adjust the speed of compressor motors for residential heat pumps evaluated both cost effectiveness and seasonal performance to determine their feasibility for residential use (Mohan and Ramsey 1986). Studies have also been conducted to investigate and improve motor efficiency when the motor is powered by six-step and pulse-width-modulated (PWM) inverter drives (Lloyd 1982). However, little has been done to optimize the refrigeration side of a variable-speed heat pump operating under part-load conditions. Detailed information revealing optimal airflow and refrigerant flow control for variable-speed systems is sparse.

A series of tests was conducted in the laboratory to demonstrate optimal refrigerant flow and airflow control for a continuously variable speed heat pump (CVSHP). A range of condenser exit subcooling and indoor airflow for each discrete compressor speed was tested to observe the effect on coefficient of performance (COP), capacity, and component efficiencies. An optimal ratio of compressor motor speed to indoor blower speed and the optimal level of subcooling at the condenser exit as a function of compressor speed were determined and compared with earlier test results for capillary tube flow control.

As the speed of the compressor was increased per test, any excess charge was held in the accumulator, which served as a receiver to maintain optimal charge level. Analytical results developed by Rice* indicated that the condenser exit subcooling could be reduced at constant charge with no noticeable loss in COP despite saturated conditions at the inlet to the compressor. Therefore, optimal airflow and refrigerant subcooling at the condenser exit were experimentally determined at discrete compressor speeds using a constant charge. This helped to reduce the total number of tests performed. Afterward, the minimum charge for each compressor speed was determined that duplicated previous experimentally determined optimal subcooling and airflow conditions. These later laboratory results revealed the COP to be within 2% of COP measured for tests having saturated conditions at the inlet to the compressor.

On command, the DAS and host computer monitored refrigerant-side and air-side steady-state data at 10-second intervals over a period of 0.5 hour for use in calculating average COP and capacity. Calculated values of heat pump heating-mode COP and capacity based on air-side measurements were within 3% of calculated values based on refrigerant-side measurements. For the cooling mode, the COP and capacity based on air-side measurements were within 6% of calculated values based on refrigerant-side measurements.

SYSTEM EFFICIENCY—HEATING MODE

Steady-State Tests: 40 F (4.4°C) Outdoor Air Temperature

Refrigerant-side COP and capacity measurements observed at 40 F (4.4°C) outdoor air temperature and for various levels of refrigerant subcooling at condenser exit per compressor speed are plotted in Figures 1 and 2 as a function of indoor blower speed.

Efficiency at 60-Hz Compressor Speed. For tests conducted at 60-Hz compressor speed, decreasing the condenser exit subcooling from 30 F to 20 F (17 C° to 11 C°) improved COP by 4%. Reducing the condenser exit subcooling to 10 F (6 C°) had less than a 1% effect on COP and capacity. However, less charge is required at the lower level of subcooling because the subcooled portion of the condenser, acting as a receiver, decreases. The increase in COP observed by lowering the refrigerant subcooling from 30 F to 10 F (17 C° to 6 C°) is caused by a reduction of condensing to evaporating temperature difference. This decreased the compressor power draw approximately 3% and increased capacity by 2%. For the tests conducted at 60-Hz compressor speed, the optimal blower speed yielding best COP was observed to be 50 Hz regardless of the level of condenser exit subcooling. However, as seen in Figure 2, increasing the blower speed from 50 Hz to 60 Hz caused a 4% increase in capacity with only a 1% drop in COP.

Efficiency at 30-Hz Compressor Speed. Test results at 30-Hz compressor speed revealed a 4% increase in COP as subcooling at the condenser exit was lowered from 20 F to 10 F (11 C° to 6 C°). The further lowering of subcooling to 5 F (3 C°) caused a slight drop in COP and capacity (Figures 1 and 2). The optimal blower speed at 30-Hz compressor speed and 10 F (6 C°) subcooling was 40 Hz. Yet varying the blower speed from 35 Hz to 45 Hz caused less than a 1% drop in COP, while capacity varied from 17.2 kBtu/h to 18 kBtu/h (5 kW to 5.3 kW), respectively, for the tests at optimal level of refrigerant subcooling.

Efficiency at 15-Hz Compressor Speed. Best COP measurements, shown in Figure 1, indicate approximately an 8% increase in COP when the compressor speed was lowered from 60 Hz to 30 Hz. However, when the speed was lowered from 30 Hz to 15 Hz, the measured COP dropped 20% relative to that observed at 30 Hz. This drop in COP at 15 Hz can be attributed to (1) a loss in heat exchanger efficiency, and (2) a drop in the isentropic efficiency of the compressor,† as explained in the following paragraphs.

The outdoor heat exchanger effectiveness‡ calculated over the two-phase region dropped sharply from 0.95 at 60-Hz compressor speed to 0.36 at 15-Hz compressor speed. The effectiveness of the indoor heat exchanger remained fairly constant from 60 Hz to 30 Hz but dropped from 0.85 to 0.72 as compressor speed was lowered to 15 Hz. The refrigerant mass flow rate decreased from 340 lb/h to 200 lb/h to 100 lb/h (154 kg/h to 91 kg/h to 45 kg/h) as the compressor speed was slowed from 60 Hz to 30 Hz to 15 Hz, respectively. The reduction of mass flow rate decreased

*Personal communication with C. K. Rice, Oak Ridge National Laboratory, Building Equipment Research Program, August 1986.

†Compressor isentropic efficiency is defined here as the ratio of isentropic compressor output (measured across the shell) to the measured power input.

‡Heat exchanger effectiveness is the ratio of the actual rate of heat transfer in a given heat exchanger to the maximum possible rate of heat exchange.

comfort can be maintained below the balance point with a lower auxiliary heat requirement. These results indicate that the optimally controlled CVSHP can satisfy comfort requirements at least as well as the single-speed unit can while better following house load.

SYSTEM EFFICIENCY—COOLING MODE

Cooling-mode tests were conducted at indoor ambient test conditions of 80 F (26.7°C) dry-bulb and 67 F (19.4°C) wet-bulb (ARI Standards 1984). The outdoor dry bulb was held constant at 82 F (27.7°C) and 95 F (35°C) per respective test series.

Steady-State Tests: 82 F (27.7°C) Outdoor Air Temperature

Results of tests conducted at 82 F (27.7°C) outdoor air temperature, shown in Figure 5, display the optimal levels of refrigerant subcooling and indoor blower speed at each respective compressor speed.

Efficiency at 60-Hz Compressor Speed. At 60-Hz compressor speed, optimal COP was observed at 45-Hz indoor blower speed. Optimal refrigerant subcooling was 20 F (11 C°). Increasing the subcooling to 30 F (16.7 C°) increased condensing temperature and pressure and increased the difference between the condensing and evaporating temperatures. This caused an increase in compressor power with only a slight increase in capacity resulting in a drop in COP. Decreasing the subcooling to 10 F (6 C°) further reduced condensing temperature and pressure, and the evaporator capacity (Figure 6) dropped as compared with that measured for tests at 20 F (11 C°) refrigerant subcooling. Results indicate that the drop in cooling capacity is more pronounced than the drop in compressor power and causes COP to drop (Figure 5). For testing at optimal blower and subcooling, the measured sensible heat ratio (SHR) was 0.71, which is acceptable for humidity control by industry standards.

Efficiency at 30-Hz Compressor Speed. Indoor blower speeds of 30 Hz to 40 Hz yielded near equal COP measurements for tests conducted at 30-Hz compressor speed. Decreasing the condenser exit subcooling from 20 F to 10 F (11 C° to 6 C°) resulted in a 6.5% increase in COP. The compressor power draw was 10% higher at 20 F (11 C°) subcooling than that at 10 F (6 C°) subcooling due to the increase of temperature lift across the compressor. Further lowering the refrigerant subcooling to 5 F (3 C°) caused a loss in evaporator capacity, as seen in Figure 6. At the blower speed of 40 Hz measured SHR was 0.83, and at 30 Hz measured SHR was 0.72.

Efficiency at 15-Hz Compressor Speed. At 15-Hz compressor speed the optimal blower speed yielding best COP did not yield best latent capacity. In terms of best COP, the indoor blower speed was 35 Hz, yet the latent capacity was negligible. Lowering the indoor blower speed from 35 Hz to 16.7 Hz lowered the SHR from 1.00 to 0.77, thus improving the moisture-removing capability of the evaporator coil. The indoor airflow at 16.7-Hz blower speed was 361 cfm (170 L/s), which may be too low for proper air distribution through a supply air duct system. Further study is required to determine minimum airflow requirements for maintaining proper humidity control in a residence.

The effect of refrigerant subcooling on COP and capacity was greatest at 15-Hz compressor speed, as seen in Figures 5 and 6. A refrigerant subcooling of 10 F (6 C°) at the condenser exit caused COP measurements to fall below performance observed at 30-Hz compressor speed. However, at a condenser exit subcooling of 2.5 F (1 C°), COP improved above optimal COP measurements at 30-Hz compressor speed. This is a result not previously seen in heating-mode operation. This improved performance is the result of improved part-load cooling-mode heat exchanger efficiency at 15-Hz compressor speed as compared with that of the heating mode. In the cooling mode, the effectiveness of both indoor and outdoor coils is approximately constant as a function of compressor speed; in the heating mode, the effectiveness of the heat exchangers drops. The cooling-mode refrigerant mass flow rate is double the heating-mode mass flow rate at 15-Hz compressor speed because of the higher working temperature. Thus, with a higher refrigerant velocity through the coils in the cooling mode coupled with the higher working ambient temperature, apparently there was no reduction in refrigerant-side heat flux in the cooling mode as compared with the heating mode.

Comparison with Capillary Tube. Figure 7 shows a comparison of cooling-mode performance for the heat pump with the original capillary tubes and with the variable-area metering valves. Measured air-side COP for tests conducted at 15-Hz compressor speed and 82 F (27.7°C) outdoor air temperature was much improved compared with previous results with the same heat pump operating with capillary tube flow control. The improvement in both COP and SHR is easily seen in Figure 7. The COP increased 12% and the SHR was lowered from 0.92 to 0.77, indicating much improved

observed to yield optimal or near optimal COP for compressor speeds of 30 Hz to 90 Hz operating in 40 F and 10 F (4.4°C and −12°C) outdoor air ambient temperatures. For these heating-mode conditions, the indoor blower could actually be set at 45-Hz speed with less than a 2% drop in COP measured at the optimum blower speeds.

Best COPs in the cooling mode required a greater change in blower speed and refrigerant subcooling at each compressor speed tested. Cooling-mode indoor blower speed varies from 45 Hz at 60-Hz compressor speed to 16.7 Hz at 15-Hz compressor speed (Figure 5). Refrigerant subcooling for optimal COP is 20 F (11 C°) at 60-Hz compressor speed and 82 F and 95 F (27.7°C and 35°C) outdoor air temperature, while at 45-Hz through 15-Hz compressor speed optimal subcooling varies from 20 F to 2.5 F (11 C° to 1 C°). These results reveal that greater trade-offs will be required in the cooling mode for blower speed and humidity control, which satisfy comfort yet still provide a minimal first cost for CVSHPs. However, results do indicate that (1) at a 15-Hz compressor speed setting the indoor blower at 16.7 Hz yielded much improved humidity control, (2) at 82 F (27.7°C) the indoor blower can be set at 35 Hz with near optimal COP and good humidity control, and (3) at 95 F (35°C) a blower speed of 45 Hz would yield near optimal COP and capacity with acceptable humidity control.

The optimal steady-state performance of the CVSHP with variable-area refrigerant flow control was compared with a similar single-speed unit having capillary tube flow control. Results in Table 2 show that at 60-Hz compressor speed the cooling- and heating-mode capacity of the CVSHP is comparable with that of the single-speed unit. COP improved 12%, and the energy efficiency ratio (EER) improved 5% over the efficiency of the single-speed unit. Better control of the refrigerant flow improved refrigerant-side capacity and allowed a reduction of indoor blower speed resulting in the increase in COP and EER. Comparison of the CVSHP performance, having optimal airflow and refrigerant flow control settings, with that of a single-speed unit indicated that the CVSHP satisfies both heating and cooling comfort requirements while better following house load through capacity modulation. Reducing the compressor speed by half (60 Hz to 30 Hz) reduced heating capacity at 40 F (4.4°C) outdoor air temperature to 55% of the 60-Hz value and cooling capacity at 82 F (27.7°C) outdoor air temperature to 62%. At quarter speed (15 Hz), the cooling-mode capacity was reduced to 32% of the 60-Hz value. Overspeeding the compressor from 60 Hz to 90 Hz increased capacity output by 44% of capacity for 60-Hz compressor speed at 10 F (−12°C) outdoor air temperature.

The trends of COP, blower/compressor speed, and optimal levels of subcooling would be similar for other reciprocating compressors and inverter drives. Factors such as compressor valve design, heating of the suction gas, opening and closing delays of the compressor valves and characteristic efficiency of the inverter drive would change the absolute value of COP; yet the trends would remain the same. However, the results of this study may differ with advanced compressor and compressor drives. Separate studies by Senshu et al. (1985) and Itami et al. (1982) showed that the efficiency trends of scroll and rotary compressors differ from trends of reciprocating compressors. Isentropic and volumetric efficiencies of rotary and scroll compressors improve as speed is increased from 60 Hz through approximately 90 Hz; however, the isentropic and especially the volumetric efficiencies of reciprocating compressors drop from 60-Hz to 90-Hz speed. Senshu et al. showed the volumetric efficiency of a scroll compressor much improved over a conventional reciprocating compressor for pressure ratios ranging from 2 to 4.

Further work is continuing to validate a CVSHP computer model developed by Rice* at ORNL. Once validated against laboratory data presented in this paper, the model will be used to determine system efficiency trends as a function of speed using advanced compressors and high-efficiency drives.

CONCLUSIONS

1. For the CVSHP tested, the compressor speed in the heating mode should be limited to 30 Hz for best efficiency. However, in the cooling mode the compressor speed can be reduced to 15 Hz with observed improvement in efficiency.
2. The variable-area refrigerant flow control maximized refrigerant-side capacity and increased steady-state COP approximately 10% as compared with the capillary tube refrigerant flow control.
3. Much improved cooling-mode humidity control was observed at 15-Hz compressor speed with variable-area flow control.
4. Too low a refrigerant velocity through the heat exchangers decreased effective refrigerant to air heat transfer and caused COP to drop for heating tests conducted at 15 Hz as compared with 30-Hz compressor speed.

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TABLE 1

Capacity and Delivery Air Temperature for a Single-Speed Heat Pump and the Continuously Variable Speed Heat Pump (CVSHP)

Outdoor temperature [F (°C)]	House load[b] [kBtu/h (kW)]	Single-speed heat pump[a]		Laboratory data for CVSHP with optimal COP		
		Capacity [kBtu/h (kW)]	Supply air [F (°C)]	Capacity [kBtu/h (kW)]	Supply air [F (°C)]	Compressor speed (Hz)
40 (4.4)	11.3 (3.3)	32.1 (9.4)	94.8 (34.8)	31.2 (9.1)	98.5 (37.0)	60
				17.0 (5.0)	92.4 (33.5)	30
				11.5 (3.4)	85.7 (29.8)	20
				8.5 (2.5)	81.6 (27.5)	15
10 (−2.1)	30.3 (8.9)	16.1 (4.7)	82.4 (28.0)	22.6 (6.6)	92.8 (33.9)	90
				17.3 (5.0)	86.7 (30.4)	60

[a]Manufacturer's data for the same heat pump operating with a capillary tube flow control.
[b]Calculated heating load for an 1800-ft^2 (167-m^2) house in Knoxville, TN, having Housing and Urban Development minimum insulation.

TABLE 2

Measured Optimal Steady-State Performance at 60-Hz Compressor Speed Compared with Measured Data for the Same Test Unit Operating as a Single-Speed Unit

Unit	Throttle	Cooling mode[a]			Heating mode[b]		
		Airflow [cfm (m^3/s)]	Capacity [kBtu/h (kW)]	EER[c]	Airflow [cfm (m^3/s)]	Capacity [kBtu/h (kW)]	COP[d]
Single speed	Capillary	1275 (0.60)	32.0 (9.4)	7.3	1280 (0.60)	31.2 (9.1)	2.55
Variable speed	Variable area	957 (0.45)	30.7 (9.0)	7.7	1043 (0.49)	31.2 (9.1)	2.88

[a]Cooling tests conducted at Air-Conditioning and Refrigeration Institute ambient test conditions.
[b]Heating tests conducted at 40 F (4.4°C) outdoor air, 70 F (21°C) return air temperatures.
[c]Energy efficiency ratio.
[d]Coefficient of performance.

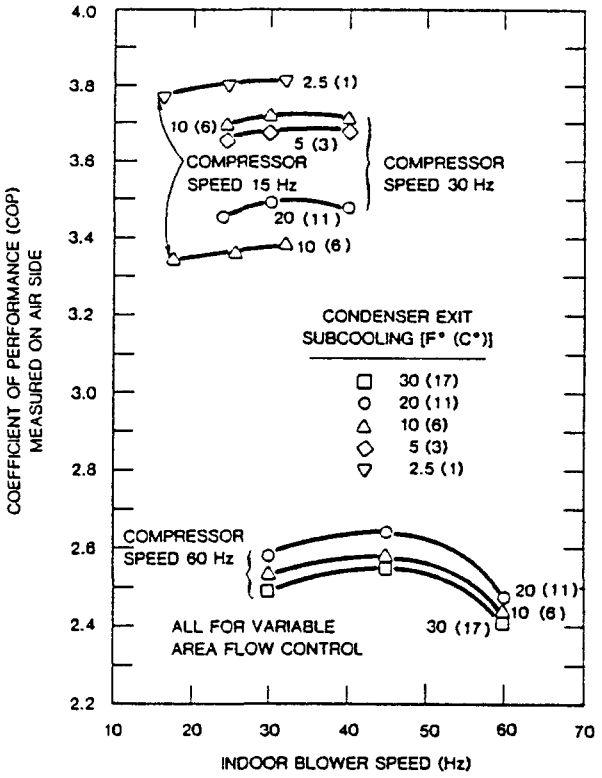

Figure 5. COP measured at 82 F (27.7°) outdoor air temperature for the variable speed heat pump with variable area refrigerant flow control

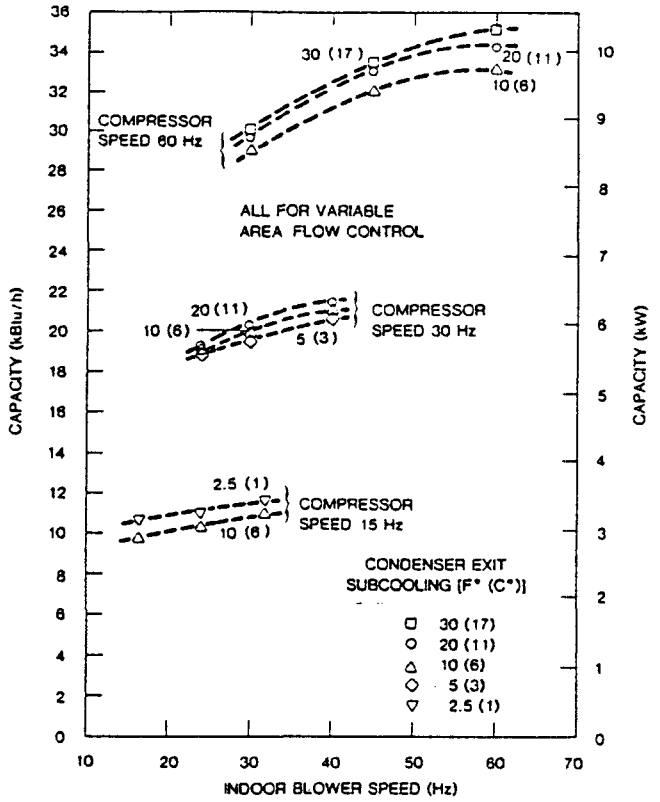

Figure 6. Capacity measured at 82 F (27.7°C) outdoor air temperature for the variable speed heat pump with variable area refrigerant flow control

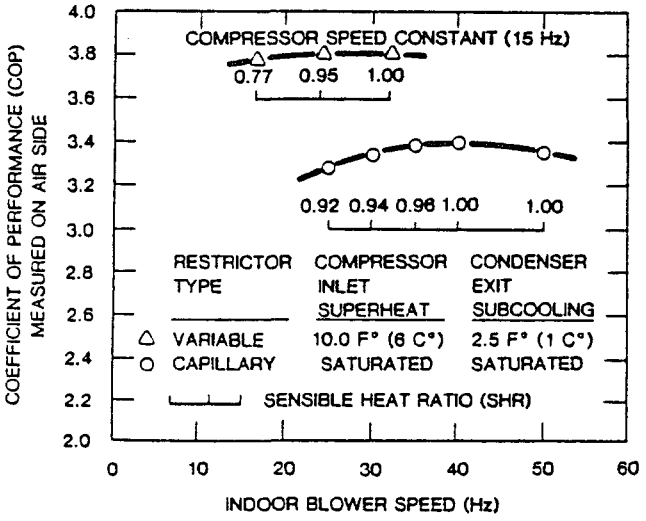

Figure 7. COP and SHR observed for variable area flow control vs. capillary flow control, outdoor air temperature 82 F (27.7°C)

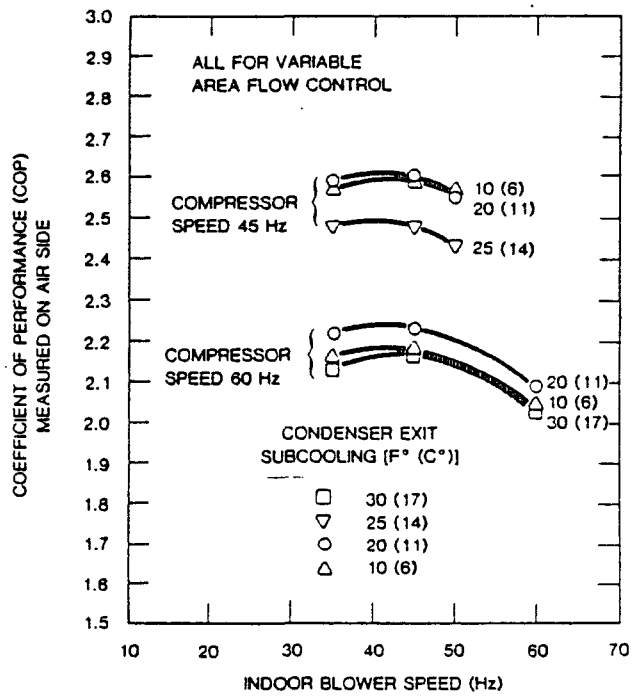

Figure 8. COP measured for the variable speed heat pump operating at 95 F (35°C) outdoor air temperature

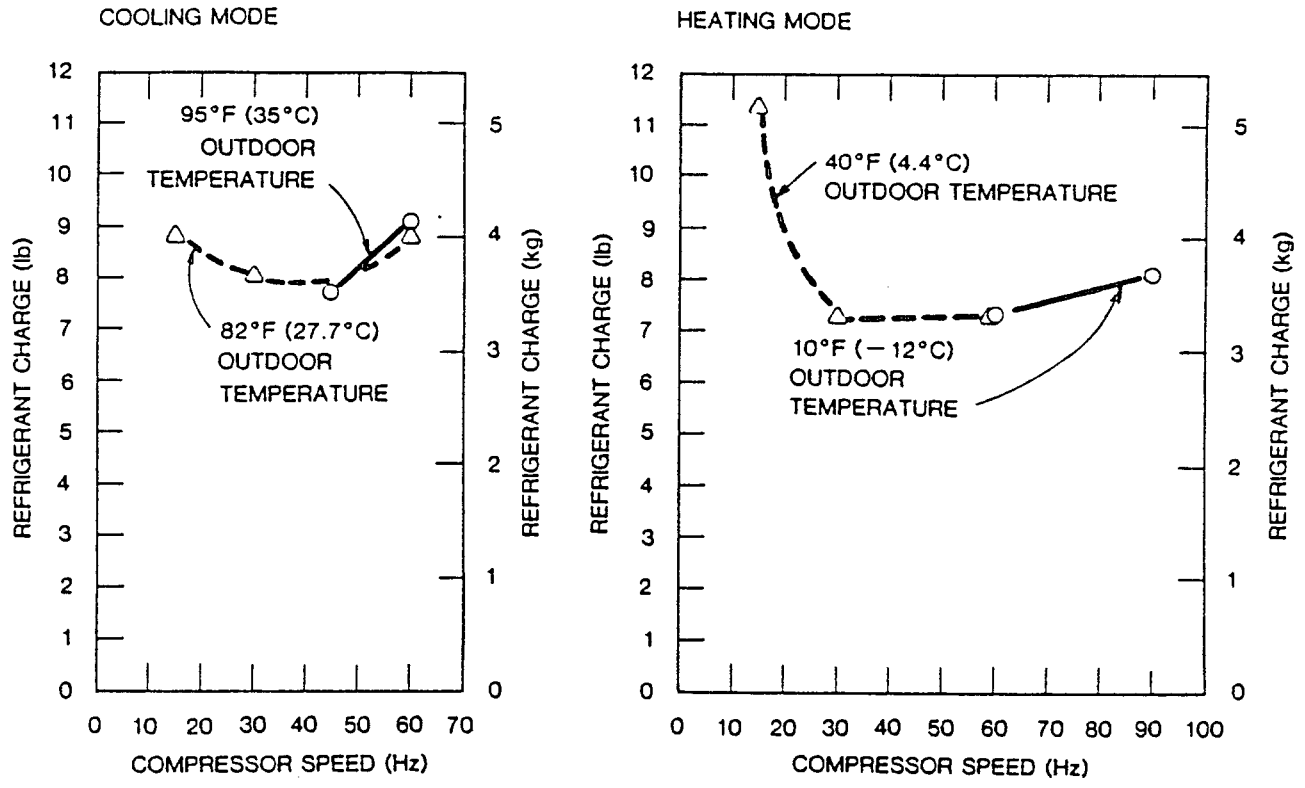

Figure 11. Optimal refrigerant charge required for both low superheat at the inlet of the compressor and optimal level of refrigerant subcooling at the condenser exit

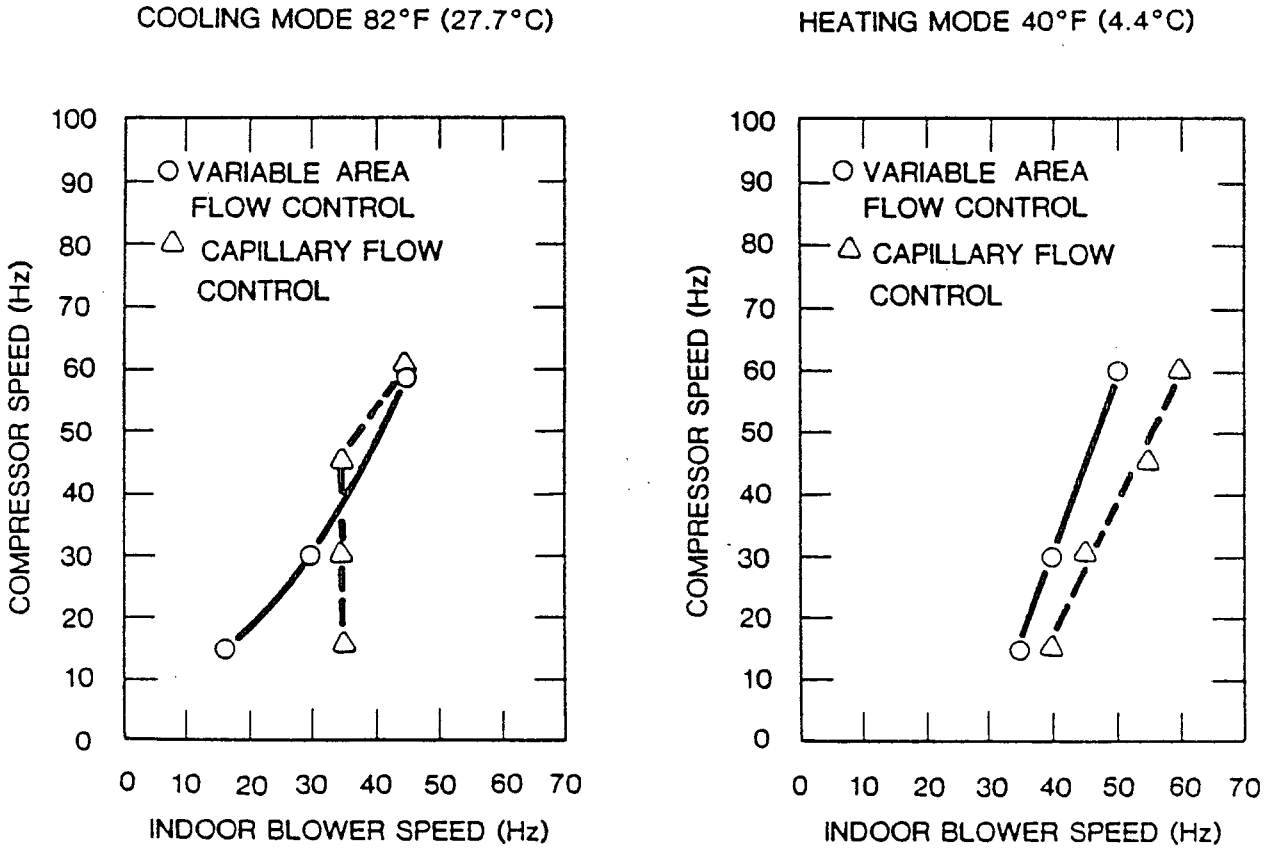

Figure 12. Optimal ratio of indoor blower motor speed to compressor motor speed for the test heat pump operating with variable area flow control and capillary flow control