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A Performance Evaluation of a Variable Speed, Mixed Refrigerant Heat Pump



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Department of Commerce
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Gaithersburg, MD 20899



Prepared for:

U.S. Environmental Protection Agency
Air and Energy Engineering Research Laboratory
Research Triangle Park, NC 27711

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A Performance Evaluation of a Variable Speed, Mixed Refrigerant Heat Pump

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U.S. Department of Commerce
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DISCLAIMER

This study was conducted utilizing a ductless split heat pump manufactured by the National Division of the Matsushita Company of Japan. It was selected because of its uniqueness in design; that is, it was the only heat pump available that incorporates both a variable speed compressor and a zeotropic refrigerant mixture to vary the capacity. At the time of this publication, this particular design is no longer commercially available because one of the refrigerants used (R13B1) has a very high ozone depletion potential. The NIST in no way endorses or criticizes this product, but rather simply reports on its performance under the test conditions noted for the purposes of obtaining an estimate of the potential benefits the unique design characteristics may offer.

ABSTRACT

The performance of an innovative heat pump, equipped with a distillation column to shift the composition of a zeotropic refrigerant mixture, was evaluated. The results of the DoE rating tests and seasonal energy calculations are reported with the main cycle refrigerant compositions. No composition shifting of the circulating refrigerant mixture was observed. To demonstrate the potential value of composition shifting, an ideal vapor compression cycle computer program was used to predict what the system performance would have been had composition shifting occurred. Seasonal energy usage calculations based on the computer predictions demonstrated that the effect of composition shifting on HSPF was very small, increasing slightly with climate zone. However, the savings in auxiliary heat was found to be substantial. In cooling mode, computer predictions showed pure R22 to have a SEER approximately two percent higher than a mixture of 20% R13B1 and 80% R22 by weight.

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TABLE OF CONTENTS

TABLE OF CONTENTS.....vi
LIST OF FIGURES.....vii
LIST OF TABLES.....viii
1. INTRODUCTION.....1
2. BACKGROUND.....5
 2.1 ZEOTROPIC MIXTURES.....5
 2.2 DYNAMIC RESPONSE TO CHANGES IN SOURCE TEMPERATURE.....10
3. REVIEW OF PREVIOUS RESEARCH.....11
4. UNIT DESCRIPTION.....17
5. DESCRIPTION OF TEST APPARATUS.....21
6. TEST RESULTS.....25
7. DISCUSSION OF RESULTS.....29
8. RECOMMENDATIONS FOR FUTURE WORK.....37
9. REFERENCES.....38
APPENDIX A - STEADY STATE HEATING MODE ERROR ANALYSIS.....41
APPENDIX B - STEADY STATE COOLING MODE ERROR ANALYSIS.....45

LIST OF FIGURES

1.	Building load and building system capacity versus outdoor temperature for a single speed heat pump operating with a pure refrigerant.....	1
2.	Building load versus system capacity for a variable speed heat pump operating with a pure refrigerant.....	3
3.	Building load versus system capacity for a single speed heat pump with composition shifting of a zeotropic mixture.....	3
4.	Temperature versus concentration equilibrium diagram for a binary zeotropic mixture at a constant pressure.....	5
5.	Typical flow diagram for a single stage equilibrium distillation.....	6
6.	Enthalpy versus concentration diagram for a single stage equilibrium distillation process.....	8
7.	Temperature versus concentration diagram for a single stage equilibrium distillation process.....	8
8.	Typical distillation column used for a continuous rectification process.....	9
9.	Unit piping schematic.....	18
10.	Test apparatus schematic.....	22

LIST OF TABLES

1. Heating test results.....27

2. Cooling test results.....28

3. Discharge temperature for all tests.....31

4. Comparison of computer predictions with actual
test data for pure R22.....33

5. Results of SEER calculation for three speed
air-conditioner for two refrigerant compositions.....34

6. Variation of HSPF and amount of auxiliary heat required
for a variable speed heat pump employing composition
control in various climate regions.....35

7. Variation of HSPF and amount of auxiliary heat required
for a single speed heat pump employing composition
control in various climate regions.....36

1. INTRODUCTION

The heating capacity of a single speed air-to-air heat pump is directly proportional to the outside (evaporator) temperature. As the outside temperature falls, the suction pressure and the suction temperature also fall. This causes both the suction vapor specific volume and the compression ratio to increase. The system heating capacity is thereby reduced and the compressor work input is increased. The building heat load, on the other hand, is directly proportional to the temperature difference between the indoor and outdoor air. Consequently, the building heat load increases as the outside temperature falls.

The building heat load and system capacity are plotted as functions of the outside temperature in Figure 1. The outside temperature at which the building load equals the system capacity is referred to as the balance point. If the outdoor temperature is below the balance point, the system capacity will be insufficient to satisfy the heating needs of the structure. In order to maintain the indoor temperature, the system capacity will have to be supplemented by an auxiliary energy source.

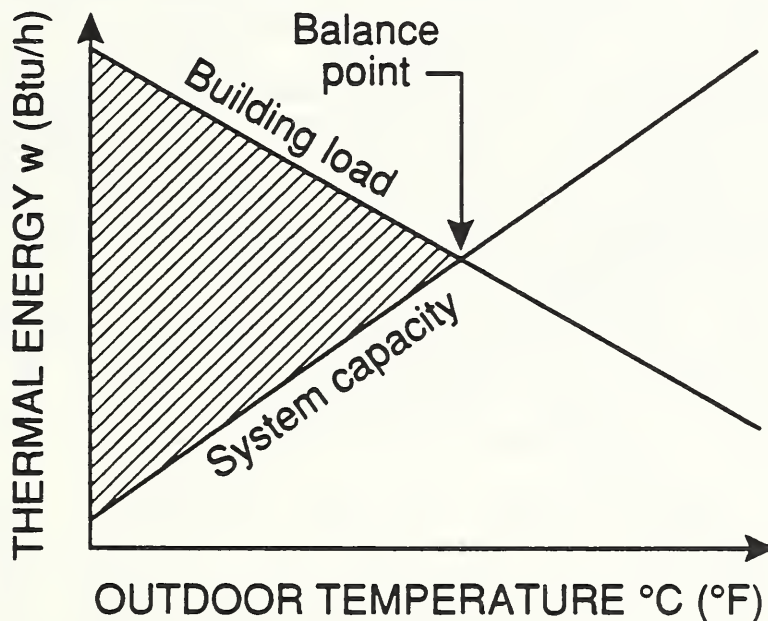


Figure 1. Building Load and System Capacity versus Outdoor Temperature for a Single Speed Heat Pump Operating with a Pure Refrigerant

The auxiliary energy required for an entire heating season is the difference between the seasonal building load and the seasonal heat pump output below the balance point. The auxiliary energy is usually supplied by electric resistance heating, which has a coefficient of performance (COP) of one. Since the COP of a heat pump is virtually always greater than one, the heating seasonal performance factor (HSPF) can be increased by reducing the auxiliary heat required. Although the increase in HSPF is usually modest (depending on the climatic region), the reduction in peak load to the electric utility can be significant. This energy demand reduction benefit is often magnified if the electric power plant uses a less efficient electric generation system to meet the peak demand.

To reduce the amount of auxiliary heat required, the heat pump capacity must be increased to match the building load as the outside temperature falls. The only commercially available heat pumps capable of matching system capacity to the building load have been those which vary the volumetric capacity of the compressor. This can be accomplished several ways; the most common methods are two-speed compressor motors and frequency inverters for variable speed compressor motors. The system capacity can also be controlled by varying the composition of a zeotropic refrigerant mixture.

The capacity increase that can be achieved by changing the composition of a zeotropic refrigerant mixture has been demonstrated by many investigators. However, the complexities involved with providing continuous composition control have thus far limited its commercial applications. The effects of increasing the heat pump capacity to match the building load are shown for a variable speed system in Figure 2. These same effects are also shown for a single speed system capable of shifting the composition of a zeotropic refrigerant mixture in Figure 3. In Figure 3, it is assumed that the heat pump would operate on the pure refrigerant until the outdoor temperature falls to the pure refrigerant balance point (a). Figures 2 and 3 show that the result of increasing the capacity will be a lower balance point (from a to b), reduced auxiliary energy requirement and an increase in the HSPF.

The heat pump tested in this study is equipped with a distillation column to provide continuous control over the refrigerant composition. The unit is charged with a zeotropic refrigerant mixture of 80% R22 and 20% R13B1 by weight. The distillation column is intended to optimize the composition of the circulating refrigerant for different operating conditions. In the cooling mode, the distillation column should separate and store the more volatile R13B1 component. In this way, the system takes advantage of the lower operating pressures and higher COP of pure R22. In heating mode, the capacity is increased to match the building load by shifting the refrigerant composition towards greater percentages of R13B1. The properties of the resulting

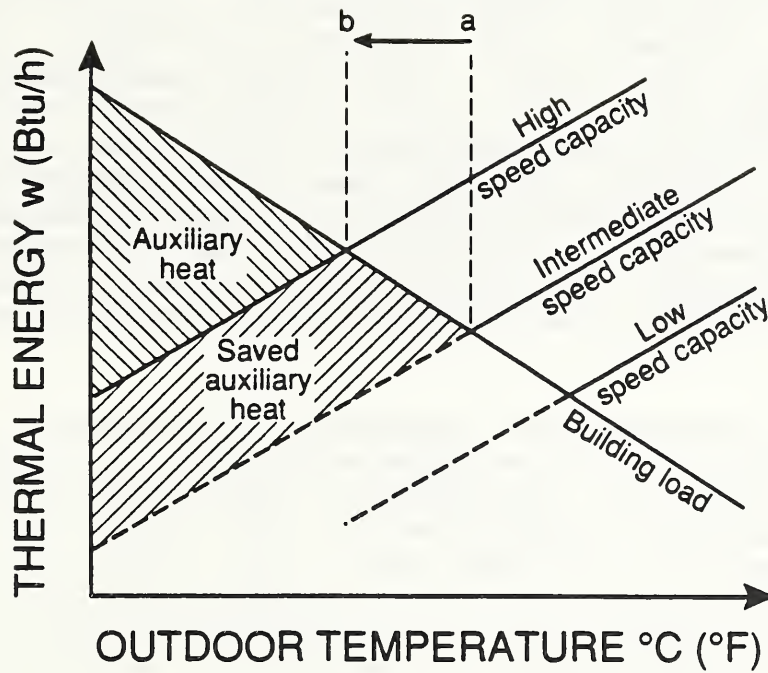


Figure 2. Building Load versus system capacity for a variable speed heat pump operating with a pure refrigerant

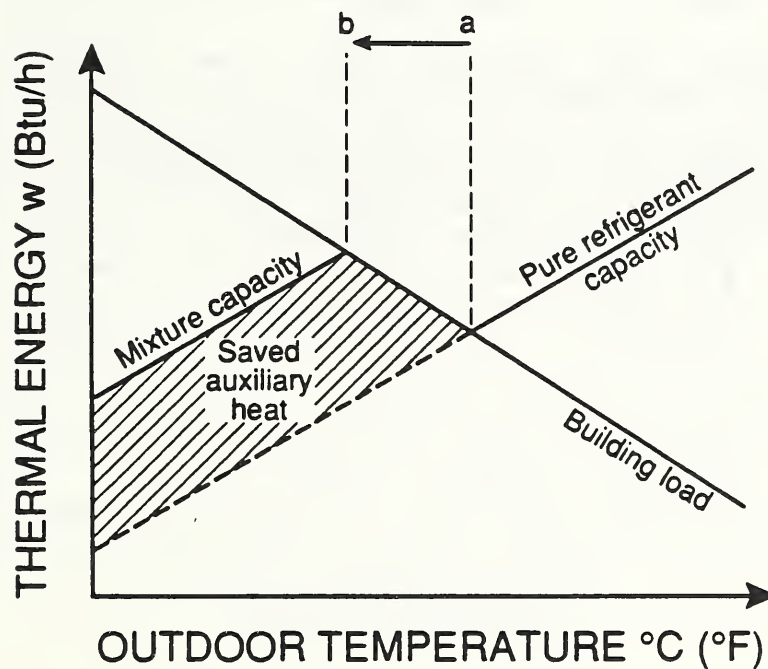


Figure 3. Building Load versus system capacity for a single speed heat pump with composition shifting of a zeotropic refrigerant mixture

refrigerant mixture are much better suited to the low temperature heating application than those of pure R22. Specifically, the high suction vapor pressure and density of R13B1 increases the refrigerant mass flow rate. The higher refrigerant flow rate increases the capacity sufficiently to compensate for the mixture's lower latent heat of vaporization (which tends to decrease the capacity). The maximum capacity for a given outside temperature results when the circulating refrigerant composition equals that of the initial charge (i.e., 20% R13B1/80% R22 by weight). The unit is also equipped with a variable speed compressor, which gives the system an additional method of capacity control, as shown in Figure 2.

There were two primary purposes for undertaking this study. The first objective was to determine the extent to which the distillation column can control the composition of the zeotropic refrigerant mixture. Additionally, the study was intended to demonstrate that controlling the composition of a zeotropic refrigerant mixture can increase the HSPF and reduce the seasonal auxiliary energy requirement. For the cooling mode, the effect of composition control and speed variation were examined by performing seasonal energy efficiency ratio (SEER) calculations. Since composition control and variable speed capability are both methods of controlling capacity, their effects on system performance are complementary. In order to isolate the effects of composition control from those of variable speed operation, they were studied both separately and in conjunction with each other.

To determine the capacity and power inputs necessary to perform the HSPF and SEER calculations, an experimental test rig was constructed in accordance with ASHRAE Standard 116-1983 [1]. The test conditions and corresponding compressor speed selections were as prescribed in the Department of Energy/National Institute of Standards and Technology (DoE/NIST) variable speed rating procedure [2]. The DoE/NIST procedure also describes a method for performing HSPF and SEER calculations for variable speed systems. The DoE/NIST procedure was written for systems which have continuously variable speed control. However, the heat pump used in this study was only capable of varying its speed in discrete intervals. The HSPF and SEER calculation procedures were modified to account for discrete interval speed control. To ascertain the extent of composition shifting, refrigerant composition samples were taken during all tests. The samples were analyzed with a gas chromatograph.

2. BACKGROUND

2.1 Zeotropic mixtures

Zeotropic mixtures are those which exhibit a temperature change as they condense or evaporate at constant pressure. An additional characteristic of zeotropic mixtures is that as they condense or evaporate the compositions of the liquid and vapor phases are not equal. A temperature versus concentration equilibrium phase diagram for a binary zeotropic mixture is shown in Figure 4.

To better understand the vaporization process for a binary zeotropic mixture, consider a closed container which is entirely filled with a liquid solution represented by point A in Figure 4. It is assumed that some means for keeping the container pressure constant is provided. If the container is now heated, the first bubble of vapor will form at B and has the composition at F. As more of the mixture is vaporized, the composition of the liquid and vapor phases continues to change. If the process was stopped at

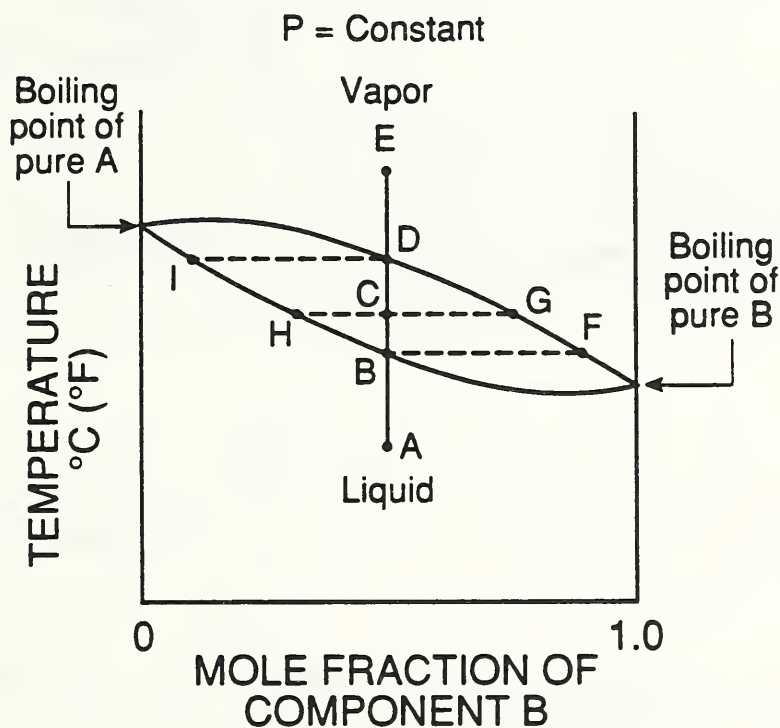


Figure 4. Temperature versus concentration equilibrium diagram for binary zeotropic mixture at a constant pressure

some arbitrary point, the compositions of the equilibrium vapor and liquid phases would be G and H respectively. The last drop of liquid vaporizes at D and has the composition at I. Superheating the mixture follows the path from D to E. Unlike the single vaporization temperature of a pure substance, the mixture has vaporized over a temperature range from B to D. If the superheated vapor at E is cooled, all of these state points reappear in reverse order.

There are several methods of utilizing the liquid-vapor equilibrium composition difference to separate a zeotropic mixture into its respective components. The methods which have been applied to vapor compression heat pumps are single-stage equilibrium distillation (or flash distillation) and continuous rectification. Single-stage equilibrium distillation is a process wherein a partially vaporized liquid mixture is introduced into a vessel capable of separating the resulting equilibrium phases. A typical separating vessel for this process is shown in Figure 5.

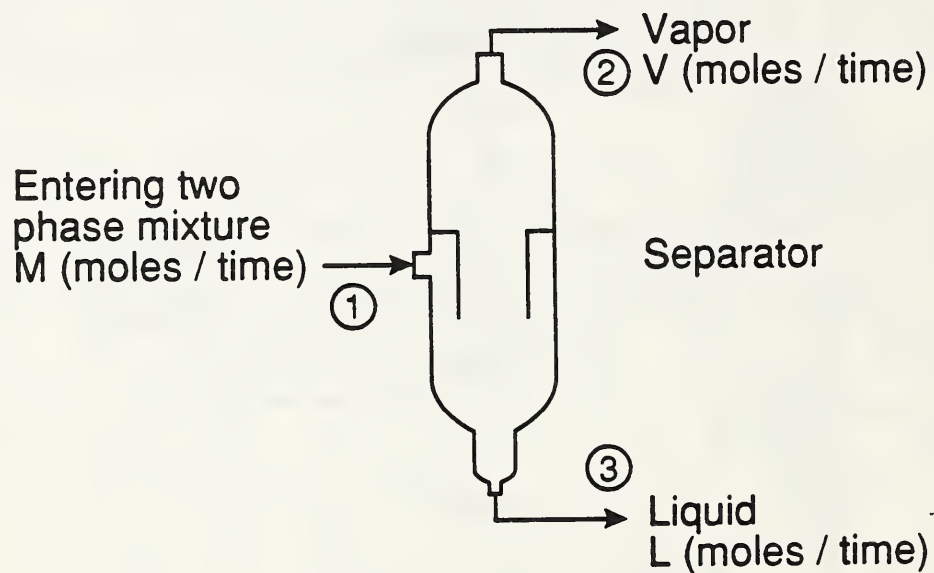


Figure 5. Typical flow diagram for a single stage equilibrium distillation

If the entering fluid is a binary zeotropic mixture of pure components A and B, and Z is the mole fraction of the more volatile component (i.e., $Z=B/[A+B]$), then the material and enthalpy balances are

$$M = V + L$$

$$M*Z_1 = V*Z_2 + L*Z_3$$

$$M*H_1 = V*H_2 + L*H_3$$

A simultaneous solution yields the expression

$$L / V = (Z_1 - Z_2)/(Z_3 - Z_1) = (H_1 - H_2)/(H_3 - H_1)$$

This equation represents a straight line on an enthalpy-concentration diagram. Since the effluent streams are assumed to be in equilibrium, the process can also be represented on a temperature-concentration diagram. The enthalpy-concentration and temperature-concentration diagrams for this process are shown in Figures 6 and 7 respectively. Figures 6 and 7 show that the vapor effluent will be rich in the more volatile component, whereas the liquid will be rich in the less volatile component. This process requires simple hardware and no active control, but it is limited in the degree of separation that can be achieved.

Continuous rectification is a multistage countercurrent distillation operation. For most binary mixtures, this method can separate the mixture into its components with any desired purity, assuming the rectification column is large enough. The analysis of a continuous rectification process is considerably more complex than a single-stage equilibrium distillation process. Therefore, it will be given a brief qualitative description only. The interested reader can find a detailed treatment in references [3] and [4].

A typical distillation column used in a continuous rectification process is shown schematically in Figure 8. The distillation column is composed of perforated plates or other types of packing material. A vapor solution V ascends through the column and a liquid solution L descends. The purpose of the plates is to bring the liquid and vapor solutions into intimate direct contact. This is necessary to achieve the high rates of heat and mass transfer necessary for efficient separation. The vapor rising through the column has some liquid condensed from it, while the descending liquid has some vapor evaporated from it. As this occurs, the concentration of the more volatile component becomes progressively stronger in the ascending vapor, and weaker in the descending liquid. In principle, both of these methods are the same with the latter involving more hardware but being capable of producing greater composition shifts.

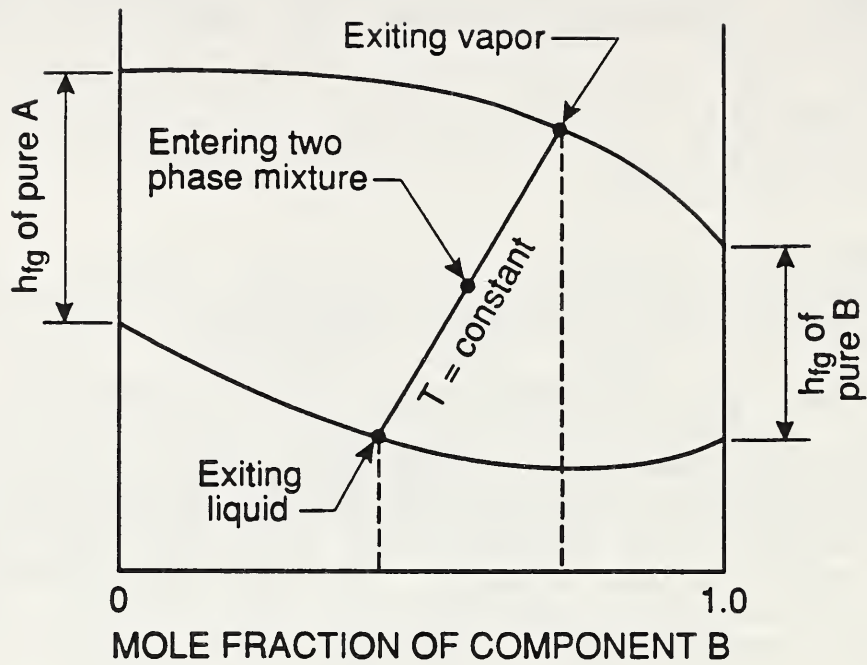


Figure 6. Enthalpy versus concentration diagram for a single stage equilibrium distillation process

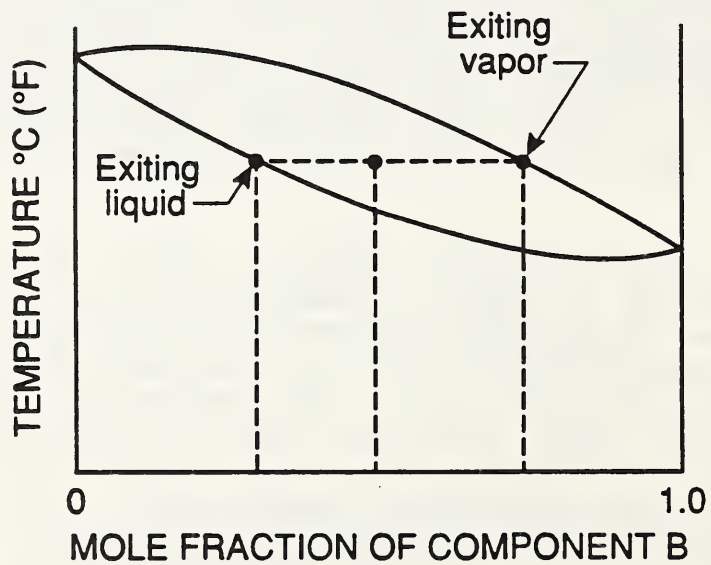


Figure 7. Temperature versus concentration diagram for a single stage equilibrium distillation process

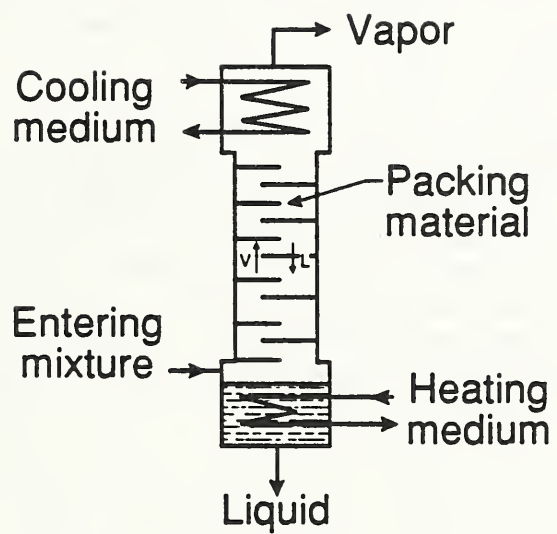


Figure 8. Typical distillation column for a continuous rectification process

2.2 Dynamic response to changes in heat source temperature

The main purpose of composition shifting is to increase the heat pump capacity to match the building load as the outside temperature falls. To accomplish this the system must affect a composition shift in response to a change in heat source temperature. To understand how this can be accomplished, it will be instructive to examine how a pure refrigerant heat pump responds to a change in source temperature. Consider a system with a constant area expansion device operating in the heating mode at steady state conditions. As the outside temperature falls, the evaporator pressure and temperature will decrease commensurately. Since the indoor temperature remains relatively constant the high side pressure will also remain relatively constant. Therefore, the compressor will pump less refrigerant due to the lower suction vapor density and increased compression ratio. However, the expansion device will continue to pass the same amount of refrigerant since it is relatively insensitive to changes in low side conditions [5,6].

Because the expansion device is now passing more refrigerant than the compressor is pumping, the refrigerant charge will migrate to the low pressure side of the system where it will collect in the suction accumulator. As the refrigerant migration proceeds, the condenser will contain progressively less liquid. The migration of refrigerant cannot proceed indefinitely, so the system must adjust itself to achieve a new steady state. Fortunately, the expansion device mass flow rate is directly proportional to the high side pressure and the liquid subcooling. The high side pressure has remained constant, but the amount of subcooling is determined by the amount of liquid in the condenser. Therefore, as the condenser is depleted of liquid the system loses subcooling. Since the amount of refrigerant passed by the expansion device is directly proportional to the amount of subcooling, the gradual loss of subcooling causes a gradual decrease in expansion device mass flow rate. When the expansion device mass flow rate becomes equal to the compressor mass flow rate, a new steady state equilibrium condition will be reached. This new equilibrium condition will have less subcooling, and may even have two phase refrigerant entering the expansion device.

3. REVIEW OF PREVIOUS RESEARCH

One of the earliest experimental studies of the performance of various refrigerants and zeotropic mixtures was conducted by McHarness and Chapman [7]. They constructed an experimental water source test rig and evaluated the operating performance of several pure refrigerants and zeotropic mixtures of these pure components. Since the test rig was not designed for composition shifting, the refrigerant was charged in the composition that was to be tested. The objective of this study was not to solve the practical problems of how composition shifts would be realized in a commercial heat pump. Rather, the study showed that zeotropic mixtures offer significant possibilities for optimizing the refrigerant properties for a given application.

The simplest approach to composition shifting is single-stage equilibrium distillation. It is a passive method which requires little or no hardware modifications to a basic heat pump system. Heat pumps equipped with a capillary tube or short tube restrictor will progressively flood more liquid into the suction accumulator as the outside temperature decreases (see background section for a detailed explanation of this). If the system is charged with a zeotropic mixture, the equilibrium concentration difference between the liquid and vapor will cause the accumulator to store the less volatile refrigerant in the liquid phase. Thus, the vapor in the accumulator will be rich in the more volatile component. Since it is predominantly the vapor that returns to the compressor, the circulating refrigerant will become enriched in the more volatile component and the capacity will be increased.

This method of composition shifting is attractive because the only additional hardware required is a larger suction accumulator. For heat pumps equipped with fixed area expansion devices, this method is also self-regulating, since the accumulator will nevertheless flood at low outdoor temperatures. However, the degree of composition shift attainable is limited by the equilibrium vapor-liquid composition difference and the percentage of the total refrigerant charge that can be stored in the accumulator. As the percentage of the total system charge that can be stored in the accumulator increases, the magnitude of the composition shift will also increase.

An early investigation of single stage-equilibrium distillation was conducted by Cooper and Borchardt [8]. Their experimental apparatus was a secondary refrigerant calorimeter coupled to a water-cooled condenser. For a refrigerant mixture with an initial composition of 60% R13B1 and 40% R152a by weight, the study claimed a composition shift to 82% R13B1. The system capacity and heat source temperature at this composition were not reported. In an additional series of tests, the investigators reported system capacities for three refrigerant compositions of

50, 60 and 70% R13B1 by weight (mixed with R152a). The authors reported a 60% increase in compressor capacity over this range of compositions. The composition shifts in these capacity tests were not brought about by flooding into the suction accumulator. Instead, the system was charged with the respective mixtures and then operated with a dry suction accumulator.

In a subsequent study [9], Cooper experimentally demonstrated single-stage equilibrium composition shifting in a commercially available air source split-system heat pump. The influence of heat source temperature on the circulating refrigerant composition was evaluated for a refrigerant mixture of 70% R13B1 and 30% R152a. For source temperatures of 8.3 °C (47 °F), -8.3 °C (17 °F) and -17.8 °C (0 °F), the percentage of R13B1 was 75.5, 81.1 and 82.9 respectively. In these tests, system capacities were not calculated. Instead, the air temperature rise across the indoor coil was measured. Compared to pure R22, the mixture showed a 50% increase in the indoor coil air temperature rise at the -17.8 °C (0 °F) source temperature.

In addition to publishing these experimental results, Cooper suggested several hardware modifications to facilitate composition shifting. For example, he recommended that the suction accumulator should be large enough to store one-third to one-half of the initial charge. It is generally true that a larger charge will allow a greater percentage of the refrigerant to be stored in the accumulator, provided the accumulator is large enough. Therefore, a larger charge is desirable in heating mode to provide maximum composition shifting. However, this amount of refrigerant may be more than the system can accommodate in the cooling mode. To address this problem, Cooper suggested adding a high side receiver to the system.

Another recommendation was made regarding the choice of the expansion device. At low outdoor temperatures, the system may flood the accumulator to such an extent that all subcooling in the condenser is lost. This means that two phase refrigerant will be entering the expansion device, which imposes an efficiency penalty on the system. The efficiency loss occurs because any refrigerant that is not condensed prior to entering the expansion device cannot produce any useful refrigerating effect. It does, however, absorb the work of compression. To alleviate this problem, Cooper recommended that two fixed area expansion devices should be used. One should be sized for the proper amount of subcooling in the high temperature heating mode, and the other should be sized to keep a minimum amount of subcooling at low outdoor temperatures. This idea was theoretically sound, but was impractical because of the additional valves required. As an alternate approach, Cooper suggested using an electronically controlled expansion valve.

Mulroy and Didion [10] also studied single-stage equilibrium distillation in a commercially available split-system air source

heat pump. The following four test series were conducted in this investigation: (1) heat pump unmodified, (2) charge and expansion device area optimized, (3) possible composition shifts evaluated, and (4) high side receiver added. Tests were first conducted with pure R22 in the unmodified heat pump to determine a baseline condition for comparison. After optimizing the expansion device area and charge size, a mixture of 65% R13B1 and 35% R152a by weight gave a 6% increase in heating capacity as compared to pure R22 at a source temperature of -8.3 °C (17 °F). The extent of composition shift for these tests ranged from the charged value of 65% to a maximum of 75.9% R13B1.

An additional series of tests was performed to evaluate if the composition shift could have been increased by adding a high side receiver. This modification resulted in a greater capacity enhancement relative to R22. At a -8.3 °C (17 °F) source temperature, the capacity was increased by 14% over that of R22. In these tests, the percentage of R13B1 ranged from the charged value of 65% to a maximum of 77.5%.

Gromoll and Gutbier [11] constructed a water source heat pump to investigate single-stage equilibrium distillation for mixtures of R22 and R114. Their system was equipped with a high side receiver and a suction accumulator. Additionally, the suction accumulator contained a submerged coil which was cooled by external means. Since a constant pressure regulator was used for an expansion device, the test data was reported with the evaporator pressure as a parameter. Capacity tests were conducted for mixtures with weight fractions of 50 and 20 percent R22. The heat source temperature ranged from 0 °C (32 °F) to 25 °C (77 °F), for evaporator pressures of one to 3.5 atmospheres. The heat sink temperature was kept constant at 40 °C (104 °F) for all tests. For a charged composition of 50% R22, at an evaporator pressure of three atmospheres, the composition shift held the capacity constant as the source temperature fell from 20 °C (68 °F) to 5 °C (41 °F). The authors recommended that the suction accumulator and the receiver each be capable of holding approximately four times the amount of refrigerant needed for the cycle.

When a heat pump affects a single-stage equilibrium distillation with a low side accumulator, the minimum capacity occurs when the low side accumulator is empty. In this condition, the system operates with the charged mixture composition. When the low side accumulator contains the maximum amount of liquid, the system will operate at its maximum capacity. The circulating refrigerant now contains the maximum possible amount of the more volatile component. In summation, the system is capable of modulating its capacity from a minimum at the charged composition (low side accumulator empty) to a maximum when the low side accumulator is full.

As a means of extending the capacity modulation range, Vakil

[12] proposed a system capable of storing refrigerant on both the high and low sides of the system. The high side storage was accomplished by adding a liquid-vapor separation vessel (similar to the one shown in Figure 5) immediately after the condenser. The separator vapor effluent, which is richer in the more volatile component, flows into a high side accumulator where it is stored. The separator liquid effluent, which is richer in the less volatile component, continues to circulate on its way to the expansion device. When the high side accumulator is full, the system operates at a minimum capacity with a circulating composition that is richer in the less volatile component than the original charge. The range of composition modulation was thereby extended over a system which employed only low side accumulation.

Vakil experimentally verified the increased range of composition shift of the two-accumulator system by comparing it with a low side accumulation system. For a zeotropic refrigerant mixture of 50% R22 and 50% R114 by mole, the two-accumulator system had a minimum capacity at a circulating composition of 17.6% R22 by mole. Both systems had a maximum capacity at a circulating composition of 82.2% R22 by mole. The results showed overall capacity modulation ratios of 1.5 for the low side accumulation system and 2.6 for the two-accumulator version. Vakil patented the two-accumulator system as well as several variations of the concept which are listed in the references [13,14,15,16].

Another method of extending the possible range of composition shift was proposed by Gerdsmeier and Kruse [17]. By utilizing a liquid-vapor separator between two condensers, they were able to store both refrigerant components on the high side of the system. The first condenser was sized so that it would be just large enough to completely desuperheat the refrigerant. The operation of this system will be described by referring to the state points in Figure 4. Storage of the less volatile component is accomplished by sending the liquid (of composition I) from the first condenser to an extraction receiver for storage. The vapor (of composition D) flows to the second condenser where it is liquified and recirculated. This process increases the composition of the more volatile component in the circulating refrigerant, which increases the capacity.

To store the more volatile component, all of the circulating refrigerant flows through both condensers followed by another liquid-vapor separator. The vapor from this separator (of composition F) is sent to the extraction receiver where it is liquified and stored. The liquid (of composition B) is recirculated through the cycle. This process decreases the composition of the more volatile component in the circulating refrigerant, which decreases the capacity.

Tests were conducted for a refrigerant mixture of R22 and R114 in a weight fraction of 50% R22. For a constant heat source

temperature, the ratio of the maximum capacity to the minimum capacity was 1.5. At the minimum capacity, the mass fraction of the more volatile R22 was decreased to 0.20 in the circulating refrigerant. At the maximum capacity, the mass fraction of R22 was increased to 0.60. The separation quality was poor for storage of the less volatile component because of the difficulty in stopping the condensation process at point D. If the condensation proceeds too far towards point C, the separated liquid will not be rich enough in the less volatile component to produce a significant shift.

Although composition shifting by single-stage equilibrium distillation is a simple method, it does require a large refrigerant charge to produce a significant composition shift. Because of this limitation, another method of controlling the mixture was suggested by Schwind [18]. This involved the use of a distillation column equipped with a heat source to generate vapor. The system was suggested for a mixture of R13B1/R12 but was not limited to that mixture. A heater was placed in the sump (generator) of the distillation column so that, when energized, the more volatile R13B1 vapor would be driven out of the sump and rise through the column. The gradual reflux with the R12 liquid would consequently cause the stored liquid to be nearly pure R12. In this manner the system would enrich the main cycle refrigerant composition with a greater percentage of R13B1. This type of system does not require an excessive amount of refrigerant and can achieve greater composition shifts than single-stage equilibrium distillation. However, the cost and complexity of such a system may limit its commercial use.

Heat pumps which utilize zeotropic mixture composition control also have the potential for applications other than space heating. For example, dual function heat pumps which combine space heating with domestic water heating are excellent candidates for refrigerant mixtures. Pure R22 dual function heat pumps are currently commercially available, but the water heating mode has some operational limitations. Specifically, precautions must be taken to compensate for the high pressure of R22 at the high condensing temperatures of the water heating application.

An experimental heat pump which utilized a composition shift to limit the discharge pressure during high condensing temperature operation was proposed by Nakatani et al. [19]. This system used a distillation column with a heated storage receiver to separate and store the less volatile refrigerant component. The unit was designed to store the less volatile component during periods of low condensing temperature operation to achieve maximum capacity. During periods of high condensing temperature operation, the unit operates on the mixture as charged, thereby limiting the discharge pressure. Although the capacity decreases as the circulating refrigerant composition is shifted towards the less volatile component, the range of safe operating conditions has been

increased. The study was conducted for binary refrigerant mixtures of R22 with R134a, R152a, R142b, R123 and R114. The study concluded that a mixture of 75% R22 and 25% R142b by weight provided the highest COP and the best separation characteristics.

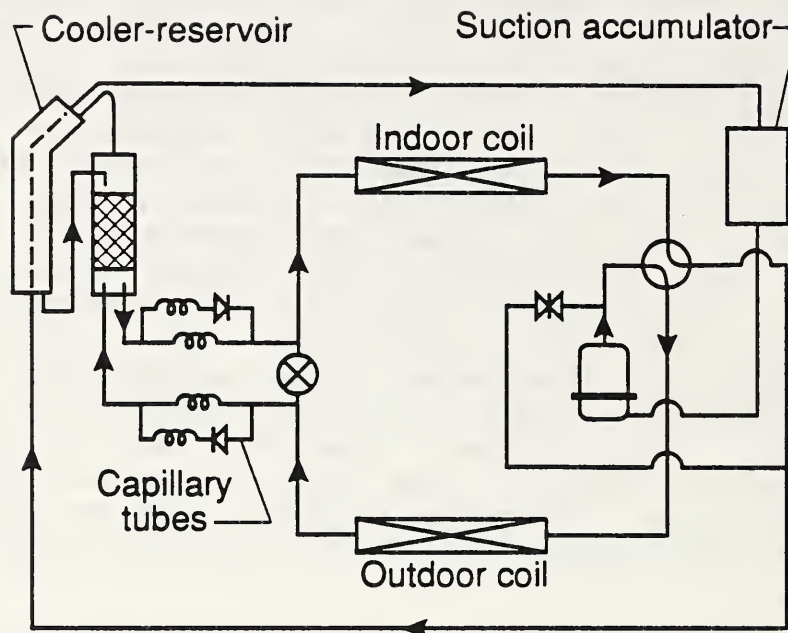
4. UNIT DESCRIPTION

The unit tested was a ductless split residential heat pump, with a variable speed compressor and a battery powered infrared remote controller and thermostat. The system operates with a zeotropic refrigerant mixture of 20% R13B1 and 80% R22 by weight. It is also equipped with a distillation circuit which is connected in parallel with the main circuit expansion valve. The distillation circuit enables the unit to shift the main cycle operating composition by separating and storing the more volatile R13B1 component. The unit does not operate on a reverse cycle defrost system like conventional heat pumps, but uses a hot gas bypass system. This method defrosts the outdoor coil without removing energy from the dwelling. An evaluation of this defrosting method was considered to be beyond the scope of this study. The interested reader can find a detailed explanation in reference [20]. The system also utilizes a microprocessor controlled expansion valve. The valve is driven by a permanent magnet stepping motor. Complete piping diagrams with refrigerant flow arrows for both the heating and cooling modes are shown in Figure 9 [21].

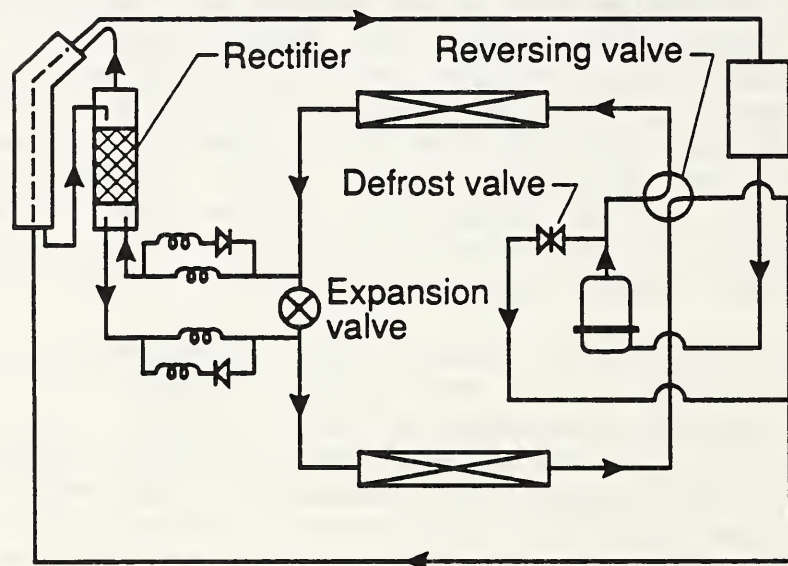
Referring to this figure, the rectification cycle is seen to consist of four capillary tubes, a distillation column and a cooler-reservoir. The distillation column is a 210 mm long section of 15 mm diameter copper pipe. The column is filled with a packing material which promotes the high degree of liquid-vapor contact necessary for efficient heat and mass transfer. The cooler-reservoir is constructed of two concentric copper pipes which have outside diameters of 12.7 mm (0.5 in.) and 25.4 mm (1 in.). The suction vapor passes through the inner tube while the annular space is connected in parallel with the rectification column. Additionally, two of the capillary tubes are fitted with check valves. The check valves are arranged so that refrigerant flows through the tubes which contain them only in the cooling mode.

For rectification to occur, the refrigerant must enter the rectification column as a two phase mixture [21]. Since the rectification column contains no heat source, the pressure drop across the inlet capillary tubes must be large enough to bring the refrigerant to its saturation pressure. The capillary tubes are sized so that flashing will only occur if the liquid subcooling is not too large. If the subcooling is too great, the pressure drop will not bring the refrigerant to its saturation pressure and subcooled liquid will enter the rectification column. This will render the rectification column inactive and stop the rectification process.

Since the amount of liquid subcooling is the predominant factor governing rectification, it is important to understand what determines how much subcooling will be present. For a single speed



(a)



(b)

Figure 9. Unit piping schematics (a) cooling mode, (b) heating mode

heat pump with a constant area expansion device, a change in the heat source temperature was shown to produce a refrigerant migration. Consequently, the amount of liquid in the condenser changes as does the amount of subcooling. The sequence of events is considerably more complex for a heat pump with a variable area expansion device and a variable speed compressor. The first step in analyzing this situation would be to determine the sensor inputs and logic algorithm the microprocessor uses to control the expansion valve. This information is of a proprietary nature and was only partially available. The problem is further complicated by the fact that the electronically controlled expansion valve is in parallel with two pairs of capillary tubes. In order to simply describe the theory of operation for each operating mode, the combination of the two expansion devices (i.e., capillary tubes and electronic expansion valve) shall be referred to as a single effective expansion device. With this conceptualization in mind, the theory of operation in the cooling and heating modes can be easily described for a given compressor speed.

In order to increase cycle efficiency in the cooling mode, the rectification cycle shifts the composition of the main cycle refrigerant by storing R13B1 in the cooler-reservoir. In this manner, the main cycle operates with nearly pure R22. The rectification cycle is activated by adjusting the main cycle expansion valve. When the expansion valve is completely (or nearly) closed a large portion of the refrigerant mass flow is forced through the capillary tubes into the rectification column. The effective expansion device resistance is now such that the system will have a minimum of subcooling, and the resulting pressure drop will cause the refrigerant entering the rectifier to flash. The resulting two phase mixture will contain vapor rich in the more volatile R13B1 component. Repeated vaporization and condensation causes this concentrating process to continue throughout the height of the column. The end result is for the R13B1 vapor to collect at the top of the column, while the nearly pure R22 liquid collects at the bottom of the column. The R13B1 vapor then moves over into the cooler-reservoir where it is condensed and stored, while the R22 liquid returns to the main cycle through the second pair of capillary tubes.

When operating in heating mode at low outdoor temperatures it becomes advantageous to match the building load by increasing the system capacity. This is accomplished by stopping the rectification process so that the main cycle operates with the mixture as charged. During this period of operation, the main cycle expansion valve is open normally and the check valves prevent the refrigerant from flowing through one capillary tube in each pair. In this condition, the flow resistance of the effective expansion device is such that the subcooling will be sufficient to prevent the refrigerant from flashing in the inlet capillary tube. As a result, the refrigerant entering the rectifier is completely liquid and the rectification process ceases. The rectification

circuit now behaves as if it were a liquid receiver, storing only a minimal amount, and the system operates on the charged composition.

When operating in heating mode at high outdoor temperatures, rectification is desirable to increase cycle efficiency (when high capacity is not required). Heating mode rectification is accomplished in a manner similar to cooling mode rectification. However, as the outside temperature falls, there will be a temperature range over which partial rectification occurs. The extent of rectification will be controlled by regulating the phase of refrigerant entering the rectifier with the electronic expansion valve.

This unit is also equipped with a frequency inverter to vary the compressor speed. The inverter is not infinitely variable, but is instead capable of varying the compressor speed in discrete intervals. The unit's front panel displays the operating compressor frequency in Hertz (Hz). The correlation between the displayed frequency and the actual revolutions per minute is not known. Therefore, a specific compressor speed will be referred to in terms of the displayed Hz. In cooling mode, there are four available speeds of 30, 50, 70, and 100 Hz. In heating mode, there are five available speeds of 30, 50, 70, 100, and 130 Hz.

5. DESCRIPTION OF TEST APPARATUS

To evaluate the capacities and power inputs necessary for performing seasonal energy calculations (i.e., HSPF and SEER), an experimental test rig was constructed in accordance with ASHRAE Standard 116-1983. To facilitate testing, the indoor and outdoor portions of the unit were each placed in separate adjoining environmental chambers. Both chambers are capable of exercising independent control over the air dry bulb temperature and humidity content. ASHRAE Standard 116-1983 requires that the primary method of determining system capacity be the indoor air enthalpy method. This standard also requires that the capacity from at least one steady state test be corroborated by an independent secondary method. For this study, the outdoor air enthalpy method was chosen as the secondary test method. The capacity determined by the secondary method must agree with the primary method to within six percent.

The indoor air enthalpy method consists of attaching an insulated duct to the unit air discharge so that the exiting air properties and flow rate can be measured. A complete drawing of the indoor test apparatus is shown in Figure 10. The first portion of the duct consists of a mixing section to insure that all air properties are uniform across the duct cross-section. Ref. [22] gives a thorough explanation of the fabrication and operating principles of the air mixers. After mixing, the air is passed over a thermocouple grid consisting of 17 copper-constantan thermocouples located in the center of approximately equal areas. The purpose of these thermocouples is to determine both the effectiveness of the mixing section and the average air temperature in the duct. ASHRAE Standard 116-1983 requires that the temperature profile in the duct shall be uniform within a tolerance of $T=T(\text{avg}) \pm 0.42 \text{ }^\circ\text{C}$ ($0.75 \text{ }^\circ\text{F}$). The sensible temperature change across the indoor unit coil was determined by a nine junction thermopile connected between the duct thermocouple grid and the unit air inlet.

After the thermocouple grid, the air is passed through a flow measurement apparatus. This consists of a receiving chamber and exhaust chamber, which are separated by a partition fitted with an elliptical nozzle. The air flow rate was determined by measuring the static pressure drop across the nozzle with an inclined manometer. The manometer has a least scale division of 0.254 mm (0.01 in.) of H_2O and a full scale of 31.75 mm (1.25 in.) of H_2O . A thermocouple was mounted in the nozzle throat to determine the air density.

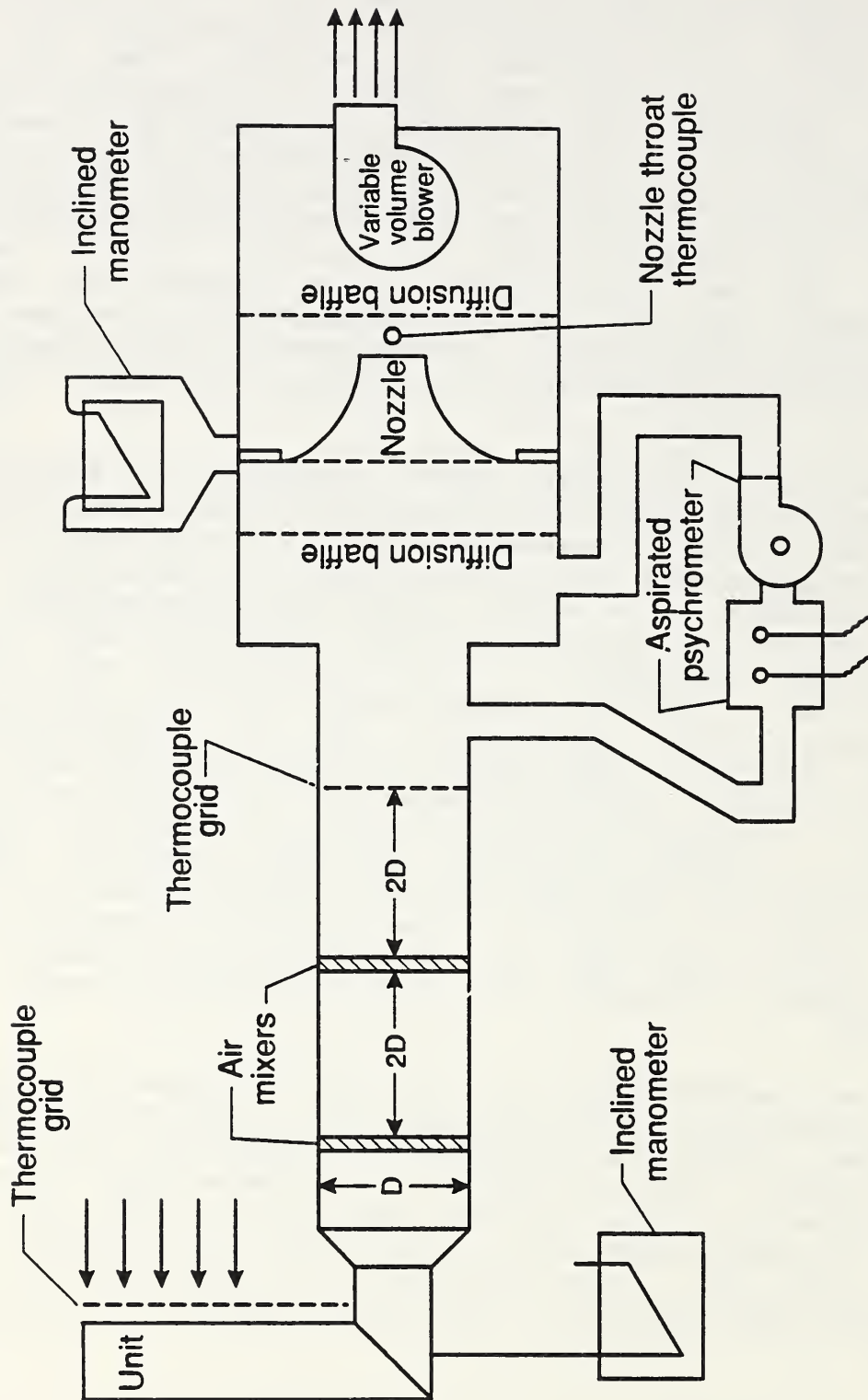


Figure 10. Test apparatus schematic

The unit being tested for this project contains its own indoor fan and was designed for use without ducts. In order to simulate this condition in the laboratory the static pressure difference between the duct and the surrounding atmosphere at the unit outlet must be zero. This is accomplished by adjusting a variable volume blower, mounted at the exhaust end of the tunnel, until the desired static pressure difference at the unit outlet is obtained. This static pressure difference was also measured with an inclined manometer having a least scale division of 0.254 mm (0.01 in.) of H₂O and a full scale of 31.75 mm (1.25 in.) of H₂O.

The temperature and humidity content of the air entering the unit was determined by a thermocouple grid attached to the unit inlet and an aspirated thermocouple psychrometer. The humidity content of the air after leaving the unit is determined by diverting a portion of the flow through another aspirated thermocouple psychrometer. This portion of the flow is then returned to the air flow measuring apparatus receiving section since it is part of the total volumetric flow rate to be measured.

ASHRAE Standard 116-1983 specifies that the change in humidity ratio across the indoor coil shall be the primary method of determining latent capacity. However, this method relies on the wet bulb temperature measurement to calculate the humidity ratio. Due to the sensitivity of this calculation to small errors in wet bulb measurement this method was used as a secondary check only (see error analysis in appendix). The primary method of latent capacity measurement used for this study was condensate collection. A 250 ml graduated cylinder was used to collect the condensate for successive ten minute intervals until the rate of collection no longer changed with time. The accuracy of the graduated cylinder was estimated to be ± 1 ml. Throughout the course of testing this method of measurement was demonstrated to have a much better degree of repeatability.

Since the unit tested was equipped with a frequency inverter for variable speed operation, a solid state watt meter utilizing a Hall effect watt transducer was used. This type of watt meter is especially well suited to power measurement when non-sinusoidal waveforms are involved. The accuracy of the watt meter was 1.5% of the full scale reading. Refrigerant pressures were measured in several locations with pressure transducers calibrated on a standard dead weight tester. These transducers were accurate to within 0.11% of full scale. All thermocouples were attached to a data acquisition system which utilizes an electronic thermistor reference junction compensation system. The thermocouples were calibrated by adjusting a potentiometer located on a reference junction compensation circuit until the thermocouple reading agreed to within ± 0.056 °C (± 0.1 °F) of a mercury thermometer with 0.12 °C (0.2 °F) divisions. The accuracy of all thermocouple readings was estimated to be ± 0.278 °C (± 0.5 °F).

A concentration sample port was also installed in the system discharge line so that the main cycle refrigerant composition could be analyzed. Samples were drawn from the discharge line to ensure that only vapor refrigerant enters the sample bottle. Refrigerant samples drawn from the system were analyzed with a gas chromatograph.

ASHRAE standard 116-1983 requires that an independent secondary method of capacity measurement be employed and that this secondary method shall agree with the primary method to within six percent. For this study the outdoor air enthalpy method was chosen. This method was employed by constructing an exact duplicate of the indoor air enthalpy measuring apparatus and attaching it to the outdoor portion of the unit in a manner similar to that shown for the indoor apparatus in Figure 10. When operating in cooling mode, the heat rejected to the outdoor air stream as measured by the outdoor test apparatus minus the compressor input energy (in consistent units) equals the indoor total cooling capacity. In heating mode the heat removed from the outdoor air stream plus the compressor input energy equals the indoor heating capacity. The outdoor air enthalpy secondary method was utilized for all tests except for heating at $-8.3\text{ }^{\circ}\text{C}$ ($17\text{ }^{\circ}\text{F}$) outdoor temperature. The method was abandoned for this test to avoid the difficulties of low temperature psychrometry.

6. TEST RESULTS

The test results for the heating and cooling modes are summarized in Tables 1 and 2, respectively. The DoE/NIST test procedure requires complete control over the unit's compressor speed. However, it was difficult to bypass the unit's electronic control system to obtain complete control over the compressor speed. Therefore, a method was devised to allow false room temperature signals to be sent to the unit controller. This was done by replacing the unit's room temperature sensor, a negative temperature coefficient thermistor, with a potentiometer. This enabled the compressor speed to be held constant at certain values that could be more easily sustained. The maximum, intermediate and minimum values that were used in heating and cooling modes are as follows.

In cooling mode, only three speeds, 30 Hz, 50 Hz and 70 Hz, could be held long enough to achieve steady state operation. A fourth cooling mode speed, 100 Hz, would run only at start-up and could not be held for more than approximately forty-five minutes. The 100 Hz speed was therefore neglected. Although there were five possible speeds in heating mode, only the maximum (130 Hz), minimum (30 Hz), and an intermediate speed of 70 Hz could be sustained for steady state operation. As a result, the defrost test had to be performed at the next highest available speed, rather than the speed required by the DoE/NIST procedure. The DoE/NIST procedure requires that the defrost test be conducted at the same speed as the intermediate speed cooling test (50 Hz). Since this speed could not be held steady, the defrost test was conducted at 70 Hz.

The impact of modifying the defrost test procedure is insignificant in terms of the results. This can be explained by the fact that the unit, when running at 70 Hz, did not develop any frost on the outdoor coil. Additionally, the electronic control system did not initiate a defrost period. All other conditions being equal, a slower compressor speed would result in a higher evaporator surface temperature. Thus, if the outdoor coil did not develop any frost at a speed of 70 Hz, it can be inferred that it would not have developed frost at the slower speed of 50 Hz.

Compliance with the DoE/NIST test procedure was further complicated by the unit's user controlled variable speed indoor blower. The DoE/NIST test procedure states that the air flow rate of the indoor and outdoor sections shall be governed by the system controls. However, it also states that the indoor side air quantity shall meet the requirements of section 5.1.3.3 of ARI Standard 210/240-89 [23]. The ARI Standard states that equipment with indoor fans not intended for use with field installed duct systems (free discharge) shall be rated at the indoor-side air quantity delivered when operating at 0 mm (0 in.) of water external static pressure as specified by the manufacturer. Although the

external static pressure requirement was easily satisfied, the indoor fan speed selection would have affected the test results.

The fan speed dilemma was alleviated by operating the system at its maximum indoor fan speed for all tests. The maximum fan speed was chosen because it was the closest to the 1.062 standard m^3/min (37.5 SCFM) per 293.1 W (1000 Btuh) of rated capacity that is specified in section 5.1.3.3 of ARI Standard 210/240-89. This practice resulted in a nearly constant indoor air flow rate for each mode of operation (see Tables 1 and 2). However, the maximum heating mode fan speed was higher than the cooling mode. It should also be noted in Table 1 that the 8.3 °C (47 °F) minimum compressor speed test has a considerably lower indoor air flow rate. This condition could not be avoided since the unit's control system was found to be capable of overriding the user selected fan speed. It is believed that the control system bases its decision to override the user selected fan speed on some minimum value of the discharge air temperature.

TABLE 1

HEATING TEST RESULTS

Outdoor Temp. °C (°F)	Test Type	Capacity W (Btu/h)	Power Input (W)	Comp. Speed	Air Flow m ³ /min (CFM)	Composition % R13B1 by weight	Refrig. Press.		COP
							kPa	psia	
16.7/13.6 (62/56.5)	SS	1821 (6200)	381	MIN	9.91 (350)	21.0	772 (112)	1496 (217)	4.78
8.3/6.1 (47/43)	SS	4009 (13650)	1366	MAX	10.34 (365)	21.0	545 (79)	2137 (310)	2.94
8.3/6.1 (47/43)	SS	2538 (8640)	710	INT	10.19 (360)	20.5	586 (85)	1758 (255)	3.58
8.3/6.1 (47/43)	SS	1989 (4390)	400	MIN	4.96 (175)	19.4	634 (92)	1606 (233)	3.22
1.7/-1.1 (35/30)	DEF	2138 (7280)	678	INT	10.05 (355)	---	496 (72)	1586 (230)	3.15
-8.3/-9.4 (17/15)	SS	3416 (11630)	1683	MAX	10.34 (365)	---	317 (46)	1924 (279)	2.03

NOTES

- Indoor conditions for all heating tests were 21.1°C (70 °F) dry bulb and a wet bulb no greater than 15.6 °C (60°F).
- SS = steady state test; DEF = defrost test.
- Composition of initial charge was 21.7% R13B1 and 78.3% R22 by weight.
- Unit has five possible compressor speeds in the heating mode: 30, 50, 70, 100, 130 Hz.
- Test compressor speeds were: MAX = 130 Hz, Int. = 70 Hz, Min. = 30 Hz.
- The specified temperatures are dry bulb/wet bulb except for the DEF test which is dry bulb/dew point.

TABLE 2

COOLING TEST RESULTS

Outdoor Temp. °C (°F)	Test Type	Capacity W (Btu/h)			Power Input (W)	Comp Speed	Air Flow m ³ /min (CFM)	%R13B1 by weight	Refrig. Press		COP (EER) $\frac{\text{Btu}}{\text{W}\cdot\text{h}}$
		Sens	Lat	Total					kPa (psia)	Suct. Dis.	
95 (35)	SS	6670 (1959)	2340 (687)	9010 (2646)	949	MAX	270 (7.65)	23.3	99 (683)	288 (1986)	2.781 (9.49)
87 (30.6)	SS	5580 (1639)	1420 (417)	7000 (2056)	562	INT	265 (7.50)	21.0	110 (758)	240 (1655)	3.652 (12.5)
82 (27.8)	SS	6740 (1980)	2520 (740)	9260 (2720)	821	MAX	270 (7.65)	22.1	92 (634)	245 (1689)	3.306 (11.3)
82 (27.8)	SS	4840 (1422)	210 (62)	5050 (1483)	360	MIN	265 (7.50)	20.9	122 (841)	216 (1489)	4.112 (14.0)
67 (19.4)	SS	5230 (1536)	590 (173)	5820 (1709)	302	MIN	265 (7.50)	---	117 (807)	179 (1234)	5.647 (19.3)

NOTES

- Indoor conditions for all cooling tests were 26 °C (80 °F) dry bulb and 19.4 °C (67 °F) wet bulb.
- SS = steady state test.
- Composition of initial charge was 21.7% R13B1 and 78.3% R22 by weight.
- Unit has three compressor speeds in cooling mode: MAX = 70 Hz, INT = 50 Hz, MIN = 30 Hz.

7. DISCUSSION OF RESULTS

The distillation column was expected to achieve the greatest degree of composition shifting that is practically possible in a residential heat pump. However, the unit did not demonstrate composition shifting in any of the tests (see Tables 1 and 2). To ensure that the system was fairly evaluated under typical field conditions, the unit was installed strictly according to the manufacturer's recommendations. To further ensure a fair evaluation, instrumentation was added to the unit in a manner that did not require any system hardware changes. Due to the compact design and small size of this system, only a minimum of instrumentation could be added without making piping changes to the unit.

The economy of instrumentation that was utilized to ensure a fair performance evaluation made it difficult to determine why no composition shifting occurred. However, a possible explanation may be that there was an excessive amount of liquid subcooling. It has already been established that rectification cannot occur unless the refrigerant enters the column as a two phase mixture (see unit description). Since the rectification column contains no heat source to partially vaporize the entering refrigerant, the inlet capillary tube pressure drop must be sufficient to cause flashing. The pressure drop required to cause flashing is directly proportional to the amount of liquid subcooling. Therefore, excessive subcooling could prevent the refrigerant from flashing in the capillary tubes and cause subcooled liquid to enter the column. This causes the rectification column to become flooded with liquid which makes rectification impossible.

Subcooling occurs naturally when a portion of the condenser is completely filled with liquid. The factors which determine how much subcooling is present are the amount of refrigerant charge, the operating conditions, the expansion valve setting and the compressor speed. The system was charged with refrigerant according to the manufacturer's specifications, which were obtained from the unit nameplate. The charge specification was 1040 grams of R22 and 260 grams of R13B1. The actual charge added to the unit was 1033 grams of R22 and 286 grams of R13B1. This was determined by weighing the refrigerant cylinders before and after charging on an electronic scale with an accuracy of ± 1 gram. The small discrepancy between the actual amount charged and the amounts listed on the nameplate was assumed to be negligible. Although it is possible that excessive subcooling could be caused by too much refrigerant, the unit was charged according to the manufacturer's guidelines. Therefore, it is unlikely that overcharging caused any excessive subcooling that may have existed.

Another possibility which could prevent rectification is the expansion valve setting. In addition to its effect on liquid

subcooling, it must also be remembered that the expansion valve is connected in parallel with the rectification column. Since the capillary tubes present a fixed resistance to the flow of refrigerant, the mass flow entering the rectification column is determined by the expansion valve setting. If the expansion valve resistance is low in comparison with the capillary tube resistance, most of the refrigerant will bypass the rectification circuit. A reduced mass flow in the capillary tubes leading to the rectification column would result in a lower pressure drop, which could prevent the refrigerant from flashing in the tubes.

Because the expansion valve was hermetically sealed and electronically driven, there was no way to control or even determine the valve position. Therefore, it could not be ascertained if the valve setting was preventing rectification. In an effort to resolve this problem, a series of questions regarding the expansion valve operation was sent to the manufacturer. It was learned that the electronic control system modulates the expansion valve to maintain the discharge temperature below 88 °C (190 °F). The discharge temperature is limited by causing saturated vapor to enter the compressor. This is accomplished by opening the expansion valve enough to flood the suction accumulator with liquid. It is assumed that this control strategy was chosen because of a concern for the adverse effect of high discharge temperatures on system reliability. The discharge temperatures, which are shown in Table 3, were well below 88 °C (190 °F) for most of the tests in which rectification should have occurred. Assuming that an increased expansion valve opening prevented rectification, it is unclear why the control system would not close down the expansion valve when the discharge temperature was low enough.

TABLE 3

DISCHARGE TEMPERATURE FOR ALL TESTS

Outdoor Temp. °C (°F)	Comp. Speed	Operating Mode	Composition % R13B1 by weight	Discharge Temperature °C (°F)
35 (95)	MAX	COOL	23.3	88.7 (191.7)
30.6 (87)	INT	COOL	21.0	73.1 (163.6)
27.8 (82)	MAX	COOL	22.1	82.2 (179.9)
27.8 (82)	MIN	COOL	20.9	52.6 (126.8)
19.4 (67)	MIN	COOL	----	49.9 (121.9)
16.7 (62)	MIN	HEAT	21.0	64.9 (148.9)
8.3 (47)	MAX	HEAT	21.0	90.8 (195.4)
8.3 (47)	INT	HEAT	20.5	77.6 (171.6)
8.3 (47)	MIN	HEAT	19.4	63.0 (145.4)
1.7 (35)	INT	HEAT	----	-----
-8.3 (17)	MAX	HEAT	----	86.4 (187.6)
35 (95)	MAX	COOL	PURE R22	67.7 (153.9)
27.8 (82)	MAX	COOL	PURE R22	66.5 (151.7)
-8.3 (17)	MAX	HEAT	PURE R22	89.1 (192.4)

NOTES

1. The discharge temperature was not recorded during the 1.7 °C (35 °F) defrost test. During this test, the data acquisition system was programmed to skip unnecessary channels to increase the scan rate.

Since the distillation column proved ineffective in controlling the refrigerant composition, the seasonal energy calculations (i.e., HSPF and SEER) performed with actual test data shed little light on the value of composition control. To circumvent this problem, a vapor compression cycle computer simulation program was used to simulate the system performance as if it would have shifted composition. This program, developed at the National Institute of Standards and Technology [24], is capable of predicting ideal cycle performance for both pure refrigerants and zeotropic mixtures.

The computer program was used to simulate an ideal vapor compression cycle with isentropic compression, exchanging heat with two isothermal heat reservoirs. The actual test data were for the charged composition of approximately twenty percent R13B1 by weight. Computer calculations were conducted for all the test heat source and sink temperatures at this refrigerant composition. Another set of computer calculations was then made at these same source and sink temperatures with pure R22 refrigerant as the cycle working fluid. The ratio of the ideal cycle pure R22 capacity per cubic foot of compressor displacement to that of the mixture was then calculated. This ratio was then multiplied by the capacity obtained from the actual test data to get the predicted R22 capacity at the same conditions. The same procedure, using the actual test data, was used to predict the power consumption of a cycle operating on pure R22.

This method of capacity prediction relies on the assumption that the ideal cycle can be used to predict relative changes in system performance. When operating between the same two isothermal heat reservoirs, the absolute value of the capacity and COP of a real cycle will be much lower than that of an ideal cycle. The percent change, however, will be approximately the same. In an effort to verify this assumption, three tests were performed with pure R22 as the system working fluid. The results of these tests are shown along with the computer predictions for the same operating conditions in Table 4.

Once the results of the capacity and power predictions are obtained, seasonal energy calculations may be performed to determine what the changes in HSPF and SEER would be had the composition shifting occurred. These calculations were conducted with computer programs written at NIST for pure refrigerant heat pumps. The programs had to be modified to handle the type of capacity curve that would be present in a system which modulates the refrigerant composition. Additional modifications to the programs were also required. For example, the unit has only three discrete cooling speeds, whereas the program was originally written for systems which have continuously variable speed control. The results of these calculations for the cooling mode are shown in Table 5.

TABLE 4

COMPARISON OF COMPUTER PREDICTIONS WITH ACTUAL
TEST DATA FOR PURE R22

Outdoor Temp. °C (°F)	Operating Mode	Computer Predictions		Pure R22 Test Results		% diff in capacity	% diff in power
		Capacity W (Btu/h)	Power W	Capacity W (Btu/h)	Power W		
35 (95)	COOL	2514 (8578)	885	2351 (8021)	861	6.9	2.8
27.8 (82)	COOL	2569 (8767)	762	2610 (8904)	751	-1.5	1.4
-9.3 (17)	HEAT	3197 (10909)	1553	3265 (11141)	1597	-2.1	-2.8

NOTES

1. % difference = [(computer prediction - test results)/test results] * 100

TABLE 5

RESULTS OF SEER CALCULATION FOR THREE SPEED AIR-CONDITIONER
FOR TWO REFRIGERANT COMPOSITIONS

	Seasonal COP	SEER (Btu/W*h)
Actual test data with system operating on mixture as charged	3.871	13.21
Pure R22 prediction	3.948	13.47
% difference from mixture % diff. = [(mix -R22)/mix]*100	1.97	1.97

NOTES

1. The dimensionless seasonal COP is used here to represent the SI equivalent of SEER.

In the heating mode, similar calculations were conducted to determine the effect of composition shifting on the HSPF and the amount of auxiliary heat required. These results are summarized in Table 6. The program was not modified for the fact that in heating mode the unit has only five discrete speeds. It was found that the difference between five speed operation and infinitely variable speed operation was negligible. From Table 6, it is evident that the increase in HSPF with composition shifting is insignificant. However, the percentage amount of auxiliary heat saved is considerable. This savings is valuable as a reduction of the peak demand on electric utilities.

An additional series of calculations were performed to determine the effect of composition control on HSPF and the amount of auxiliary heat required for a single speed heat pump. These results are summarized in Table 7. The actual test data for maximum speed operation was used as the basis for extrapolating to other refrigerant compositions. For the purpose of the HSPF calculation it was assumed that the system capacity line would be like that shown in Figure 3. In both Tables 6 and 7 it should be noted that the column with 20% R13B1 is derived from actual test data, whereas the columns for other compositions are derived from computer predictions.

TABLE 6

VARIATION OF HSPF AND AMOUNT OF AUXILIARY HEAT REQUIRED
FOR A VARIABLE SPEED HEAT PUMP EMPLOYING COMPOSITION
CONTROL IN VARIOUS CLIMATE REGIONS

		Pure R22	20% R13B1 80% R22 by weight	30% R13B1 70% R22 by weight
Region III	W*h/W*h HSPF (Btu/W*h)	3.07 (10.48)	3.074 (10.49)	3.075 (10.49)
	% diff. in HSPF from pure R22	---	0.13	0.15
	Aux. heat W*h	3.0	1.5	1.2
	% diff. in RH. from pure R22	---	50	60
Region IV	W*h/W*h HSPF (Btu/W*h)	2.734 (9.328)	2.750 (9.382)	2.753 (9.392)
	% diff. in HSPF from pure R22	---	0.58	0.69
	Aux. heat W*h	22.6	13.4	10.8
	% diff. in RH. from pure R22	---	40.7	52.2
Region V	W*h/W*h HSPF (Btu/W*h)	2.340 (7.986)	2.361 (8.056)	2.365 (8.071)
	% diff. in HSPF from pure R22	---	0.88	1.06
	Aux. heat W*h	64.2	44.4	36.6
	% diff. in RH. from pure R22	---	30.8	43.0

NOTES

1. All calculations were made for the maximum DHR as calculated from the -8.3 °C (47 °F) nominal speed test. See reference [2].
2. Column for 20% R13B1 is derived from actual test data. Other columns are derived from computer projections.

TABLE 7

VARIATION OF HSPF AND AMOUNT OF AUXILIARY HEAT REQUIRED
FOR A SINGLE SPEED HEAT PUMP EMPLOYING COMPOSITION
CONTROL IN VARIOUS CLIMATE REGIONS

		Pure R22	20% R13B1 80% R22 by weight	30%R13B1 70% R22 by weight
Region III	W*h/W*h HSPF (Btu/W*h)	2.238 (7.637)	2.258 (7.705)	2.268 (7.738)
	% diff. in HSPF from pure R22	---	0.89	1.32
	Aux. heat W*h	78.0	59.5	50.3
	% diff. in RH. from pure R22	---	23.7	35.5
Region IV	W*h/W*h HSPF (Btu/W*h)	1.98 (6.755)	2.011 (6.863)	2.027 (6.916)
	% diff. in HSPF from pure R22	---	1.60	2.38
	Aux. heat W*h	273.2	225.2	200.1
	% diff. in RH. from pure R22	---	17.6	26.8
Region V	W*h/W*h HSPF (Btu/W*h)	1.785 (6.091)	1.814 (6.190)	1.827 (6.233)
	% diff. in HSPF from pure R22	---	1.63	2.33
	Aux. heat W*h	432.9	368.6	336.3
	% diff. in RH. from pure R22	---	14.9	22.3

NOTES

1. All calculations were made for the maximum DHR as as calculated from the -8.3 °C (47 °F) nominal speed test. See reference [2].
2. Column for 20% R13B1 is derived from actual test data. Other columns are derived from computer projections.

8. RECOMMENDATIONS FOR FUTURE WORK

Although this system failed to produce any composition shifting, the concept of rectification is still valid from the standpoint of auxiliary energy savings. The logical next phase of this project would be to remove all of the electronic controls from the system and replace the expansion valve with one that is manually operated. Modifications should also include the addition of sufficient diagnostic instrumentation to the rectification circuit. A provision should also be made to incorporate a controllable heat source to the rectification column entrance. These modifications should allow the complete determination of all the factors which govern rectification. These factors would include but not be limited to the compressor speed, expansion valve setting, refrigerant charge, rectifier heat input and the operating conditions.

When the problems with rectification have been resolved, an experimental comparison should be conducted between variable speed operation and zeotropic mixture composition shifting as a means of capacity control. The results of such a study may be recommendations as to whether composition control or variable speed or a combination of the two is best for a given application. Such a recommendation would also have to consider the effects of system first cost, complexity and maintenance.

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APPENDIX A

ERROR ANALYSIS OF STEADY STATE HEATING TEST

If we let the result R be a function of n independent variables V_1, V_2, \dots, V_n and each of the independently measured variables has an uncertainty $\Delta V_1, \Delta V_2, \dots, \Delta V_n$, then the uncertainty in the result ΔR is given by:

$$\Delta R = \left[\left(\frac{\partial R}{\partial V_1} \Delta V_1 \right)^2 + \left(\frac{\partial R}{\partial V_2} \Delta V_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial V_n} \Delta V_n \right)^2 \right]^{1/2} \quad (1)$$

This equation can be nondimensionalized to express the uncertainty on a percentage basis by dividing through by R .

$$\frac{\Delta R}{R} = \left[\left(\frac{\partial R}{\partial V_1} \frac{\Delta V_1}{R} \right)^2 + \left(\frac{\partial R}{\partial V_2} \frac{\Delta V_2}{R} \right)^2 + \dots + \left(\frac{\partial R}{\partial V_n} \frac{\Delta V_n}{R} \right)^2 \right]^{1/2} \quad (2)$$

From ASHRAE Standard 116-1983, the equation for the steady state heating capacity Q_H (Btu/h) is:

$$Q_H = 60 \cdot 1096 \cdot C \cdot A (0.24 + 0.444 W) (T_2 - T_1) \left[\frac{P_S P_B}{RT_n (1 + W)} \right]^{1/2} \quad (3)$$

where:

C = nozzle discharge coefficient

A = nozzle throat area (ft²)

W = humidity ratio (lb H₂O/lb da)

$T_2 - T_1$ = indoor coil air temperature rise (°F)

P_S = static pressure drop across nozzle (in. H₂O)

P_B = nozzle throat absolute barometric pressure (in. Hg)

R = ideal gas constant for air (in. Hg - ft³/lbm - °R)

T_n = nozzle throat air temperature ($^{\circ}$ R)

Calculate the required partial derivatives:

$$\frac{\partial Q_H / \partial A}{Q_H} = \frac{1}{A}$$

$$\frac{\partial Q_H / \partial W}{Q_H} = \frac{0.111 W + 0.162}{0.222 W^2 + 0.342 W + 0.12}$$

$$\frac{\partial Q_H / \partial (T_2 - T_1)}{Q_H} = \frac{1}{T_2 - T_1}$$

$$\frac{\partial Q_H / \partial P_s}{Q_H} = \frac{1}{2P_s}$$

$$\frac{\partial Q_H / \partial P_b}{Q_H} = \frac{1}{2P_b}$$

$$\frac{\partial Q_H / \partial T_n}{Q_H} = \frac{1}{2 T_n}$$

Rewriting equation (2) for $\Delta Q_H / Q_H$ gives:

$$\begin{aligned} \frac{\Delta Q_H}{Q_H} = & \left[\left(\frac{\partial Q_H / \partial A}{Q_H} \Delta A \right)^2 + \left(\frac{\partial Q_H / \partial W}{Q_H} \Delta W \right)^2 \right. \\ & + \left. \left(\frac{\partial Q_H / \partial (T_2 - T_1)}{Q_H} \Delta (T_2 - T_1) \right)^2 + \left(\frac{\partial Q_H / \partial P_s}{Q_H} \Delta P_s \right)^2 \right. \\ & \left. + \left(\frac{\partial Q_H / \partial P_b}{Q_H} \Delta P_b \right)^2 + \left(\frac{\partial Q_H / \partial T_n}{Q_H} \Delta T_n \right)^2 \right]^{1/2} \end{aligned} \quad (4)$$

Equation (4) can be evaluated to give the uncertainty in Q_H as a percentage of the experimental value for Q_H . Using the data from the 47 $^{\circ}$ F maximum speed heating test, eq. (4) can be

evaluated as follows:

The necessary measured values and constants are:

$$A = 0.0873 \text{ ft}^2$$

$$W = 0.0087 \text{ lb H}_2\text{O/lb da}$$

$$T_2 - T_1 = 29.9^\circ\text{F (measured by thermopile)}$$

$$P_s = 1.045 \text{ in. H}_2\text{O}$$

$$P_B = 29.82 \text{ in. Hg}$$

$$T_n = 564^\circ\text{R}$$

$$C = 0.99$$

$$R = 0.75434 \text{ (in. Hg - ft}^3\text{)/(lbm - }^\circ\text{R)}$$

The estimated uncertainties are:

$$\Delta A = \pm 0.00013 \text{ ft}^2$$

$$\Delta W = \pm 0.00018 \text{ lb H}_2\text{O/lb da}$$

$$\Delta(T_2 - T_1) = \pm 0.5^\circ\text{F}$$

$$\Delta P_s = \pm 0.01 \text{ in. H}_2\text{O}$$

$$\Delta P_B = \pm 0.01 \text{ in. Hg}$$

$$\Delta T_n = \pm 0.5 \text{ }^\circ\text{R}$$

The estimated uncertainty of a temperature difference would normally be greater than that of a single measurement. For thermocouples, the uncertainty of the difference is larger because there are two voltage measurements required. If a thermopile is used instead, then only one voltage measurement is required. Therefore, the error of a thermopile is not increased over that of a single absolute temperature measurement.

Evaluating the partial derivatives and substituting into eq. (4) yields:

$$\frac{\Delta Q_H}{Q_H} = [(0.00149)^2 + (0.00024)^2 + (0.01672)^2 + (0.00478)^2 + (0.00017)^2 + (0.00044)^2]^{1/2}$$

$$\frac{\Delta Q_H}{Q_H} = 0.0175 = 1.75\%$$

APPENDIX B

ERROR ANALYSIS OF STEADY STATE COOLING TEST

From ASHRAE Standard 116-1983, the equation for steady state cooling capacity, Q_C (Btu/h) is:

$$Q_C = Q_S + Q_L \quad (5)$$

where:

$Q_S =$ sensible cooling capacity (Btu/h)

$Q_L =$ latent cooling capacity (Btu/h)

The uncertainty in Q_C is given by eq. (1) as:

$$\Delta Q_C = \left[\left(\frac{\partial Q_C}{\partial Q_S} \Delta Q_S \right)^2 + \left(\frac{\partial Q_C}{\partial Q_L} \Delta Q_L \right)^2 \right]^{1/2} \quad (6)$$

since $\partial Q_C / \partial Q_S = \partial Q_C / \partial Q_L = 1$

$$\Delta Q_C = [(\Delta Q_S)^2 + (\Delta Q_L)^2]^{1/2} \quad (7)$$

From ASHRAE Standard 116-1983, Q_S and Q_L are given by:

$$Q_S = M \cdot C \cdot A [0.24 + 0.444 W] (T_2 - T_1) \left[\frac{P_S P_B}{R T_n (1 + W)} \right]^{1/2} \quad (8)$$

where:

$$M = 60 \cdot 1096$$

This is the same as the steady state heating capacity given by eq. (3). Therefore, the uncertainty in Q_S is given by eq. (4).

$$\frac{\Delta Q_S}{Q_S} = \frac{\Delta Q_H}{Q_H} = 0.0175 \quad (9)$$

$$Q_L = 63600 \cdot 1096 \cdot C \cdot A \left[\frac{P_S P_B}{R T_n (1 + W_n)} \right]^{1/2} (W_R - W_n) \quad (10)$$

where:

W_R = room humidity ratio

W_n = nozzle humidity ratio

Eq. (10) can be simplified as follows:

$$K = 63600 \cdot 1096 \cdot C$$

Eq. (10) becomes

$$Q_L = K \cdot A \left[\frac{P_S P_B}{R T_n (1 + W_n)} \right]^{1/2} (W_R - W_n) \quad (11)$$

The required partial derivatives are:

$$\frac{\partial Q_L}{\partial A} = K \left[\frac{P_S P_B}{R T_n (1 + W_n)} \right]^{1/2} (W_R - W_n)$$

$$\frac{\partial Q_L}{\partial P_S} = \frac{K \cdot A}{2} \left[\frac{P_B}{P_S R T_n (1 + W_n)} \right]^{1/2} (W_R - W_n)$$

$$\frac{\partial Q_L}{\partial T_n} = \frac{-K \cdot A}{2 T_n^{3/2}} \left[\frac{P_S P_B}{R (1 + W_n)} \right]^{1/2} (W_R - W_n)$$

$$\frac{\partial Q_L}{\partial P_B} = \frac{K \cdot A}{2} \left[\frac{P_S}{P_B R T_n (1 + W_n)} \right]^{1/2} (W_R - W_n)$$

$$\frac{\partial Q_L}{\partial W_R} = K \cdot A \left[\frac{P_S P_B}{R T_n (1 + W_n)} \right]^{1/2}$$

$$\frac{\partial Q_L}{\partial W_n} = -K \cdot A \left[\frac{P_S P_B}{R T_n} \right]^{1/2} \left[\frac{2 + W_R + W_n}{2 (1 + W_n)^{3/2}} \right]$$

Rewriting eq. (1) for ΔQ_L gives

$$\Delta Q_L = \left[\left(\frac{\partial Q_L}{\partial A} \Delta A \right)^2 + \left(\frac{\partial Q_L}{\partial P_S} \Delta P_S \right)^2 + \left(\frac{\partial Q_L}{\partial P_B} \Delta P_B \right)^2 + \left(\frac{\partial Q_L}{\partial T_n} \Delta T_n \right)^2 + \left[\left(\frac{\partial Q_L}{\partial W_R} \Delta W_R \right)^2 + \left(\frac{\partial Q_L}{\partial W_n} \Delta W_n \right)^2 \right]^{1/2} \quad (12)$$

The necessary measured values and constants from the 95°F maximum speed steady state cooling test are:

$$\begin{aligned} A &= 0.0873 \text{ ft}^2 \\ P_S &= 0.6 \text{ in. H}_2\text{O} \\ P_B &= 29.7 \text{ in. Hg} \\ T_n &= 519.3 \text{ }^\circ\text{R} \\ W_R &= 0.01118 \text{ lb H}_2\text{O/lb da} \\ W_n &= 0.00945 \text{ lb H}_2\text{O/lb da} \\ C &= 0.99 \\ R &= 0.75434 \text{ (in. Hg - ft}^3\text{)/(lbm - }^\circ\text{R)} \\ Q_S &= 6670 \text{ Btu/h} \\ Q_L &= 2340 \text{ Btu/h} \end{aligned}$$

The estimated uncertainties are:

$$\begin{aligned} \Delta A &= \pm 0.00013 \text{ ft}^2 \\ \Delta P_S &= \pm 0.01 \text{ in. H}_2\text{O} \\ \Delta P_B &= \pm 0.01 \text{ in. Hg} \\ \Delta T_n &= \pm 0.5 \text{ }^\circ\text{F} \\ \Delta W_n &= \Delta W_R = \pm 0.00018 \text{ lb H}_2\text{O/lb da} \end{aligned}$$

Evaluating the partial derivatives and substituting into eq. (12) yields:

$$\Delta Q_L = [(3.29466)^2 + (18.43744)^2 + (0.37247)^2 + (11.06513)^2 + (230.2016)^2 + (-230.39885)^2]^{1/2}$$

$$\Delta Q_L = 326.234 \text{ Btu/h}$$

Expressing the uncertainty as a percentage of Q_L gives

$$\frac{\Delta Q_L}{Q_L} = \frac{326.234}{2340} = 0.1394 = 13.94\%$$

From eq. (9)

$$\Delta Q_S = 0.0175 (6670 \text{ Btu/h}) = 116.725 \text{ Btu/h}$$

From eq. (7)

$$\Delta Q_C = [(116.725)^2 + (326.234)^2]^{1/2} = 346.5 \text{ Btu/hr}$$

Expressing the uncertainty as a percentage of Q_C gives

$$\frac{\Delta Q_C}{Q_C} = \frac{346.5}{6670 + 2340} = 0.0385 = 3.85\%$$

UNCERTAINTY OF LATENT CAPACITY AS MEASURED BY CONDENSATE COLLECTION

$$Q_L = 0.00212 \cdot h_{fg} \cdot \rho \cdot V$$

where

$$\begin{aligned} h_{fg} &= \text{latent heat of vaporization for saturated H}_2\text{O at } 40^\circ\text{F} \\ &= 1070.9 \text{ Btu/lbm} \end{aligned}$$

$$\rho = \text{density of saturated liquid H}_2\text{O at } 40^\circ\text{F} = 62.42 \text{ lbm/ft}^3$$

$$V = \text{volume rate of condensate collection (ml/min)}$$

$$\Delta Q_L = \left[\left(\frac{\partial Q_L}{\partial V} \Delta V \right)^2 \right]^{1/2} = \frac{\partial Q_L}{\partial V} \Delta V \quad (13)$$

The estimated uncertainty in $V = \pm 1 \text{ ml}$

Evaluating eq. (13)

$$\Delta Q_L = 141.67 \text{ Btu/h}$$

Expressing the uncertainty as a percentage of Q_L yields

$$\frac{\Delta Q_L}{Q_L} = \frac{141.67}{2340} = 0.0605 = 6.05\%$$

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