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HEAT PUMP DESIGN, ANALYSIS, AND APPLICATION

edited by
K. E. HEROLD
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FOREWORD

This volume consists of 12 papers which span a wide range of interests related to heat pump technology. The papers were submitted for the symposium on Heat Pump Design, Analysis and Application of the 1991 ASME Winter Annual Meeting. The symposium is sponsored by the Heat Pump Technical Committee of the Advanced Energy Systems Division of ASME.

The Heat Pump Technical Committee (HPTC) was founded in 1987 by Dr. Fang C. Chen who acted as the first Chairman. The HPTC is currently led by Dr. Keith E. Herold of the University of Maryland, Chair, Dr. Karen den Braven of the University of Idaho, Vice-Chair, Dr. Vince Mei of Oak Ridge National Laboratory, Paper Review Coordinator and Dr. Lin-Tao Lu of Rheem Manufacturing Co., Secretary. The HPTC has grown considerably during 1990–1991 and we anticipate continued growth because of renewed interest due to many factors. We invite participation from anyone interested in heat pump technology. We are particularly interested in attracting engineers from industry. Anyone interested in heat pump technology is welcome to attend our meetings (yearly meetings at ASME WAM) and participate in our activities (main activities are sponsoring technical sessions and bestowing best paper awards).

The Heat Pump Technical Committee exists to facilitate development of widely diverse heat pump technologies. These include vapor compression, absorption, adsorption, magnetic and many other technologies. Interest in alternative heat pump technologies has been growing in recent years with increased environmental concerns. The phase out of CFC's has caused a major disruption in the vapor compression heat pump industry. Concern about the greenhouse effect has encouraged renewed interest in energy efficiency. Both the CFC issue and the greenhouse effect concern have led to consideration of heat pump alternatives.

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LIST OF REVIEWERS

The following individuals donated their time and talents as reviewers for the papers in this symposium volume. The quality of the papers is a reflection of the efforts made by the reviewers and the session organizers would like to express our appreciation to each reviewer.

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TRANSIENT DEHUMIDIFICATION CHARACTERISTICS OF A HEAT PUMP IN COOLING MODE

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ABSTRACT

Much of the seasonal cooling operation of the heat pump occurs at part-load conditions when the unit cycles on and off to meet the cooling load. The seasonal efficiency under part-load conditions of the heat pump is typically estimated from a laboratory measurement of the degradation coefficient (C_D). Manufacturers are only required to estimate C_D at a single test condition where the indoor coil performs sensible cooling only. The effects of transient dehumidification losses are not accounted in estimating the seasonal efficiency. In hot and humid climates, dehumidification performance of a heat pump is as important as the sensible cooling performance. Therefore, a series of tests were designed to quantify the part-load dehumidification characteristics of a 3-ton residential air-to-air heat pump.

The tests include: cycling rates from 0.8 to 10 cycles per hour (cph), percent on-times of 20, 50, and 80 %, indoor dry-bulb temperature between 22.2 °C and 26.7 °C, and indoor relative humidity between 20 to 67%. The outdoor conditions and the indoor air flow rate were constant for all test runs. All experiments were performed in psychrometric chambers under controlled conditions.

The dehumidification process started between 60 to 150 seconds after start-up depending on the test conditions. During start-up, the losses in the latent capacity were greater than the losses in the sensible capacity. The dehumidification response increased with indoor dry-bulb temperature at constant relative humidity and decreased with indoor temperature at constant dew-point temperature.

INTRODUCTION

In the cooling mode the heat pump absorbs heat from the air inside a residence and rejects it to the outside air. In the process it performs two functions: sensible cooling and dehumidification. In hot and humid climates the dehumidification capabilities of the heat pump are important in achieving and to maintaining comfort in the conditioned space [Katipamula et al., 1987, 1988].

Upon compressor start-up, the cooling capacity of a heat pump increases to steady-state over several minutes. The slow response leads to average capacities and efficiencies which are lower than the steady-state values. Since heat pumps operate at a part-load condition for long periods of time, understanding their transient dehumidification response is essential to quantify both comfort and efficiency.

There are number of variables which affect the transient performance of a heat pump: (i) percent on-time, (ii) the thermostat cycling rate, (iii) the indoor dry-bulb temperature, (iv) the outdoor dry-bulb temperature and (v) indoor relative humidity. In this study the affects of percent on-time, cycling, indoor dry-bulb and humidity on the transient dehumidification performance of a heat pump are quantified.

LITERATURE REVIEW

A number of researchers have recognized the significance of the transient losses in capacity and efficiency at start-up (Kelly and Bean 1977; Parken et al., 1977; Bullock and Reedy, 1978; Murphy and Goldschmidt, 1979; Hart and Goldschmidt, 1980; Baxter and Moyers 1985; Miller, 1985, O'Neal and Katipamula, 1991, Katipamula and O'Neal 1991).

Kelly and Bean (1977), tested a 5-ton heat pump installed in a residence in Washington D.C. In cooling mode, the degradation due to part-load operation on performance reported to as high as 18%. The degradation of the latent capacity due to part-load operation was not reported.

Murphy and Goldschmidt (1979), noted that the losses in performance due to transient effects is most likely attributed to refrigerant dynamics rather than the thermal mass of the heat exchanger coils. They concluded that the degradation coefficient is not unique but depends on the transient response under a fixed set of test conditions. Hart and Goldschmidt (1980), suggested that the transient response is not a simple function of the thermal mass of the heat exchanger coils and that the degradation of performance was different for the heating and cooling modes of a heat pump.

Numerous researchers have attempted to study the transient response of heat pumps in a controlled environment: Parken et al. (1977), Murphy and Goldschmidt (1984), Bullock and Reedy (1978), Miller (1985), Katipamula and O'Neal (11).

Parken et al. (1977) studied the transient performance of a 10.6 kW (3-ton) heat pump in a laboratory for both heating and cooling at different cycling rates and loads. At 20% on-time (i.e. the on-time to total cycle time ratio), the decrement of the part-load cooling capacity from the steady-state cooling capacity changed from 10% at a cycling rate of 0.8 cph to 35% at 4 cph. For an on-time of 80%, the decrement in cooling capacity ranged from 4% at 0.8 cph to 8% at 4 cph. The compressor power was relatively unaffected by cycling rate.

Parken et al. (1985) evaluated the cooling performance of three heat pumps in the field. The results indicated that the cooling load and part load factor (PLF) were independent of the outdoor temperature and dependent on the cycling rate and the percent on-time. The study also found that

$$\frac{\dot{Q}^l_{ss}}{\dot{Q}^s_{ss}} \neq \frac{\dot{Q}^l_{cyc}}{\dot{Q}^s_{cyc}} \qquad (1)$$

where, the quantities $\dot{Q}^l_{ss}$ and $\dot{Q}^s_{ss}$ denote latent and sensible steady-state capacities, respectively and $\dot{Q}^l_{cyc}$ and $\dot{Q}^s_{cyc}$ the latent and sensible cooling done over a complete cycle. This shows that the cyclic latent capacity and cyclic steady state capacities reach steady-state at different rates. However, the authors could not quantify the latent dehumidification performance in the field because they were only measuring the drop in the dry-bulb temperature across the indoor coil.

Previous study by the authors (Katipamula and O'Neal 1991) indicated that the PLF can be as low as 0.65 at low percent on-times and high cycling rates. It took between 6 to 15 minutes for the system to reach steady-state depending on the operating conditions. The study also suggested that the relationship between PLF and cooling load factor (CLF) is non-linear. Since the relationship was non-linear the C_D at 50 percent on-time was lower than at 80 or 20 percent on-times (O'Neal and Katipamula 1991; and Katipamula and O'Neal 1991).

A number of conclusions can be drawn from the literature reviewed:

(1) The losses due to transient effects can be as much as 20%.
(2) It takes 6–15 minutes to achieve steady-state after start-up.
(3) The transient response is affected by the number of on-off cycles and percent on-time during each cycle.
(4) The mass of the heat exchangers affects transient losses.
(5) The off-cycle migration of the refrigerant from the condenser to the evaporator causes significant losses in capacity.
(6) The relationship between CLF and PLF is non–linear.
(7) The compressor power was relatively unaffected due to part load operation.
(8) The transient performance is independent of the outdoor temperature.

Although a number of researchers have recognized the importance of transient losses in heat pump operation, mostly it has been in the heating mode. In cooling mode, the research was focused to quantifying the effects of heat exchanger mass, the off-cycle phenomena on the transient sensible capacity, and effects of percent on-time and cycling rate on the cooling performance. One of the major issue that has not been addressed so far is transient dehumidification performance. Thus there appears to be a need to better characterize the transient dehumidification performance of heat pumps.

EXPERIMENTAL FACILITY

To characterize the transient dehumidification response of heat pumps over a range of operating conditions, a series of experiments with a heat pump were devised. All the tests were performed in the psychrometric test facilities constructed and maintained according to the American Society of Heating and Air-Conditioning Engineers (ASHRAE) specifications (1983). The rooms can simulate both indoor and outdoor conditions required for testing the performance of the split system heat pumps.

A nominal 10.6 kW (3 ton) capacity split-system residential air-to-air heat pump was used for testing. The details of the indoor and outdoor heat exchangers are tabulated in Table 1. The test heat pump had a 10.6 kW (3 ton) capacity reciprocating compressor. The schematic of the indoor and the outdoor coils is shown in Figure 1.

Temperatures and pressures were recorded at many locations in the test loop (Figure 1). In addition, the refrigerant flow rate, the air flow rate, the temperature drop and the change in moisture content of the supply air across the indoor heat exchanger were recorded. Other points where temperature was measured on the test loop include: surface temperature of the accumulator and compressor and surface temperature of the liquid and vapor lines going to each of the four circuits of the outdoor heat exchanger.

Table 1 - Details of Outdoor and Indoor Heat Exchange Coils.

	Outdoor Coil	Indoor Coil
Shape	Horse Shoe Vertical	Vertical
Number of Rows	Two Rows, Four Circuit	Four Rows, Four Circuit
Ref. Tubing	0.95 cm dia Copper	0.95 cm dia Copper
Fin Spacing	7.9/cm	4.7/cm
Fin Type	Wavy	Wavy
Passes	Thirty	Seventeen
Frontal Area	1.63 m^2	0.35 m^2
Rated Flow	53.8 m^3/min	35.4 m^3/min

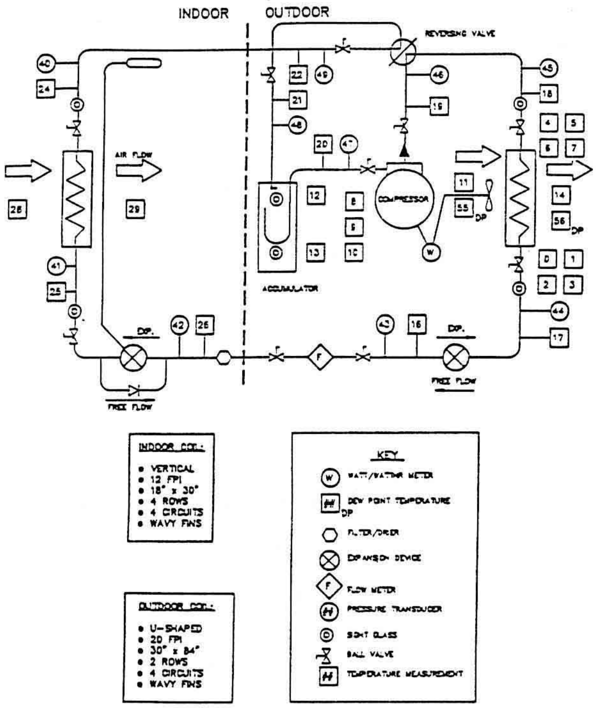

Figure 1 – Schematic of the Test Setup.

The sensor signals from the test points shown in Figure 1 were collected and converted to engineering units by a data logger. During each transient test of the heat pump, data processed by the data logger was transferred to a personal computer where it was stored. The fastest collection and storage rate for the cyclic testing was seven seconds per scan. The scan rate was adjusted such that at start-up the scan rate was seven seconds and gradually increased to 30 seconds after five minutes.

METHODOLOGY

The first step of the test procedure was to set the system refrigerant charge according to criteria specified by the manufacturer of the heat pump. The charge was set while the heat pump was operating in the cooling mode with an indoor dry-bulb temperature and humidity of 26.7 °C and 50%, respectively, and an outdoor dry-bulb temperature of 35 °C. The refrigerant charge was adjusted to obtain a 6.7 °C sub-cooling of refrigerant leaving the outdoor coil. The steady-state and cyclic tests recommended by the Department of Energy (DOE) were then run.

For all measurements, the psychrometric rooms were operated for more than one hour to allow the rooms and the equipment inside them to stabilize to the pre-set conditions before initiating a test run. Rooms were always maintained within the tolerance prescribed by the ASHRAE Standard (1983). For cyclic tests, three complete cycles were run to allow for consecutively repeatable cycles. As prescribed in the test procedure, all the cyclic tests except for the last cycle of a test were run with the continuous indoor fan operation (i.e. during both the on and off-cycles). The cooling capacity was measured by two methods: indoor air enthalpy and the refrigerant enthalpy. All measurements and calculations were in compliance with the ASHRAE Standard (1983).

The overall performance of the heat pump operating in cycles was characterized by measuring total cyclic capacity, total cyclic energy consumption, time required to reach steady-state, and cyclic energy efficiency ratio (EER_{cyc}). In addition to overall system performance, the details of the transient refrigerant and the air-side dynamics were analyzed carefully to completely understand the transient behavior of the heat pump.

EXPERIMENTAL TESTS

There are number of variables which affect the cyclic performance of heat pumps. This study included three different percent on-times (20, 50 and 80%) and broad range of cycling rates, including those which may be typically expected in a residence. It is often assumed that a typical residential thermostat cycles 1.8 cycles per hour (cph) at 20% on-time and 80% on-time and 3 cph at 50% on-time (1972). The indoor dry-bulb temperatures varied between 22.2 C and 26.7 C and indoor relative humidity varied between 20 to 67%. The outdoor conditions and the indoor air flow rate (34 m^3/min) were constant for all test runs.

TEST RESULTS

Steady-State

To compare the steady-state results with those from the manufacturer a base case steady-state test (Test A) was run. The indoor and outdoor rooms were maintained at 26.7 and 35 °C, respectively. The indoor relative humidity was maintained at 50% and the air flow rate was 34 m^3/min. The changes in the temperatures at steady-state condition were within the tolerance prescribed by the ASHRAE Standard (1983).

The total instantaneous cooling capacity obtained from the air enthalpy and refrigerant enthalpy calculation varied less than five percent during cooling operation. This agreement was within the ASHRAE test standard requirements (1983) and was observed in all test data.

Typical Cyclic Test Run

At start-up, the compressor reached steady-state speed almost instantaneously. Therefore, the suction and discharge pressures approached the steady-state values quickly (Figure 2). The refrigerant in the evaporator at start-up was saturated. The sudden drop in the suction pressure caused two phenomena. First, the refrigerant began to vaporize within the evaporator which rapidly cooled the evaporator. This phenomenon was characterized by the rapid increase in sensible capacity during the first 75 to 90 seconds after start-up (Figure 3). Figure 3 also shows the rate of moisture removal across the evaporator coil. Since two cycles were run before data were collected, the evaporator coil had moisture left at the end of the second cycle and was also at a lower temperature than the indoor room at the time of start-up. Therefore, the moisture removal is negative at start-up due to moisture being evaporated from the wet evaporator coil into the air stream.

The second phenomenon was the buildup of liquid refrigerant in the accumulator during start-up. This can be seen by studying the surface temperatures of the accumulator and the compressor (Figure 4). The compressor shell temperature, at the top, indicates that refrigerant inside the shell boiled due to sudden drop in suction pressure at start-up. Any refrigerant that boiled inside the compressor shell and any liquid that reached the accumulator was lost temporarily for cooling purpose, and contributed to the cycling losses.

After an initial rise, the power dropped to about 3.3 kW and remained constant (Figure 4). The initial rise in the power consumption corresponded to the boiling of refrigerant in compressor shell. The integrated cyclic cooling capacity and energy consumption were obtained by a numerical integration of the instantaneous values over the time of the cycle. The integrated power and capacity were then used to estimate the EER_{cyc}.

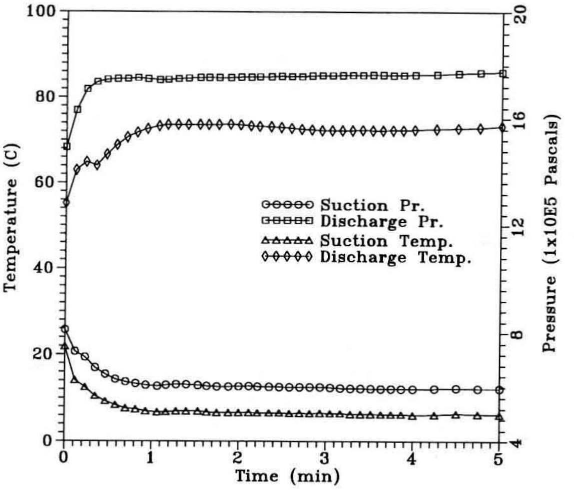

Figure 2 – Compressor Suction and Discharge Temperatures and Pressures.

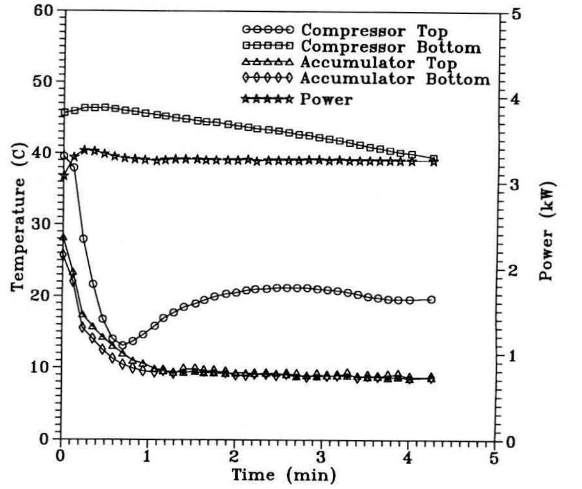

Figure 4 – Compressor and Accumulator Surface Temperatures.

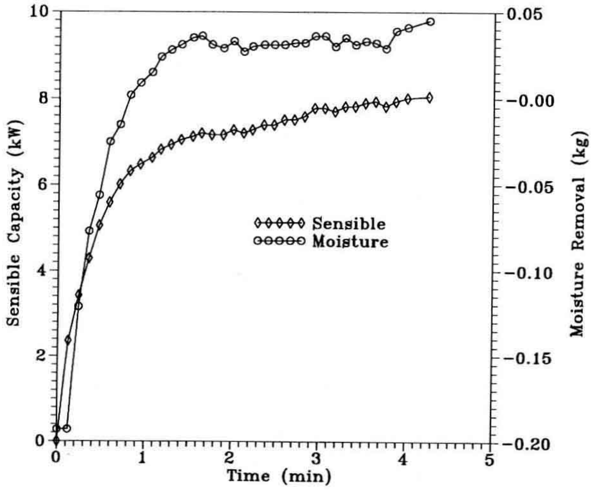

Figure 3 – Sensible, Latent, and Total Capacity for Typical Cyclic Tests.

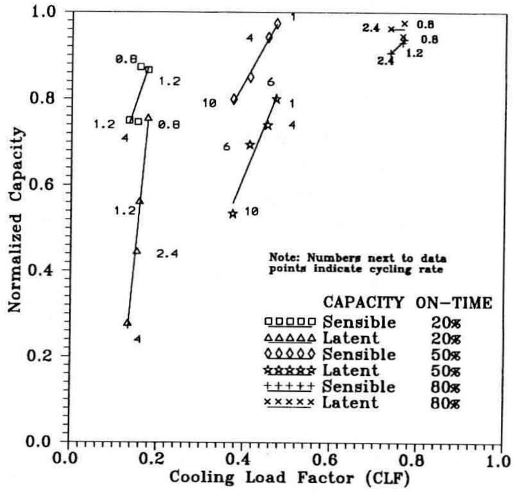

Figure 5 – Change of Normalized Sensible and Latent Capacities at Various Cycling Rates and Percent on-times.

RESULTS

One way to present results is in a non-dimensional or normalized form. For cycling effects on heat pumps, it is useful to normalize the latent (Q^L_{norm}) and sensible (Q^S_{norm}) capacities in terms of the steady-state values:

$$Q^L_{norm} = \frac{\frac{1}{t}\int_0^{t_{on}} \dot{Q}_L dt}{\dot{Q}_{Lss}}$$
$$Q^S_{norm} = \frac{\frac{1}{t}\int_0^{t_{on}} \dot{Q}_S dt}{\dot{Q}_{Sss}} \tag{2}$$

Figure 5 shows the normalized latent and sensible capacities at various percent on-times and cycling rates. The solid line represent the best fit through the data points. The normalized capacity is the ratio of the instantaneous capacity and the steady-state capacity at the same indoor and outdoor conditions as that of the cyclic test (Eq. 2). The loss of latent capacity due to cycling was greater than the loss of sensible capacity. As the cycling rate increased, the deviation of the latent capacity from the expected steady-state capacity increased. The deviation also increased as the percent on-time decreased. Although the effect of cycling rate and percent on-time on the sensible capacity was not severe as latent capacity the trends were similar. For instance, at 20 percent on-time and cycling rate of 4 cph the system lost over 70% for the steady-state latent capacity while the drop in sensible capacity was only about 15%.

The response of the moisture removal across the evaporator coil is shown in Figure 6 for three different indoor relative humidities (and 80% percent on-time). Initially, at start-up, the moisture removal was negative. Before condensation can occur on the evaporator, the surface temperature on the heat exchanger must drop below the dew-point of the air. When the system starts, the refrigerant temperature and the surface temperature of the evaporator are greater than the air dew-point temperature. Because there is moisture on the heat exchanger left from the previous cycle, some of this moisture will evaporate from the heat exchanger until the heat pump is able to reduce the evaporator surface temperature below the air dew-point. Thus, there is the negative moisture removal during the first 75 to 90 seconds (Figures 6 and 7). The steady-state moisture removal increased with increasing indoor relative humidity. The moisture removal reached steady-state quicker with increasing indoor relative humidity. It took 6 minutes, after start-up, for the moisture removal to reach steady-state for an indoor relative humidity of 67%, whereas it took almost 12 minutes at 45% Rh. There was no dehumidification below 45% Rh.

The moisture removal at 20% on-time is shown if Figure 7. Although the trends are similar to the 80 percent on-time case, the moisture removal becomes positive much earlier. It took less than 60 seconds for the moisture removal to become positive. Since the run time was only 7 minutes, the system never reached steady-state

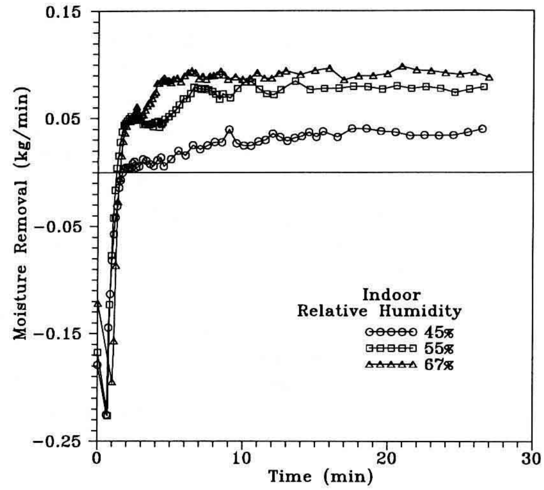

Figure 6 – Response of Moisture Removal Across the Evaporator Coil at Various Indoor Relative Humidities (80% on-time).

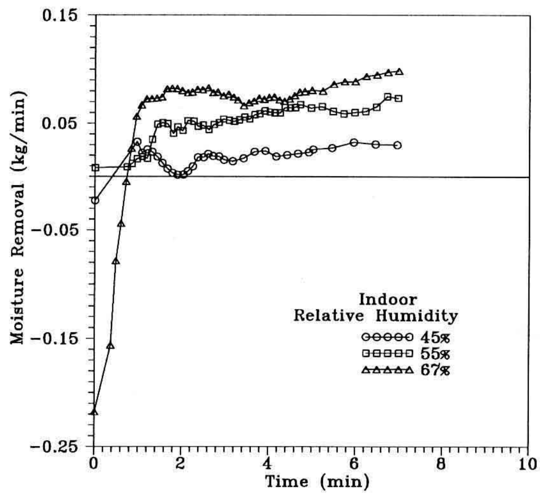

Figure 7 – Response of Moisture Removal Across the Evaporator Coil at Various Indoor Relative Humidities (20% on-time).

and the moisture accumulation on the evaporator coil was less than when the unit was run for 27 minutes (at 80% on-time). Therefore, the dehumidification process started early.

Dehumidification begins when the average surface temperature of the indoor coil falls below the dew-point temperature of the supply air. Since the dew point temperature of the supply air increases with increase in supply air relative humidity (at constant supply dry-bulb temperature), the potential for the dehumidification process also increases. Therefore, with increase in indoor relative humidity the dehumidification process will start much quicker. This leads to higher cyclic latent capacities. Although the moisture removal is negative for longer time period with increase in percent on-time, the cyclic latent capacity also increases with percent on-time.

The normalized latent capacity increased linearly with an increase in the indoor relative humidity (Figure 8) for a given percent on-time. The solid lines represent the best fit through the data points. The slope of the lines increased with decrease in percent on-time. With the increase in indoor relative (at constant indoor dry-bulb) the unit reached steady-state faster. Therefore, at a given percent on-time, the normalized latent capacity increased with relative humidity. With increase in the percent on-time, at constant indoor relative humidity, the run time of the unit increases and the unit has more time to reach steady-state. Therefore, the normalized capacities increase. The trends of the normalized sensible capacity were similar to that of the latent capacity (Figure 9). However, the slopes of the curve at a given percent on-time were smaller.

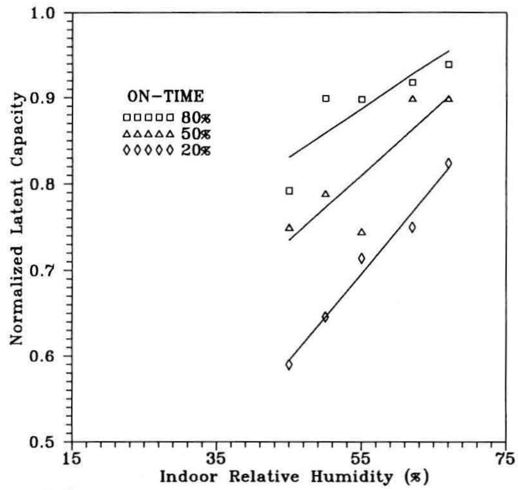

Figure 8 – Change in Normalized Latent Capacity With Indoor Relative Humidity.

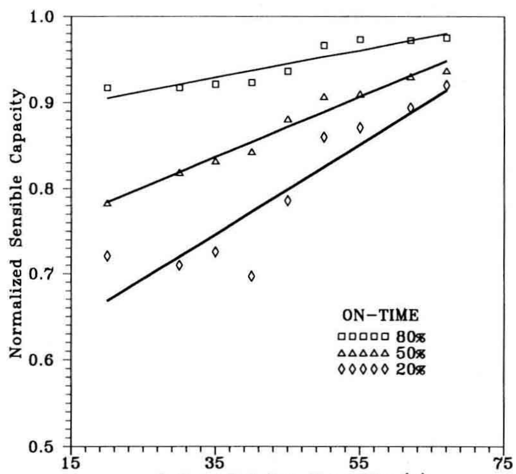

Figure 9 – Change in Normalized Sensible Capacity With Indoor Relative Humidity.

The response of the moisture removal at 50 percent on-time for various indoor dry-bulb temperatures and constant indoor relative humidity is shown in Figure 10. Moisture removal increased with an increase in the indoor dry-bulb temperature. Because moisture in the air-stream increases with an increase in indoor temperature at constant relative humidity, the moisture removal capacity also increases. The moisture removal was negative for the first 75 to 90 seconds after start-up. Similar to 50 percent on-time, the moisture removal increased with an increase in indoor temperature at 80 percent on-time. However, the moisture removal was negative for longer period of time (90 - 120 seconds).

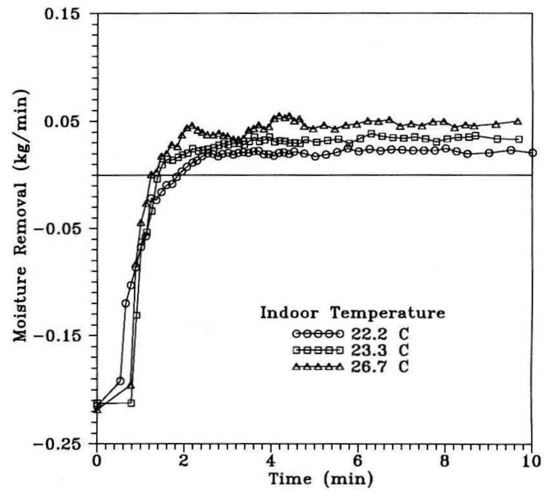

Figure 10 – Response of Moisture Removal at Various Indoor Dry-Bulb Temperatures and Constant Relative Humidity (50% on-Time).

The normalized latent capacity decreased with decreasing indoor temperature (Figure 11). In contrast, the normalized sensible capacity remained almost unchanged with indoor dry-bulb temperature for a given percent on-time (Figure 12). The normalized latent capacity showed larger slope than the normalized sensible capacity. The results for the 50 percent on-time showed a larger slope and variation than the 80% on-time. Since the run time of the unit at 50 percent on-time is smaller than at 80 percent on-time the slopes are larger.

At constant dew-point conditions, the moisture content in the air-stream is also constant. Therefore, to study the transient response of the heat pump under such conditions, a series of tests was run with constant indoor dew-point temperature (14.4 °C). The sensible capacity at the end of the cycle for both 80 and 50 percent on-times increased with an increase in indoor temperature. Moisture removal at the end of the cycle for both 80 and 50 percent on-times decreased with an increase in the indoor temperature. The time at which dehumidification started increased with an increase in indoor temperature. At 22.2 °C indoor temperature, it started after 60 seconds, whereas at 26.7 °C, it started after 120 seconds.

There was a slight increase in normalized latent capacity with indoor temperature for both 80 and 50 percent on-times (Figure 13). The normalized sensible capacity decreased slightly at 80 percent on-time and increased

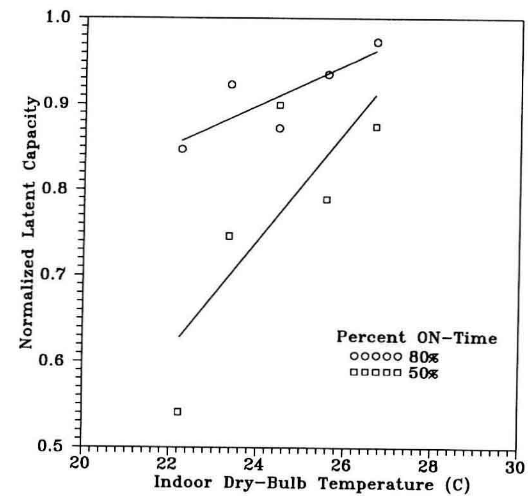

Figure 11 – Change in Normalized Latent Capacity With Indoor Dry-Bulb Temperature and Constant Indoor Relative Humidity.

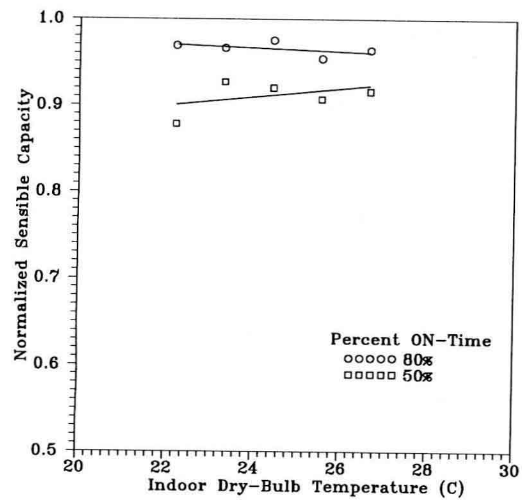

Figure 12 – Change in Normalized Sensible Capacity With Indoor Dry-Bulb Temperature and Constant Indoor Relative Humidity.

slightly with indoor temperature at 50 percent on-time (Figure 14).

Finally, a relationship was developed between normalized capacity and percent on-time and indoor relative humidity. The capacity varies exponentially with percent on-time (Figure 3) and linearly with indoor relative humidity (Figures 8 and 9). Therefore, a non-linear regression analysis was performed on the experimental data. The regression model relating the normalized capacity to the percent on-time and indoor relative humidity is of the form:

$$cap_{nor} = 1. - exp(-\alpha \times pon) + \beta \times rh_i \times exp(-\alpha \times pon) \quad (3)$$

where, α and β are regression constants and pon and rh_i are percent on-time and indoor relative humidity, respectively. This model is consistent with previous models proposed by Goldschmidt and Murphy [6] and O'Neal and

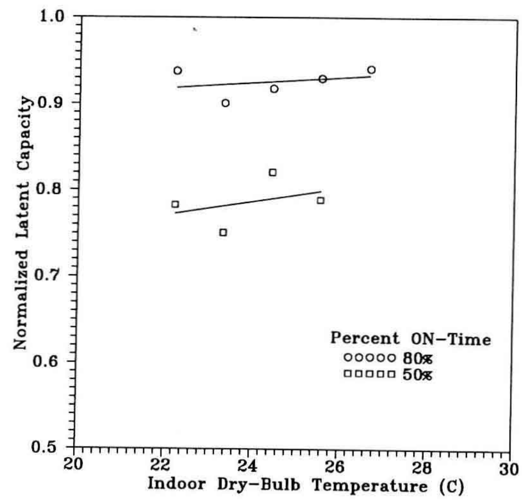

Figure 13 – Change in Normalized Latent Capacity With Indoor Dry-Bulb Temperature and Constant Indoor Dew-Point Temperature.

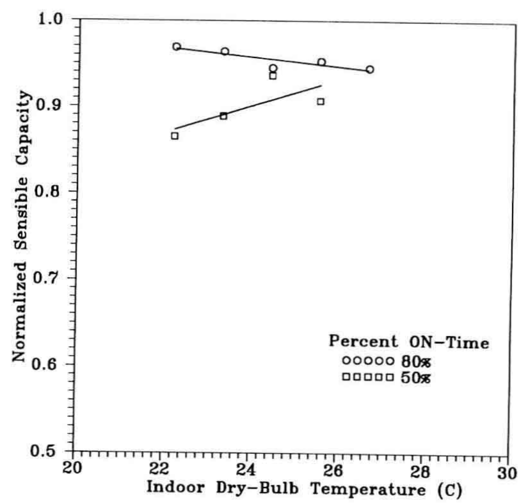

Figure 14 – Change in Normalized Sensible Capacity With Indoor Dry-Bulb Temperature and Constant Indoor Dew-Point Temperature.

Katipamula [10]. This equation is valid for percent on-times greater than zero. The values for α and β which resulted in the best fit for the normalized latent and sensible capacity data are shown in Table 2. The value of α and β are system constants and may vary with the system. The predicted normalized latent capacity was in good agreement with the actual (Figures 15). The predicted normalized sensible capacity was also in good agreement; however, there was more scatter below 0.7 (Figure 16).

Table 2 - Regression Constants.

Capacity	α	β
Latent	1.391	1.121
Sensible	2.779	1.343

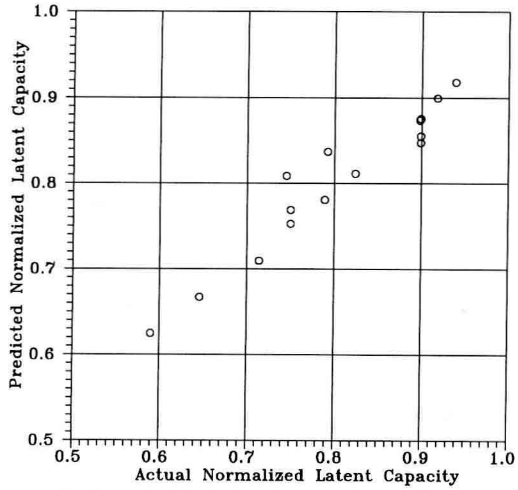

Figure 15 – Comparison of Predicted with Actual Normalized Latent Capacity

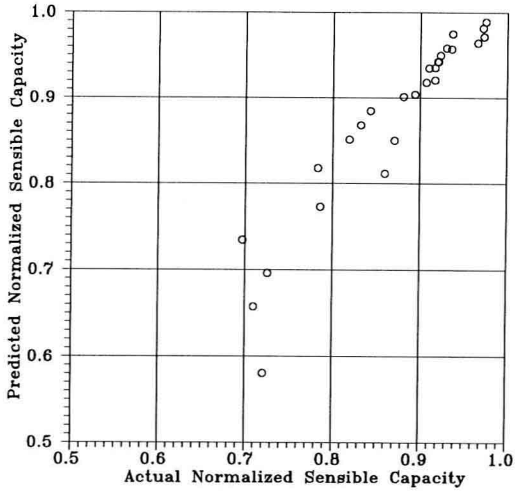

Figure 16 – Comparison of Predicted with Actual Normalized Sensible Capacity

CONCLUSIONS

To characterize the transient dehumidification performance of heat pumps in the cooling mode, an experimental and analytical investigation was performed. A series of tests were performed at varying operating conditions and the transient dehumidification performance was characterized. This study illuminated several trends required to characterize both transient sensible and dehumidification performances of the test heat pump:

(1) For almost all the tests, moisture was added at start-up and dehumidification began 60 to 150 seconds after start-up depending on the ambient conditions. Therefore, it appears that if the heat pump is ON for less 2 minutes it will add rather than remove moisture from the air.

(2) The latent capacity took 6 to 15 minutes to reach steady-state after start-up. As the on/off cycles increased it took longer for the latent capacity to reach steady-state. Also as the indoor relative humidity decreased (at constant indoor dry-bulb temperature) the latent capacity took longer to reached steady-state.

(3) The loss in latent capacity was greater than the loss in the sensible capacity with a decrease in the unit's run time and increase in the number of on/off cycles. If the conditioning unit is over-sized, which is a common practice, the unit will cycle longer. Therefore, the unit's dehumidification performance will be poor and the unit may not meet the comfort criteria.

(4) It appears that the current methodology used to estimate the seasonal performance, based on a single test run, may not be able to quantify the seasonal dehumidification performance adequately. At high cycling rates and low percent on-times the seasonal performance will be over predicted.

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INEXPENSIVE METHOD FOR PERFORMING CONTINUOUS DUTY CYCLE TESTING OF ENGINE-DRIVEN GAS HEAT PUMPS

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ABSTRACT

The use of recirculating air loops as a simple, cost-effective approach for durability testing of gas heat pumps is described. Several alternative test methods are also described and evaluated. These testing approaches provide a means for running the heat pump all available hours while simulating the actual running conditions expected in field service. The recirculating air loops have been operating employing two different duty cycles based on historical weather data for Chicago and Salt Lake City. The operation and limitations of the testing approach are also described.

INEXPENSIVE METHOD FOR PERFORMING CONTINUOUS DUTY CYCLE TESTING OF ENGINE-DRIVEN GAS HEAT PUMPS

INTRODUCTION

The Gas Research Institute is currently sponsoring work to develop a 5-ton engine-driven gas heat pump (GHP) for light-commercial applications. Aside from first cost and performance, two of the most critical factors affecting commercial success of space conditioning products are long life and high reliability. However, evaluation of system life and reliability requires extensive operating experience under expected "duty cycle" conditions.

Due to the excessive time required to accumulate actual operating experience in the field, it was necessary to develop a means of obtaining "accelerated" experience in order to gain confidence in system durability, reliability, and overall operation over a wide variety of operating conditions. A typical engine-driven GHP, on average, is expected to operate between 2,500 and 4,000 hours annually, whereas continuous year-round testing has the potential to subject the GHP to 8,760 hours per year or greater than two to three times that possible in the field. One approach, the recirculating air loop method, has been successfully implemented to gain operating experience with the engine-driven GHP over the shortest time-frame. The recirculating air loop method was chosen over other techniques that control engine and compressor loading due to considerations of cost and ease of testing. The primary benefit of this test method is the ability to simulate GHP operating conditions, thus enabling year-round continuous GHP operation under normal duty cycle conditions for most climates.

ALTERNATIVE HEAT PUMP LOADING SCHEMES

A number of approaches were considered in the selection of an endurance test method capable of subjecting the critical components of a GHP--the engine and compressor--to operating conditions expected in the field while accumulating operational time in the shortest time-frame. The most important factors in selecting the endurance test approach were:

- Accuracy of load control with minimal operator intervention and maintenance. The need for accurate load control can be seen by examining Figure 1, which shows a greater than two to one ratio of engine and compressor load over the range of outdoor temperatures. The lives of engine and compressor components are quite sensitive to the applied loads. For instance, life is inversely related to the cube of the load for rolling element bearings.

- Confidence in maintaining proper component operating conditions. For example, in the case of the compressor, it is important to avoid creating conditions such as excessive loading, liquid slugging or lubricant starving of the compressor. Testing of a complete heat pump helps to avoid such problems.

- Need to minimize development time and cost. Obtaining cost-effective operating experience in the shortest period of time is always a key factor in selecting an endurance test approach.

The simplest endurance testing approach is to install the GHP in a field application, heating and cooling a residence or building space. However, since operation of the GHP is dictated by weather conditions, operating hours are limited. Alternatively, the GHP