SPLIT-SYSTEM RESIDENTIAL HEAT PUMP WITH SEPARATE DEHUMIDIFICATION OPERATING MODE

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Residential air conditioners and heat pumps operate when the room thermostat senses a temperature above the control set point temperature. Systems are designed to provide dehumidification simultaneously with cooling operation and split-system components can be selected to bias performance toward greater dehumidification. Even so, there are times when the incidental moisture removal capacity of the system is not sufficient to keep indoor humidity within desired limits. This situation is common during the spring and autumn, when the sensible cooling load is low and, in humid climates, can occur throughout the cooling season.

This paper reviews the background and development of a split-type heat pump system with three operating modes: heating, cooling, and dehumidifying. An analytical method was developed to assess the effectiveness and operating cost of active dehumidification options. Several technologies were compared and contrasted using this evaluation tool. A split-system heat pump that flexibly controls both indoor temperature and humidity was built and tested. Laboratory and field performance results are presented.

Key Words: ***dehumidification, heat pump, air conditioner, humidity, reheat, comfort, health***

1 INTRODUCTION

Conventional vapor compression air conditioners and heat pumps provide cooling in response to a room temperature deviation from the thermostat set point temperature. Design practice has evolved over the years such that a reasonable amount of latent cooling is accomplished simultaneously with sensible cooling. Air-Conditioning and Refrigeration Institute certification program rules limit the amount of indoor air circulation to 218 cubic meters per hour for each kW of delivered cooling (450 cfm/ton) as one way of forcing some dehumidification to occur at the rating point condition. Most residential heat pumps operate with a sensible heat ratio of 0.70 to 0.80, at rating conditions.

Humidity is not actively controlled during the cooling season. When moisture gains (from internal sources and infiltration) exceed the incidental moisture removal capability of the heat pump as it works to meet the sensible load, the indoor humidity can exceed desirable levels. This condition frequently occurs during the spring and autumn when there is a low sensible cooling load. It can also occur in humid climates, all through the cooling season, when the infiltration of humid outdoor air overwhelms the latent capacity of the heat pump. This condition can occur in spite of efforts by equipment specifiers to deal with the local climatic conditions by carefully selecting and commissioning equipment.

This paper reviews the background and development of a split-type heat pump system with three operating modes: heating, cooling, and on-demand dehumidification. The authors developed an analytical method to assess the effectiveness and operating cost of various residential active dehumidification options. Several technologies were compared and contrasted using this evaluation tool. Subsequently, a split-system heat pump was developed to flexibly control both indoor temperature and humidity. A new control algorithm was developed and implemented in a thermostat capable of sensing temperature and humidity. This system's laboratory and field performance results are presented in this paper.

2 SUMMER COMFORT

2.1 ASHRAE Summer Comfort Zone

ASHRAE Standard 55 (ASHRAE 2004) provides a guideline for summer indoor temperature and humidity conditions to assure sedentary (or slightly active) occupants will find the environment thermally acceptable. This Summer Comfort Zone is shown on the psychrometric chart in Fig. 1.

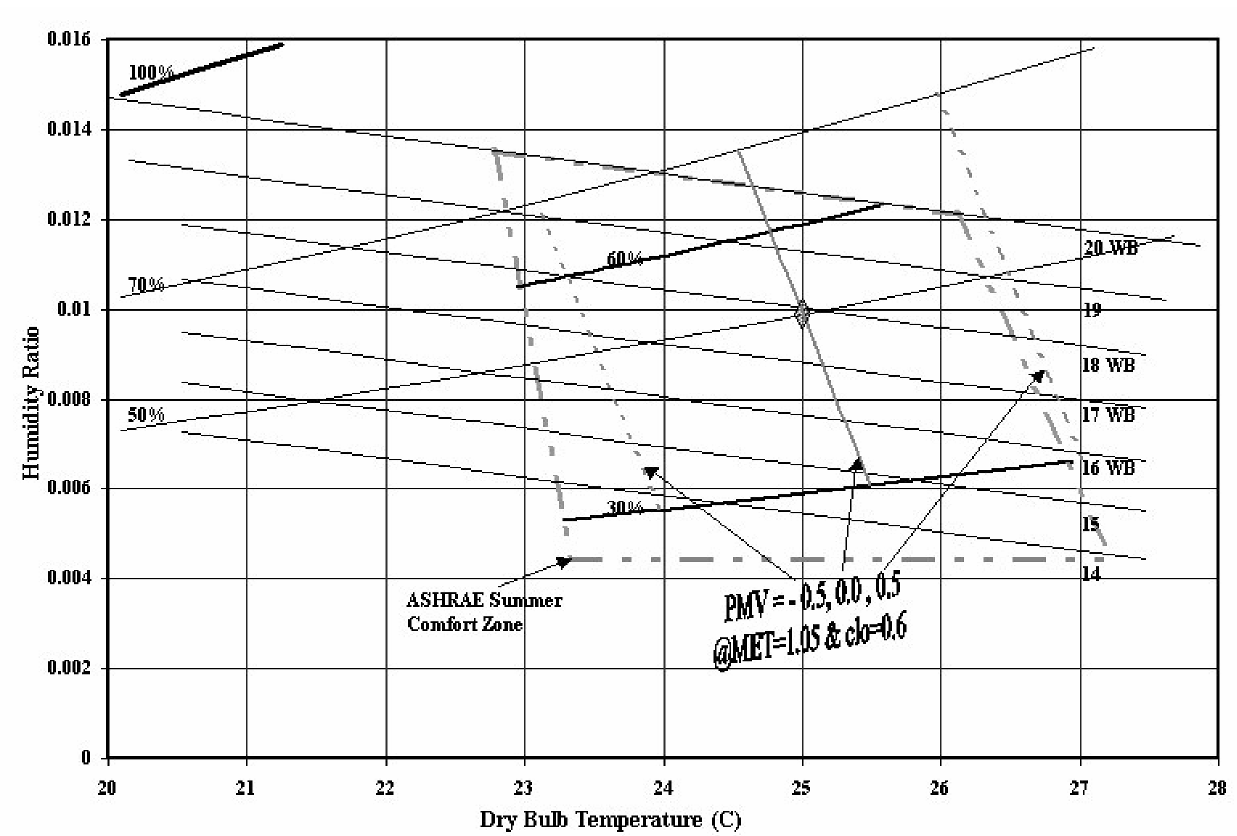

Fig. 1. ASHRAE Summer Comfort Zone and Zero PMV +/- 0.5

The lines of constant temperature are vertical in the psychrometric chart. The left and right edges of the Summer Comfort Zone slant downward and to the right. The interpretation is that when humidity is lower, a higher sensible temperature is acceptable for the same comfort sensation. (Note that the left and right edges are not parallel.) The upper boundary of the Summer Comfort Zone coincides with the 20º C wet bulb temperature line. The lower boundary is set to assure a minimum absolute humidity level since it has been found that physical discomfort (dry eyes, skin, etc.) can occur when humidity is too low. (For most climates that warrant comfort cooling, excessively low summertime indoor humidity is not an issue.)

Also shown are three lines indicated as PMV = -0.5, PMV = 0.0 and PMV = 0.5. The Predicted Mean Vote (PMV) is an indication of thermal acceptability of a space. PMV is an empirical function derived from physics of heat transfer and the thermal responses of people in climate chamber tests. PMV establishes a thermal strain based on environmental conditions and attaches a comfort vote to that amount of strain. Zero indicates neutral; a positive value indicates warmth and negative means coolness. The assumptions made about activity level and clothing worn by the occupants affect the PMV. For this study, we have assumed the occupants are only engaged in light activity and are wearing casual summer apparel. If the predicted environmental conditions are within a PMV range –0.5 and +0.5, then 80% of occupants will find the thermal environment acceptable. The central line (PMV= 0.0) and sideband PMV lines correspond fairly closely to the Summer Comfort Zone. The central line crosses 50% relative humidity at approximately 25º C. A diamond-shaped marker is shown at this location in Fig. 1. This environmental condition has been somewhat arbitrarily selected to represent an ideal summer indoor condition.

The ASHRAE Summer Comfort Zone is a very broad region and it is normally a goal to maintain fixed indoor conditions at a point near the center of the Zone. ASHRAE recommends limiting indoor RH to 60% but, for some people, health issues (allergies) argue for maintaining lower indoor humidity. In particular, limiting average indoor humidity to approximately 50% tends to reduce odors, and allergens. One commonly cited study showed that the Critical Equilibrium Humidity (CEH) for the most common US dust mite, Dermatophagoides farinae, is just over 55% RH throughout the normal range of indoor termperatures (Arlian 1981). The second most common North American species, Dermatophagoides pteronyssinus, has a higher CEH (Crowther 2000). It is the fecal matter of these mites that causes a common allergic reaction in people. The US Environmental Protection Agency (EPA) recommends keeping indoor humidity "…ideally between 30% and 50%" to avoid production of mold allergens (EPA 2002). These facts argue for revising our goals for indoor environmental conditioning.

2.2 Assessing Cooling Season Indoor Conditions

2.2.1 Seasonal simulation of temperature and humidity dynamics

In order to assess the performance advantages of different residential cooling/dehumidification schemes a simulation program was developed for this project. The core of the simulation program is a customized version of the Florida Solar Energy Center whole-building model (version 3.0). This program was selected because it takes into account moisture transport to and from the building materials. Equipment performance was supplied by "maps." Typical meteorological year (TMY2) hourly data sets from 1961 – 1990 National Solar Radiation Data Base were interpolated to obtain 3-minute time step data. Seventeen cities were used in the initial evaluations. (Due to space limitations, only data for Houston, TX is presented in this paper.)

A house, with the following construction and load characteristics, was modeled:

- Single-story, "L" shaped ranch style home
- 232 square meters conditioned floor area
- 10.2 cm slab-on-grade foundation with carpeted floors
- Frame/Brick wall construction, 38 hr*degK/Watt*hr (R-11)
- Window area 15% of floor area, Transmissivity coefficient of 0.7 (except Florida, 0.54)
- Ceiling insulation of 102 hr*degK/Watt*hr (R-30)
- Duct in attic, insulation of 21 hr*degK/Watt*hr (R-6)
- Natural infiltration of 0.35 ACH
- Lighting, equipment and "people" loading were scheduled using information from an LBNL study of typical usage patterns

A conventional residential split-system heat pump was selected to provide baseline performance information. The home was modeled in Houston using a 25ºC thermostat setpoint. Each hour of the cooling season was assessed for indoor humidity and temperature and then the data were plotted on a psychrometric chart. Figure 2 shows how the hourly indoor conditions are scattered around and even outside the Summer Comfort Zone. Most hourly temperatures are aligned with the vertical 25ºC dry bulb temperature line. The points more than 0.5ºC to the right of the line are hours in which the equipment could not meet the sensible cooling load. (It is a sign that the equipment is properly sized for the house that there are a few hours each summer when this occurs.) There is a wide variation in the humidity ratio over the season. In one sense, this is to be expected since humidity is not sensed by the control system and whatever dehumidification occurs is actually incidental to a thermostat call for sensible cooling. As can be seen, there are many hours of operation for which the indoor relative humidity exceeds 60%.

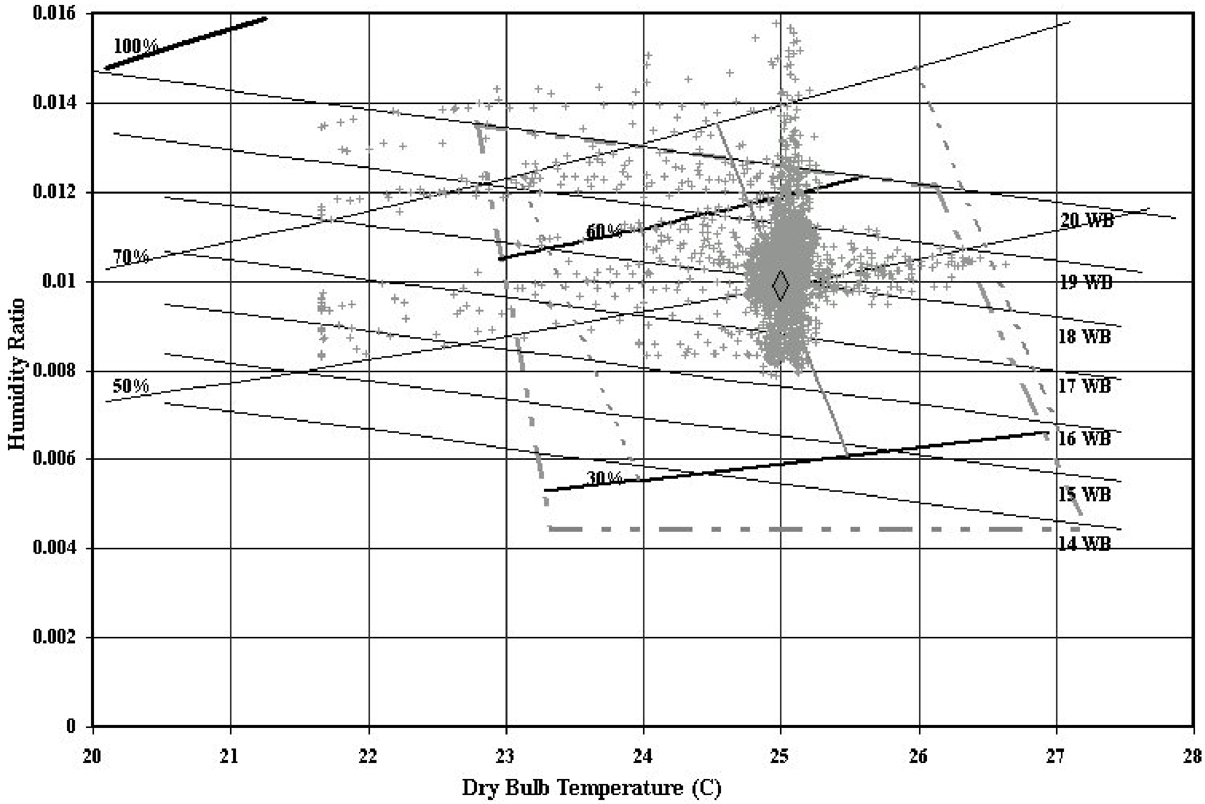

Fig. 2. Houston Cooling Season Hourly Results for a Conventional Residential Split-System Heat Pump

2.2.2 Seasonal performance index for indoor conditions

There are many potential methods for improving the ability of heat pumps to limit indoor humidity excursions. Before starting to model system and cycle variations we needed a quantitative way to compare the results. The system we developed assigns a penalty factor for each cooling season hour. The penalty factor is near zero when the average conditions for an hour are near 25º C and 50% relative humidity. The penalty system has these features:

- Excursions along the 0.0 PMV line and in the direction of lower humidity have low penalties
- Excursions away from the 0.0 PMV line are penalized moderately
- Excursions in the direction of higher humidity are significantly penalized.

Averaging the values of all the hourly penalties produces a seasonal performance index. A low index is better than a high one. The penalty system we selected is very non-linear in order to provide good resolution between the different systems analyzed.

Figure 3 illustrates the penalty factor map. We have used contour lines to show penalty factor values. The central portion of the lower half of the ASHRAE Summer Comfort Zone becomes the target zone for system operation and hours spent there receive low penalties. Hours in the upper one-third of the Comfort Zone are penalized more heavily because we know that odors, dust mites and mold are suppressed by keeping the humidity in check.

Another characteristic of this penalty system, used for Cooling-Only simulations, is that hours spent with good humidity levels but at temperatures cooler than the left boundary of the Comfort Zone are not heavily penalized. The rationale is that an auto-changeover thermostat would trigger the heating system and warm the space to restore thermal comfort.

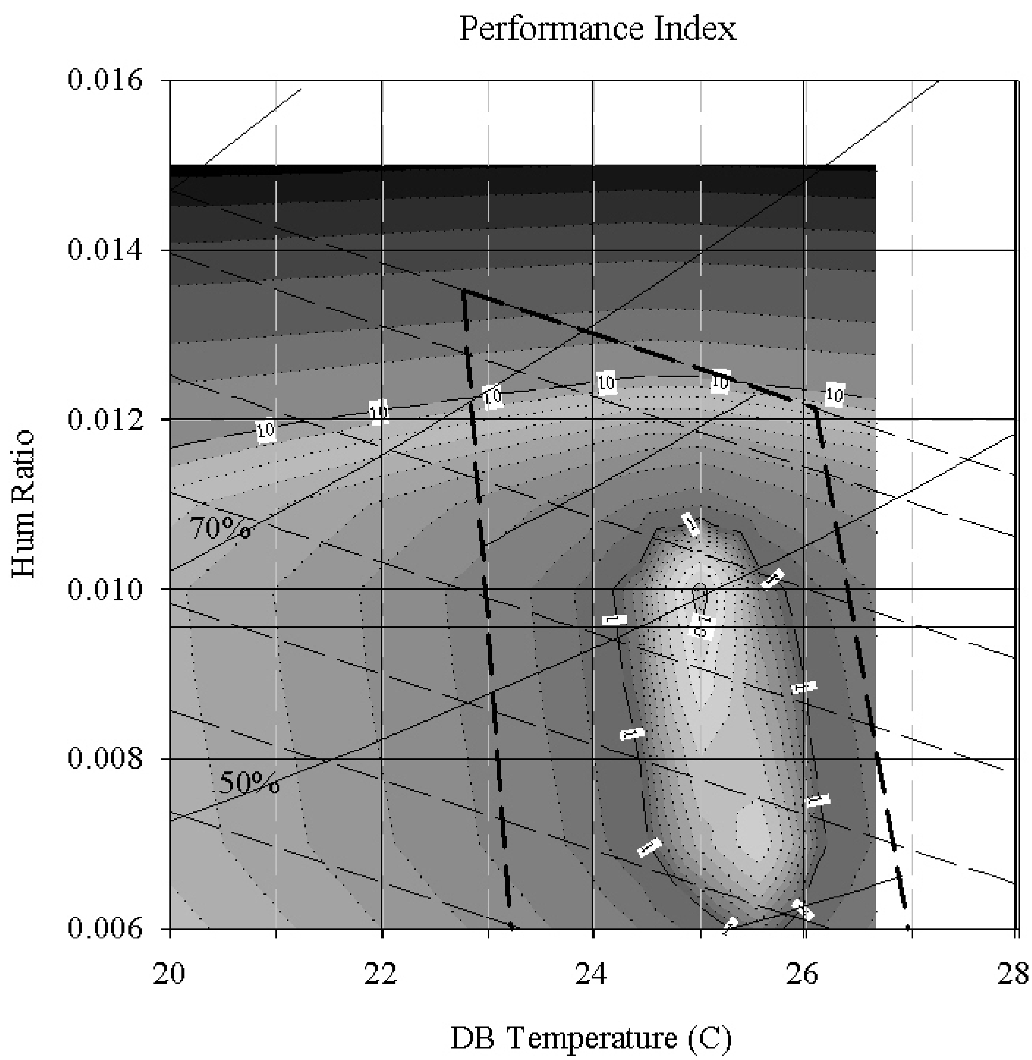

Fig. 3. Penalty Factor Map Superimposed Upon a Psychrometric Chart

2.2.3 Seasonal simulation results for several different heat pump systems

A high-efficiency, two-stage heat pump reduces the cooling season electrical energy usage by 27% but it does not change the character of the indoor conditions much, as can be seen by comparing Fig. 2 and Fig. 4. (The high-efficiency, two-stage heat pump uses 65% of normal indoor airflow, while operating on low-stage, to maintain latent moisture removal capability.)

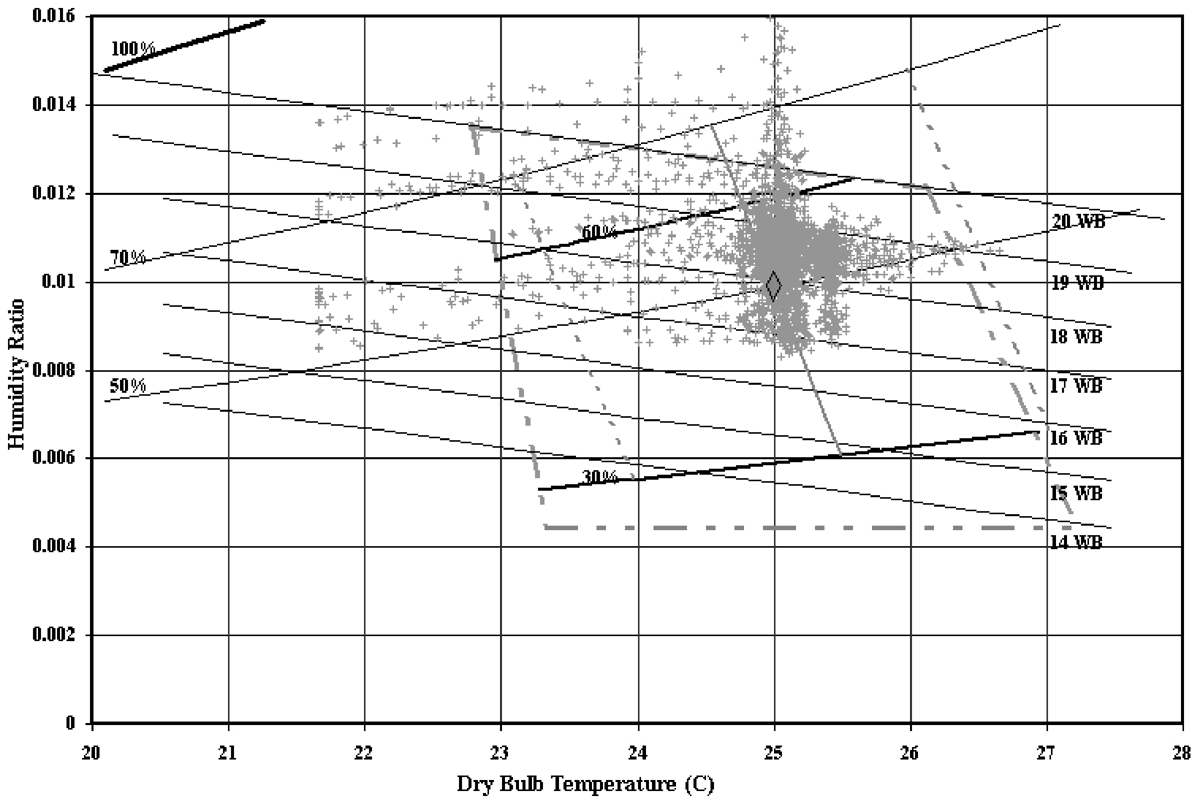

Fig. 4. Two-Stage, High-Efficiency Heat Pump

A recent innovation in the residential air conditioning market is the combination thermostat/humidistat. Some versions incorporate microprocessor-based logic that temporarily lowers the temperature setpoint under high humidity conditions. Figure 5 shows results of using one of these advanced thermostats with a high-efficiency, two-stage air conditioner. For these simulations, the cooling setpoint is 25ºC and the humidistat is set at 50% RH. Up to 1.1ºC overcooling is enabled when the indoor relative humidity is above setpoint. The difference between Fig. 4 and Fig. 5 is dramatic: the number of summer hours above 60% RH is considerably reduced

The last psychrometric chart, Fig. 6, shows the seasonal operation of a high-efficiency, two-stage, split-system heat pump that has an enhanced dehumidification mode of operation. This mode uses condenser reheating of the supply air to shift the sensible-to-total heat ratio (S/T) down to 0.25 and lower. The control logic used is as described above except during high indoor humidity conditions, the enhanced dehumidification mode is used after the thermostat setpoint has been met. Again, there is a dramatic reduction in the number of summer hours with elevated humidity. Hours with absolute humidity above 0.011 are greatly reduced. A psychrometric chart provides a convenient way of understanding this. Sensible cooling is represented as a movement to the left on the chart. Moisture removal is represented as movement straight down on the chart. A conventional heat pump will cause a trace on the chart that is downward, but mainly to the left. The heat pump with an enhanced dehumidification mode has an S/T of 0.25 and will leave a trace on the chart that is mainly downward. It becomes possible to do a more thorough drying process on the indoor air before reaching thermal comfort limits.

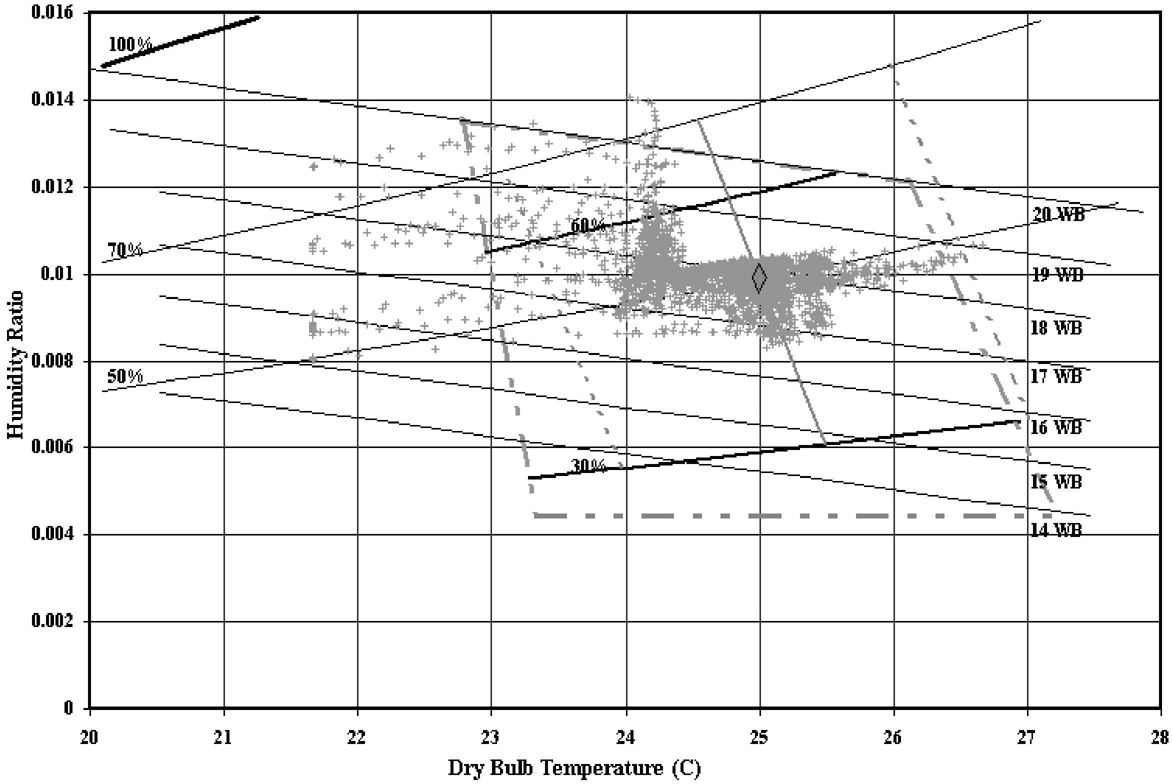

Fig. 5. Two-Stage, High Efficiency Heat Pump, 1.1º C Overcooling Allowed for Dehumidification

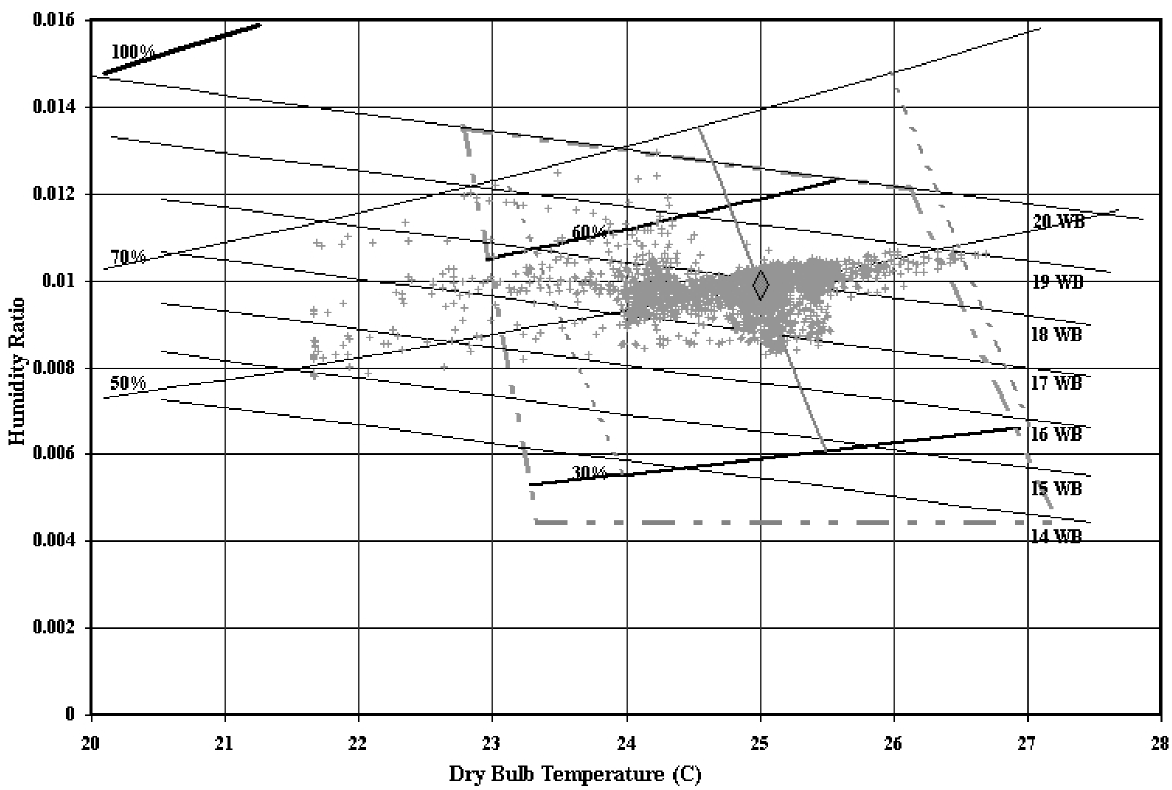

Fig. 6. Two-Stage, High Efficiency Heat Pump with Condenser Reheating, 1.1º C Overcooling Allowed

3 TESTING IN ENHANCED DEHUMIDIFICATION MODE

Figure 7 depicts the system that was assembled and tested in a psychometric chamber. The system consists of a typical heat pump system with additional components that allow normal operation in cooling and heating modes, and, in addition, operation in an enhanced dehumidification mode. The system also contained a refrigerant charge compensator and components that allowed means for storing excess refrigerant during the enhanced dehumidification mode of operation and for returning charge during normal operating mode.

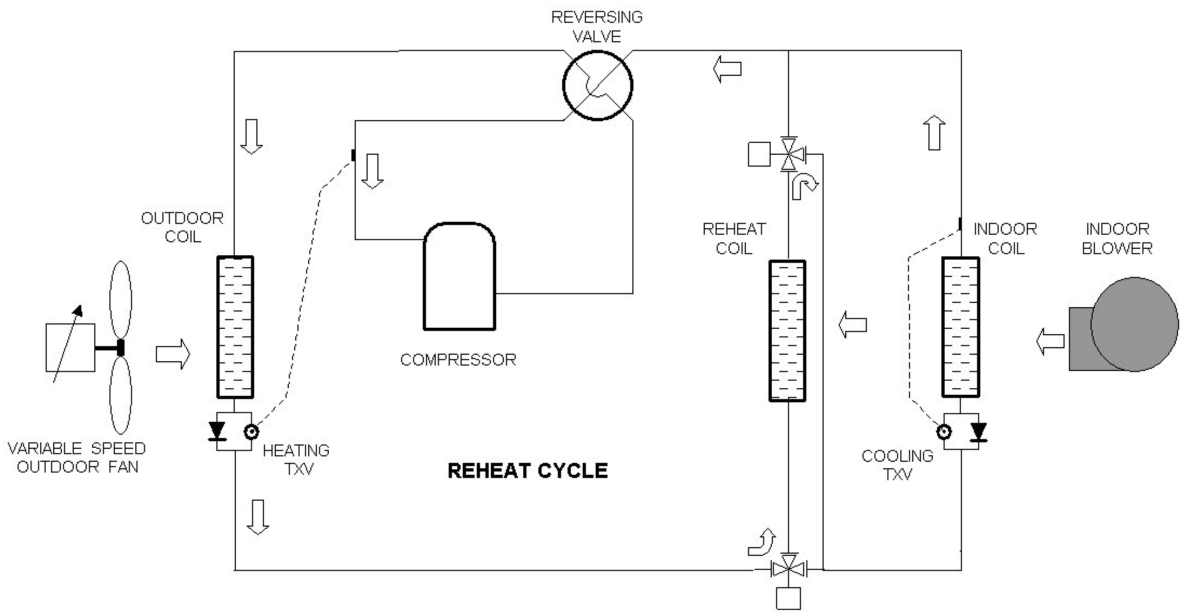

Fig. 7. Variable Condenser Reheat Heat Pump

The tested system had variable speed motors on both the outdoor and indoor sides of the system. The compressor had capability of operating at two capacity steps. During enhanced dehumidification operation, the compressor was at the high capacity step, the indoor blower was run at a reduced airflow, and the outdoor fan was operated at a reduced speed.

During dehumidification operation, refrigerant from the condenser coil is not completely condensed. The refrigerant completes condensation and is subcooled as it passes through the reheat coil. Return air passes normally through the evaporator coil, and then is warmed as it passes through the reheat coil. Air supplied to the space has thus been dehumidified, but without a greatly reduced dry bulb temperature. The impact is to allow significant dehumidification of the space without overcooling. This operating mode addresses loads that are primarily latent. A recent AHSRAE Journal article (Taras 2004) contrasts the dehumidification performance of “two-phase mixture” reheating systems and other, more common, reheating cycles. The installation advantages of the system shown in Fig. 7 are that it is a “two-pipe” system (lending itself to residential “split” applications) and that it works on heat pumps as well as conventional air conditioners.

Figures 8 and 9 illustrate the performance of the system under varying outdoor ambient temperatures for a typical indoor temperature, but with a higher than desired relative humidity. Figure 8 shows the improvement in S/T that is achieved over a conventional system. The system provides a performance characteristic that effectively reduces the humidity in the space. This is desirable when the sensible cooling load is low, but a significant humidity load exists.

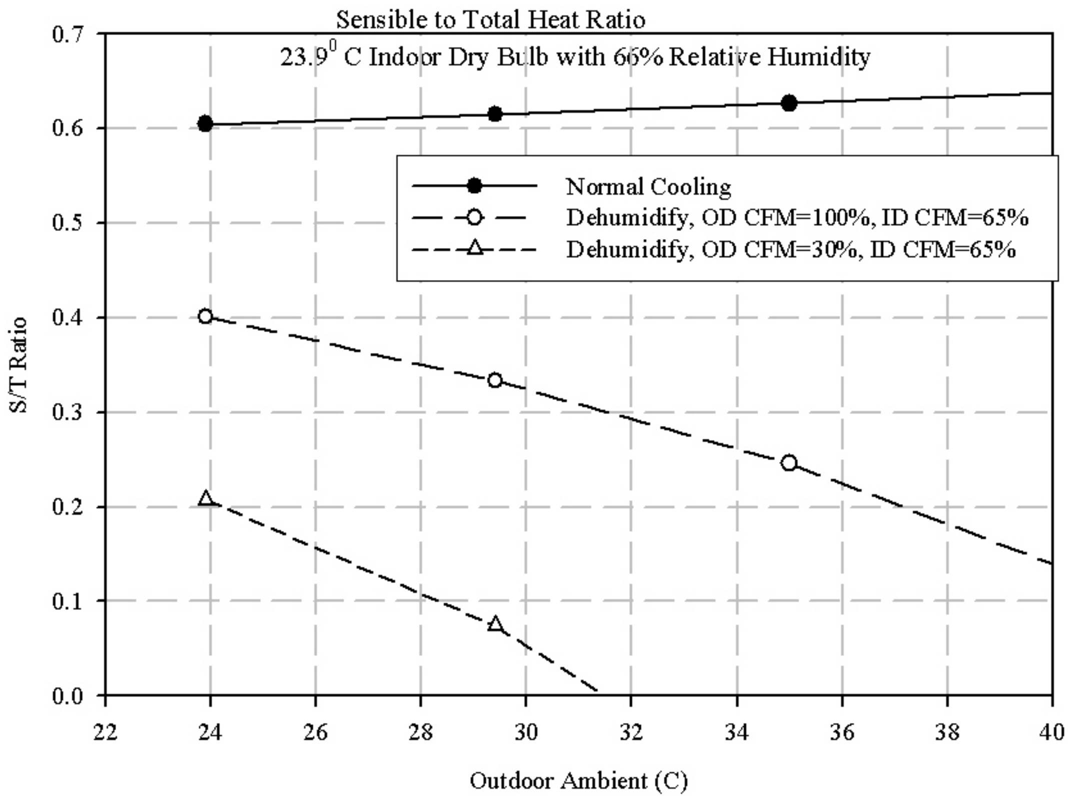

Fig. 8. Sensible To Total Heat Ratio for Variable Condenser Reheat Heat Pump

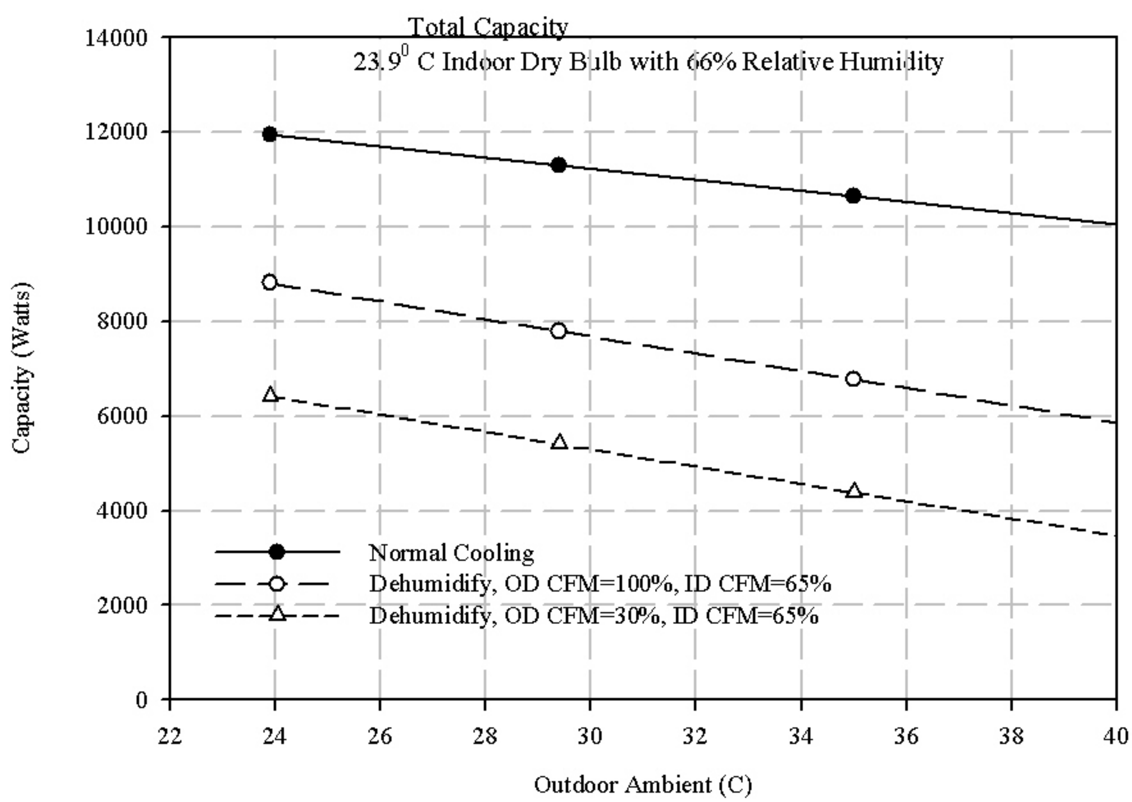

Fig. 9. Cooling Capacity for Variable Condenser Reheat Heat Pump

4 FIELD DATA COLLECTED ON VARIABLE CONDENSER REHEAT HEAT PUMP

An air conditioning system, using variable condenser reheat to provide an enhanced dehumidification mode, was installed in a home near Houston, Texas in March 2004. The home is a large, two-story structure with separate air conditioning systems for each floor. We installed the enhanced dehumidification accessory on the 14kW system that serves the lower floor. Minute-by-minute data was collected on this home throughout the 2004 cooling season. The data shown in Fig. 10, below, illustrates operation of this system over a 12-hour period in which the outdoor temperature was mild, but the latent load was high due to rain. Repeated brief calls for low-stage cooling occurred during the morning. The indoor humidity was incrementally reduced. A rain shower occurred at about 9:30 AM and, before mid-day, the moisture load began to overwhelm the latent capacity of the lightly-loaded system. A sustained enhanced dehumidification call came just before noon and the effect on indoor humidity (and lack of effect on indoor temperature) can be seen for that hour of operation. Approximately 1:30 PM, the sun came out and the sensible load on the structure started to rise. For the rest of the afternoon, intermittent low-stage cooling operation kept the indoor temperature and humidity under control.

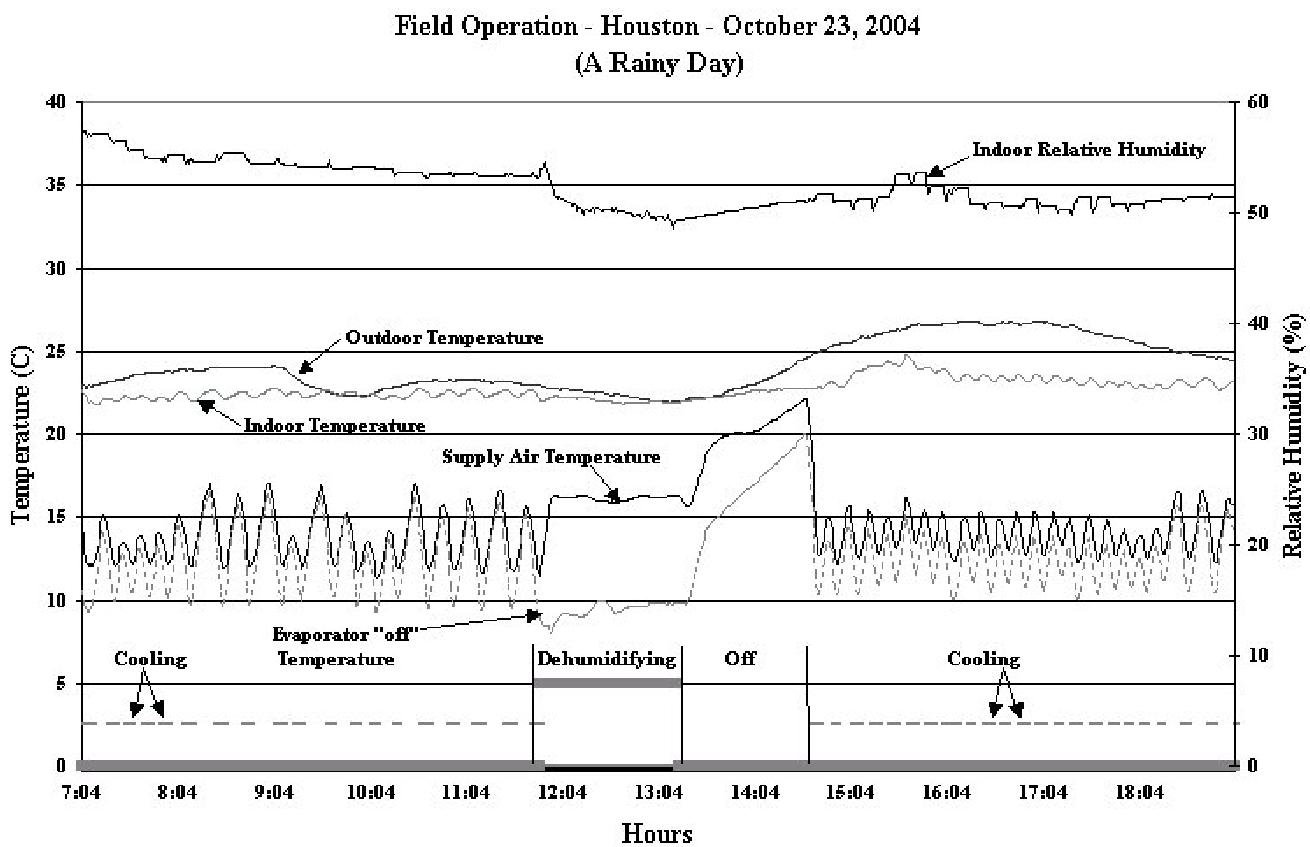

Fig. 10. Field Data Collected October 23, 2004 from Variable Condenser Reheat Air Conditioner Installed In Houston, Texas (Note: *Supply Air Temperature* & *Evaporator "off" Temperature* thermocouples only yield valid signals while indoor blower operates.)

This installation has been an interesting one to observe due to the interaction of the two systems. The upstairs unit sees a consistent sensible load and runs longer. The homeowner also uses a thermostat setpoint of 22ºC (cool by most people's standards). In this installation, the enhanced dehumidification mode is not activated very often: generally, it operates when the sensible load is very low, as in the above 12-hour sequence.

5 ESTIMATION OF ENERGY IMPACT

The range of systems that we considered in our study (again, Houston climate) is summarized in Fig. 11. At one extreme is a 13 SEER (US NAECA 2006 minimum efficiency) heat pump using electrical resistance reheating when relative humidity is above 50%. We included this system as a point of reference – no one would do this! It has the best (lowest) Seasonal Performance Index, but the energy consumption is almost double that for the 13 SEER heat pump. The high-efficiency two-stage heat pump with variable condenser reheat gives the next best Seasonal Performance Index. The seasonal energy consumption is significantly elevated compared to a high-efficiency two-stage heat pump that does not actively dehumidify. The two systems with mid-range Seasonal Performance Indices are high-efficiency systems that enable overcooling when humidity is sensed to be above setpoint. Energy penalties are modest and there is a marked difference in humidity control, compared to the standard offering. When the performance of all of these systems is plotted on a graph, it is possible to see the non-linear trend in energy consumption resulting from tighter and tighter control on indoor humidity.

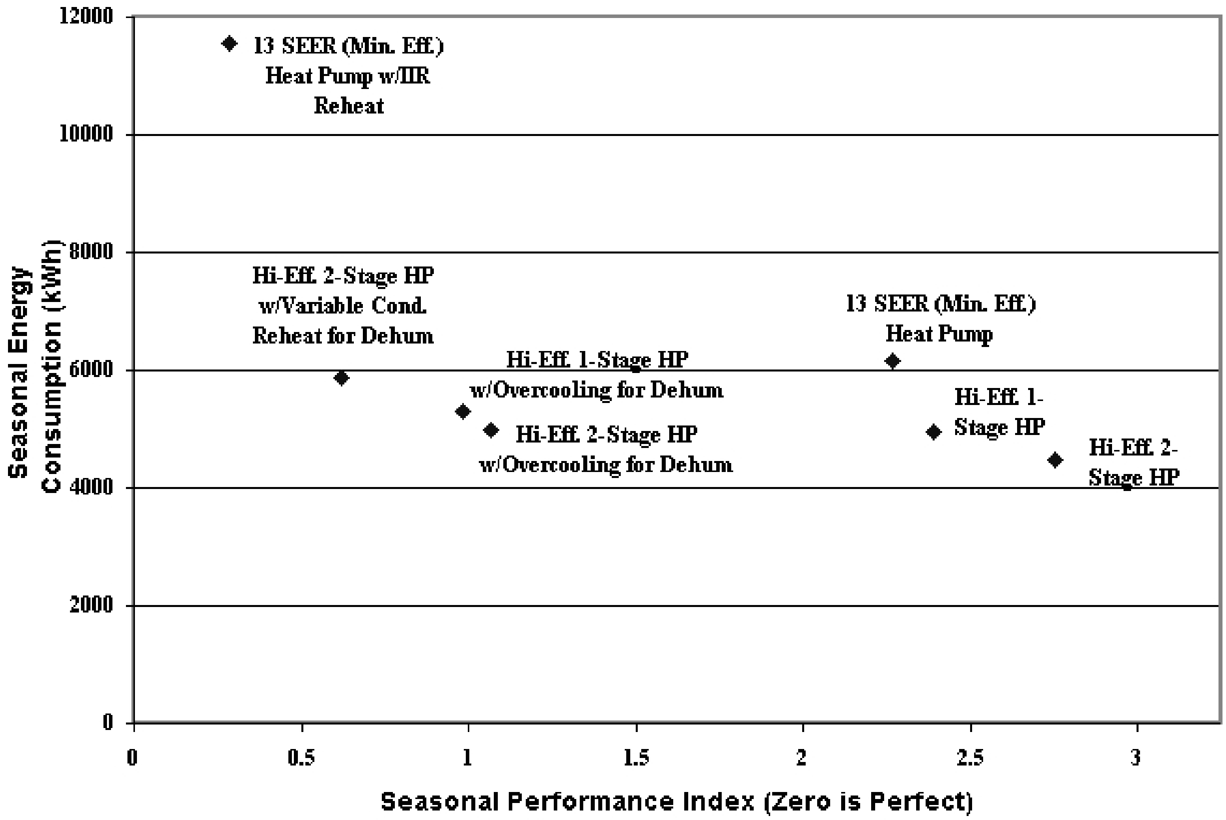

Fig. 11. Energy Consumption vs. Seasonal Performance Index

6 CONCLUSIONS

The variable condenser reheating enhancement to a conventional heat pump provides a new dimension in control of indoor humidity. This system was developed to be easily applied to a range of conventional heat pumps and to be compatible with regionally different heat pump installation customs. It integrates the dehumidification function into the central air circulation system of the home without undue complexity or cost. We have estimated the energy impact of tighter humidity control in a climate (Houston, Texas) that is known for persistent indoor humidity problems. It is possible to provide a greatly improved indoor environment and still use less energy than a 2006 minimum efficiency (US) heat pump. The driver for enhanced humidity control can be comfort and/or the desire to retard unwanted odors, mold and allergen production. This system fits what may be a significant niche.

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