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Field Evaluation of a Residential Hydronic Distribution System in the Cooling Mode

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ABSTRACT

Residential ducts have been characterized by major inadequacies that include heat conduction to unconditioned spaces, leakage, and infiltration through supply and return ducts. These problems are considered to result in substantial amounts of energy usage and poor thermal comfort. In addition, distribution problems contribute to high peak electricity demand. Previous research efforts have demonstrated large potential energy savings from distribution system improvements since more than 50% of existing homes have ducted systems. In an attempt to improve distribution system efficiencies, the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) has undertaken a large research effort to issue Standard 152P, which addresses measurement techniques for determining the efficiency of distribution systems in unconditioned and conditioned spaces. Presently, the standard addresses forced-air distribution systems (heating and cooling) and hydronic heating systems but (with one limited exception) no cooling. However, one of the next steps in the process is to begin incorporating sections into the standard that can address a broad range of hydronic cooling systems. One of the intended purposes of this field test is to incorporate the results into the ASHRAE Standard 152P writing process. The study is also expected to provide information on the space conditioning equipment/hydronic distribution system energy performance and comfort control to establish a baseline performance and determine areas for future research.

Delivery effectiveness and distribution efficiency were measured to compare the performance of the hydronic distribution system to other types of conventional distribution systems. The measured delivery effectiveness values from the field test varied from 81.4% to 92.2% over the range of outdoor temperatures. The distribution efficiency, which includes thermal regain from the distribution losses, varied from 87.5% to 92.5%, meaning that only 12.5% to 7.5% of the cooling effect that left the space conditioning equipment failed to reach the conditioned space. Measurements for indoor temperature and relative humidity (RH) were also made during the testing to determine the ability of the hydronic system to maintain adequate comfort levels. The results showed that the indoor relative humidity was controlled within a range of 46%-52% RH, while the outdoor relative humidity varied from 25% RH to 100% RH. The study also showed that the indoor temperature/relative humidity combinations were well within the limits of comfort as specified by ASHRAE.

INTRODUCTION

In an effort to provide information to aid in developing measurement techniques for determining hydronic distribution system efficiencies for ASHRAE Standard 152P, a field test was undertaken to measure the distribution losses associated with the use of a hydronic system during the cooling season. The information gained from the research effort will also be used to (1) determine the energy efficiency of the space conditioning equipment for comparison with equipment used with other distribution system types, such as conventional forced-air and high-velocity forced-air distribution systems; (2) determine the ability of the hydronic cooling system to maintain the relative humidity and dry-bulb temperature within the comfort range as specified by ASHRAE; and (3) measure the parasitic losses (pump and fan power) at different operating conditions to compare them against the

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fan and blower losses for other types of space conditioning equipment, such as electric heat pumps.

Hydronic distribution systems have been used for many years in residential applications to transfer energy from the HVAC equipment to the conditioned space. The majority of hydronic distribution systems are located in the northeastern U.S. where they are primarily used in convective baseboard and radiant floor applications. It is rare to see hydronic systems used for cooling due to perceived problems with condensate removal and noisy, inadequate fan coil designs. The field test will address these issues by investigating a hydronic distribution system with an improved condensate removal system and advanced fan coil units.

There are two main standards of reporting duct efficiency as outlined in ASHRAE Standard 152P (ASHRAE 1997a). These are *delivery effectiveness* and *distribution efficiency*. Delivery effectiveness is defined as the ratio of the thermal energy transferred to or from the conditioned space to the thermal energy transferred at the equipment/distribution system heat exchanger. While delivery effectiveness is an important measure, it fails to fully represent the fraction of the supplied energy that reaches the conditioned space to satisfy the building load. Distribution efficiency, defined as the ratio between the energy consumption by the equipment if the distribution system had no losses and the energy consumed by the same equipment connected to the distribution system, takes into account the effects of thermal regain, the interaction of unbalanced duct leakage with natural infiltration, and the impact, if any, of the distribution system on the equipment efficiency. Thermal regain accounts for energy lost by the ducts to unconditioned space that is effectively recovered by the building through reduction of losses from the conditioned space to the buffer space due to a temperature change resulting from the duct losses. The interaction of unbalanced duct leakage with natural infiltration changes the building load by either pressurizing or depressurizing the building. This, in turn, results in reducing or increasing the amount of energy that must be supplied by the space conditioning equipment to satisfy the building load (Francisco et al. 1998).

BACKGROUND

The delivery effectiveness of forced-air heating and cooling systems is greatly affected by the type of distribution system and its location, e.g., attic, crawl space, or basement. In 1983, 49% of existing residential heating and cooling systems in U.S. households relied on forced-air ducts to supply conditioned air to the building (Andrews and Modera 1991). However, the percentage of homes with ducts is increasing as indicated by more recent information that shows approximately 96% of new construction uses ducted distribution systems (NAHB 1999). Some of the drawbacks of ducted systems are that they require large amounts of space, tend to be noisy, are extremely prone to leakage, and can result in maldistribution of air and large infiltration losses. In addition, dust collection and the growth of mold and mildew inside ducts can cause indoor air quality problems (Kesselring 1993).

Estimates for energy losses for ducts in unconditioned and partially conditioned spaces are 35% and 20%, respectively (Gupta et al. 1995). Further, the problems in ducted distribution systems may contribute to high peak electricity demands. Air leakage from ducts contributes to thermal energy loss and may lead to pressure differences that could cause pollutants such as radon to infiltrate the conditioned space. Hydronic distribution systems are a means of reducing the energy losses associated with forced-air ducted systems and improving the delivery effectiveness of a building's HVAC system (Sarkisian et al. 1990). A hydronic system requires considerably less hydraulic distribution energy than a forced-air system since the working fluid is incompressible. In addition, there are no leaks in hydronic systems to contribute to increased infiltration and thermal losses. The estimated potential national energy savings from all energy sources for reducing distribution losses associated with ducted systems by improving their efficiency to that of a hydronic distribution system is 1.33 quadrillion Btu (Baskin et al.).

Hydronic heating and cooling is also viewed as a technology that could increase the market share for zone control and thus reduce the energy consumption of the system even further by providing conditioned air only to the spaces as needed. Hydronic distribution systems have individual room control with quick response to thermostat settings and freedom from recirculation of air from other conditioned space. Present zone control techniques for forced-air systems utilize dampers that are cumbersome, unreliable, and costly. With individual fan-coil units in each room, zone control is more easily accomplished, resulting in better air distribution and improved comfort throughout the room.

METHODOLOGY

The house used in the field test was a 4300 ft^2 (400 m^2), two-story home with a basement located in Newark, New Jersey. The majority of the distribution piping is located in the basement with additional piping located in an uninsulated, unvented crawl space and a garage. The basement ceiling was insulated to R-11. There were eight fan coil units throughout the house consisting of four 3500 Btu/h (1026 W) capacity units and four 10,000 Btu/h (2931 W) capacity units. The fan coil units are equipped with variable-speed, brushless-dc motors that consume 8.5 W (small fan coil) and 15 W (large fan coil) at the highest speed. The speed is controlled by the thermostat setting. Air enters from the bottom of the fan coil unit and exits though the top of the unit. The distribution piping is 1-inch (2.5 cm) CPVC with R-2.75 closed-cell foam insulation. Condensate is removed from the fan coil units by means of a PVC drain pipe that drains to a common PVC line piped to a drain. The chiller is a 48,000 Btu/h (14,069 W) unit equipped with two variable-speed condenser fans and a pump to circulate the chilled water throughout the distribution system. The chiller also employs a 20-gallon insulated storage

tank to reduce cycling of the compressor during periods of low ambient temperatures.

The distribution system, chiller, fan coil units, and house were instrumented to determine distribution losses, equipment efficiencies, and indoor and outdoor ambient conditions. The distribution system losses for the fan coil units with pipes located in the basement were determined by first measuring the distribution loss to the farthest fan coil unit in the system from the flow rate and temperature difference. Next, the distribution losses for the other fan coil units were proportioned to the loss associated with the farthest coil according to their flow rates and piping lengths. Distribution losses for the fan coil units with pipes located in an uninsulated, unvented crawl space and a garage were calculated in a similar manner as for the farthest piping run. Water flow rates for each fan coil unit were measured using high-precision turbine flow meters with an accuracy of ±1%. Temperature measurements were made from thermocouples inserted into the piping. The thermal energy input by the chiller in Btu/h (W) was measured with a turbine flow meter and thermocouples at the inlet and exit to the chiller. Condenser fan, fan coil unit, and pump power was measured with watt transducers to determine the parasitic losses. The fan coil units were also instrumented with RTDs at the air inlet and outlet. Using information from laboratory studies of the air flow rate as a function of energy consumption for the fan coil units, the inlet and outlet temperature measurements, together with the power draw, enabled a calculation of the sensible capacity for each fan coil unit. Indoor and outdoor temperature and humidity measurements were made at a single point inside and outside the house. The indoor sensor was placed in a centrally located two-story foyer. The outdoor sensor was placed in a shaded area and shielded from rain. The outdoor ambient conditions were compared against airport weather data, where the airport was within 8 miles of the home. Data were taken at 30-second intervals and summed over a three-hour period.

EXPERIMENTAL PLAN

Since hydronic cooling systems are not presently included in ASHRAE Standard 152P, measurement techniques for determining the efficiency of the distribution system are undefined. In determining the instrumentation requirements for the field test, plans concentrated on the ability to calculate the measures of efficiency that are similar to those for forced-air systems, such as delivery effectiveness and distribution efficiency. This would enable comparisons to the results for forced-air distribution systems and aid in future revisions to ASHRAE Standard 152P. In addition to the efficiency of the distribution system, instrumentation was also included on the chiller to measure the energy efficiency ratio (EER), which is a standard basis for comparison used in rating the performance of forced-air space conditioning equipment.

Delivery Effectiveness

The delivery effectiveness is defined in ASHRAE Standard 152P as the ratio of the thermal energy transferred to or from the conditioned space to the thermal energy transferred at the equipment-distribution system heat exchanger. Accurate field measurements of the energy transferred at each fan coil unit were quite difficult given the uncertainty associated with airflow rate measurements and relative humidity measurements across the coil. In order to more accurately determine the delivery effectiveness, the approach taken was to calculate the distribution losses and subtract them from the energy supplied by the chiller. The distribution losses for the hydronic system are inherently more accurate since they only involve temperature and water flow measurements, which are more accurate than airflow measurements. The distribution loss was calculated for each length of piping in the basement from a common manifold to each fan coil unit and back. As previously explained, the distribution losses associated with the fan coil units with piping in the basement were measured for the farthest run and then proportioned for the other units according to flow rate and piping length. The losses in the farthest run were determined from the following equation:

$$Q_{LOSS} = K \times V_{DIST} \times C_p \times (T_{OUT} - T_{IN}) \qquad (1)$$

where Q_{LOSS} is the energy loss in Btu/h (W), K is a conversion constant (500.7 lb_m·min/gal·h [1000 kg·J/L·kJ]), V_{DIST} is the water flow rate through the pipe in gal/min (L/s), C_p is the specific heat of water in Btu/lb_m·°F (kJ/kg·K), T_{OUT} is the outlet water temperature in °F (°C) measured at the end of the piping prior to the fan coil unit, and T_{IN} is the inlet water temperature in °F (°C) from the manifold.

In addition to the piping in the basement, there were two additional fan coil units with piping runs through an unvented, uninsulated crawl space and a garage. These were also determined using Equation 1. The distribution losses for all the piping were then summed up to determine the total distribution losses for the system. Next, the energy provided by the space conditioning equipment (chiller) was determined from the following equation:

$$Q_{CHILLER} = K \times V_{CHILLER} \times C_p \times (T_{IN} - T_{OUT}) \qquad (2)$$

where $Q_{CHILLER}$ is the energy provided by the space conditioning equipment in Btu/h (W), K is a conversion constant (500.7 lb_m·min/gal·h [1000 kg·J/L·kJ]), $V_{CHILLER}$ is the water flow rate through the chiller in gal/min (L/s), C_p is the specific heat of water in Btu/lb_m·°F (kJ/kg·K), T_{IN} is the inlet water temperature to the chiller in °F (°C), and T_{OUT} is the outlet water temperature from the chiller in °F (°C).

The delivery effectiveness was then determined by subtracting the total distribution system losses (as summed from the losses for all the piping in basement, crawl space, and garage) from the total energy provided by the chiller (Equation 2) and dividing by the total energy provided by the chiller (Equation 2) as shown in the following equation:

$$DE = (Q_{CHILLER} - \Sigma\, Q_{LOSS})/Q_{CHILLER} \quad (3)$$

where *DE* is the delivery effectiveness in percent. The energy terms in Equation 3 are in Btu/h (W).

Distribution Efficiency

The distribution efficiency takes into account the portion of the thermal loss (gain) for the distribution system that is recaptured by the building to reduce/increase the load, depending on its location, e.g., basement, crawl space, etc. Thermal regain factors for ducts in different locations are shown in Table 1 (ASHRAE 1997a). The distribution efficiency for the hydronic distribution system was determined from the following equation:

$$\eta_{DIST} = DE + ((\Sigma\, (Q_{LOSS} \times F_{REGAIN}))/Q_{CHILLER}) \quad (4)$$

where η_{DIST} is the distribution efficiency in percent, *DE* is the delivery effectiveness in percent calculated from Equation 3, Q_{LOSS} is the distribution loss for each section of piping in Btu/h (W), F_{REGAIN} is the thermal regain factor for different locations as defined in Table 1, and $Q_{CHILLER}$ is the energy provided by the chiller in Btu/h (W) calculated from Equation 2. As shown in Equation 4, the distribution losses in each of the different pipe locations (crawl space, basement, garage) times their respective thermal regain factors are summed and divided by the chiller power, then added to the delivery effectiveness to give the total distribution system efficiency.

Equipment Efficiency

In addition to efficiency measurements for the distribution system, measurements were also made to determine the efficiency of the space conditioning equipment. Typically, equipment efficiencies are reported in terms of the EER, which is simply the energy supplied by the equipment in Btu/h divided by the input power in watts. For the chiller, the energy supplied was determined from Equation 2 and the total input power was measured for the compressor, condenser fans, fan coil units, and pump.

EXPERIMENTAL RESULTS

The field test is designed to assess the performance of the hydronic distribution system and space conditioning equipment under a variety of outside ambient temperatures and relative humidities. Of particular interest is the performance of the hydronic distribution system under the following conditions: (1) low ambient temperatures/high humidities, (2) high ambient temperatures, and (3) low ambient temperatures, regardless of humidity. Forced-air distribution systems can experience problems with maintaining comfort at low ambient temperatures with high humidities. This problem occurs when the space conditioning equipment does not run long enough to adequately dehumidify the air in the building. High ambient temperatures result in a reduced distribution efficiency for forced-air systems in unconditioned space due to high conduction losses from the duct to the ambient air. Finally, low ambient temperatures result in the parasitic losses in hydronic systems becoming a larger portion of the overall power consumption and thus reducing the equipment EER.

The testing occurred over a period from late July to early September. Two modes of testing were performed. The first mode, which is the one reported in this study, included running all the fan coil units and opening all the doors to the rooms. In the second mode, which will be reported in a later study, some of the fan coil units were turned off and doors to the rooms were shut to simulate zoning.

TABLE 1
Thermal Regain Factors

Location	Thermal Regain Factor
Garage	0.05
Insulated-ceiling basement	0.30
Crawl space, unvented, uninsulated	0.60

Delivery Effectiveness

Delivery effectiveness is plotted in Figure 1 as a function of the outdoor air temperature. For the test period, the outdoor ambient temperature varied from 57.2°F (14.0°C) to 91.9°F (33.3°C). As shown in Figure 1, the delivery effectiveness varied from a high of approximately 92.2% to a low of 81.4%. In general, the delivery effectiveness trended lower as the outdoor temperature increased. This reduction in delivery effectiveness occurs as the result of the increased conduction losses (distribution losses) from the higher temperature difference between the water temperature in the pipe and the ambient air temperature surrounding the pipe. As shown in Figure 2, these distribution losses for the piping in the basement, garage, and crawl space increase from approximately

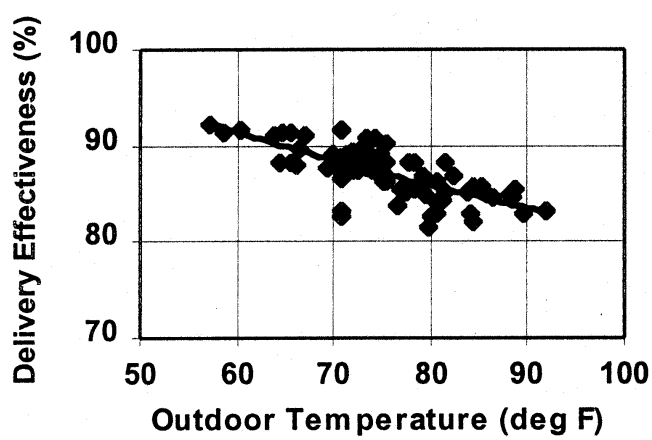

Figure 1 *Delivery effectiveness vs. outdoor temperature.*

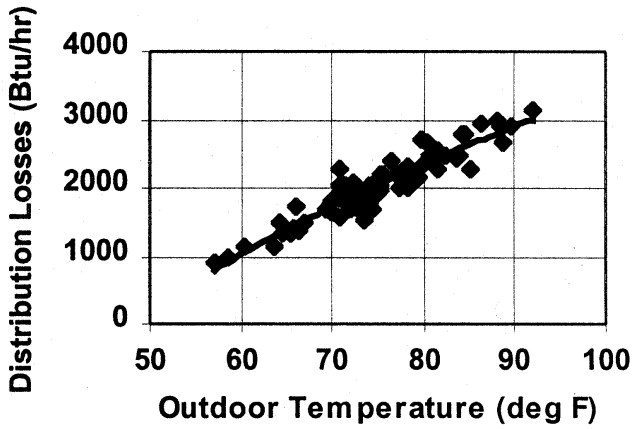

Figure 2 *Distribution losses vs. outdoor temperature.*

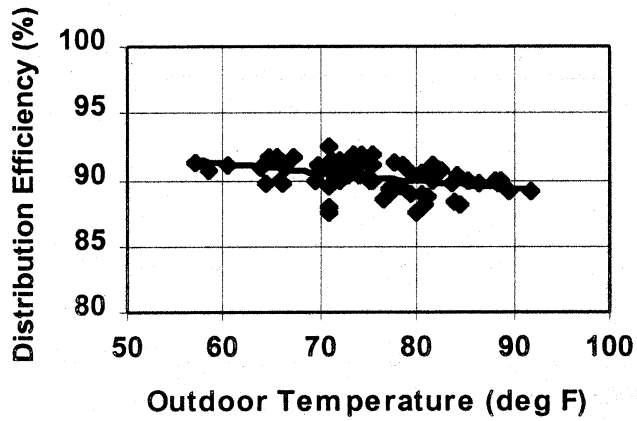

Figure 3 *Distribution efficiency vs. outdoor temperature.*

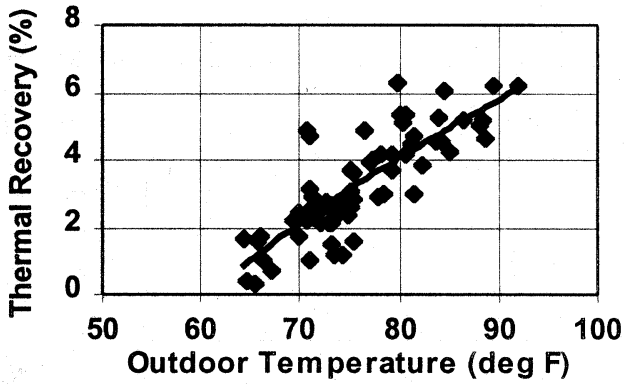

Figure 4 *Thermal recovery vs. outdoor temperature.*

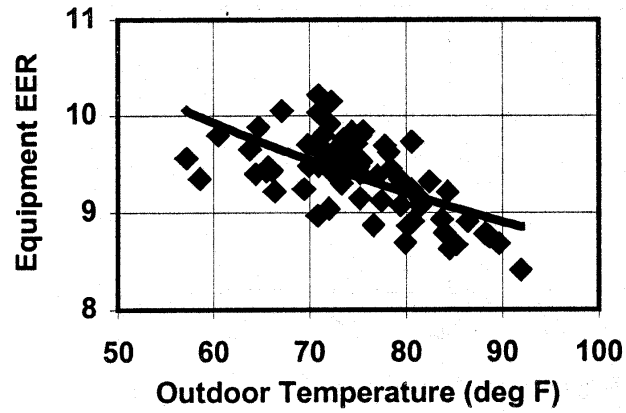

Figure 5 *Equipment EER vs. outdoor temperature.*

900 Btu/h to 3100 Btu/h (264 W to 909 W) as the outdoor air temperature increases.

Distribution Efficiency

The distribution efficiency is plotted in Figure 3 as a function of outdoor air temperature. The distribution efficiency ranged from a high of approximately 92.5% to a low of 87.5%. As in the case for the delivery efficiency, the distribution efficiency also trended lower with increasing outdoor air temperatures. There was a total of 479 ft (146 m) of distribution system piping in the house. The majority of the distribution system piping (74.9%) was located in a basement with an insulated ceiling. From Table 1, the thermal regain factor for this application is 0.30. The remaining distribution losses that were partially recovered were in an uninsulated, unvented crawl space (0.60 thermal regain factor), which accounted for approximately 4.2% of the distribution piping length, and in a garage (0.05 thermal regain factor), where 20.9% of the total distribution piping was located. Comparing the trend lines for the delivery effectiveness and the distribution efficiency, the distribution efficiency decreases at a much lower rate than the delivery effectiveness as the outdoor air temperature increases. This is the result of the second term in Equation 4, the thermal recovery from the distribution losses divided by the chiller energy, increasing as the outdoor temperature increases, as shown in Figure 4. This increased thermal recovery occurs because the distribution losses in the crawl space are increasing at a much higher rate than the losses in the basement and garage as outdoor temperature increases. Compounding this effect is the fact that the regain factor for the crawl space is much higher (0.60) than that for the basement (0.30) or garage (0.05). At ambient temperatures less than 70°F (21°C), the thermal recovery accounts for less than 2% of the total distribution efficiency. As the outdoor temperature increases above 70°F (21°C), the thermal recovery increases to as much as 6% of the total distribution efficiency.

Equipment Efficiency

The EER for the chiller was measured to enable a comparison against other types of space conditioning equipment that is used in residential distribution systems, such as electric heat pumps. Electric heat pumps are covered under the National Appliance Energy Conservation Act (NAECA 1987) and are required to meet certain minimum efficiency levels. The typi-

cal rating point for electric heat pumps operating in the cooling mode is 82.0°F (27.8°C). At that temperature, the minimum EER for electric heat pumps is 10.0 as mandated by the U.S. Department of Energy (DOE). Although hydronic heat pumps and chillers are not covered under the DOE standard, it is useful to compare their performance against air-source equipment to evaluate the overall performance of the equipment/distribution system. The equipment EER as a function of outdoor temperature for the chiller is shown in Figure 5. The EER trends from a high of approximately 10.2 down to 8.4 at an ambient temperature of 91.9°F (33.3°C). At 82.0°F (27.8°C), the chiller EER is approximately 9.2 as estimated from the trend line in Figure 5. Thus, its EER is 8.0% below the minimum efficiency for air-source space conditioning equipment. The lower EER is mainly due to the lower evaporating temperature at which the chiller must operate in order to chill the water down to 45.0°F (7.2°C). Typical entering air temperatures for air-source equipment are approximately 55.0°F-60.0°F (12.8°C-15.6°C), which result in reduced compressor power requirements as compared to the hydronic chiller.

Parasitic losses associated with the condenser fan, fan coil units, and circulating pump have been a concern for hydronic cooling systems. The circulating pump is the main source of parasitic power loss since it operates continuously. For the hydronic system tested, the circulating pump energy consumption was 243 W. By comparison, the total energy consumption of the condenser fans and the fan coil units ranged between 49 W and 134 W, depending on the compressor run time. Figure 6 is a plot of the parasitic losses as a function of compressor run time. The parasitic losses are reported as a percentage of the overall energy consumption of the chiller and fan coil units. As shown in Figure 6, the parasitic losses decrease with increasing compressor run time. The losses, ranging from 16% to 24%, are comparable to those for the blower and condenser fan in an electric heat pump. Thus, even with the circulating pump running continuously, the parasitic losses are not too high. One improvement would be to replace the existing pump motor with a brushless-dc or some other high-efficiency motor to further reduce the parasitic losses.

Comfort

Although energy consumption is the main focus of this study, of secondary importance is the ability of the hydronic system to maintain comfort in the conditioned space. In Figure 7, the indoor relative humidity is plotted as a function of the outdoor relative humidity. Figure 7 shows that the indoor relative humidity remains fairly constant (46%-52% RH) over a wide range of outdoor relative humidity levels (25%-100% RH). Thus, the hydronic system is able to maintain comfortable indoor relative humidity levels over a wide range of outdoor relative humidity levels.

To further investigate the ability of the hydronic distribution system to maintain comfort, the indoor relative humidity

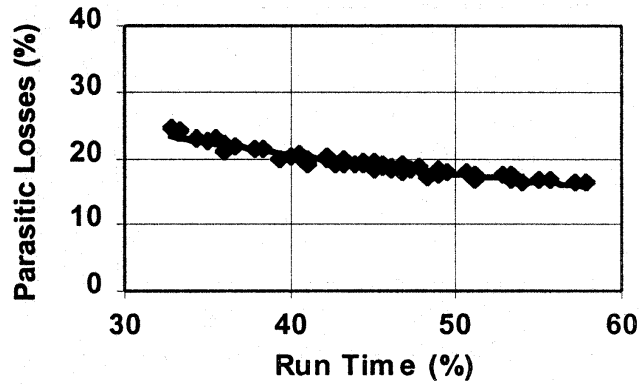

Figure 6 *Parasitic losses vs. run time.*

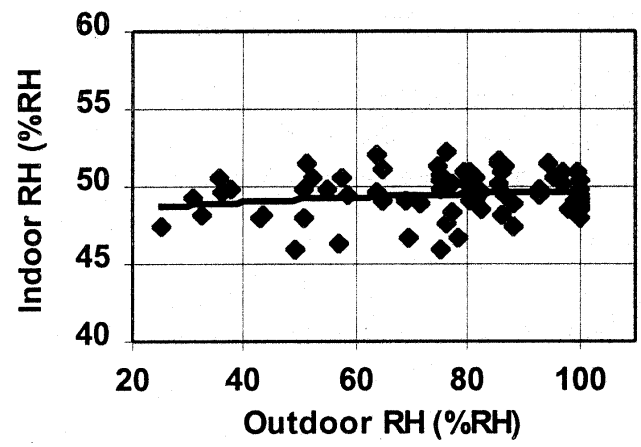

Figure 7 *Indoor RH vs. outdoor RH.*

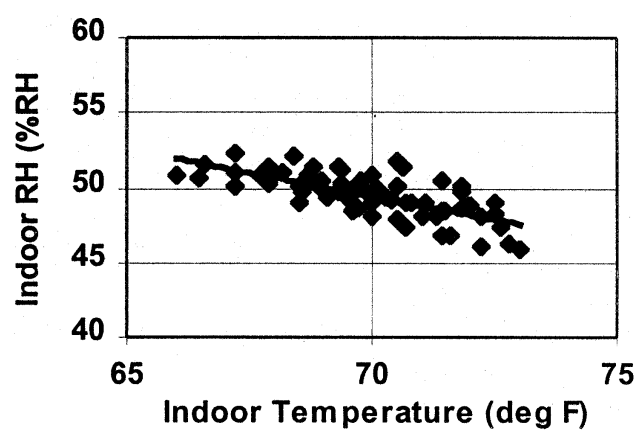

Figure 8 *Indoor RH vs. indoor temperature.*

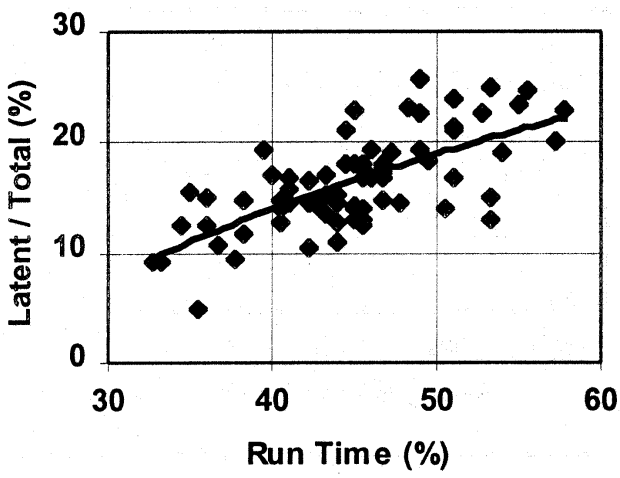

Figure 9 *Latent/total vs. run time.*

was plotted as a function of indoor dry-bulb temperature in Figure 8. Using the ASHRAE comfort zone conditions for temperature and humidity, the conditions in the home are maintained in the middle of the humidity range and at the low end of the temperature range (ASHRAE 1997b). The temperatures actually fall within the winter comfort conditions, which suggests that the indoor temperatures could be increased to save energy and comfort would still be maintained.

The slope of the trend line in Figure 8 shows that as the indoor temperature increases, the relative humidity is decreasing. From the ASHRAE comfort zone conditions, this is exactly the way a system should control humidity as the outdoor temperature increases in order to reduce energy consumption and still maintain comfort (ASHRAE 1997b). In order to investigate how the hydronic system was able to reduce the relative humidity as indoor temperature increased, a plot (Figure 9) was made of the latent/total capacity of the system as a function of percent run time for the space conditioning equipment. Figure 9 shows that as the run time increased (indicating that the building load is increasing), the latent portion of the total load is also increasing. This indicates that the fan coils are removing more moisture as the system runs longer to meet the building load.

CONCLUSIONS

Several significant findings were concluded from the field test. It was determined that the distribution efficiency of the hydronic system was quite high, ranging from 87.5% to 92.5%. Typical distribution system efficiencies for residential construction are in the range of 60%-70% (Modera 1993). Thus, the hydronic system is a significant improvement over conventional forced-air distribution systems and would result in large energy savings since the space conditioning equipment would have to provide less energy to meet the building load.

One of the major criticisms for hydronic systems has been the parasitic losses associated with running the circulation pump continuously and having several fans running in the individual fan coil units. The fan coil units and the condenser fans in this study were equipped with variable-speed, brushless-dc motors that enabled the energy consumption to be greatly reduced. Even though the circulation pump was still allowed to run continuously, the high-efficiency fans enabled the parasitic losses to be reduced to a level comparable to those for forced-air space conditioning equipment operating with conventional motors. Further reductions in parasitic losses should concentrate on using high-efficiency motors for the pump or control strategies for cycling the pump with the compressor. However, cycling the pump could result in higher relative humidity levels in the building.

Measurements for the chiller EER indicated that there is room for improvement to increase the energy efficiency to levels comparable for forced-air equipment. The chiller EER was approximately 9.2 at the 82.0°F (27.8°C) rating point for forced-air equipment. The value is 8.0% below the minimum 10.0 EER for forced-air equipment. The energy input to the hydronic system includes the compressor, condenser fan, circulation pump, and all the fan coil units. As previously mentioned, the fan motors have already been replaced with high-efficiency models. Thus, the only opportunity for improving the EER is to reduce the energy consumption for the pump, improve the compressor efficiency, or improve the heat exchangers to reduce the temperature difference.

The hydronic system did an excellent job of maintaining comfort in the house, even under outdoor conditions of high humidity and low dry-bulb temperatures. The results showed that the set point temperature could have been raised to further reduce the energy consumption while still maintaining the comfort conditions in the house.

FUTURE PLANS

The results of this study were only for the cooling season. Plans are to convert the chiller to a hydronic heat pump and test the hydronic distribution system in the heating mode. In addition, the circulation pump will be replaced with a higher efficiency unit. The test house is presently equipped with a radiant floor heating system. Therefore, tests will be conducted by switching from one system to the other so that direct comparisons can be made for the energy consumption and comfort levels for each type of distribution system. At some point, measurements will be made of temperature stratification in the rooms to determine how well the fan coil units distribute the air throughout the rooms as opposed to the radiant floor heating system.

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