Fresh Air Moisture Control with a Water-Source Heat Pump and Enthalpy Recovery System

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ABSTRACT

A nominal 2-ton capacity ground-source heat pump was tested in cooling mode with and without fresh air to characterize its moisture removal and cooling capabilities in both cases. Tests were first performed with the heat pump only, to characterize its performance in accordance with ARI Standard 330*, and later, in the cases using fresh air, tests were performed with and without an enthalpy exchange unit. Fresh air was supplied at a humid state above that of the conditioned air and in accordance with the guidelines of* ASHRAE Standard 62-1999*. Without fresh air, under ARI conditions, the heat pump performed in line with expectations. At the cooling standard rating condition—77°F fluid temperature and 80/67°F (dry bulb/wet bulb) return air state—the measured air-side capacity was 32,815 Btu/h with a condensation rate of 9.76 lb/h. Following ARI testing, fresh air at temperatures between 55ºF and 90°F and at 90% RH was introduced at a rate of 320 cfm, with and without the enthalpy exchange system. The heat pump was tested in cooling mode with two indoor air states—ARI and 70°F/50% RH. Rates of moisture introduction from fresh air, moisture removal from the enthalpy exchange wheel, and moisture removal by the heat pump were all monitored, as were sensible transfers of heat in both cooling and heating modes. Resulting from this study is a characterization of the moisture removal and sensible cooling abilities of the combined system working in a variety of modes under a variety of indoor and fresh air conditions. With information gained from this parametric study, coupled with knowledge of the thermal loading characteristics of the conditioned space, a designer is better able to match equipment capabilities with desired classroom conditions.*

INTRODUCTION

Background

The impetus for the work covered in this report is twofold. First, a ground (water) source heat pump (of nominal 2-ton capacity) was tested under a variety of conditions to characterize its heating and cooling abilities. This type of unit could be used in a school classroom, where the required capacity would depend on the size of the classroom and its load characteristics. These tests were performed without fresh air as an expanded set of the ARI test standard for water-source heat pumps. Second, the heat pump was supplied with ~300 cfm of fresh air, with and without heat recovery, and tested under a variety of fresh air conditions. Through such testing, the capabilities of the equipment—that is, the sensible and latent capacities—have been characterized. This knowledge of the equipment, combined with building and climate details, will allow a designer to more accurately match cooling and heating systems with buildings requiring fresh air.

Ground-source heat pumps have been shown to be very effective devices for heating and cooling buildings. The inherent tendency of the ground to maintain moderate temperatures throughout the extremes of the winter and summer makes it an effective heat sink for heat pump equipment. Compared with air-source heat pumps, ground-source units maintain capacity and efficiency during extreme-temperature days in both the heating and cooling seasons. In the summer, this characteristic gives them superior latent heat removal capabilities, making them better equipped to handle large dehumidification loads.

ASHRAE Standard 62-1999, *Ventilation for Acceptable Indoor Air Quality,* outlines the minimum levels of fresh air that are required in a variety of occupied spaces to minimize

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adverse health effects related to stale air. The standard does not address thermal comfort or energy efficiency. According to Standard 62, ventilation may be performed through natural means—for example, open windows—or mechanical means. In either case, when there is a large disparity between the temperature or humidity conditions of the fresh air and the desired conditions for the indoor space, large amounts of energy may be required to condition the fresh air.

There are several techniques for preconditioning fresh air (cooling it in the summer and heating it in the winter) that center around exchanging heat between the fresh air and the exhaust air. Since air contains both noncondensing gases (e.g., nitrogen, oxygen) and condensable water vapor, the energy exchange between airstreams may be either sensible, latent, or both. Heat wheels, as they are known in the heating, ventilating, and air-conditioning industry, are wheels with a sealed, baffled design that rotate between two airstreams, absorbing heat from one and delivering it to the other. A desiccant wheel is coated with a water-adsorbing material, such as silica gel, and transfers moisture from one air stream to another. Finally, an energy wheel—or enthalpy wheel—is a baffled wheel coated with desiccant material that transfers both moisture and sensible heat between airstreams.

Humidity Dependence of Airborne Contaminants

The moisture level of indoor conditioned air affects both the comfort and the health of occupants. Air that is too dry (relative humidity [RH] below ~30%) causes dryness of the skin and rapid drying of the mucus membranes that has been linked to an increase in the susceptibility to colds. Overly humid air (RH greater than ~70%) decreases the effectiveness of the body's sweat-based evaporative cooling system and may lead to problems such as heat stroke or heat exhaustion. When humidity in a building is either too low or too high, the comfort of the occupants is compromised, leading in many cases to shorter attention spans, lower productivity, and increased irritability.

In addition to the tangible comfort problems associated with extreme humidity levels, many airborne contaminants—bacteria, viruses, fungi, and mites—are more abundant at humidity extremes (Sterling 1985). Many types of airborne bacteria and viruses increase in abundance in air with RH conditions of below 30% and above 70% and show a marked decrease around 50% RH. In addition, dust mites are not abundant below 50% RH and proliferate rapidly above 50% RH. Fungi and mildew generally do not appear in air below 80% RH but flourish in the 90–95% range. All of these airborne pathogens are believed to cause a wide range of adverse reactions in people, although the details of the biological mechanisms are not necessarily entirely understood. Considering these factors, it is generally recommended that winter humidity levels be maintained above 30% and that summer levels be held below 70%.

PRELIMINARY ANALYSIS

Cooling Demands

The southeastern United States has a notoriously humid climate, particularly from May to September. In such a climate, the exchange of fresh outside air for stale inside air—as outlined in ASHRAE Standard 62-1999—can, in many circumstances, create huge and often unaccounted-for loads that must be dealt with by cooling systems. For school classroom settings, Standard 62 recommends 15 cfm of fresh air per person.

With some simple assumptions, the degree to which fresh air adds to the air-conditioning requirements of a school classroom is easily illustrated. A classroom filled with 20 people will require 300 cfm of fresh air according to Standard 62. If the desired indoor conditions are 80°F and 53% RH, while the outside air conditions (the source of the fresh air) are 95°F and 85%RH, then the respective air enthalpies are 32 Btu/lb (indoor) and 57.5 Btu/lb (outdoor). The fresh air at these conditions, coming into the classroom at 300 cfm, adds a load of approximately 31,000 Btu/h to the classroom air-conditioning requirements. At the same time, an equivalent amount of cooling is effectively being supplied to the atmosphere. Similar, but reverse, problems occur during the heating season when cold fresh air must be heated to the desired indoor condition. The cooling season problems, however, are particularly acute for several reasons: (1) the hottest point of the day occurs during the time when the building typically is occupied—the middle of the afternoon, and (2) the efficiency and cooling capacity of air-source equipment decreases as outdoor temperature increases.

In addition to the enthalpy content of fresh ventilation air, its absolute humidity level is important. It is possible—perhaps on a rainy spring day—for the temperature and enthalpy of the outside air to be lower than that of the conditioned air but for the absolute humidity to be higher. In this situation, and in any situation where the absolute humidity of the fresh air exceeds that of the conditioned air, unless the air is treated, the moisture content of the conditioned air will eventually reach the moisture content of the ventilation air. The moisture content of the fresh air typically exceeds that of the conditioned air in situations when there is a cooling demand. In this case, air-conditioning equipment may be able to handle the additional moisture from fresh air; however, this is a delicate balance, and problems can arise in the direction of both increased cooling load (warmer days) and decreased cooling load (cooler days). Table 1 is a matrix showing the air states of two hypothetical days, a cool, humid spring day and a hot, humid summer day.

The case of the warmer day applies to the hypothetical case at the beginning of this section, where the added demand on cooling equipment was ~31,000 Btu/h for an assumed 20-person classroom. What is less evident is the proportion

TABLE 1
Possible Fresh Air Conditions and Corresponding Cooling Requirements

Cool Spring Day Raining Temperature: 60ºF ϕ: 100% ω: 0.012 (lb H_2O/lb air)	**Desired Room Conditions:** Temperature: 70ºF ϕ: 40-70% ω: 0.0065-0.011	**Notes:** Dehumidification = > necessary Without dehumidification: Absolute humidity = > 0.012 Relative humidity = > 100%
Hot Summer Day Muggy Temperature: 95ºF ϕ: 80% ω: 0.027	**Desired Room Conditions:** Temperature: 80ºF ϕ: 40-60% ω: 0.0085-0.013	**Notes:** Dehumidification = > necessary Without dehumidification: Absolute humidity = > 0.027 Relative humidity = > 100%

of that load contributed by moisture. Under the conditions cited, 23 lb/h of evaporated moisture are introduced to the conditioned space through the fresh ventilation air. At 970 Btu/lbm of condensation energy for water, the moisture load from fresh air is 22,800 Btu/h, or ~74% of the entire fresh air load. Single-coil air-conditioning systems are generally not equipped to handle such proportions. In this case, an enthalpy recovery system for fresh air not only can provide for substantial energy savings but also can greatly extend the range of fresh air conditions under which the cooling equipment can maintain the desired indoor air state.

The case of the cooler day is a bit different in that while the air-conditioning equipment may be able to handle the necessary moisture condensation easily, there is no load demand for cooling. Indeed, from an enthalpy and temperature standpoint, the fresh air is actually cooling the room. Unless the internal heat loads from people, computers, lights, and such sources are enough to justify operation of the cooling (and moisture-condensing) equipment, then it will remain off, and the fresh air absolute humidity level will dominate. In this case, also, the use of an enthalpy recovery system can enhance the range of fresh air conditions under which the cooling system can maintain the desired indoor air state. However, in cases where there is no cooling load in the conditioned space—with or without enthalpy recovery—the absolute humidity level will drift up toward the level of the fresh air.

Thermodynamics of Heat Recovery

As discussed in the introduction, energy can be exchanged between the fresh and exhaust airstreams as sensible heat, latent heat, or both. In all three cases, there must be a driving potential between the airstreams that causes the transfer. For sensible heat, the potential is the difference in airstream temperatures; and for latent heat, the potential is absolute humidity. When both types of heat transfer are taking

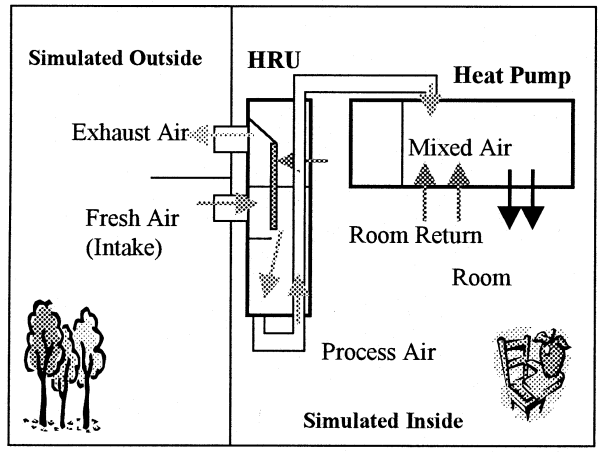

Figure 1 *Schematic of test setup.*

place, the combined potential of temperature and absolute humidity is captured as the single property, enthalpy.

For a total energy recovery system—enthalpy wheel—the energy balance equations follow:

$$Q_f = \varepsilon \cdot m_f \cdot (h_{f,o} - h_{f,i})$$
$$Q_e = \varepsilon \cdot m_e \cdot (h_{e,i} - h_{e,o})$$

and

$$Q_f = -Q_e$$

where

Q = heat transfer to/from the heat exchanger
ε = heat exchanger effectiveness
m = mass flow rate
Cp = specific heat
h = enthalpy

Subscripts

f = fresh
e = exhaust
i = inlet
o = outlet

Referring to Figure 1, the effectiveness of a total enthalpy recovery process is defined as

$$\varepsilon = m_{2\to x} \cdot (h_x - h_2) \div m_{4\to y} \cdot (h_4 - h_2)$$

where the process $2 \to x$ is the lower enthalpy airstream and process $4 \to y$ is the higher enthalpy stream. In the limiting case, where the effectiveness is 100% (ε = 1.0), and the mass flow rates are equal, then

$$h_x = h_y = (h_2 + h_4) \div 2 .$$

TEST SETUP

To test the water-source heat pump, an appropriate, controllable water source was constructed that was capable of producing fluid flow rates and temperatures meeting ARI

TABLE 2
Air and Brine Conditions for ARI Testing

Test type	Brine temperature	Air dry-bulb temperature	Air wet-bulb temperature
1. Standard rating, cooling	77ºF (25ºC)	80ºF (26.7ºC)	67ºF (19.4ºC)
2. Part-load, cooling	70ºF (21.1ºC)	80ºF (26.7ºC)	67ºF (19.4ºC)
3. Maximum operating, cooling	100ºF (37.8ºC)	95ºF (35ºC)	71ºF (21.7ºC)
4. Low-temperature start, cooling	32ºF (0ºC)	80ºF (26.7ºC)	67ºF (19.4ºC)

specifications and heat pump requirements. For ARI testing, saltwater brine was used; for all other tests, water from the municipal water system was used.

The heat pump was nominally rated at 2 tons. In a classroom setting, it requires water connections through ¾ in. female copper fittings that are accessible through the compressor compartment on the left side of the unit. Typically, these connections would feed through the back of the unit and connect, through the wall, to a water-to-ground heat exchange loop. In the laboratory setup, the water connections were made through the front of the unit for ease of connection and instrumentation. No connections would be visible to people in the classroom in an actual setup. The right side of the unit houses dual forward-curved blowers that supply approximately 1100 cfm of airflow across a refrigerant-to-air heat exchanger. Return air from the conditioned space is drawn in at the base of the unit to a mixing chamber, where it may or may not be mixed with fresh ventilation air that can enter through a swinging door on the back side of the chamber. Air from the mixing chamber is then drawn through a filter, through the blowers, through the heat exchanger, and out the top of the unit.

The heat recovery unit (HRU), uses a total enthalpy recovery wheel to exchange energy between an exhaust airstream and an intake airstream. Fresh air is drawn from the rear of the unit through an opening (which would be ducted to the outside in a classroom setup) and through one half of the rotating enthalpy wheel. This semi-conditioned (process) air then travels through the fresh-side blower and is delivered to the fresh air intake of the heat pump. On the exhaust side, air is drawn from the conditioned space through the other side of the rotating enthalpy wheel, passes through the exhaust-side blower, and is rejected out the rear of the unit. Again, this exhaust air would typically be ducted to the outside, with care taken to ensure that it is not drawn back through the intake plenum.

The heat pump and the heat recovery unit were mounted inside a two-room environmental chamber, one side of which simulated indoor (classroom) conditions, and the other simulated outdoor (fresh) air conditions—see Figure 1. Ideally, in a classroom setting, the water-source heat pump and the HRU would attach lengthwise and share a common mixing plenum in the wall space behind them. Because of spatial restrictions in the environmental chamber, the two units were mounted at a 90° angle and ducted together as necessary. In the case of ARI and heat pump characterization testing, the fresh air intake gate in the rear panel of the water-source heat pump was taped shut, and the HRU was not operated, ensuring that all the air entering the mixing chamber came from the conditioned space. When the HRU was included in testing, the swinging gate was opened to the process duct of the HRU, allowing a measured amount of fresh air (unconditioned or semi-conditioned) to enter the mixing chamber. For unconditioned fresh air tests, the HRU enthalpy wheel was switched off so that it stopped rotating and so that after equilibrium was reached with the airstream, the intake air would be unconditioned. For conditioned intake air tests, the enthalpy wheel was left on, and the HRU was allowed to run normally.

Instruments were placed at many points in the water supply system, the heat pump, and the HRU. Temperatures were measured either with type-T thermocouples or 100 W resistance temperature detectors (RTDs). Refrigerant pressure was measured with capacitive type transducers; power was measured with a watt transducer; air flow was measured via differential pressure; relative humidity was measured with thin-film type transducers; and water (brine) mass flow was measured by a timed weight method.

The order of testing was as follows: (1) ARI standard tests (using 10% NaCl brine), (2) WSHP characterization tests, (3) cooling with fresh air with 70°F DB/50% RH conditioned space air, (4) cooling with fresh air with 80°F DB/53% RH conditioned space air.

WATER-SOURCE HEAT PUMP PERFORMANCE AND CHARACTERIZATION TESTING

ARI Testing

ARI standard 330-93 has the purpose of establishing requirements for testing and rating ground-source closed-loop heat pumps with less than 135,000 Btu/h of cooling capacity at ARI standard rating conditions. Four types of tests from this ARI standard were performed for cooling conditions. *Standard rating conditions* (SRC) tests measure the steady-state operation of the heat pump at standard ARI return air conditions. *Part-load rating* (PLR*)* tests also are steady-state tests, similar in nature to the SRC tests, but with a less adverse water temperature. The *maximum operating conditions (MOC)* tests are again steady-state but with extreme water and return air temperatures. Finally, the *low-temperature start and operate* (LTSO*)* tests are performed with cold water and ARI standard rated air. Table 2 outlines the specific requirements for water temperature and return air state for each of the four ARI tests performed. Table 3 contains selected data measurements and

TABLE 3
Selected Cooling Results from ARI Tests

Test Type	Air Capacity	Brine Capacity	Power Consumption	COP	Air Flow	Supply Air Temperature
	Btu/h	Btu/h	W		cfm	ºF
#1 SRC—cooling	32,815	43,166	2,822	3.4	930	57.2
#2 PLR—cooling	34,000	44,377	2,706	3.7	942	56.8
#3 MOC—cooling	31,335	42,534	3,269	2.8	947	65.0

TABLE 4
Selected Results for Parametric Cooling Tests

Brine Temperature	Air Capacity	Brine Capacity	Condensation Rate	Power Consumption	COP	Air Flow	Supply Air Temperature
	Btu/h	Btu/h	lb/h	W		cfm	ºF
50ºF (10ºC)	37,391	46,398	10.8	2,317	4.7	938	55.6
60ºF (15.6º)	35,512	45,487	10.4	2,488	4.2	940	56.3
70ºF (21.1ºC)	33,371	43,945	9.7	2,672	3.7	944	57.9
80ºF (26.7ºC)	32,072	42,526	9.0	2,842	3.3	942	58.5
90ºF (32.2ºC)	30,268	41,180	8.7	3,009	2.9	942	59.2
100ºF (37.8ºC)	30,035	39,340	7.7	3,210	2.7	944	59.6

calculated values for capacity and coefficient of performance (COP) for ARI tests 1, 2, and 3, which are all steady-state tests.

Additional Parametric Tests

Additional characterization testing was performed by changing the brine temperature while holding other variables constant (e.g., air temperature, airflow rate, brine flow rate). The brine temperature was increased in 10º increments from 50°F to 100°F. In each case, the indoor return air conditions were maintained at ARI SRCs. Selected results from the six cooling tests are shown in Table 4.

WATER-SOURCE HEAT PUMP FRESH AIR TESTING

Outline of Testing

Cooling mode testing of the combined WSHP/HRU system was performed in three sets with specified indoor air states and water temperatures. In each of the three testing styles, a baseline trial was performed in which no fresh air was introduced to the mixing chamber of the WSHP; only in the case of cooling with 70°F indoor conditions was this baseline test not a duplicate of an earlier test.

Cooling Test Results

In the cooling tests, interest was focused primarily on the rates of moisture introduction to and exit from the conditioned space and on the rate of moisture condensation by the WSHP. The actual cooling capacity of the WSHP when running with fresh air is not directly measurable because the exact mixed air conditions—temperature and humidity—could not be measured. However, by measuring the supply airflow rate; the fresh airflow rate; and the air states (temperature/RH) of return air, fresh (or semi-conditioned) air, and supply air, the actual cooling capacity is calculated according to the following energy balance:

$$m_p h_p + m_r h_r = m_m h_m$$

where

m = mass flow rate of air
h = enthalpy of air
x_p = process air
x_r = return air
x_m = mixed air

Through the measured value of collected condensate, the latent capacity of the WSHP is known, and the sensible heat ratio is calculated. The rate of moisture introduction to the room is calculated as the multiple of the mass flow rate of fresh air and the humidity ratio of semi-conditioned or fresh air (for

TABLE 5
System Performance at ARI Indoor Conditions and High Humidity Fresh Air Conditions (Cooling Season)

Outdoor Conditions	Fresh Air Moisture Rate	Moisture Captured by HRU	Condensation Energy Rate*	Moisture Exhausted**	Moisture Remainder	Power Use by HRU	Effective HRU EER
ºF/%RH	lbm/h	lbm/h	Btu/h	lbm/h	lbm/h	W	
70/90	21.5	3.1	3,008	14.1	4.3	374	8.0
80/90	28.6	6.2	5,978	14.1	8.3	374	16.0
90/90	37.1	9.9	9,593	14.1	13.1	374	25.7

*Equivalent for captured moisture
**At ARI conditions — 80ºF, 53% RH

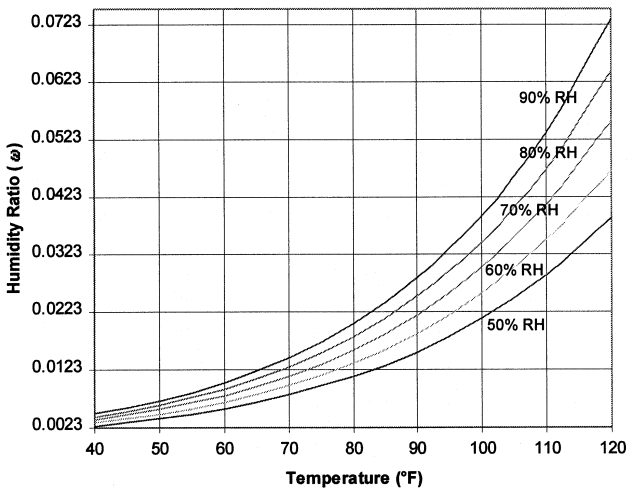

Figure 2a *Humidity ratio vs. temperature and relative humidity.*

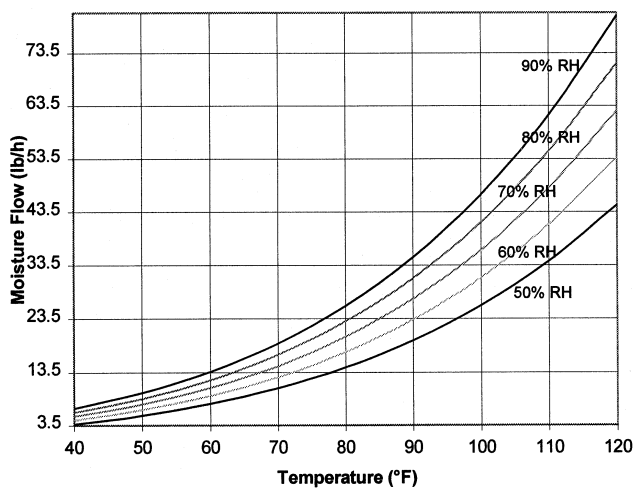

Figure 2b *Moisture flow vs. temperature and relative humidity—for a 300 cfm airflow rate.*

tests with and without the enthalpy wheel operating). The rate of moisture exhausted from the room is equal to the mass flow rate of exhaust air—which is figured by assuming the same volumetric flow rate as the intake air—multiplied by the humidity ratio of conditioned space return air. The net rate of moisture introduction is simply the difference between the rate of introduction and the rate of exhaust. Figure 2a shows humidity ratio as a function of air temperature and humidity, while Figure 2b shows the corresponding moisture flow rate associated with 300 cfm of air. Table 5 presents the results of the combined WSHP/HRU system performance at three humid fresh air conditions, 70°F, 80°F, and 90°F, all at 90% RH and 300 cfm. In each case, the indoor return air state was held at 80°F/53% RH.

Column 4 of Table 5 shows the equivalent condensation energy of the moisture captured by the HRU. This is the amount of latent cooling effect that is saved by the HRU that would otherwise need to be provided for by the air conditioner. The moisture remainder in column 6 is the amount of moisture that must be handled by the air-conditioning coil when incorporating the HRU. It is equivalent to column 2 minus column 3, minus column 5. Without the HRU, the air-conditioning moisture load would equal the *fresh air moisture rate* (column 2) minus the *moisture exhausted* (column 5). The last column, *Effective HRU EER*, is figured by dividing the latent heat of condensation of the captured moisture (column 4) by the power input to the HRU (column 7). It is apparent that as the ventilation air conditions deviate from the room conditions—ARI in this case—the relative effectiveness of the enthalpy recovery system increases greatly. This is rather intuitive since if one were to replace 80°F mildly humid air with 81°F mildly humid air, then enthalpy recovery would scarcely be needed. However, when the enthalpies of the ventilation air and the conditioned air differ greatly—as is the case in the three conditions listed in table 8—then substantial effectiveness can be gained with an enthalpy recovery system. Without an enthalpy recovery system, all the condensation energy must be absorbed by a cooling system, which generally does not exceed 10-12 EER.

DISCUSSION OF FRESH AIR TESTING

One way to interpret these results is to observe the rates of moisture introduction and removal. If a room is to be maintained at a certain condition, then the rates of moisture intro-

duction and removal must be equal. As the outdoor (fresh air) conditions get warmer with the same RH, the absolute humidity increases, and so does the rate of moisture introduction to the room. Also, as the indoor air condition warms (at an approximately constant RH), the rate of moisture exhausted increases, as does the capacity and rate of condensation of the heat pump. Without enthalpy exchange, the rate of moisture introduction quickly becomes a problem when outdoor conditions are more humid than indoor conditions. Since the only moisture-removing mechanism is the heat pump (air conditioner), any humidity brought into the room in excess of what is being exhausted—the net rate of moisture introduction—must be removed by the heat pump.

Hypothetical Classroom Case

Consider a hypothetical case, a classroom of 30 people in New Orleans, that is to be maintained at 75°F and 53% RH. From ASHRAE climactic data, the 0.4% extreme humid condition in New Orleans is 90°F DB and 81°WB. Assume the classroom nominally requires a two-ton cooling load at such conditions, without fresh air. As well, assuming a moisture load per person in the classroom to be 0.2 lb/h, the internal moisture load (if other sources are neglected) will be ~6.0 lb/h. In this case—without fresh air—a two–ton air conditioner could handle this load from both a sensible and a latent standpoint. Complications arise when fresh ventilation air at 15 cfm per person is introduced. The 450 cfm of total fresh air brings with it a sensible heat load of 7,041 Btu/h and a latent load of 21,249 Btu/h or 21.9 pounds of water vapor. This together more than doubles the total cooling load to 52,290 Btu/h, and more than triples the latent load to 27.9 lb/h (27,063 Btu/h). Effectively, the addition of fresh ventilation air has increased the required cooling system size from ~2 tons to 4 or 4.5 tons.

If an enthalpy recovery system is used to mitigate the cooling load penalty brought on by the 450 cfm of fresh air, a very large savings in energy consumption is possible, as is a substantial reduction in the required size of the cooling equipment. If the enthalpy exchange system is able to recover 40% of the otherwise penalizing heat from the fresh airstream, then the contributions to cooling load will be reduced in turn to 2,957 Btu/h sensible load and 9.15 lb (8875 Btu/h) latent load, reducing the total cooling load by more than 22% to 40,458 Btu/h. As such, the power consumption and the required size of the cooling system both drop by 22%, reducing both the operating cost and the capital cost of the cooling system; there is, however, added capital cost associated with the addition of the enthalpy exchange system.

In general, with knowledge of the load characteristics of the room—for example, number of people, lighting load, computer load—one could gauge when cooling and heating would be called for and determine what room states could be maintained given certain outdoor conditions. It appears that the enthalpy exchange system greatly enhances the flexibility of the heat pump, particularly in cooling mode, and that it substantially reduces the required cooling capacity, in turn lowering the capital cost of the cooling unit and the operating cost of the overall system.

Postscript

After a thorough analysis of the data, it was discovered that moisture and airflow rates around the enthalpy wheel were not always balancing as expected. Further exploration revealed that the HRU process air fan cabinet had reasonably significant air leaks at most joints. Negative pressure created by the fan pulls in air through many of the loosely sealed joints on the cabinet. In addition, on the positive pressure side of the fan, air leaks out into the conditioned space and into the fresh air plenum space. Primarily, these leaks come from conditioned air that acts to dilute the process air after it has been conditioned. Additionally, there may be leaks between the process and exhaust airstreams; these, however, were indiscernible without a more extensive detection technique. Such air leaks affect both the effectiveness of the HRU and the actual flow rate of fresh ventilation air. Because the interest was in characterizing the equipment, the leaks inherent in the structure of the cabinet were allowed to remain.

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