ABSTRACT

This paper presents salient results of a utility-sponsored research project whose major objective was to identify all-electric technical means for energy-efficient independent control of sensible and latent cooling in residential and small commercial buildings, and to assess their technical and economic potential, including utility impact.

INTRODUCTION

Dehumidification has become an increasingly large fraction of the total cooling load in many new buildings, as heat gains through the envelope have been reduced but internal moisture generation and the need for ventilation have remained. At the same time, efforts to gain cooling efficiency in residential-size air conditioners have not in general differentiated between temperature reduction (sensible cooling) and dehumidification (latent cooling), so that an imbalance often exists between the amount of each type of cooling that is needed and what is delivered. Dehumidification does occur in small air-conditioning systems, but it is uncontrolled. This lack of control can result not only in poor thermal comfort and unnecessarily high energy costs, but in extreme cases may result in premature deterioration of building materials.

Utilities are at the same time being faced with increasingly troublesome summer peaking problems. These summer peaks tend to be coincident with maximum air-conditioning loads. The possibility was suggested that by independently controlling temperature and humidity ways might be found to ameliorate the peak electrical loads imposed on utilities by the residential and small commercial air-conditioning sector.

A utility-sponsored research project was therefore initiated, with the major objective to identify all-electric technical means for energy-efficient independent control of sensible and latent cooling in residential and small commercial air-conditioning systems. Several generic all-electric approaches to simultaneous control of sensible and latent cooling were analyzed:

1. Reheat of process air after it leaves the cooling coil. This approach involved some of the sensible cooling, (temperature reduction), thereby increasing the fraction of delivered cooling that is latent. Two energy-efficient approaches were identified: evaporator run-around and the subcooler/reheater (see below).

2. Reducing the temperature of the process air leaving the cooling coil. This allows the air conditioner to condense a greater amount of moisture from the process air than would otherwise be the case.

3. Novel equipment technologies. New approaches were sought that might offer advantages over traditional methods. Several candidate approaches were studied; however, none was found to be attractive for small-building applications.

4. System integration. Synergistic interaction between the equipment and the building can, under the right conditions, be used to improve dehumidification and reduce utility peak loads.

This paper gives a brief overview of the equipment options in the first two categories that were judged to have merit, and then focuses on two system-integration approaches that use both equipment and building characteristics to optimize performance.

EQUIPMENT OPTIONS

The equipment options below were found to be capable of significantly improving dehumidification performance over that of conventional air conditioners with little or no decrease in energy efficiency. In some cases, energy efficiency is expected to be enhanced. Dehumidification was expressed in terms of the sensible heat ratio (SHR), the ratio of the sensible cooling to the total cooling provided by the unit. Low SHR corresponds to good dehumidification.

Evaporator Run-Around using Heat Pipes

This reheat technology (2) uses heat pipes to extract heat from the air entering the indoor cooling coil of the air conditioner. The heat is rejected into the stream of air leaving the indoor coil. This process reduces the sensible cooling capacity of the machine and adds a nearly equivalent amount to the latent cooling. The system is currently being marketed in Florida.

Subcooler/reheater

In this approach, (3) an additional refrigerant-to-air heat exchanger is located in the ductwork.

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The conditioned air leaving the cooling coil is reheated by heat from the subcooler/reheater, while the extraction of this heat subcools the warm condensed liquid refrigerant. This results in a net gain in latent cooling capacity, with a corresponding reduction in sensible cooling.

In addition to the enhanced dehumidification experienced in the cooling mode of an air conditioner, there is also a potential improvement in efficiency in the heating mode, if the concept is used in a reversing heat pump coupled with a lower-than-normal air flow rate. This results in a system that can be retrofitted in houses without ductwork to provide central air conditioning.

Each of the equipment options discussed above can be used in the CH/RP control strategy discussed below, because of their improved dehumidification capabilities.

THE CH/RP CONTROL STRATEGY

Within the general approach of system control, as opposed to equipment modification, a novel control strategy was developed that can make a significant contribution to the reduction of peak cooling loads. We have called this the CH/RP Control Strategy (pronounced "chirp"). The acronym stands for Control Humidity/Reduce Peak. It works as follows. An air conditioner capable of supplying a large fraction of its cooling as dehumidification is used. Any of the technologies discussed above could be applied here. During the
The strategy does depend on the availability of equipment with a lower delivered SHR than is required to maintain the design conditions of 78°F dry bulb and 50% relative humidity. The peak demand, under these conditions is 58°F and the humidity ratio at 0.0103. Because of the low SHR of 0.0103, the equipment is controlled by a dewpoint sensor instead of a thermostat to maintain the humidity ratio within the bounds of the comfort zone.

The tradeoff is therefore as follows. As SHR is reduced, the greater will be the reduction of peak demand, but the shorter will be the time during which this peak demand reduction will be maintained within the limits defined by the comfort zone.

Quantitative Analysis of the CEOF Strategy

The following analysis assumes a peak cooling load of 36,000 Btu/hr, with a load SHR of 0.75, meaning that the sensible cooling load is 36,000 and the latent cooling load is 9,000 Btu/hr. This is the assumed load at design summer conditions of 78°F dry-bulb and humidity ratio = 0.0103. As the indoor temperature is reduced, the sensible cooling load will increase because the temperature difference between the inside and the outside of the house is increasing. This increase may be estimated at 100 Btu/hr for each degree F temperature depression (corresponding to a cooling balance point of 68°F).

Then, if the house is precooled to 72°F and allowed to rise to 79°F during the peak, the average temperature depression is about 3°F, leading to an average sensible load of 37,000 × 3,000 = 30,000 Btu/hr. The load SHR will be slightly higher (0.77) during this strategy's operation because of the slight addition to the sensible cooling load.

We now assume the house to be cooled by an air conditioner with an SHR equal to s, a total cooling capacity of 36,000 Btu/hr, and run at a fractional on-time f. The sensible and latent cooling provided per hour will be

Sensible cooling = 36,000 fs

Latent cooling = 36,000 f (1-s)

The unmet sensible load will be

30,000 - 36,000 fs

and the unmet latent load will be

9,000 - 36,000 f (1-s)

We now find the value of f for which the unmet latent load is equal to zero. Then we use Equation (1) to calculate the unmet sensible load. The latter quantity, divided by the sensible thermal capacitance, gives the rate of rise of the temperature. The rate of rise is then divided into the allowed temperature increase (7°F) to give the time the system remains comfortable. These calculations are as follows:

9,000 - 36,000 f (1-s) = 0

f = 0.25/(1-s)

As an example, suppose the equipment provides an SHR of 0.65, or 0.10 less than the load. Then the fractional on-time is 0.714. The unmet sensible load is 20,000 - 36,000 × 0.714 or 12,290 Btu/hr. If the thermal mass of the house is 10,000 Btu/°F, the rate of temperature rise is then

12,290 Btu/hr ÷ 10,000 Btu/°F = 1.23 °F/hr

The time the house remains in the comfort zone is then 7°F ÷ 1.23°F/hr, or 5.7 hr. Table 1 presents results for other SHR's.

We may also calculate the savings on peak load, assuming a given value for the energy efficiency ratio (EER) of the equipment. This will be the peak power draw (in kw) ÷ EER multiplied by one minus the fractional on-time (1-f). For the above example, assuming an EER of 10, the peak demand reduction will be

16.10 X (1-0.714) = 1.07 kw.
It is also possible to quote a fractional peak demand reduction, independent of the EER, which is defined as the peak demand reduction divided by the full-load peak demand. This is equal simply to 1-f.

In examining Table 1 it should be noted that the tradeoff between peak load reduction and time in the comfort zone is dependent on specific assumptions concerning the peak cooling load and the sensible thermal capacitance of the house. If energy conservation measures are taken to reduce the cooling load (insulation, overhangs, low-emissivity glass, radiant barriers, high-EER equipment) this will lengthen the time within the comfort zone for a given SHR, but it will also reduce the additional kW demand savings available via application of the control-on humidity strategy. For example, if the peak cooling load is two tons, or 24,000 Btu/hr, the design load SHR is still 0.75, and the increase in sensible load due to the precooling is proportional to the peak load (i.e., 667 Btu/hr per degree reduction), then the parameters shown in Table 1 alter their values to those shown in Table 2.

Comparison of Tables 1 and 2 shows that while reduction of the peak cooling load by one-third (from three to two tons) increases the time in the comfort zone and decreases the peak kW demand reduction for a given equipment SHR, if we look at the relation between time in the comfort zone and peak kW demand reduction, with SHR as a dependent variable, we find there is much less difference between the three-ton and two-ton peak load cases. This is shown in Figure 3. From this figure, one sees that the time in the comfort zone that is associated with a given peak load reduction is nearly the same for both cases. A 1 kW peak load reduction is associated with a 6 hr comfort window in the 2-ton case and 5.5 hr for the 3-ton case. What is different is the equipment SHR needed to produce the given peak reduction. For the 2-ton case, an SHR of 0.58 is required, whereas for the 3-ton case, an SHR of 0.66 will suffice.

### Table 1. Comfort Window Parameters for Various Equipment Sensible Heat Ratios (3-Ton Design Cooling Load, 10,000 Btu/hr Sensible Thermal Capacitance)

<table>
<thead>
<tr>
<th>SHR</th>
<th>Fractional</th>
<th>Unmet Sensible Cooling [%/hr]</th>
<th>Rate of Temp. Rise</th>
<th>Time in Comfort Zone [%/hr]</th>
<th>Peak Load Reduction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.70</td>
<td>0.83</td>
<td>9000</td>
<td>0.90</td>
<td>7.8</td>
<td>0.60</td>
</tr>
<tr>
<td>0.65</td>
<td>0.71</td>
<td>13300</td>
<td>1.33</td>
<td>5.3</td>
<td>1.03</td>
</tr>
<tr>
<td>0.60</td>
<td>0.62</td>
<td>16500</td>
<td>1.65</td>
<td>4.2</td>
<td>1.35</td>
</tr>
<tr>
<td>0.55</td>
<td>0.56</td>
<td>19000</td>
<td>1.95</td>
<td>3.7</td>
<td>1.60</td>
</tr>
<tr>
<td>0.50</td>
<td>0.50</td>
<td>21000</td>
<td>2.10</td>
<td>3.3</td>
<td>1.80</td>
</tr>
</tbody>
</table>

### Table 2. Comfort Window Parameters for Various Equipment Sensible Heat Ratios (2-Ton Design Cooling Load, 10,000 Btu/hr Sensible Thermal Capacitance)

<table>
<thead>
<tr>
<th>SHR</th>
<th>Fractional</th>
<th>Unmet Sensible Cooling [%/hr]</th>
<th>Rate of Temp. Rise</th>
<th>Time in Comfort Zone [%/hr]</th>
<th>Peak Load Reduction [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.70</td>
<td>0.83</td>
<td>6000</td>
<td>0.61</td>
<td>11.5</td>
<td>0.60</td>
</tr>
<tr>
<td>0.65</td>
<td>0.71</td>
<td>8900</td>
<td>0.89</td>
<td>7.9</td>
<td>0.69</td>
</tr>
<tr>
<td>0.60</td>
<td>0.62</td>
<td>11000</td>
<td>1.10</td>
<td>6.4</td>
<td>0.90</td>
</tr>
<tr>
<td>0.55</td>
<td>0.56</td>
<td>12700</td>
<td>1.27</td>
<td>5.5</td>
<td>1.07</td>
</tr>
<tr>
<td>0.50</td>
<td>0.50</td>
<td>14000</td>
<td>1.40</td>
<td>5.0</td>
<td>1.20</td>
</tr>
<tr>
<td>0.40</td>
<td>0.42</td>
<td>16000</td>
<td>1.60</td>
<td>4.4</td>
<td>1.60</td>
</tr>
<tr>
<td>0.30</td>
<td>0.36</td>
<td>17500</td>
<td>1.75</td>
<td>4.0</td>
<td>1.54</td>
</tr>
</tbody>
</table>
Figure 3 can be used for sensible thermal capacitance values other than 10,000 Btu/F and for EER values other than 10 Btu/W as follows:
Multiply the desired peak load reduction by the quotient of the thermal capacitance divided by 10,000.
Then draw a vertical line until the curve for the 2-ton or 3-ton peak load is reached. By interpolation, we read off the required SHR. Then draw a horizontal line from either of these points to the vertical axis and read off the number. The time in the comfort zone is this number multiplied by the quotient of the thermal capacitance divided by 10,000.
For example, suppose the EER is 15 and the thermal capacitance is 12,500 Btu/F, and suppose further that the desired peak-load reduction is 1 kW. In this case we would use 1.0 x 15/10 = 1.5 kW as our entry point on the x-axis. Next, we draw a vertical line until the curve for the 2-ton peak load is reached. By interpolation, we read off the required SHR. For the 3-ton load this is 0.57 and for the 2-ton load it is 0.33. Drawing a horizontal line from either of these points intersects at about 4 hours (a bit more for the 3-ton case and a bit less for the 2-ton case). Multiplying 6 hours by the ratio 12,500/10,000 yields 5 hours as the time in the comfort zone for our example.

DEHUMIDIFICATION OF FORCED VENTILATION AIR
This option differs from the others in that it incorporates the design of the building structure as well as of the cooling equipment. The necessary characteristic of the structure is that it: has a very low natural air infiltration rate, 0.1 air changes per hour (ACH). Although ordinary housing typically has infiltration of 0.5 to 1.0 ACH, this rate of 0.1 ACH has been demonstrated in unventilated housing emanating from Scandinavia, (7) and Scandinavian building codes set upper limits in the range 0.15-0.20 ACH (8). There is no reason why it could not be duplicated in the U.S., although it will be many years before such housing comprises a significant portion of our housing stock. Health and comfort demand greater ventilation; this is supplied by a forced ventilation system in which warm air from the outside is passed over the cooling coil before it mixes with the house air. Dehumidification to the point where it is predicted that any remaining cooling demands will be almost entirely sensible. Dehumidification ceases to be a problem. Significant peak load reduction is also expected.

That this option involves the design of the whole house, rather than just the cooling equipment, is at once its major drawback and its main advantage. It is a drawback because it restricts the application to houses designed and constructed in conformity with the strategy. The advantage lies in the optimum performance made possible through the inclusion of all portions of the building system to the design.

It is noted here that this strategy allows the dehumidification of the ventilation air to be done in the most advantageous manner. This air, which is generally more humid than the conditioned room air, is brought over the cooling coil immediately upon entering the house, before it is mixed with the room air. More water can be extracted from it than if it were first mixed with the room air and then brought over the cooling coil, as is done in conventional air conditioning.

The ventilation characteristics used in the study are those found in a study of an energy-efficient Danish house constructed and monitored at Brookhaven National Laboratory (7). This house was found to have a very low air infiltration rate of 0.1 air changes per hour (ACH). To this we have added a forced ventilation system that provides an additional 0.4 ACH, to bring the total ventilation rate to 0.5 ACH. In line with previous recommendations for comfort and health, this forced ventilation air is brought in from the outside through a duct, which distributes the air to the rooms. A second duct extracts an equivalent amount of air to be rejected, maintaining a neutral pressure balance in the house. The reject air is taken from points of accumulation of odors and moisture (kitchen range hood, bathrooms).

At some convenient location the ducts are designed to pass close by one another. At this point, refrigerant-to-air heat-exchange coils are located, one in each duct. These coils form the evaporator and the condenser of a heat pump. In summer, the heat pump serves as an air conditioner/dewhumidifier, with heat and moisture extracted from the incoming ventilation air, and heat rejected to a domestic hot water tank and to the outgoing air. The system is designed such that any remaining cooling demand will be almost entirely sensible.

Recent advances in the design of the dehumidification and air-conditioning system have led to the construction of houses in which the air is dehumidified and then brought over the cooling coil.

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We now analyze the effect on cooling loads of the natural air infiltration rate, providing for forced ventilation to bring the overall ventilation rate up to a minimum standard, and dehumidifying this forced ventilation air before it is mixed with the house air.

The advanced-design house is compared with a conventional house of identical proportions and building envelope thermal characteristics (aside from infiltration). Each house is assumed to be 1500 ft² in area with 8-ft ceilings, for a total air volume of 12,000 ft³. The advanced-design house has a natural infiltration rate of 0.1 ACH and a forced ventilation rate of 0.4 ACH; while the conventional house has natural infiltration only, at a rate V. The value of V is a parameter which is allowed to vary between 0.5 and 1.0 ACH, in order to assess the impact of varying degrees of infiltration in conventional housing. The advanced-design house has the above-described heat pump to recover heat and provide cooling and dehumidification. The conventional house, having no forced ventilation system, does not have the heat pump, either.

Assumed Parameter Values. The sensible cooling load is composed of internal, transmission, and infiltration components. The latent load is composed of internal and infiltration components only (diffusion of water vapor through a typical building envelope being negligible).

The conventional house is characterized by the following parameters:

1. Transmission and internal sensible heat gain as a function of outdoor temperature. Here two of the above components are lumped together in a single function (see below).
2. Internal latent heat gain. This was assumed to equal 1000 Btu/hr, equivalent to 23 pints of moisture per day (9).
3. Number of air changes per hour. Knowing the outdoor air conditions, this provides the infiltration sensible and latent loads. Varied between 0.5 and 1.0 ACH.

The advanced-design house requires two additional parameters to characterize the effect of the heat pump, and forced and natural ventilation are split. We therefore require six parameters:

1. Transmission and internal sensible heat gain function of outdoor temperature (same as conventional house; see below).
2. Internal latent heat gain (same as conventional house).
3. Number of natural air changes per hour (0.1 ACH).
4. Number of forced air changes per hour (0.4 ACH).
5. Apparatus dew point or heat pump evaporator. Here taken equal to 45 F.
6. Bypass factor of heat pump evaporator. This is a measure of the fraction of intake air that is not cooled by the evaporator. A typical value is 0.15.

In addition to these parameters, one needs to know the design sensible heat gain function of outdoor temperature and humidity. We take, as we have before, 78 F dry bulb and 50% relative humidity (58 F dewpoint) as our standard indoor condition. This is a measure of the fraction of intake air that is not cooled by the evaporator. A typical value is 0.15.

Figure 4 shows the results dramatically. In the conventional house, with natural ventilation that is uncontrolled, the infiltration rate may fluctuate. We have shown curves for infiltration rates of 0.5 and 1.0 ACH and four outdoor humidity levels, a significant latent cooling load exists, which may or may not be taken care of by the air conditioner, depending on what sensible cooling loads exist. For the forced ventilation case, however, humidity ceases to be a problem, regardless of the outdoor condition. Even for the high humidity ratio of 0.017, the net latent cooling load is only 750 Btu/hr. This contrasts with values of 4500 Btu/hr for the conventional house with the same total ventilation rate (0.1 ACH) or 7400 Btu/hr for the conventional house with a ventilation rate of 1 ACH.

That humidity is controlled so well should not be a complete surprise. By controlling the
Figure 4. Net latent cooling load vs. humidity ratio for conventional and advanced-design houses

amount of infiltration air, the gross latent cooling load is kept within bounds, and then by dehumidifying this air in the most advantageous manner, it is reduced still further.

A second benefit of the tight-house approach is that the size of the air conditioner needed to satisfy the net cooling load is much reduced. The cooling and dehumidification done by the forced-ventilation heat pump replaces some of the capacity that otherwise would be needed. For example, at the condition 95 F outdoor dry bulb and 45% relative humidity, the air conditioning capacity required for the advanced-design house is one ton (12,000 Btu/hr) less than for the conventional house with 1 ACH of infiltration. This reduced capacity can be taken as a credit towards the cost of the forced ventilation heat pump.

The total design cooling load is then the sum of the above, or 21,110 Btu/hr, of which 16,860 is sensible and 4,250 Btu/hr is latent.

We calculate the latent and sensible cooling provided by the heat-recovery heat pump as follows, on the basis of the assumed 45 F apparatus dewpoint and 15% bypass factor. We note that the humidity ratio of the outdoor air at the design condition is 0.0321, and the humidity ratio of saturated air at 45 F is 0.0063. Then, for each pound of air passed through the heat pump, the latent heat removed is

\[ 1.060 \text{ Btu/lb} \times (0.0321-0.0063) \text{ lb water/lb dry air} \times (1-0.15) = 9.73 \text{ Btu/lb} \]

while the sensible heat removed is

\[ 0.244 \text{ Btu/lb-F} \times (95-45) \text{ F} \times (1-0.15) = 10.37 \text{ Btu/lb} \]

The mass of air passing through the heat-recovery heat pump per hour is

\[ 12,000 \text{ ft}^3 \times 0.4 \text{ ACH} \times 0.075 \text{ lb/ft}^3 = 360 \text{ lb/hr} \]

Hence, the latent and sensible heat removal rates of the heat-recovery heat pump are 3500 Btu/hr and 3730 Btu/hr, respectively. This reduces the net cooling load, to be met by the supplementary air conditioner, to 17,680 Btu/hr, of which all but 750 Btu/hr is sensible.

In what follows we made the same assumptions concerning the thermal capacitance of the house and the EER of the air conditioners as were made in the previous section, namely, EER = 10 for both the heat-recovery and supplemental heat pumps, and sensible thermal capacitance equal to 20,000 Btu/F.
The peak demand for air conditioning and domestic hot water in the advanced-design house is now 21,147/10 = 2.12 kW.

If the conventional house is assumed to have a peak cooling load 50% greater than that of the advanced-design house, if this load is met at an EER of 5, and if in addition an average hot water load of 2,000 Btu/hr is met with electric resistance (EER = 1.34), the peak demand will be 3.41 kW. Although precise estimates of peak load reduction will depend on the details of house design, the magnitude of the reduction estimated here, 1.43 kW, is viewed as well within the current state of the art.

Additional Peak-Load Reduction

An additional strategy for further peak-load reduction, called on combi-strategy, is that the house is precooled and dehumidified during the pre-peak evening hours, so that at the start of the peak period the temperature is 74 F and the relative humidity is 40% (humidity ratio 0.0071). Let us now assume that for the duration of the utility peak, the supplemental air conditioner is turned off, and only the heat recovery heat pump and the ventilation air fan are left on. Under these conditions the unmet latent cooling load will be the design latent cooling load of 4,250 Btu/hr less the latent cooling provided by the heat-recovery heat pump (3,500 Btu/hr) or 750 Btu/hr. The unmet sensible cooling load will be the design sensible cooling load of 16,860 Btu/hr less the sensible cooling provided by the heat-recovery heat pump (3,730 Btu/hr) or 13,130 Btu/hr.

A calculation of the rate of rise of the temperature and humidity ratios, assuming 10,000 Btu/F thermal mass, predicted that the house will remain in the comfort zone for five hours. By utilizing this strategy, the total electricity demand for cooling with dehumidification has been reduced to that required by the heat-recovery heat pump, which is equal to its total capacity divided by the EER, or (3,500 + 3,730)/2 + 753 W = 0.72 kW.

If the sensible thermal capacitance of the house is less than 10,000 Btu/F thermal mass, predicted that the house will remain in the comfort zone for five hours. By utilizing this strategy, the total electricity demand for cooling with dehumidification has been reduced to that required by the heat-recovery heat pump, which is equal to its total capacity divided by the EER, or (3,500 + 3,730)/2 + 753 W = 0.72 kW.

The strategy described here bears a superficial resemblance to the CH/RP strategy discussed in the preceding section. In both strategies the house is precooled before the onset of the peak period, and during the peak some of the air-conditioning heat-recovery heat pump and the ventilation air fan are left on. Under these conditions the unmet latent cooling load will be the design latent cooling load of 4,250 Btu/hr less the latent cooling provided by the heat-recovery heat pump (3,500 Btu/hr) or 750 Btu/hr. The unmet sensible cooling load will be the design sensible cooling load of 16,860 Btu/hr less the sensible cooling provided by the heat-recovery heat pump (3,730 Btu/hr) or 13,130 Btu/hr.

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