INDEPENDENT CONTROL OF SENSIBLE AND LATENT COOLING IN SMALL BUILDINGS[1]

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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, (1) 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 buildings, and to assess their technical and economic potential, including utility impact.

Several generic all-electric approaches to simultaneous control of sensible and latent cooling were examined:

1. Reheat of process air after it leaves the cooling coil. This approach cancels 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 (Figure 1). 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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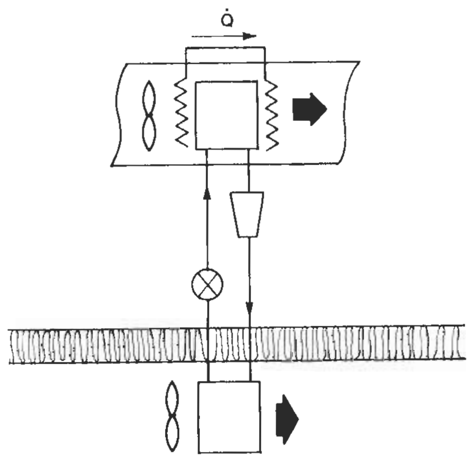

Figure 1. Reheat via evaporator run-around

downstream of the cooling coil (Figure 2). 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.

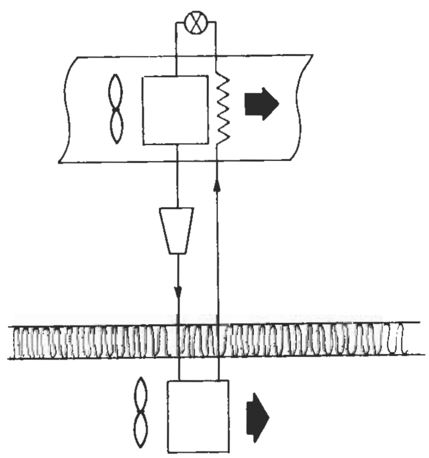

Figure 2. Reheat using a subcooler/reheater

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 reversible heat pump, because of the added heat-exchange surface in the ductwork. This improvement, estimated at approximately 5%, is significant in New York State because we have many more heating hours than cooling hours. It can more than pay for the modification over the life of the equipment, making the improved dehumidification a free or negative-cost option in northern climates.

<u>Reduce Temperature of Air Leaving Evaporator</u>

In vapor-compression cooling, dehumidification is performed by cooling the process air below its dewpoint, forcing it to give up some of its water vapor. The more one cools the air below the dewpoint, the more water vapor is removed. This approach to increased dehumidification is one that the industry has used for years in commercial buildings. It is not, however, without its drawbacks. One disadvantage is the cost penalty that has generally been associated with the approach, since lowering the leaving air temperature involves either reducing the evaporator temperature (associated with reduced efficiency and increased power costs) or increasing the depth of the evaporator coil (resulting in some increase in fan power and significant increase in equipment first cost). Another disadvantage is the diminishing returns effect inherent in the fact that for each unit of water vapor removed from a parcel of air, the more difficult it is to remove the next incremental unit.

Several candidate systems were identified that address some of these problems. One approach (4,5) is based on electronically commutated blower motors (ECM's). The use of ECM's permits total system efficiency to increase even as the fraction of cooling that goes to dehumidification increases, the reverse of the normal pattern. Another approach (6) uses a deep evaporator coil 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.

Another approach within this general category is ice storage. By making ice during off-peak hours and then using it for cooling during the peaks, utility load factors can be improved significantly. This is an ongoing technology in commercial buildings. Because of drawbacks of small-scale economics, and because of the cost of metering houses on a time-of-use basis, ice storage has not penetrated the residential market, though it may do so in the future.

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 hours preceding the on-peak period, the house is cooled from its normal setpoint of 78 F to 72 F, at the bottom end of the summer comfort zone. When the on-peak period begins, control is shifted from the thermostat to a dewpoint sensor, which causes the air conditioner to operate only enough to maintain the humidity ratio within the bounds of the comfort zone. Because of the enhanced dehumidification capability of the unit, it runs a smaller fraction of the time than a standard unit. Because of the lesser running time and because less sensible cooling than normal is provided per running hour, the temperature of the house will begin to rise. This is why the precooling is needed. In effect, the thermal mass of the house acts as a storage of sensible cooling, while the dehumidification, which is harder to store, is provided on call. The length of time the control strategy can remain in force before the temperature rises past the upper limit of the comfort zone is determined by the thermal mass of the house and the split between sensible cooling and dehumidification provided by the equipment.

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 dewpoint under these conditions is 58 F and the humidity ratio is 0.0103. At the beginning of the on-peak period, with the dry-bulb temperature in the house at 72 F and the humidity ratio not greater than 0.0103, the equipment is controlled by a dewpoint sensor instead of a thermostat to maintain the humidity ratio at 0.0103. Because of the low SHR delivered by the equipment, it does not run as long to maintain the humidity as it would if the SHR were higher. It therefore does not meet all of the sensible cooling load. The temperature therefore rises, even while the humidity ratio is held constant.

The following system characteristics are crucial. The lower the SHR, the less running time is needed to maintain the humidity ratio constant, and therefore the lower will be the average power draw seen by the utility. Since it is presumed that a large number of these machines are operating with random on-cycles, it is the average power draw of each unit during the peak period, and not the maximum power draw, that determines the aggregate peak demand of these units as a class.

However, the lower the SHR and running time, the less sensible cooling will be done, and the faster will be the temperature rise in the house.

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 CH/RP Strategy

The following analysis assumes a peak cooling load of three tons (36,000 Btu/hr), with a load SHR of 0.75, meaning that the sensible cooling load is 0.75 x 36,000 or 27,000 Btu/hr, and the latent cooling load is 9,000 Btu/hr. This is the assumed load at design indoor 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 1000 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 27,000 + 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

$$\text{Sensible cooling} = 36{,}000\ fs \tag{1}$$

$$\text{Latent cooling} = 36{,}000\ f\ (1-s) \tag{2}$$

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 30,000 - 36,000 x 0.714 x 0.65 or 13,290 Btu/hr. If the thermal mass of the house is 10,000 Btu/F, the rate of temperature rise is then

$$\frac{13{,}290\ \text{Btu/hr}}{10{,}000\ \text{Btu/F}} = 1.32\ \text{F/hr}$$

The time the house remains in the comfort zone is then 7 F divided by 1.32 F/hr, or 5.3 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) 36/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

$$36/10 \times (1-0.714) = 1.03\ \text{kW}.$$

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

Equipment Sensible Heat Ratio	Fractional Ontime	Unmet Sensible Cooling (Btu/hr)	Rate of Temp. Rise (F/hr)	Time in Comfort Zone (hr)	Peak Load Reduc-tion (kW)	Fractional Peak Reduction
0.70	0.83	9000	0.90	7.8	0.60	0.17
0.65	0.71	13300	1.33	5.3	1.03	0.29
0.60	0.62	16500	1.65	4.2	1.35	0.38
0.55	0.56	19000	1.90	3.7	1.60	0.44
0.50	0.50	21000	2.10	3.3	1.80	0.50

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

Equipment Sensible Heat Ratio	Fractional Ontime	Unmet Sensible Cooling (Btu/hr)	Rate of Temp. Rise (F/hr)	Time in Comfort Zone (hr)	Peak Load Reduc-tion (kW)	Fractional Peak Reduction
0.70	0.83	6100	0.61	11.5	0.40	0.17
0.65	0.71	8900	0.89	7.9	0.69	0.29
0.60	0.62	11000	1.10	6.4	0.90	0.38
0.55	0.56	12700	1.27	5.5	1.07	0.44
0.50	0.50	14000	1.40	5.0	1.20	0.50
0.40	0.42	16000	1.60	4.4	1.40	0.58
0.30	0.36	17500	1.75	4.0	1.54	0.64

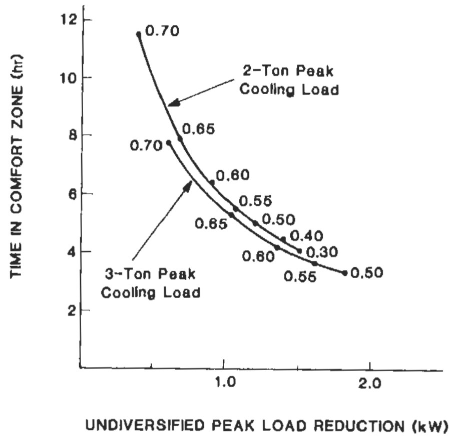

Figure 3. Time in comfort zone vs. peak load reduction, with equipment sensible heat ratio (numbers on curves) and peak cooling load as parameters. Equipment EER=10, House sensible thermal capacitance=10,000 Btu/F

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 EER divided by 10. Then draw a vertical line to intersect the 3-ton or 2-ton peak load curve. Read off the required SHR. Then draw a horizontal line 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- or 3-ton peak load is reached. By interpolation, we read 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 4 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 have 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, a rate of 0.1 ACH has been demonstrated in industrialized 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. This enhances dehumidification to the point where it is predicted that any remaining cooling demand 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 optimized performance made possible through the inclusion of all portions of the building system in 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 which was constructed and monitored at Brookhaven National Laboratory (7). This house was found to have a natural air change 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 minimums recommended 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/dehumidifier, 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 in such a way that the domestic water has first priority for heat rejection, but when the domestic water is heated to or near the setpoint temperature, the remaining condenser heat is rejected to the outgoing air.

The system can also serve a heating function, either by reversing the directions of air flow or by including a 4-way valve in the system. The latter is likely preferable, in that it would allow the air-flow paths to remain the same winter and summer.

Humidity Analysis

We now analyze the effect on cooling loads of the strategy of designing the house to have a very low 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 ft^2 in area with 8-ft ceilings, for a total air volume of 12,000 ft^3. 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 as functions 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 of 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 indoor design dry bulb 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. One also needs to know the outdoor dry- bulb temperature and humidity. Here we have considered the twelve combinations of three dry-bulb temperatures (75 F, 85 F, 95 F) and four humidity ratios (0.0086, 0.0114, 0.0143, and 0.0171), covering the broad range of conditions to be found during the cooling season.

Obtaining Useful Results. Since we do not know the transmission and internal sensible heat gain function, we have searched for a set of outputs that captures the essence of the latent load impact, but which is independent of the sensible heat gain function.

The most informative of such outputs is the net latent cooling load. For the conventional house, this is the total latent cooling load, including the internal moisture and the infiltration gain. For the advanced-design house, this is the latent cooling load remaining after subtracting the moisture removed by the heat pump. Both quantities can be plotted as functions of the humidity ratio of the outdoor air, in grains of water per pound of dry air.

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. At the higher 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 0.0171, the net latent cooling load is only 750 Btu/hr. This contrasts with values of 4300 Btu/hr for the conventional house with the same total ventilation rate (0.5 ACH) or 7600 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

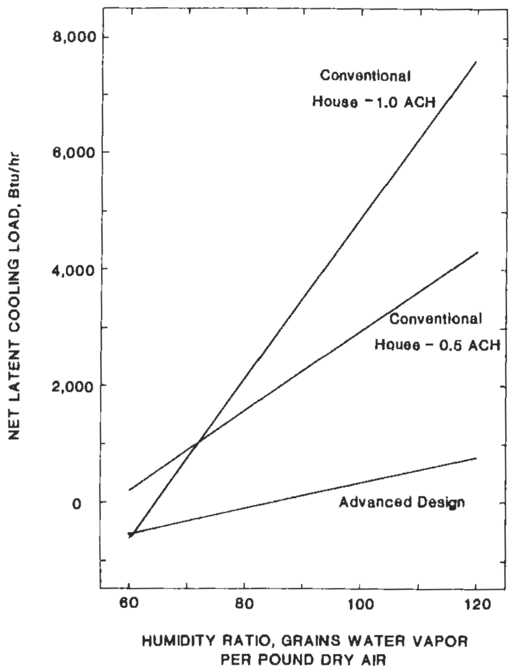

Figure 4. Net latent cooling load vs. humidity ratio for conventional and advanced-design houses

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.

Reduction of Peak Cooling Load

It was possible to calculate the net latent cooling load in the previous section without knowing the precise functional relationship between sensible heat gain and outdoor temperature. If we are to estimate the peak cooling load, however, we need to know this function. Clearly, the relationship will depend on the design of the house and on climate. In order to obtain a baseline estimate of the peak load reduction, we assume a gain of 600 Btu/hr for each degree F increase in outdoor temperature, together with a 70 F base for cooling. This leads to a design cooling load (at 95 F dry bulb, 78.5 F wet bulb) as follows:

1. Sensible gain (internal plus transmission) = 600 Btu/hr-F x (95-70) F — 15,000 Btu/hr

2. Internal latent heat gains (see above) — 1,000

3. Infiltration sensible gain (0.1 ACH) 0.1 x 12,000 ft^3 x 0.075 lb/ft^3 x 0.244 Btu/lb-F x (95-78) F — 370

4. Infiltration latent gain (0.1 ACH) 0.1 x 12,000 ft^3 x 0.075 lb/ft^3 x 1060 Btu/lb water vapor x (0.0171-0.0103) lb water vapor/lb dry air — 650

5. Forced ventilation sensible gain (0.4 ACH) Similar to Item 3 — 1,490

6. Forced ventilation latent gain (0.4 ACH) Similar to Item 4 — 2,600

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 0.0171, 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 Btu/lb x (0.0171-0.0063) lb water/lb dry air x (1-0.15) = 9.73 Btu/lb

while the sensible heat removed is

0.144 Btu/lb-F x (95-45) F x (1-0.15) = 10.37 Btu/lb

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

12,000 ft^3 x 0.4 ACH x 0.075 lb/ft^3 = 360 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 13,880 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 10,000 Btu/F.

The peak demand for air conditioning and domestic hot water in the advanced-design house is then 21,147/10 = 2,115 W = 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 10, and if in addition an average hot water load of 2,000 Btu/hr is met with electric resistance (EER = 3.41), the peak demand will be 3.75 kW. Although precise estimates of peak load reduction will depend on the details of house design, the magnitude of the reduction estimated here, 1.63 kW, is viewed as well within the current state of the art.

Additional Peak-Load Reduction

An additional strategy for further peak-load reduction can be used. In this strategy, the house is precooled and dehumidified during the pre-peak morning 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 ratio, 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 and hot water 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)/10 = 723 W = 0.72 kW.

If the sensible thermal capacitance of the house is less than 10,000 Btu/F, or if the peak period is more than five hours, a mixed strategy involving some operation of the supplemental air conditioner could be used. The effective peak demand in this case would be between 0.72 kW and 2.12 kW, depending on conditions.

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 equipment is either not used or used for fewer hours than would otherwise be the case. The strategies differ in the following important respects:

- CH/RP can be applied in any house, requiring only a low-SHR air conditioner for its application. Dehumidification of forced-ventilation air requires a tight envelope (0.1 ACH), and is therefore restricted to the next generation of housing.
- CH/RP does not, of itself, provide any energy savings. Dehumidification of forced-ventilation air provides significant savings on energy.
- The potential reduction in peak cooling load is greater with dehumidification of forced-ventilation air than with CH/RP.

To sum up, CH/RP is a near-term option applicable to both retrofit and new construction. Dehumidification of forced-ventilation air, while feasible today, will require a shift in design philosophy in the American housing industry that will take some time to penetrate deeply into our housing stock.

CONCLUSIONS

1. Several technology options, some of which have recently come into the market, have the capability of enhancing the fraction of cooling that is delivered as dehumidification, and at the same time either increasing efficiency or at least not degrading it significantly.

2. A novel control strategy has been developed that can reduce utility peak loads by 30 to 50%. This strategy, which we have called CH/RP (control humidity/reduce peak) works by precooling the house during the pre-peak hours and then controlling on humidity rather than temperature during the on-peak hours. An air conditioner with a lower sensible heat ratio than the house actually needs is required to implement the strategy.

3. A house with a very low air infiltration rate (0.1 air changes per hour) has the potential for improved dehumidification performance and reduced utility peak loads. Ventilation air is brought in through a single portal and passed over a cooling coil before being mixed with the house air. This maximizes the dehumidification that this coil can do, to the point that the remaining cooling load is nearly all sensible.

Houses of this type are routinely constructed in Scandinavia, and there is no reason why the American housing industry cannot adopt similar practices. Scandinavian houses, however, are oriented toward heating only, with no provision for cooling. We conclude that the building envelope and ventilation system lend themselves naturally to cooling with enhanced dehumidification, so that this type of house could become the future standard in the United States. However, it will be many years before this type of house becomes a significant fraction of the housing stock.

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