AIR CONDITIONING—A LUXURY OR A NECESSITY IN GREEN BUILDINGS?

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INTRODUCTION

Air conditioning is firmly entrenched in modern U.S. lifestyles, with over 80% of all new houses being centrally heated and cooled. This trend will likely remain in place even as energy prices increase along with pressure to reduce global carbon emissions. Air conditioning has been the major factor behind the explosive growth of the Sunbelt in the south and southwest. Las Vegas, Phoenix, Houston, Miami and many other cities would be only small regional centers if not for the availability of air conditioning for businesses and residences. Air conditioning is becoming more accepted even in areas where it was previously shunned, such as parts of Europe which experienced the severe heat wave in 2002 that was responsible for the deaths of thousands. Given its prominence in our modern society, we must learn to use air conditioning in an intelligent and efficient manner so we can benefit from the quality of life it provides, but without bankrupting our energy resources and permanently altering our global environment.

AIR CONDITIONING ENERGY CONSUMPTION

The vast majority of air conditioning systems for residential and small commercial systems use the standard vapor compression cycle with an HFC or HCFC refrigerant. Nearly all of these systems are electrically driven in the United States, though there are significant numbers of engine driven heat pumps in Japan. There are some absorption systems in larger sizes, but very few in the 20-ton (70kW) or smaller capacity range. The majority of residential and small commercial A/C systems are categorized as unitary products, meaning that the system is completely assembled at the factory and shipped to the site as a unit. Many of these unitary products are two-piece split systems and require refrigerant piping to be field installed between the indoor and the outdoor sections. A small but growing part of the unitary market uses water source heat pumps, either with a boiler/cooling tower arrangement in some commercial applications or with ground source heat exchangers. These systems require more field fabrication than air source systems, since they require additional indoor water piping and/or excavation or drilling for the ground heat exchangers.

Regardless of the exact air conditioning system configuration, the overall energy usage can be expressed in the same general way, as:

$$E_{AC} = \frac{Q_{Load}}{\eta_D \eta_{AC}} \tag{1}$$

where E_{AC} is the energy consumption of the air conditioning system, Q_{Load} is the heat removed (load) from the conditioned space, η_D is the efficiency of the distribution system, and η_{AC} is the efficiency of the air conditioning system. The air conditioning efficiency may be a dimensioned value, such as the commonly used SEER, which has units of Btu/W-h, and thus converts the units of Q_{Load} expressed in Btu into the energy units of E_{AC} (W-h).

IMPACT OF THE BUILDING LOAD

The simple form of Equation (1) provides some important basic insights into what should be first priorities regarding air conditioning energy conservation. First and foremost, to reduce air conditioning energy costs, one should first target the air conditioning load of the building. It can easily be seen that a 10% reduction in Q_{Load} should produce a 10% reduction in

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A/C energy consumption. Reductions in Q_{Load} can result from many design changes, including, but not limited to:

- Better envelope insulation
- Lower U-value windows or use of solar reflective films
- Reduced window area
- Improved window orientation (less East/West facing glazing)
- Tighter building envelope to reduce air infiltration
- Reduced hours of A/C operation (setback thermostat)
- Higher setpoint thermostat setting by use of ceiling fans
- Reduced internal loads due to lights, appliances, etc.

Energy conservation retrofits of existing buildings will produce different decisions about which form of load reduction is cost effective compared to what might be possible for new construction. For instance, additional envelope insulation is often limited to the attic of existing buildings, but can be incorporated into all parts of a new building design. Window area and orientation are usually not options that are available in an energy conservation retrofit. Similarly, the airtightness of many buildings is difficult to improve significantly in a retrofit since many of the sources of the air leaks are concealed and may be difficult to even identify. The traditional application of caulk and weather stripping around windows and doors typically results in less than a 10% reduction in building air infiltration. Specifications for new building construction can be effective at reducing air infiltration if appropriate sealing measures are specified during construction. Built-in appliances and lighting systems may have useful lives of 20 years or more and usually cannot be economically replaced simply on the basis of energy efficiency.

While most aspects of the building cooling load can be considered completely separate from the design and operation of the A/C system, one must never think that the two are unrelated. Even on a first-cost basis, an additional investment in energy reduction technologies can be partially offset by an incremental reduction in the cost of the A/C and distribution systems. Such reductions in equipment size and capacity occur in step increments, so each individual energy conservation item may not match up to a corresponding saving in cooling equipment system cost. However, as the cooling load is reduced, the cooling system can be downsized with some reduction in first cost. The smart building designer first looks for those efficiency improvements that have the least capital costs and over which he has some control. For instance, in a small commercial building, the incremental cost to go from a double pane, low-e picture window to a triple pane window may be a factor of two or more. A similar reduction in cooling load may be obtained by simply reducing the size of the window somewhat or by use of low solar heat gain glazing. Structural design changes could also be implemented, such as window orientation or overhangs, but often at some incremental construction costs.

With the expectation that future energy prices will continue on a general uptrend, architects, building designers, and contractors have a responsibility to implement features that are in the best long-term interest of the building owner (or whoever pays the utility bills). Unfortunately, the building owner and the person that pays the utility bills are often not the same person. In this instance there is a natural conflict between capital investments required for energy conservation and reductions in utility costs versus lowest construction cost. A building owner that will lease the space has pressures to keep construction costs to a minimum so that lease rates are competitive with the existing competing building stock. This situation usually results in buildings that use code minimum designs and construction practices and have the lowest efficiency A/C systems that can be purchased. Until energy costs become important enough for a tenant to scrutinize them in advance of signing a lease, code minimum design and construction practices will continue. Fortunately, these code minimum requirements are generally trending upwards in their energy stringency with each code cycle, and so provide some degree of protection against buildings becoming too expensive to operate long before the end of the useful life of the structure.

One of the energy reduction design changes listed above is actually a system operational change, though it results in a reduced cooling load. Changing the thermostat setpoint or the use of setback/setup during unoccupied periods will produce reduced heating/cooling loads by reducing the indoor-to-outdoor temperature difference, either temporarily or on a constant basis. One of the myths of HVAC is that thermostat setback causes the system to use even more energy as it brings the space back to the setpoint temperature. Setting the thermostat to a higher temperature for a long enough period of time to allow the space temperature to reach the new setpoint temperature will save energy. The longer that the higher setpoint can be maintained, the longer more energy will be saved. Figure 1 shows temperature versus time curves for an ideal thermostat setup change over a finite time period. The area represented by the incremental change in ΔT divided by the total integrated ΔT for that time period would represent the fractional energy savings that would result. In reality, efficiency changes caused by the different return air temperature and compressor cycling rates would also affect energy savings, perhaps either positively or negatively. Obviously, the longer the setup temperature period and the more the temperature is allowed to float, the greater the energy savings. Since much of the cooling requirements are for solar heat gains and internal loads, calculation of cooling energy savings from thermostat setup can't reliably be determined using a simple indoor-to-outdoor temperature difference as shown in the schematic. However, this simplified approach is a reasonable approximation for heating loads, however.

ENERGY DISTRIBUTION SYSTEM EFFICIENCY

The term η_D in Equation (1) is the efficiency of the energy distribution system. Air conditioning almost always uses a forced air system, though some European countries use "cooling ceilings" in larger commercial buildings. Except for through-the-wall packaged terminal-type systems, small window units, or multi-split systems, forced air systems will usually involve some amount of ducting, often on both the supply and return sides of the air handler. This energy distribution efficiency will always be a function of the individual design. It must account for both heat transfer losses by conduction from the cool conditioned air to the surroundings, as well as the air leakage losses. ASHRAE has a standard procedure for the determination of seasonal efficiencies of energy distribution systems for residential applications (ASHRAE 2004a). An accurate calculation of this distribution efficiency is not a trivial matter for a forced air system, and would require measurements of duct system leakage using an in situ pressurization technique as well as measurement of total system air flow rate.

Conduction heat gains from the ambient air to the cool conditioned air involve a fairly straightforward calculation once the ambient air temperature is estimated from the ASHRAE standard. The ambient air temperature that the ducts are exposed to will depend on their location. Ducts located in an uncondi-

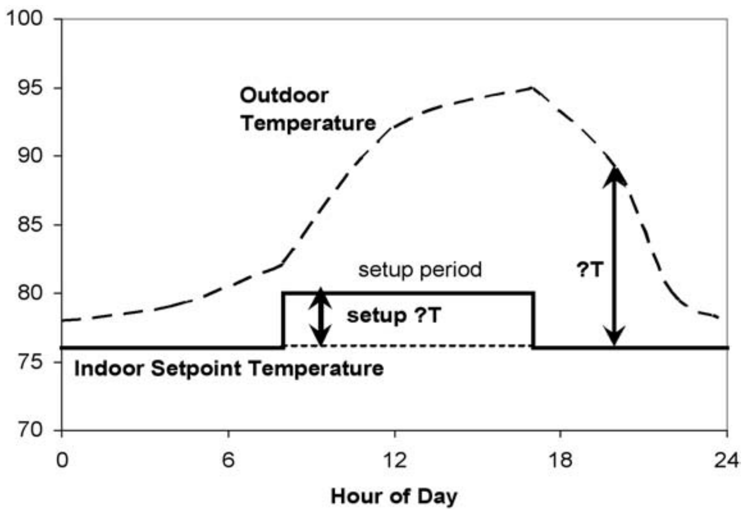


FIGURE 1. Idealized representation of thermostat temperature setup for reduction of building cooling load.

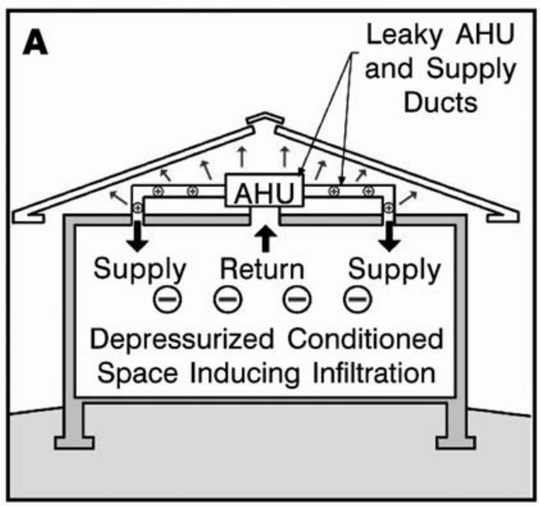
tioned basement may have little driving temperature difference for heat gains, while ducts in an unvented attic space may have a very high temperature difference during the hours of peak A/C operation. Ducts located *within* the envelope of the conditioned space may have some actual heat transfer to the space air, but those heat gains would not be counted as a loss of efficiency, since the cooling effect is still being delivered to the space (though perhaps not to the exact desired point of delivery).

It must be realized that exterior air ducts are an extension of the conditioned space, and as such should be insulated and sealed as well as the rest of the building envelope. It is common practice to apply a 1-inch layer of blanket insulation to metal ducts located in attic spaces that approach 150°F in summer, while there may be 6 to 12 inches of blanket insulation on the floor of the same attic. The duct carries 50°F air compared to the 75°F air in the conditioned space, so it experiences an even greater temperature difference than between the room and the attic air temperature. Admittedly, it would be challenging to encase ductwork in 6 to 12 inches of insulation. That alone makes it all the more important to absolutely minimize the use of exterior ductwork, and especially within attic spaces in cooling dominated climates.

While conduction heat gains to A/C ducts is an obvious source of efficiency loss, air leakage in the duct system can be shown to be the greater source of loss for most exterior ductwork. Leakage rates of 30% to 40% and higher have been reported in the literature from studies of multiple homes for both new construction and older houses (Cummings and Tooley 1989). These duct leaks are not simply losses from the supply air duct, but also from induction of outdoor (outside, attic, or crawl space) air into the return air system. Even with a tight duct system with a leakage rate of only 10% of the supply air flow, the leakage losses will typically dominate the overall distribution system losses. Rigid metal duct is perhaps the worst offender for duct leaks in residential and small commercial applications, as the seams are often not taped or sealed with mastic. Seam sealing is far more common for larger systems, perhaps because it may be included as part of the specification details. Residential and small commercial customers may not be sophisticated enough to know of such requirements, or that such details must be specified in their construction contract.

Leakage from external supply ducts is a direct loss of energy efficiency, usually in direct proportion to the percent leakage. The impact on energy loss is fairly obvious, as anyone would understand that if you leak 30% of your cooled air to the outdoors, it costs you essentially 30% more than was necessary to satisfy the space load. The magnitude of the typical leakage rates is not commonly known by homeowners or small commercial building owners or tenants. A more subtle result of duct leakage is the induction of outside air into the space, as illustrated in Figure 2. During periods with high outside dew point temperatures, this induced infiltration air can produce local humidity conditions that foster the growth of mold inside the space or even within the building structure (Lstiburek 2002). The significance of this effect is most pronounced in humid climates where the latent capacity of the cooling equipment may be only marginally adequate at design conditions, not accounting for this additional induced latent load. A similar but reversed effect is produced if the leaks occur in the return duct, causing the space to be pressurized and forcing conditioned air to leak out of the house. During heating season, this conditioned

FIGURE 2. Induced air infiltration caused by air leakage from external ductwork. (Source: ASHRAE Journal; Copyright, April 2002.)



air is more humid than outdoor air and may cause condensation problems in wall cavities, receptacles openings, or other sites where the air escapes. Significant return air leaks are very common in residential applications because air filter slots are often poorly sealed, if at all, and are typically located near the lowest pressure point in the system. Civil courts are becoming filled with mold lawsuit cases, many of whose symptoms were likely aggravated by a leaky air distribution system.

A/C SYSTEM EFFICIENCY

When someone thinks of HVAC efficiency, they usually think of the efficiency of the "box," or the vapor compression system. As has just been illustrated, a building is an interrelated system, and the operating cost of the HVAC system will depend on the building and the air distribution system as much as the actual cooling components.

Being the only component of an HVAC system that is factory assembled, the A/C unit is typically the only part of the system that is actually performance tested and which must conform to federal minimum performance requirements. Beginning in 2006, the U.S. Department of Energy increased the minimum efficiency requirements to a value of SEER 13 for unitary products with cooling capacities below 65,000 Btu/h. The previous minimum SEER value had been 10 for nearly 15 years. The SEER (seasonal energy efficiency ratio) rating procedure has been in place in essentially its current form since the early 1980s. The SEER is intended to account for seasonal effects on the operating efficiency of a unit. The rating test performs steady state efficiency measurements at two different outdoor temperatures (95°F and 82°F), as well as under a part load cycling condition at 82°F. The test assumes that the average outdoor temperature at which the unit will operate is 82°F. While probably few would argue that the SEER rating gives an accurate representation of the seasonal efficiency of a unit, it does address the major factors that affect the A/C unit's operating efficiency and so is generally accepted by the industry. The Air-conditioning and Refrigeration Institute (ARI) publishes the SEER rating and other performance data for all the unitary products of its member companies on its website (ARI 2006).

Unitary cooling products can generally be broken into two broad categories: air source equipment and water source equipment. These designations refer to the heat sink to which they reject the heat extracted from the conditioned space. Air source equipment will typically have an external condensing unit located on a pad either at ground level or on the rooftop. The condensing unit will normally contain the compressor and controls besides the condensing coil. Larger condenser coils have been one of the solutions to the higher efficiency requirements mandated by DOE. These large units now become more of a visual nuisance factor when located around expensively landscaped grounds. Larger outdoor units will normally require fencing or other containment for security or for noise control, as can be seen in Figure 3. Packaged chilled water systems tend to be

FIGURE 3. Air source condensing units located outside a building. Retaining walls are commonly required for security or noise control for larger systems.

FIGURE 4. Packaged air source chilled water condensing unit located in a secured containment area.

larger, with the water cooler located outdoors as shown in Figure 4. The condenser heat exchanger section of air source systems is a relatively simple mechanical system, since it requires only a conventional axial fan to move the ambient air over the outdoor coil.

Water source units use a circulating water stream as the heat sink. The water stream may go to a cooling tower, be a once-through water stream that may be used for irrigation or other applications (or simply dumped to a drain), or be ground coupled and circulate through tubes buried in the earth. One of the major attractions of a water source unit is that the refrigerant-to-water heat exchanger is very small compared to conventional refrigerant-to-air coils. These systems are often completely packaged in one unit with the water piping and the air ductwork connected to the unit. Water source units with a cooling tower have been successfully used in office applications for many years. Since the package units are small, they can be located very close to the zone that they serve and thus require a minimum of ducting. Each unit generally operates independently of every other unit, as they are typically controlled by individual thermostats. A central water pump and pipe loop provides the water for the units, with necessary controls for activating the cooling tower fan as the load and weather conditions may require. While this design can be used for cooling only products, the design arrangement most commonly uses heat pump units to provide simultaneous heating and cooling options. A combustion boiler is often used as the heat input during the heating season to prevent the circulating water stream temperature from dropping below about 50°F.

A major advantage of the heat pump application is that it provides 4-pipe function (independent heating and cooling capability in each zone) with 2-pipe plumbing. The water temperature in the circulating lines is in a range of 50–70°F, so the pipes do not even need to be insulated. Having multiple units also provides significant equipment diversity, where a failure in one unit affects only that one room or zone with minimal effect on the rest of the building. In an application where many similar units are used, the maintenance staff may actually keep a spare unit on hand so it can quickly replace the failed unit and get the room back in use while the faulty unit is analyzed and repaired. The efficiency of these systems can be quite good in larger applications with significant building diversity. Having some units heating while others are cooling reduces the amount of boiler or cooling tower operation, and is basically just moving energy around within the building. Since the heat pumps operate in both heating and cooling modes with COPs of four or higher, this is a very efficient way to "recycle" energy within the building without consuming additional energy resources.

One of the major negative aspects of using multiple water source units is accessibility for service. These units are often located in the ceiling space, so the service technician must be on a ladder or lift to access the unit. Accessibility is less of an issue for single story structures where a mezzanine is often built in the attic space overhead and the units are located in a somewhat central location where they can all be easily accessed. Schools will often have this concrete mezzanine located over the hallways so that the ducting length to each zone is still quite short. Another negative aspect of the multiple units is that the fan and compressor are now located near the zone. This is desirable from a ducting perspective, but undesirable from a noise perspective. Blanket insulation between the occupied space and the unit will dampen the compressor noise, but background noise is still quite common. It is important that the unit is not strapped to pipes or structural elements that may transmit its vibration throughout the building.

One of the faster growing segments of the water source market is that of ground source systems. Ground source systems (usually heat pumps for both heating and cooling functions) exchange their heat with the earth, either through a water loop or with the refrigerant lines buried directly in the earth. The water loop type is far more common since it can accommodate a variety of local soil and water conditions. Examples of the variety of ways that the ground heat exchange process can take place are shown in Figure 5. The water loop can be a closed loop variety where water is simply circulated through the piping system with a small pump, and the water exchanges heat with the earth by conduction through the walls of the pipe. The open loop variety pumps water through the system on a once-through arrangement and dumps the water to a surface water body, or reinjects it back into the aquifer from which it was pumped. Another arrangement is somewhat of a hybrid of these two, called a standing column well, where the water is circulated back to the same well bore from which it was pumped and the water exchanges heat with the earth from the borehole.

There are pros and cons to all of these systems. The closed loop type uses little pumping power since it has to overcome only the pipe friction, but will require expensive excavation to install the pipes in the earth (either in vertical boreholes, horizontal trenches, or on the bottom of a lake or other large body of water). Since the water is recirculated, there is no problem with scaling or water fouling issues, but antifreeze may be required for regions where there is a significant heating season. The open loop (water well) variety is often used for larger systems (hundreds to thousands of tons of capacity) to preclude the high cost of heat exchanger placement. However, a secondary heat exchanger loop is often used so that the heat exchanger for the source water stream can be disassembled (often a plate-frame type) for periodic cleaning. Small unitary products are typically not designed for such cleaning, so may experience a gradual degradation of performance until their capacity is no longer sufficient. The loss of efficiency and the required maintenance for such systems often make the closed loop variety the lower life cycle cost system for smaller systems. The advantage of the open loop type that uses well water is that the water source temperature is essentially constant at the local deep ground temperature. The closed loop water temperature will vary considerably due to the slower process of heat conduction in the earth around the buried pipes. The piping that is most commonly used for the ground

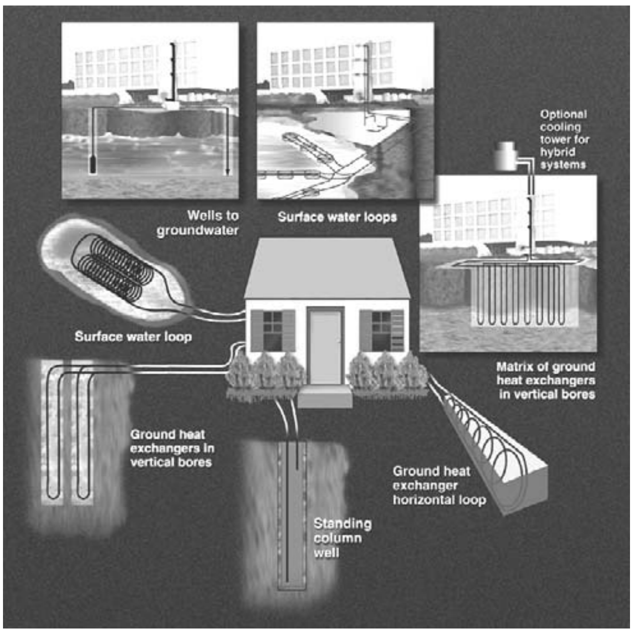


FIGURE 5. Schematic of the various ways that ground source heat exchange can be performed with heat pump systems. (Source: Sustainable Technologies for Overview of Prototype Green-Built House Design, by R. Wayne Guy and Paul Tinari, *Journal of Green Building,* vol. 1, no. 2.)

source applications is high density polyethylene that is heat fused on site. The heat fusing (or thermal welding) greatly reduces the chance for water leaks from joints in buried pipes. If a leak were to develop, that section of pipe would have to be abandoned and another installed to replace it. The closed loop variety is usually initially pressurized upon installation with no makeup water option (assuming there will be no leaks). A leak of just a few ounces from such a system will often drop the water pressure to where a pump with a low net positive suction head may begin to cavitate and eventually fail. There are some useful design guides for ground source systems, but experience in installation of this type of system is critical for low cost and reliable long-life operation.

With the apparent complexity of the ground heat exchanger, it might seem that there would be little advantage to using a ground source system over a conventional air source product. However, the reliability of properly installed ground source systems rivals or exceeds that of air source products. There are no exposed components outdoors that may be subject to weather or vandalism, and the relatively constant source water temperature does not subject the compressor to additional severe operating conditions during parts of the year. The buried piping should have an extremely long life as the HDPE plastic does not react or degrade within the life of the normal system. The circulating pump would be the weakest link in this system, but can be expected to last 10 years or more in systems that are properly designed and installed. These systems are precisely charged with refrigerant at the factory, so are optimized for peak efficiency. Refrigerant leaks are the bane of air source products, but such are rare occurrences with water source products. In recent years, water source units have developed a more substantial market in public schools where on-site service has traditionally been less reliable than in the private sector due to budget constraints.

The greatest uncertainty about ground source products is their ultimate operating efficiency. While there are three different rating standards provided by ARI (1998a, 1998b, 1998c) for water source units, heat pump performance varies considerably with the temperature of the water source. The water temperature will be highly dependent upon the design and placement of the ground heat exchanger, the local soil properties, and the local soil temperature conditions during the heating and cooling seasons. Experienced contractors should understand these factors and know how to address the local conditions. However, just as SEER is hardly an exact estimate of seasonal operating efficiency of air source products, the ARI ratings for water source equipment is no better at estimating the annual efficiency of a particular ground source installation. Experiences with ground source systems have typically demonstrated significant energy savings over comparable air source products, but much of that savings is often in the heating season when auxiliary electric resistance is required for the air source equipment. The payback period for the higher cost of installing ground source equipment is more difficult to justify on energy savings alone in locations with relatively mild weather conditions.

Alternative energy sources for HVAC systems are not yet to the point of acceptable reliability for widespread use. Solar photovoltaic can provide direct electricity to drive cooling during peak cooling conditions, but substantial cooling (and dehumidification) may be required on cloudy days or at night when the photovoltaic output is significantly reduced. Wind energy densities are not adequate for reliable power generation in much of the country on a small scale. Rather than opt for the additional cost of expensive alternative energy sources with conventional power backup, it is usually more cost effective to invest that capital in energy conservation practices that reduce the building load year round.

THE IMPORTANCE OF HUMIDITY CONTROL

The term “air conditioning” is usually associated with comfort cooling, though it technically covers the entire spectrum of comfort conditioning as specified in ASHRAE’s standard on human comfort (ASHRAE 2004b). Sensible cooling (lowering temperature) is the predominant function of A/C in many parts of the country, but dehumidification (latent cooling) is equally important in many other parts of the country. While poor temperature control may result in more comfort complaints than anything else, poor humidity control can result in expensive litigation, occupant health complications, and even building structural damage.

Most unitary air conditioning systems use temperature feedback control devices (ordinary thermostats) that operate based only on the temperature of the sensor in the device. Humidity is lowered by default as the system responds to the need for temperature control. The amount of humidity control that is provided is based on the design of the system and its components. By designing the evaporator coil to operate at a certain refrigerant saturated suction temperature for the design air flow rate, the coil will provide a certain amount of dehumidification capacity for standard entering air conditions (typically 80°F dry bulb temperature and 67°F wet bulb temperature, about 50% relative humidity). At off-design conditions, the dehumidification capacity will be more (at higher entering air humidity) or less (lower entering humidity) than this amount. The ratio of the sensible cooling effect to the total cooling effect (sensible plus latent) is referred to as the sensible heat ratio (SHR) for the cooling equipment and is usually specified for a given unit at the standard design conditions. While there is no specific requirement for what the SHR should be for different applications, most unitary products are designed to deliver a steady state SHR of about 0.70 at standard operating conditions. Thus, a 5-ton unit (60,000 Btu/h) would likely have a capacity breakdown of about 42,000 Btu/h sensible and 18,000 Btu/h latent.

The required latent capacity of a particular building will depend on the specific application and the internal moisture generation sources. In most cases, the greatest source of excess moisture in buildings will come from outdoor ventilation air, especially for commercial buildings where ventilation air is mandated by local codes that reference the ASHRAE standard on ventilation (ASHRAE 2004c). Larger buildings will often use occupancy sensors to regulate how much outdoor ventilation air is brought in, but smaller applications will typically have a damper that is set at a certain position for all time.

As an example, consider a small deli restaurant (no griddles or ovens) where the majority of the ventilation is simply based on occupancy. If there is seating for 50 customers and typically three employees, and the restaurant covers 1200 square feet with two toilets, the ventilation air rate would be about 700 cfm of outdoor air using the ASHRAE ventilation standard. The required capacity for this space and occupancy (not counting the conditioning of ventilation air) may be on the order of 7.5 tons. If this restaurant was located in Huntsville, Alabama, the outdoor design condition would be 97.4°F dry bulb and 75.8°F wet bulb temperatures using the peak sensible load conditions. The moisture content of the outdoor air is 0.0143 pounds of water vapor per pound of dry air. The indoor moisture content at 78°F dry bulb and 50% relative humidity is 0.0103. The additional load produced by the outdoor air can be found using simple psychrometric relationships:

$$Q_S = 1.08 \times \text{cfm} \times \Delta T \tag{2}$$

$$Q_L = 4840 \times \text{cfm} \times \Delta W \tag{3}$$

where Q_S is the sensible cooling in Btu/h, Q_L is the latent cooling in Btu/h, ΔT is the temperature difference in degrees Fahrenheit, and ΔW is the humidity difference in pounds of water vapor per pound of dry air. For this 700 cfm ventilation air stream with the given outdoor and indoor design conditions, the required cooling capacity to condition the outdoor air is 1.22 tons sensible, and 1.13 tons latent, or nearly 2.5 tons total with an SHR of about 0.50. The restaurant would need a 10-ton system installed, with fully 25% of its capacity needed to condition the ventilation air at design conditions. Nearly 12% of its capacity is needed just to remove excess moisture from the ventilation air.

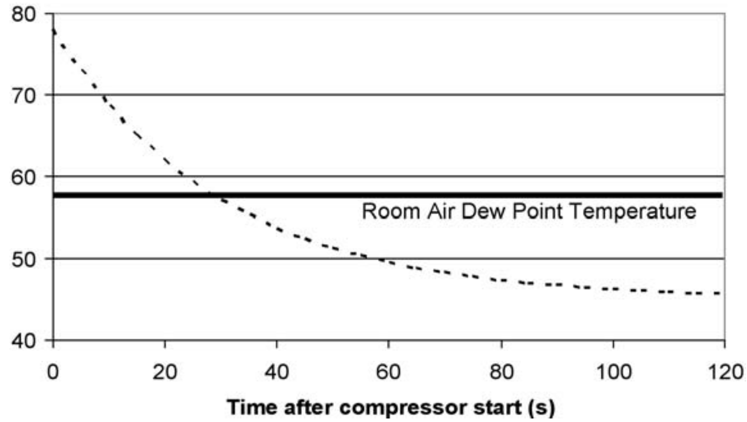
The challenge of designing such a system is the occurrence of off-design conditions. ASHRAE currently tabulates data on the basis of either design dry-bulb conditions or design wet-bulb conditions. If the peak wet bulb conditions had been used instead for these calculations (89.8°F dry bulb, 80.0°F wet bulb), the resulting ventilation air cooling requirements are 0.74 tons sensible and 2.74 tons latent, or a total of 3.5 tons. These conditions would exist, for instance, after a summer shower when the air is still warm but the presence of water greatly increases the outdoor humidity levels. For this condition, over 25% of the total system capacity (still assuming 10 tons installed capacity) is needed just for moisture control for the outdoor ventilation air. Assuming that the sensible heat ratio for the rest of the building load was 0.80, the composite SHR including the ventilation air becomes 0.58. A system designed to deliver an SHR of 0.70 will obviously not be able to maintain acceptable humidity levels under these more severe outdoor humidity conditions.

The significance of this exercise is to illustrate the importance of ventilation air in the sizing of HVAC systems for commercial applications. As more emphasis is placed on indoor air quality (IAQ), ventilation air rates have generally tended to increase. In those climates where cooling is the dominant load, humidity considerations become much more important and may require the system designer to consider peak moisture conditions rather than peak sensible cooling conditions to size the system for acceptable performance. While there are currently no single family residential building codes that require mechanical ventilation in the U.S., there will likely be some that incorporate ventilation soon. ASHRAE has now developed a residential ventilation standard that is suitable for code adoption (ASHRAE 2003). While residential ventilation may have the desired effect of improving IAQ across the board, it will very likely require a major shift in equipment sizing and humidity control from current practices using unitary equipment.

A design practice that is sometimes implemented for the sake of ventilation air requirements is the use of continuous fan operation. Unitary equipment usually uses an on-off mode of operation for the compressor, while the fan can either cycle with the compressor or run in a continuous mode. If the primary air handler is used to draw in the ventilation air, it may be set to run continuously to maintain proper ventilation levels, even during low cooling load conditions when the compressor may be required to operate less than 25% of the time. While it has been known that equipment cycling results in reduced dehumidification capacity, some recent studies have attempted to quantify this reduction for the continuous fan operating condition (Shirey, et al. 2006). In general, these researchers have found that there is actually minimal *net* dehumidification produced with continuous fan operation until the compressor is required to run at least 50% of the time. Or stated another way, at low-load off-design conditions, the cooling equipment may provide no actual humidity control whatsoever if the fan is operated continuously to provide ventilation air. Such operation during mild or rainy weather may provide the necessary conditions for the development of mold in the system ductwork or on surfaces with an affinity for moisture, such as books, leather, cloth, wood, etc.

The reason for such poor part-load dehumidification performance can be seen in a simplified schematic of the evaporator coil temperature after compressor startup, as shown in Figure 6. The dew point temperature of air at 78°F and 50% relative humidity is about 57.5°F. The actual starting temperature of the evaporator coil will depend on its location (hot attic or cool basement), internal refrigerant migration after shutdown, its size, amount of cabinet insulation, etc. However, assuming the coil approaches the return air temperature during continuous fan operation, the mean coil temperature will decrease in a manner similar to that shown in Figure 6. For the coil time constant shown, the first drop of condensate would not be produced until the coil reaches the air dew point temperature, about 30 seconds after compressor startup.

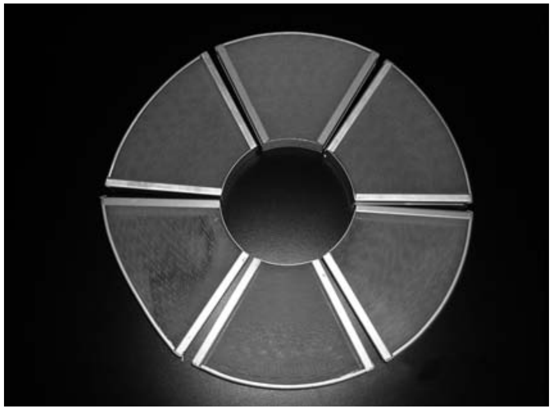
FIGURE 6. Simplified schematic of the evaporator coil startup temperature and the onset of dehumidification.



Some systems with oversized coils would have an even slower response than what is shown. At low load conditions, the compressor may run for only four to five minutes at a time. In the short time that the coil is actually dehumidifying, much of the condensate remains as droplets on the coil fins and tubes. The condensate that does drip off the coil may collect in the drain pan and remain there if the drain outlet is not at the very bottom of the pan or if the unit is not perfectly level. The work by Shirey, et al. demonstrated that at load conditions less than 50% of the system capacity, virtually all the condensate produced during the operating part of the cycle was re-evaporated back into the continuous air stream during the off-cycle.

There are solutions to the apparent conflict between providing more ventilation air to improve indoor air quality and the ability to properly control the humidity level in the space. Canadians have long used heat recovery ventilators (HRV) to provide ventilation air for their tight buildings in very cold climates. Similar devices with desiccants applied to the surfaces can be used for both sensible and latent energy recovery. These devices can be placed in the outdoor ventilation air stream to pretreat the outdoor air. In humid conditions, this pretreatment would cool and reduce the moisture content of the incoming air using the cooler and dryer exhaust air that is at room design conditions. HRVs consume little energy themselves, only a small motor that is needed to slowly turn a heat wheel in the rotary wheel types. They do introduce some additional pressure loss to the air stream, requiring some additional fan power. HRVs are becoming much more common in larger systems with high ventilation requirements, and their first cost is becoming more cost effective in smaller equipment ranges as well. Figure 7 shows the replaceable elements of a small rotary wheel HRV that have plastic heat transfer surfaces coated with a desiccant material for moisture transfer. This particular device is only about one foot in diameter and is designed for smaller applications.

FIGURE 7. Replaceable heat recovery ventilator components for moisture transfer in ventilation air.



Another method to cost effectively modify humidity is as the byproduct of producing domestic hot water with a heat pump water heater (HPWH). These devices are relatively simple devices similar to a small dehumidifier except that they reject their heat to a water stream circulating to/from the water heater tank. They extract heat from the ambient air surrounding them, and they can be placed in any area suitable for their operation (their noise level is probably a primary consideration). During cooling season, an HPWH produces desirable cooling and dehumidifying effects in the ambient air *plus* generating hot water at about half the cost of electric resistance water heaters. The amount of dehumidifying effect that is available depends on the consumption of hot water, since the operation of the HPWH is controlled by the water tank temperature. HPWHs have been in commercial use in the U.S. for over 30 years, but have received little attention from the general public or for commercial applications. Their first cost is several times that of a simple electric or gas water heater, though they typically have paybacks of two to five years, not even accounting for the beneficial cooling and dehumidifying effects in the summer. Most residential and small commercial customers do not realize how much they spend on hot water, and so tend to be exclusively first-cost sensitive. Figure 8 shows a residential HPWH installed with a conventional water heater located in an unheated garage adjacent to a basement living space. In this instance, the cooling and dehumidifying effect indirectly reduces the sensible and latent loads on the adjacent space. If the HPWH can be installed within the living space, the cooling and dehumidifying would directly reduce the A/C equipment load in summer. In such an installation, however, the HPWH would also act as an additional heating load in winter, so its net heating cost in winter would depend on the cost of the heat provided by the primary heating system. Its dehumidifying effect would still be beneficial even outside of the primary cooling season. A tightly constructed house located in many parts of the country with relatively mild spring, fall, and winter conditions will require some humidity moderation during those seasons as well.

Moisture control is not the sole domain of the A/C system or some other mechanical device. A properly designed structure can significantly reduce the moisture load on a space relative to much of our current building stock. Lstiburek and Carmody (1991) have written an extensive report based on work at Oak Ridge National Laboratory on moisture management in houses. As requirements for ventilation increase and economic pressures result in more energy efficient structures, the need for more active humidity control will increase even with structures that are well-designed for moisture management.

FIGURE 8. Air-source heat pump water heater connected to a standard hot water tank.

SUMMARY

The challenge of designing, building, and operating sustainable buildings will require the active support of a number of disciplines. There is no doubt that air conditioning will remain an important priority for homeowners and small business owners in the United States. The current trend also suggests that mechanical ventilation will play an important role in future houses and small commercial buildings. The unique challenges of ventilation in humid climates will require a more active role in humidity management beyond the current standard practice of simply designing unitary A/C systems to provide 20% to 30% latent cooling effect. Now that manufacturers have complied with DOE's higher minimum efficiency levels, they will likely begin to focus on the role of the A/C in total comfort management, which must include active humidity control. This next step will be necessary to accommodate future code requirements for residential ventilation. While it will likely come at some cost premium to current technology, the resulting system should be able to provide a more reliable means of controlling temperature and humidity in our occupied spaces. As more of these requirements come to the surface, an integrated system would likely prove the most effective way to address the various needs of the building owner. Systems that control space air temperature and humidity, and also generate domestic hot water, will likely become the benchmark of the future, where their complimentary effects will improve the overall efficiency of these processes.

REFERENCES

ARI Directory of Certified Product Performance website. 2006. http://www.aridirectory.org/ari/

ARI Standard 320, 1998a. Standard for Water-Source Heat Pumps. Air-conditioning and Refrigeration Institute. Arlington, Virginia.

ARI Standard 325, 1998b. Standard for Ground Water-Source Heat Pumps. Air-conditioning and Refrigeration Institute. Arlington, Virginia.

ARI Standard 330, 1998c. Standard for Ground Source Closed-Loop Heat Pumps. Air-conditioning and Refrigeration Institute. Arlington, Virginia.

ASHRAE Standard 55-2004b. Thermal Environmental Conditions for Human Occupancy. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA.

ASHRAE Standard 62.1-2004c. Ventilation for Acceptable Indoor Air Quality. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA.

ASHRAE Standard 62.2-2003. Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA.

ASHRAE Standard 152-2004a. Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA.

Cummings, J. B., and J. J. Tooley. 1989. Infiltration and pressure differences induced by forced air systems in Florida residences. *ASHRAE Transactions*. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA.

Lstiburek, J. 2002. Residential Ventilation and Latent Loads. *ASHRAE Journal*. April. Vol 44, no. 4, pp. 18–22. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, GA.

Lstiburek, J., and J. Carmody. 1991. Moisture Control Handbook—New, Low-rise, Residential Construction. Report ORNL/Sub/89-SD350/1. Oak Ridge National Laboratory. Oak Ridge, Tennessee.

Shirey, D. B., H. I. Henderson, and R. A. Raustad. 2006. Understanding the Dehumidification Performance of Air-Conditioning Equipment at Part-Load Conditions. Report FSEC-CR-1537-05. Florida Solar Energy Center. Cocoa, Florida.