YORK ®
A JOHNSON CONTROLS COMPANY

SOLUTION - ENERGY SERIES
REDUCING AHU ENERGY CONSUMPTION

APPLICATION GUIDE

| Supersedes: Nothing | Form: 102.20-AG13 (806) |

INTRODUCTION

Rising energy costs directly and detrimentally affect the profit/loss statements of building owners and tenants. Heating, ventilating, and air-conditioning a commercial building often consumes more energy than any other activity in the building. According to U.S. Department of Energy studies of commercial buildings, HVAC equipment typically accounts for over 40% of a building's energy usage. Given the amount of energy HVAC systems use, improvements in equipment efficiency equate to significant reductions in building operating costs.
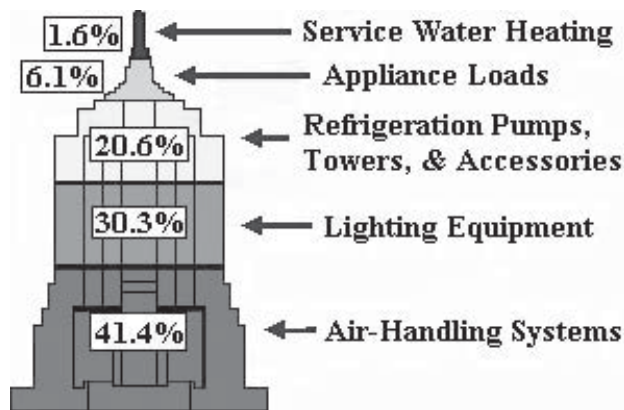
Source: ASHRAE Standard 62.1 User's Manual

FIGURE 1. ENERGY AUDIT, UNIVERSITY SCIENCE BUILDING

Energy standards and building codes have driven the design of energy efficient buildings as much as, if not more than, increased energy costs over the past decade. The "standard of care" of the HVAC industry for energy-efficient building design is ANSI/ASHRAE/IESNA Standard 90.1, the "Energy Standard for Buildings Except Low-Rise Residential Buildings." Standard 90.1 includes requirements for the building envelope, HVAC system, lighting, and other systems within a building that consume energy. Compliance with ASHRAE 90.1 is not just a good idea; in many jurisdictions it is mandatory.

The most widely used building codes in the United States are the International Code Council's (ICC) family of codes, which include the International Building Code (IBC), the International Mechanical Code (IMC), and the International Energy Conservation Code (IECC).

The IECC addresses "the design of energy-efficient building envelopes and installation of energy efficient mechanical, lighting and power systems…" There are two ways to comply with the design requirements of IECC-2003 for commercial buildings:

1) Meet the requirements of Chapter 8 of the code, which are based largely on the requirements of ASHRAE 90.1.

2) Meet the exact requirements of ASHRAE 90.1, as required by Chapter 7 of the code.

As of this writing thirty-eight states as well as the District of Columbia have adopted a version of the IEEC on the state and/or local level.

The US Green Building Council's (USGBC) Leadership in Energy and Environmental Design (LEED) Green Building Rating System has also raised awareness of the need for building designs that use energy efficiently. While several states have mandated use of the LEED rating system for government buildings, use of the system for commercial buildings is voluntary. A LEED rated building offers many benefits to owners and tenants, such as reduced operating costs, enhanced asset value and profits, improved employee productivity and satisfaction, and optimized life-cycle economic performance. These benefits have greatly increased the number of LEED registered and certified buildings over the last few years. New buildings designed for LEED certification must, as a basic prerequisite, comply with ASHRAE Standard 90.1-2004.

Clearly, energy conservation in buildings is important. Since the HVAC system is one of the primary energy users in buildings, efficient air-handling units and air distribution systems can save significant amounts of energy, reduce operating costs and comply with widely used energy standards and building codes. This application guide will focus on air handling unit design considerations that improve performance and reduce energy consumption.

AIR-HANDLING UNIT HOUSING

Two ways to reduce the heating and cooling costs in a building are to use more insulation in the walls and roof, and to reduce the air leakage (infiltration and exfiltration) through the building envelope. Increasing the insulating factor and reducing the leakage rate of the air-handling unit housing can likewise conserve energy. Air leakage and excessive heat transfer through an AHU casing causes the fans, pumps and chillers or condensing units to work harder.

Insulation

The higher the thermal resistance, or "R-value" is of the insulation in the walls of an air-handling unit, the lower the heat transfer through the walls will be. The thermal resistance of insulation is based on two factors – the thermal conductivity "K-value" of the insulation and the thickness of the insulation:

$$R = (1/K) \bullet \text{thickness} \qquad \textbf{(Eq. 1)}$$

Decreasing the thermal conductivity of the insulation, and/or increasing its thickness, increases its thermal resistance. For example, the fiberglass insulation used in many AHUs has a K-value of 0.25 btu·in/hr·ft2·°F; doubling the thickness from two inches to four inches therefore doubles the R-value:

$$R = (1/0.25) \bullet 2 = 8$$

$$R = (1/0.25) \bullet 4 = 16$$

Alternately the thermal resistance of the insulation could be increased by using denser 2" thick fiberglass, or a different material, such as a closed-cell foam insulation, with a lower thermal conductivity:

2" fiberglass: $R = (1/0.25) \bullet 2 = 8$

2" foam: $R = (1/0.16) \bullet 2 = 12.5$

When the R-value of the insulation is known, calculating the heat transfer through the casing of the air handling is a simple matter:

$$\Delta Q = \Delta T \bullet A \bullet (1/R) \qquad \textbf{(Eq. 2)}$$

Where: ΔQ = Heat flow, Btuh
ΔT = Temperature difference, F°
A = Surface area, ft^2
R = Thermal resistance, (hr·ft^2·°F)/ Btu

Example 1: Calculate the percent difference in heat flow through an AHU casing insulated with 2" foam insulation (R-12.5) and one insulated with 2" fiberglass insulation (R-8).

From Equation 2:
$\Delta Q = \Delta T \bullet A \bullet (1/R)$

Assuming the temperatures and surface areas of the two units are identical:

$$\frac{\Delta Q_{f.g.}}{\Delta Q_{foam}} = \frac{(1/R_{f.g.})}{(1/R_{foam})} = \frac{(1/8)}{(1/12.5)} = \underline{1.56}$$

Therefore, the heat transfer through the casing of an AHU insulated with 2" fiberglass will be **56% greater** than the heat transfer through the same AHU insulated with 2" foam.

For saving energy, consistent casing insulation throughout an air handling is just as important as, if not more than, the thermal resistance of the insulation. Reduced thickness or complete lack of insulation in certain sections of the AHU can lead to both increased energy consumption, and the formation of condensation on the skin of the unit. Consider a built-up AHU consisting of prefabricated frame-and-panel modules as shown in Figure 2; the frame holding the panels together can make up as much as 20-25% of the surface area of the unit. If this frame is uninsulated, the overall R-value of the AHU casing is greatly reduced. The same is true of any panels on the unit that have less than the specified thickness or R-value insulation.

To maximize thermal performance of the AHU, the frame must be insulated and sealed and the panel insulation must be consistent throughout the unit. Special care must be taken to seal all panels and framing members air-tight when fiberglass insulation is used. Leakage of
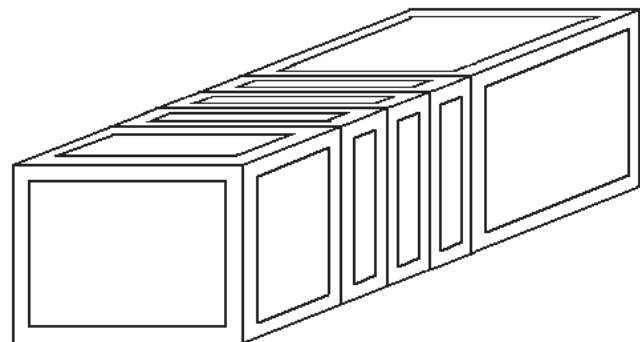
FIGURE 2. MODULAR, FRAME-AND-PANEL AHU CASING

unconditioned air into a fiberglass insulated panel or frame can cause condensation to form on the interior surfaces. This condensation can, over time, cause a collapse in the structure of the fiberglass, severely reducing its ability to insulate. Panels and frames that are insulated with injected, closed-cell foam are self-sealing and the foam insulation does not absorb water.

> **Suggested Insulation Specification Text –** All panels and structural channels shall be insulated to provide a consistent R-value throughout the unit casing. Any portion of the unit that is not insulated, or has a lower R-value than specified for the unit panels shall be the responsibility of the contractor to modify.

Leakage

Air handling unit leakage consists of infiltration and exfiltration. Infiltration is leakage of unconditioned air into the AHU from outside the casing. Exfiltration is leakage of air, typically conditioned air, from inside the AHU through the casing to the outside. Both types of leakage increase the amount of energy required to supply conditioned air to the space, and may cause other problems as well.

Minimizing infiltration reduces the flow of unconditioned ambient air entering the unit, particularly downstream of coils and filters. Uncontrolled infiltration complicates mixed air control, increases cooling and/or heating energy, and increases maintenance costs.

> **Infiltration Problems:**
>
> - Excess Coil Work
> - Potential for unfiltered air
> - Higher Maintenance Costs
> - Lack of Mixed Air Control
> - Internal Condensation

Minimizing exfiltration reduces the amount of uncontrolled air passing out through the AHU casing downstream of the supply fan. Exfiltration increases the amount of work the fan must perform to compensate for lost supply air. If the cooling or heating device is located downstream of the supply fan, the amount of work it must do increases to condition the excess air lost to exfiltration.

> **Exfiltration Problems:**
>
> - Excess Fan Work
> - Excess Coil Work
> - Potential External Condensation

In short, AHU leakage is an energy thief that often goes unnoticed until the unit forms exterior condensation (i.e. "sweats"), or the system is low on airflow. Many standard commercial AHUs have infiltration and exfiltration rates from 4% to 10% of nominal airflow depending on the operating pressures within the cabinet. That means a nominal 10,000 CFM AHU may leak 500 CFM or more at panel seams, doors, etc. If the supply airflow requirement to the occupied space is 10,000 CFM, the fan will have to work harder to produce more airflow and the coil must work harder to meet the supply air temperature.

The impact of AHU leakage will vary based upon the design of the unit and how it is used. However, in the simplest terms, for every 1% of unconditioned air that leaks into a unit, or every 1% of conditioned air that leaks out, 1% more air must be filtered, heated or cool, and moved. That means that for every 1% an air handling unit leaks it consumes 1% more energy.

Example 2: Calculate the annual energy savings associated with 1% AHU leakage compared to 5%.

Consider a building with:
- Eight (8) 16,000 CFM AHUs at 4.5" TSP.
- Two (2) 300 ton water-cooled centrifugal chillers

System *Average* Energy Use:

$$
\begin{aligned}
\text{Chiller} &= 0.55 \text{ kW/Ton} \\
\text{Pump} &= 0.10 \text{ kW/Ton} \\
\text{Cooling Tower} &= 0.05 \text{ kW/Ton} \\
\underline{\text{AHU Fan}} &= \underline{0.18 \text{ kW/Ton}} \\
\textbf{System} &= \textbf{0.88 kW/Ton}
\end{aligned}
$$

Annual HVAC Energy Use (AHEU)
   = 0.88 kW/Ton · 600 Tons · 3,120 hrs.
   = 1,647,360 kW-hrs.

Annual HVAC Energy Cost (AHEC)
   = $0.08/kW-hr · 1,647,360 kW-hrs.
   = $131,789

At 1% leakage, AHEC = 101% · $131,789
                 = $133,107

At 5% leakage, AHEC = 105% · $131,789
                 = $138,378

**Energy savings of 1% AHU leakage
compared to 5% = $5,271 per year.**

Depending upon the complexity of the air handling unit, the annual energy savings due to reduced leakage alone may amount to as much as 4% of the unit first cost. Additionally, since the air handling units are the primary energy users in a building, a 4% decrease in AHU energy consumption due to reduced leakage may be sufficient to gain an additional LEED point under the LEED Optimize Energy Performance credit, EA-1 (see table 2). The enhanced asset value, increased productivity and premium rental rates associated with LEED certified buildings further increases the value of low leakage air handling units.

TABLE 1 – LEED OPTIMIZE ENERGY PERFORMANCE

Percentage Building Energy Cost Savings Compared to ASHRAE 90.1-2004 Baseline

New Buildings	Existing Building Renovations	Points
10.5%	3.5%	1
14%	7%	2
17.5%	10.5%	3
21%	14%	4
24.5%	17.5%	5
28%	21%	6
31.5%	24.5%	7
35%	28%	8
38.5%	31.5%	9
42%	35%	10

Source: USGBC, LEED NC - Version 2.2

Due to the effect of leakage on energy usage, a specific standard, ANSI/ASHRAE/SMACNA Standard 126-2000, Methods of Testing HVAC Air Ducts, was created for ductwork throughout buildings to reduce energy usage. The standard is used to determine duct structural strength, dimensional stability, durability and leakage characteristics. The HVAC industry does not yet require a similar evaluation of AHUs, however many manufacturers already employ similar tests to evaluate their products based on a percent leakage and panel rigidity. Requiring random factory leakage testing of equipment is one more way design engineers can ensure AHUs will perform per the specifications.

Suggested Leakage Specification Text – Maximum allowable air leakage shall not exceed 1% of rated CFM when subjected to ± 8 in-w.g. static pressure. At the discretion of the customer, one random air-handling unit shall be factory leak tested.

All modern air handling units require some controls and/or sensors to operate properly and efficiently. Many times these control devices are mounted in the field by a controls contractor, which means penetrations are cut in the AHU casing by someone other than the manufacturer. Unless extreme care is taken to properly seal such penetrations, leakage and condensation

problems (i.e. sweating) can occur. Factory-mounted control devices are the logical alternative. Who better to create and seal penetrations in the unit skin than the equipment manufacturer? Whenever possible, any control devices required to be mounted in or on an AHU should be specified to be mounted at the factory, prior to an leakage testing.

Panel Deflection

Structural integrity of the air handling unit casing is directly related to its leakage rate. The more flexible the panels are on an AHU, the more likely the unit is to leak; perhaps not on the day it is installed, but over time. Repeated pressurization and depressurization of the AHU over years, months or even weeks of operation causes fatigue on the panel seams. The more the panels flex, the greater the fatigue and the quicker the failure of the seal between panels.
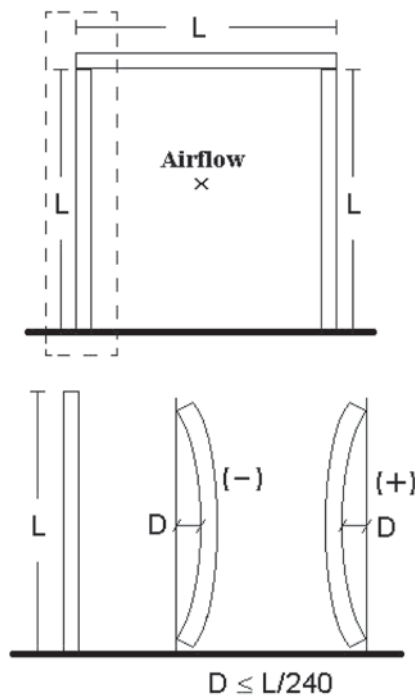
FIGURE 3. PANEL DEFLECTION

There are several techniques in practice for increasing the structural integrity of AHU panels. Using heavier gauge steel for the skin of the panels, adding stiffening ribs inside the panels and insulating the panels with rigid insulation are a few such techniques. How the panel is constructed should be of secondary concern however. How the panel performs should be the primary concern. By specifying a deflection limit for AHU panels in addition to a leakage rate, the designer and owner are assured of continued leak-tightness throughout the operating life

of the equipment. Such performance-based specifications also eliminate the tendency of manufacturers to use temporary measures, such as excessive caulking, to reduce casing leakage.

> **Suggested Deflection Specification Text –** Maximum allowable panel deflection, measured at the midpoint of the panel, shall not exceed an L/240 ratio when subjected to ± 8 in-w.g. static pressure.

AIR-HANDLING UNIT COMPONENTS

As important as tightly constructed, high R-value AHU casings are to energy conservation, proper selection of the internal components of the units is equally, if not more, important. Fans use the most energy in an AHU, however the sizing and selection of the coils, filters, dampers and every other device in the unit affects the amount of energy the fan consumes. This section will cover proper sizing and selection of internal components for maximum energy reduction.

Fans

Selecting the most efficient fans is extremely important for energy conservation in air handling units. Most air handling units use centrifugal fans because they can move large volumes of air at moderate to high pressures. Axial fans and mixed-flow fans are occasionally used; however this section will concentrate on centrifugal fans given their prevalence in AHUs.

Centrifugal fans are available with a number of blade designs – forward curved, backward curved, and airfoil, for example. These fans are also available with or without a scroll housing; unhoused centrifugal fans are often referred to as "plenum fans" or "plug fans". Each fan type is best suited to certain applications. Chapter 18 of the ASHRAE HVAC Systems and Equipment Handbook (2004) provides specific information on the application of various fan types.

Housed fans with airfoil blades are typically the most efficient of the centrifugal fans due to the aerodynamic blades and efficient conversion of velocity pressure to static pressure. Backward inclined, airfoil (AF) fans are able to move large volumes of air at high pressures. Fans with backward inclined, backward curved (BC), single thickness blades perform much the same as airfoil fans, but are slightly less efficient. Fans with forward curved (FC) blades are typically less efficient than airfoil and backward curved fans and FC fans are not capable

of operating at high pressures like AF and BC fans. In low airflow and low pressure applications however, FC fans are often more efficient than AF and BC fans. It is advisable to compare several centrifugal fan types for any application to find the most efficient.

Example 3: Calculate the annual energy cost difference between the least expensive and the most efficient fans on an AHU operating at 16,000 CFM and 4.5" TSP for 3,120 hrs. per year.

From a manufacturer's fan selection program:

Least Expensive Fan
- 20" Forward Curved Fan
- 67.3% mechanical efficiency
- 20.44 brake horsepower

Most Efficient Fan
- 28" Backward Inclined Airfoil Fan
- 80.3% mechanical efficiency
- 15.50 brake horsepower

Difference in fan energy = 4.94 hp · .746 kW/hp
= 3.68 kW

Annual Fan Energy Use = 3.68 hp · 3,120 hrs.
= **11,482 kW-hrs.**

Annual Fan Energy Cost
= $0.08/kW-hr · 11,482 kW-hr.
= **$918.56**

In all likelihood, the energy saved by selecting the most efficient fan would offset the added cost of the fan in less than one year.

Plenum fans are centrifugal fans, typically backward inclined airfoil design, which are built without the scroll housing. The absence of the housing limits the pressure against which plenum fans can operate. The mechanical efficiency of plenum fans is lower than that of comparable housed fans because the housing aids in the conversion of velocity pressure to static pressure. That's not to say that plenum fans should be avoided altogether.

Certain designs lend themselves to the use of plenum fans in lieu of housed fans. AHUs with multiple dis-

charge duct connections as shown in Figure 4 are often designed with plenum fans. The lack of a scroll housing facilitates even pressurization of the discharge plenum. Plenum fans are also often used on acoustically sensitive jobs. As a general rule, the most efficient fan on a job is also the quietest, however at low volumes and low static pressures, plenum fans are often quieter than housed fans. Acoustical lining in the discharge plenum can further reduce the sound profile of an AHU with a plenum fan.
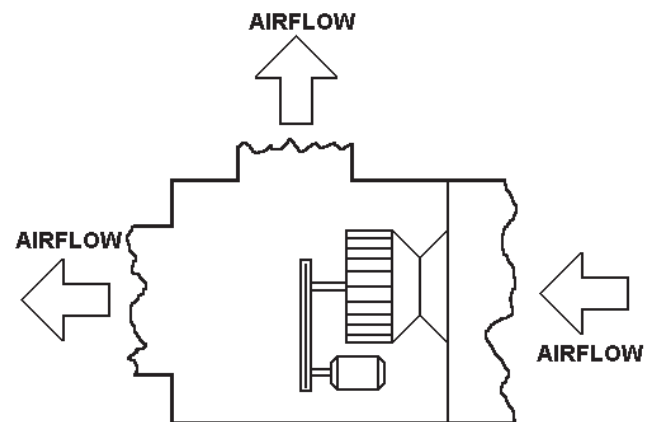
FIGURE 4. MULTIPLE DISCHARGE PLENUM FAN SECTION

The lower efficiency of plenum fans can be partially offset by a direct drive arrangement. Belt drives, which are used to drive most centrifugal fans, include inherent inefficiencies. These inefficiencies, or "belt losses", increase the horsepower required to drive the fan by five to ten percent. In certain cases eliminating belt losses by mounting the fan impeller directly on the motor shaft as shown in Figure 5 is sufficient to overcome the lower efficiency of a plenum fan.
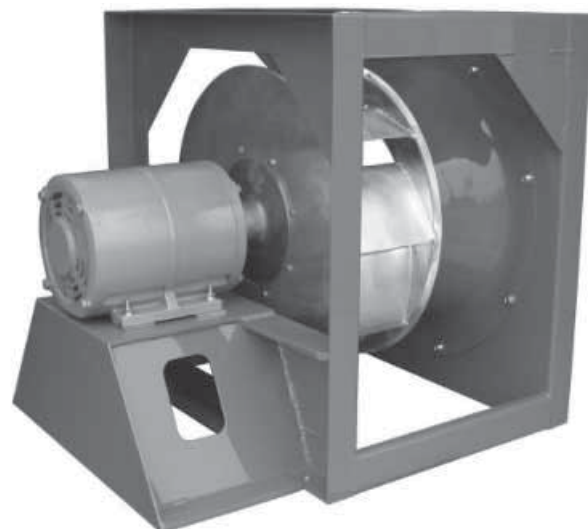

FIGURE 5. DIRECT DRIVE PLENUM FAN
(Courtesy Twin City Fan & Blower)

Variable Speed Drives

Variable air volume (VAV) control is the most cost-effective approach to AHU design. By matching air volume to actual cooling loads, VAV cuts air handling unit energy consumption by 20% to 60% versus constant air volume systems. In order to vary the air volume of a fan, it must either be mechanically throttled, or its rotational speed must be controlled. Mechanical fan volume controllers such as discharge dampers and variable inlet vanes reduce fan energy consumption somewhat, but they cannot match the saving obtainable with a variable speed drive (VSD, sometimes also called a VFD, or variable *frequency* drive).

Variable speed drives take full advantage of the relationship between fan speed and horsepower – any given reduction in fan speed results in a cubic reduction in fan horsepower. For example, a 10% reduction in fan speed results in a 27% reduction in fan horsepower. A 50% speed reduction translates to an 87% horsepower decrease. Figure 6 illustrates the relative savings offered by VSDs compared to variable inlet vanes and discharge dampers.
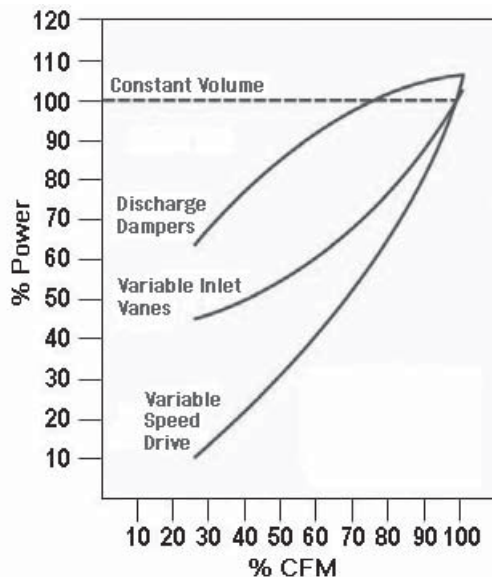
FIGURE 6. FAN VOLUME CONTROL

Section 6.5.3.2 of ASHRAE Standard 90.1 requires VAV fans with motors 15 hp and larger to meet one of three criteria:

(a) The fan is to be driven by a mechanical or variable-speed drive.

(b) The fan is to be a vane-axial fan with variable pitch blades.

(c) The fan is to have other controls and devices that will result in motor demand of no more than 30% of the design wattage at 50% of the design airflow when the static pressure setpoint equals one-third of the total design static pressure, based on the manufacturer's certified fan data.

In most cases, fans with discharge dampers or variable inlet vanes will not meet criterion (c). Plenum fans with variable inlet cones may meet the third criterion, but the fan manufacturer must provide certified data to support any such claim.

Coils and Filters

Any obstruction or appurtenance in the airstream increases the air pressure against which the supply fan must work to move the air. Coils and filters are the most common of these appurtenances since one or both are included in nearly every air handling unit. The "air pressure drop" (so called because the air pressure is higher on the upstream side of the device than the downstream side) is affected by several factors.

The air pressure drop (APD) of a coil is affected by several factors including the number of rows of tubes, the fin density, and the face velocity of the coil, as well as whether the coil condenses moisture out of the air. Table 2 lists typical air pressure drops for dry (non-condensing) coils at 500 feet-per-minute face velocity.

TABLE 2 – TYPICAL COIL AIR PRESSURE DROPS

| Rows	Dry Coil Air Pressure Drop at 500 fpm			
	90 Fins/Ft	120 Fins/Ft	150 Fins/Ft	168 Fins/Ft
4	0.30	0.35	0.41	0.47
6	0.45	0.52	0.62	0.71
8	0.60	0.69	0.83	0.94
10	0.65	0.85	1.13	1.31
12	0.78	1.03	1.36	1.57

Source: ASHRAE Standard 62.1 User's Manual

Many designers minimize the first cost of equipment by selecting the smallest practical AHU. While this practice reduces material costs, it invariably increases energy costs. When the AHU size is decreased, the face area of the coils (and filters) must also be reduced. Reducing the coil face area reduces the heat transfer surface; therefore it is usually necessary to increase the fin density and/or depth of the coil. A reduction in face area also results in an increase in face velocity and air pressure drop across the coil.

Consider a 4 row cooling coil with 120 fins per inch selected at 500 fpm that is capable of satisfying the design cooling load of an AHU. If the face area of the

coil is decreased by 17%, the face velocity increases to 600 fpm and the depth of the coil must be increased to 6 rows to provide sufficient heat transfer surface to meet the design cooling load. These changes more than double the air pressure drop across the coil from 0.35 in-w.g. to 0.71 in-w.g., which in turn increases the amount of work the fan must do.

For filters, the depth and construction of the filter, and the face velocity influence the air pressure drop across the filter. As with coils, filter APD increases as the face velocity increases. Figure 7 shows the typical effect of increased face velocity on filter APD. As shown, an increase in face velocity from 500 fpm to 600 fpm increases the APD across this particular filter from 0.53 in-w.g. to 0.72 in-w.g.
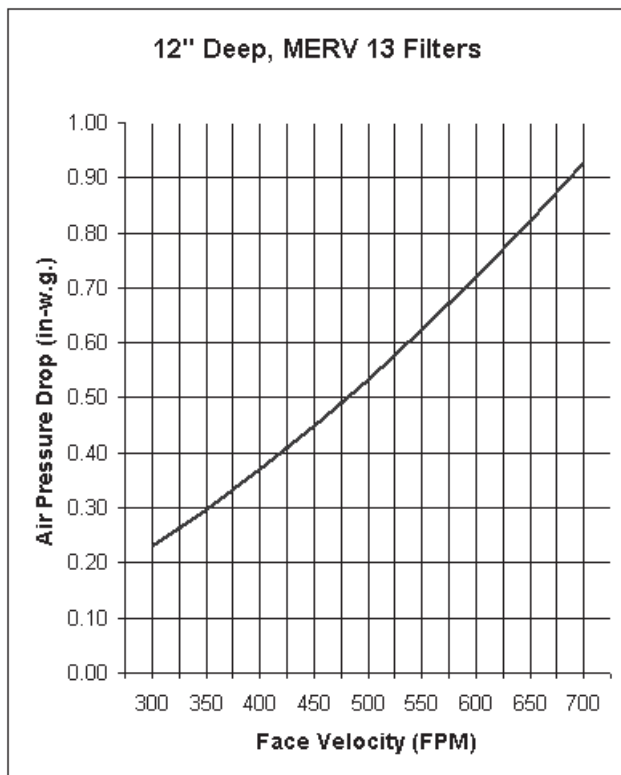

FIGURE 7. FILTER AIR PRESSURE DROP

Many AHU manufacturers offer prefilters in an angled filter bank arrangement to maximize the filter surface area and reduce the face velocity and pressure drop through the filters. Angled or "V-bank" filters can increase the surface area of a bank of 2" prefilters by as much as 50-60%, reducing the face velocity and pressure drop of the filters by commensurate amounts.

Limiting the face velocities of coils and filters to close to 500 fpm will result in significant energy cost savings. In fact, the Fan Power Limitation of ASHRAE Standard 90.1-2004 is based on selecting coils and filters at 400 fpm. Therefore, it will be difficult at best to meet this

prescriptive requirement of Standard 90.1 with coils and filters selected for 550-600 fpm.
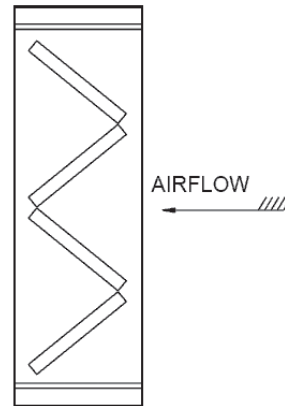
FIGURE 8. V-BANK FILTER ARRANGEMENT

Example 4: Calculate the difference in fan horsepower and annual energy cost between a 16,000 CFM air handling unit designed for 500 fpm coil/filter face velocity and one designed for 600 fpm.

From a manufacturer's AHU selection program:

500 fpm unit:
- Total APD = 4.56 in-w.g.
- Fan hp (including belt loss) = 15.73 bhp
- Unit price ≈ $18,000.

600 fpm unit:
- Total APD = 5.11 in-w.g
- Fan hp (incl. belt loss) = 22.91 bhp
- Unit price ≈ $16,900.

Difference in fan energy = 7.18 hp · .746 kW/hp
= 5.36 kW

Annual Fan Energy Use = 5.36 kW · 3,120 hrs.
= **16,723 kW-hrs.**

Annual Fan Energy Cost
= $0.08/kW-hr · 16,723 kW-hr.
= **$1337.86**

The additional $1,100 in first cost for the 500 fpm unit is offset in less than one year by the energy savings realized by the lower APD.

Larger coils and filter banks require larger air handling units; however equipment first cost can be minimized by selecting AHUs with a variable aspect ratio. Manufacturers of central station AHUs have historically offered a limited number of AHU cabinet sizes, with large differences in face velocities between sizes. Such archaic manufacturing techniques have largely been replaced by variable aspect ratio (VAR), which allows the height and width of an AHU to be adjusted in small (e.g. 3-inch) increments.

Design engineers who want a 16,000 cfm unit no longer have to choose between an oversized AHU with 445 fpm coils and a smaller unit with an excessive 539 fpm coil face velocity. Variable aspect ratio allows the designer to tailor the unit size to within 1 or 2 fpm of the desired coil velocity. This optimizes both the first cost and the energy consumption of the equipment.

Dampers

ASHRAE Standard 90.1-2004 limits the leakage rates of dampers as shown in Table 3 and Figure 9. These limitations are Mandatory Provisions of Standard 90.1. Air handling units with dampers that do not meet these leakage rates prevent the building in which they are installed from meeting ASHRAE 90.1. In the LEED NC rating system, non-compliance with a Mandatory Provision of ASHRAE 90.1 disqualifies the building from LEED certification.

**TABLE 3 –** ALLOWABLE DAMPER LEAKAGE RATES

Climate Zones	Maximum Damper Leakage at 1.0 in w.g. cfm per ft2 of damper area	
	Motorized	Non-motorized
1, 2, 6, 7, 8	4	Not allowed
All others	10	20(a)
(a) Dampers smaller than 24" in either dimension may have leakage of 40 cfm/ft2		

Source: ASHRAE Standard 90.1-2004

Damper leakage wastes as much or more energy than AHU casing leakage. Consider a 16,000 cfm AHU with a 96" high x 48" wide (500 fpm) outdoor air damper. At a leakage rate of 10 cfm/ft2, such a damper leaks 320 cfm when closed – that's equivalent to 2% of design airflow. If the casing is designed for less than 1% leakage, it only makes sense to select dampers that will not compromise the cabinet integrity. The same 96" x 48" damper with a leakage rate of 4cfm/ft2 only leaks 128 cfm, or 0.8% of design; this damper would match the energy saving quality of the cabinet.

Outdoor air dampers with integral airflow-measuring stations can save energy and ensure satisfactory indoor air quality (IAQ). ASHRAE Standard 62.1-2004 prescribes ventilation rates for a number of building types to provide indoor air quality that will be acceptable to occupants. If too little ventilation air is provided, occupants may complain of stuffiness, objectionable odors, or in extreme cases, health problems. If more
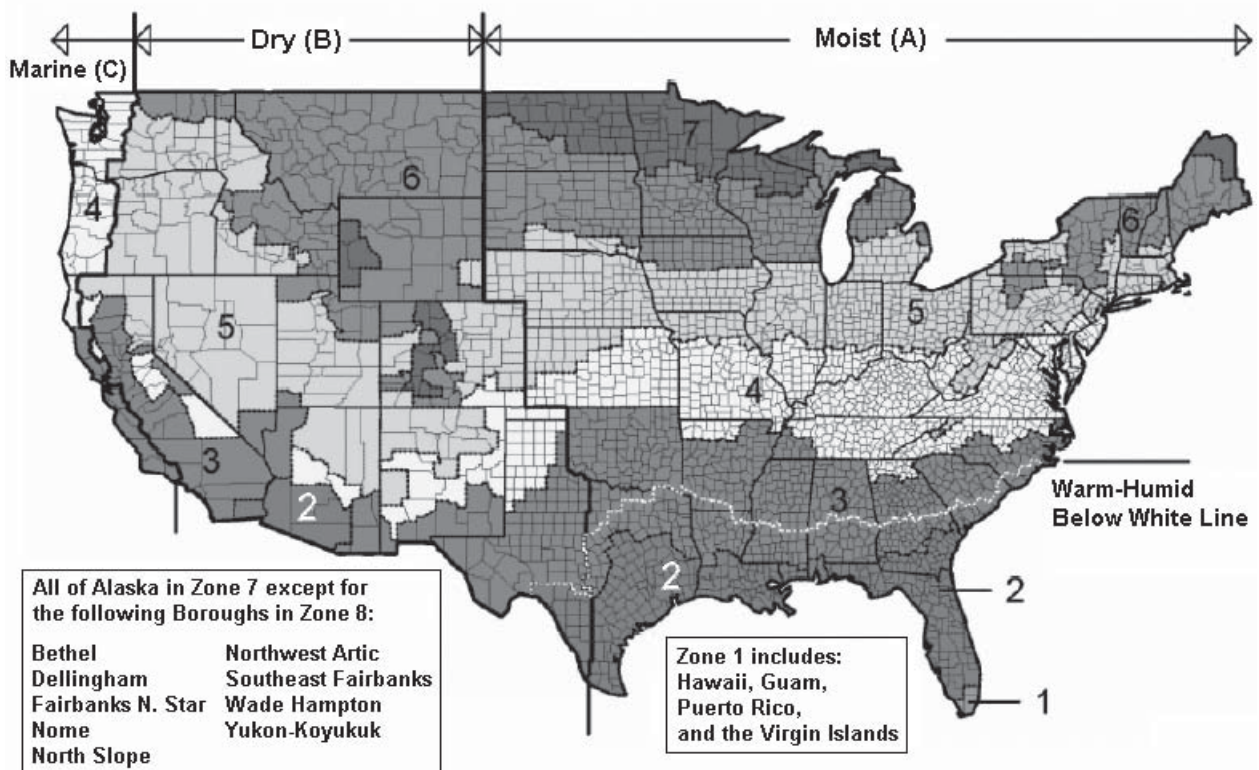
FIGURE 9. UNITED STATES CLIMATE ZONES

outdoor air than prescribed is supplied to the occupied spaces during minimum outdoor air operation, the cost of conditioning the air will increase. Measuring the outside air with an airflow-measuring station that is certified in accordance with AMCA Standards 610 and 611 maintains ventilation rates for good IAQ and eliminates heating and cooling of excess outside air when not in economizer mode. Such dampers must, of course, meet the leakage limitations set by ASHRAE Standard 90.1.

Suggested Airflow Monitoring Station Specification Text – The airflow monitoring station must be tested for pressure drop in accordance with AMCA Standard 611-95 in an AMCA registered laboratory. The airflow monitoring station must bear the AMCA Certified Ratings Seal for

Energy Recovery Devices

Exhaust air energy recovery devices conserve a great deal of energy. Rotary heat exchangers, heat-pipes, plate heat exchangers, etc., all operate on the same principle - a transfer of energy between the exhaust airstream and the supply airstream. The exhaust air preheats the supply air in the winter and pre-cools the supply air in the summer.

Some energy recovery devices transfer only sensible energy while others transfer both sensible and latent energy, the latter type of device is called a total energy recovery device. The ratio of the amount of energy transferred by the energy recovery device to the difference in energy levels of the two incoming airstreams is called effectiveness.

ASHRAE Standard 90.1 requires the use of energy recovery devices on *"individual fan systems that have both a design supply air capacity of 5000 cfm or greater and have a minimum outdoor air supply of 70% or greater of the design supply air quantity."* The devices must have an effectiveness of at least 50%. The ASHRAE requirement generally only applies to 100% outdoor air systems and those that require a high volume of ventilation air such as schools, theaters, and laboratories. However, systems with lower minimum outdoor air requirements can benefit from exhaust air energy recovery.

Example 6: Calculate the reduction in required cooling energy for a 16,000 cfm AHU with a *total* heat recovery wheel, 4,000 cfm of outdoor air (25%) at 94°Fdb and 77°Fwb; and 4,000 cfm of exhaust air at 75°Fdb and 50% RH, if both the sensible & latent effectiveness of the wheel is 60%.

At the conditions given above:
The exhaust air enthalpy = 28.15 btu/lb. = h_3
The outdoor air enthalpy = 40.38 btu/lb. = h_1

Supply air temperature:
$t_2 = t_1 - [\varepsilon \cdot V_{min} \cdot (t_1 - t_3)]/V_s$
$t_2 = 94 - [0.6 \cdot 4,000 \cdot (94 - 75)] / 4,000$
$\underline{t_2 = 82.6°F}$.

Supply air enthalpy:
$h_2 = h_1 - [\varepsilon \cdot V_{min} \cdot (h_1 - h_3)]/V_s$
$h_2 = 40.38 - [0.6 \cdot 4,000 \cdot (40.38 - 28.15)] / 4,000$
$\underline{h_2 = 33.04 \text{ btu/lb}}$.

Therefore, the supply air leaving the heat recovery wheel will be: **82.6°Fdb / 68.9°Fwb**

Cooling energy reduction:
$Q_t = \varepsilon \cdot 4.5 \cdot V_{min} \cdot (h_1 - h_3)$
$Q_t = 0.6 \cdot 4.5 \cdot 4,000 \cdot (40.38 - 28.15)$
$\underline{\mathbf{Q_t = 132,084 \text{ btu/hr} = 11 \text{ tons}}}$

Although the minimum outdoor air rate for this unit is only 25%, the energy recovery device reduces the required design cooling energy by 11 tons.

When using an exhaust air energy recovery device, design engineers should be sure to use certified equipment. Heat pipes, plate heat exchangers, and rotary heat exchangers should be certified per ARI Standard 1060 - Standard for Rating Air-to-Air Energy Recovery Ventilation Equipment. Specifying certified equipment guarantees the published performance of the device and allows the engineer to make comparisons between competitive equipment with confidence. For more

information on the principles of energy recovery, and the equations used in Example 5, see JCI/York Publication 102.20-AG6 – Air Systems Application Guide for Energy Recovery Wheels.

Suggested Exhaust Air Energy Recovery Specification Text – Thermal performance shall be certified by the manufacturer in accordance with ASHRAE Standard 84-1991, Method of Testing Air-to-Air Heat Exchangers and ARI Standard 1060-2000, Rating Air-to-Air Energy Recovery Ventilation Equipment.

SUMMARY

In these days of ever-escalating energy costs, the value of high performance air handling units can not be ignored. AHUs with low-leakage, high-thermal resistance casings and energy efficient components are a must for modern, environmentally responsible buildings. The low energy footprint of a high performance AHU reduces the cost of ownership of a building. Energy efficient HVAC equipment also enhances the asset value of a building through recognition as a "Green Building" by organizations such as the USGBC, and by providing greater occupant satisfaction and/or productivity.

As shown in this document, reduced energy consumption does not equate to huge increases in the first cost of the equipment. Thanks to modern manufacturing techniques such as Mass Customization, most of the increases in first cost required to ensure energy efficiency are so minor as to have a payback period of one year or less. The following checklist is provided as a reminder of the steps that the design engineer should take to ensure the owner receives a higher performance, energy efficient air handling unit.

High Performance AHU Checklist

☐ **Specify High Thermal Resistance:** Casing panels and structural channels fully insulated to provide a consistent R-value of 12.5 or higher throughout the AHU.

☐ **Specify Low Leakage Casing:** Leakage less than or equal to 1% of design CFM at ±8 in-w.g. static pressure.

☐ **Specify High Structural Integrity:** Panel deflection ≤ L/240 at mid-point of panel at ±8 in-w.g. static pressure.

☐ **Select High-Efficiency Fans.** Consider direct drive arrangement when plenum fans are required.

☐ **Specify Variable Speed Drives.** Use VSDs in lieu of variable inlet vanes or discharge dampers on VAV fans.

☐ **Limit Coil and Filter Face Velocities:** Select AHUs to allow for coil and filter face velocities of 500 fpm or less. Consider angle filter banks for prefilters.

☐ **Specify Low Leakage Dampers:** Maximum damper leakage rates should correspond to the mandatory requirements of ASHRAE Standard 90.1.

☐ **Specify AMCA Certified Airflow Monitoring Stations:** Airflow monitoring should be used to verify outdoor airflow rates, particularly in VAV applications. Specify airflow monitoring devices certified in accordance with AMCA Standard 611.

☐ **Consider Using Exhaust Air Energy Recovery:** Even when not prescribed by ASHRAE 90.1, energy recovery can save significant amounts of energy. Specify energy recovery devices with a minimum effectiveness of 50%, and which are certified in accordance with ARI 1060.

If you would like more information on specifying AHU performance and design requirements for a successful IAQ application, please contact your local Johnson Controls/York sales office.

REFERENCES

Energy Information Agency. 2001. *Commercial Buildings Energy Consumption Survey.* Washington, DC: U.S. Department of Energy.

ASHRAE. 2004. *ANSI/ASHRAE/IESNA Standard 90.1 – 2004 Energy Standard for Buildings Except Low-Rise Residential Buildings.* Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

ICC. 2006. *International Energy Conservation Code.* Country Club Hills, IL: International Code Council, Inc.

USGBC. 2005. *LEED-NC – Green Building Rating System For New Construction & Major Renovations, Version 2.2.* Washington, DC: U.S. Green Building Council.

ASHRAE. 2004. *ASHRAE Handbook – 2004 HVAC Systems and Equipment,* chapter 18. Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

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

ASHRAE. 2005. *ASHRAE Standard 62.1 – 2004 User's Manual.* Atlanta, GA: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

York International 2005. *Form 102.20-AG6, AIR SYSTEMS – ENERGY SERIES – Energy Recovery Wheels*, York, PA: York International Corporation.

York International 2005. *Form 102.42-SG2, Variable Speed Drives for HVAC Applications*, York, PA: York International Corporation.