UNIT VENTILATORS

A PRIMER

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INTRODUCTION

Since before the 1950s, unit ventilators have been standard elements in the physical character of elementary and high schools. For those educated prior to the 1990s, unit ventilators perched under a wall full of windows are as much a part of classroom memories as are chalk boards, crank pencil sharpeners, and recess. Their reason for being there? Outdoor air.

Introduced before World War II, the idea of a unit ventilator was to have an independent apparatus that would serve the heating and ventilation requirements of an individual classroom, office, laboratory or other similar space. Operable windows provided ventilation and cooling during autumn and spring, but the casements were closed and locked when winter arrived. Leaving a window cracked open to provide fresh air (leaking cold air in and warm air out) was always an unsatisfactory solution. The unit ventilator provided a controlled means of introducing conditioned outdoor air during the heating season. Manufacturers later added cooling coils to their list of options, increasing the unit ventilator's versatility and range of applications.

Although unit ventilators have not changed significantly in outward appearance, the internal technology has continued to advance to meet changing needs. They are fundamentally simple machines that have evolved to keep pace with increasingly complex building codes and regulations. Indoor air quality, humidity control and energy consumption have advanced to the forefront; but engineers and facility owners still rely successfully on the unit ventilator as a leading solution in meeting their requirements.

This white paper is published on the occasion of Carrier's introduction of an upgraded version of the venerable Model 40 UV/UH unit ventilator. As a primer, its purpose is to review design, construction, selection and application of unit ventilators.

THE UNIT VENTILATOR–WHAT'S INSIDE

An appealing feature of unit ventilators is their compact size. Inside a relatively small package, manufacturers assemble fans and motor, heating and cooling coils, drain pan, return air damper, outdoor air damper, face and bypass damper, air filter, piping, valves and controls. These are essentially the same components that go into a built-up air-handling unit, but in a unit ventilator they fit in the space of 8 square feet for a 500-cfm system, and up to 15 square feet for a 2,000-cfm system. Figures 1 and 2 show the components of a typical Carrier Model 40UV vertical unit ventilator.

The basic unit ventilator is a vertical model packaged for floor mounting and to fit below the window sill on an exterior wall as shown in Fig. 3. In the traditional arrangement, conditioned air discharges vertically from the cabinet top and return air enters at the front near the floor. Because a unit ventilator mounts against an outside wall, outdoor air enters the system from the back side through a louvered opening in the wall or through the bottom of the unit.

A common variation of the basic unit ventilator is the horizontal design (Model 40UH) that is installed at or above the ceiling. Conditioned air from a horizontal unit ventilator discharges either horizontally or vertically down. Recessed units installed above the ceiling level are often equipped with collars at the discharge, return and outdoor air openings to attach ductwork.

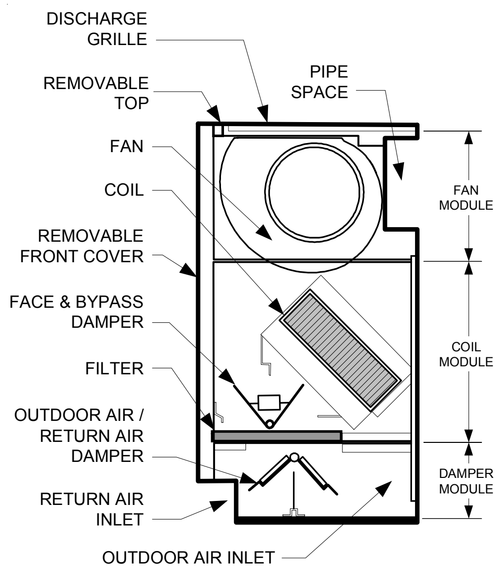

Fig. 1. Section through a typical Carrier Model 40UV showing internal components

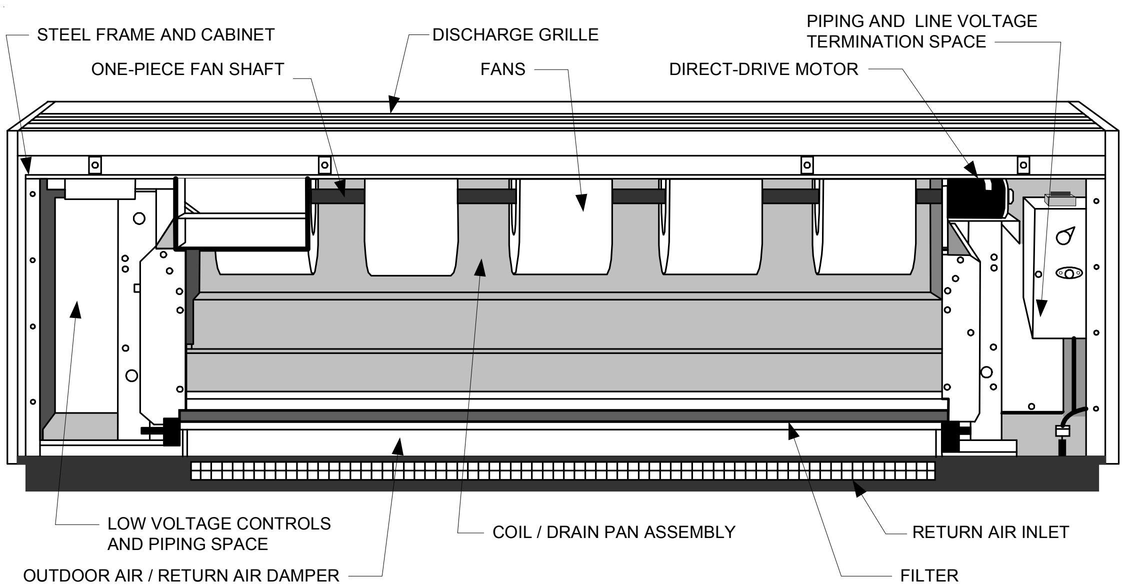

Fig. 2. Carrier Model 40UV with the front panels removed showing internal components

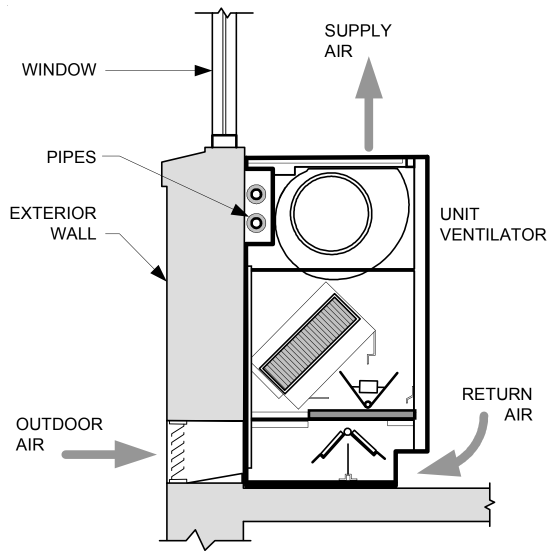

Fig. 3. Exterior wall section showing Carrier Model 40UV installed below window sill

Fans

To achieve the desired airflow distributed over the length of the cabinet and to keep the overall package compact, unit ventilators use multiple small centrifugal fans. The fans are mounted on a common shaft that is turned by a single motor. The fans are commonly double-width, double-inlet type with forward curved blades, and are the only continuously moving parts in the system. Carrier's fans are specifically designed for quiet operation.

Coils

Coil options are rarely a limitation in unit ventilator selection and application. Heating coils can be steam, hot water, or electric. Cooling coils are either chilled water or direct-expansion type. Hydronic and direct expansion coils have conventional constructions using copper tubes and mechanically bonded aluminum fins. Open-wire coils are the most frequently used electric heating elements.

Dampers

Dampers are custom designed features of unit ventilators. Because unit ventilators must be compact, traditional multiple-blade, parallel and opposed blade dampers used in large air-handling systems will not fit. Unit ventilator dampers are specially designed and fabricated to fit in the available airflow space.

Cabinets

Manufacturer's specifications often include the words, "heavy gauge steel" when describing cabinet construction for two reasons. First, the simple marketing implication is that "heavy-gauge" is perceived as being *better* than "light gauge." Second, and more important, is that the dominant market for unit ventilators is schools. Because a unit ventilator effectively becomes a classroom furnishing, used as a seat, shelf and step ladder (not recommended), it has to be well constructed, solid and able to resist the inherent abuse foreseeable in a school setting.

The basic unit ventilator (common to many manufacturers) starts with a steel frame. The fans, coils and other components mount to the frame and are enclosed with painted steel panels. Panel inside surfaces are insulated with fiberglass for both thermal and acoustic reasons. The front panels are removable to provide maintenance access, and held in place with tamper-resistant fasteners. Panel color is often an option.

Carrier's new unit ventilator model uses an innovative modular concept with a removable front and top panels. Components are assembled in three modules: fan module, coil module and damper module. Each module is individually insulated with a closed-cell urethane foam insulation rather than the cabinet panels. Also, each module is removable from the unit independent of the others.

Ancillary Space

In spite of being filled with fans, coils and other hardware, unit ventilator designers always leave space at the cabinet ends for piping, valves, wiring and controls.

Accessories

Vertical unit ventilators (Model 40UV) can be furnished with a variety of architectural accessories (e.g., cabinets, shelves, and sink units) that can be mixed in any number of ways to create a perimeter storage system for schools, hospitals and institutions.

Draft stops, which are narrow steel enclosures painted to match the unit ventilator, are another common accessory. They have a louvered top and are mounted to the wall below the window sill. As the name implies, draft stops intercept cold down drafts falling off the interior window surface and direct the cold air to the unit ventilator return. When combined with draft stops, the standard unit ventilator return air opening is blocked and the return air flows through adjoining draft stop cabinets.

APPLICATIONS

Unit ventilators find their dominant market in schools. More accurately, *classrooms* are the typical application for unit ventilators. However, they are used in any situation involving room-by-room zoning and outdoor air requirements. Other common applications include nursing homes and health care facilities.

Unit ventilators are distinguished from cabinet heaters, unit heaters and fan-coil units by their ability to introduce conditioned outdoor air to the occupied space. Their purpose is in the name: *ventilator*. A designer's ability to apply unit ventilators to different facilities is limited only by their advantages and disadvantages.

Advantages

Unit ventilators:

- Provide heating, cooling and ventilation from a single unit
- Are able to provide economic, room-by-room zoning
- Have adjustable outdoor airflow up to 100% outdoor air
- Have a small physical space requirement
- Minimize the risk and nuisance of cross-contamination between adjacent spaces from noise, odors and other contaminants
- Can provide energy efficient operation including an air-side economizer
- Enhance building-wide reliability because a single failure affects only one room, not the entire facility
- Are easy to install
- Have relatively low installed first cost
- Are easily accessible for filter replacement and maintenance
- Have simple mechanical designs and controls that can be understood by a variety of maintenance personnel
- Can be coordinated and controlled through a building automation system, or can function as a simple, thermostatically-controlled stand-alone system

Disadvantages

Unit ventilators:

- Are a source of noise—vertical unit ventilators are located in the room, sound attenuation with lined ductwork and physical separation is not available
- Have supply airflow limited to about 2,000 cfm per unit
- They are an architectural element of the occupied space and are not easily adapted to facilities where HVAC components need to be out of sight.

OUTDOOR AIR

For the modern design professional, having the correct amount of properly conditioned outdoor air is like shooting at a moving target on a foggy, moonless night. Too much outdoor air, and the facility energy costs are higher than necessary, higher than allowed by code, or worse, higher than the facility owner expected. Too little outdoor air and the occupants are uncomfortable and unhappy. In a classroom setting with too little ventilation, the students may have greater difficulty learning.

ASHRAE Standard 62, *Ventilation for Acceptable Indoor Air Quality* (ASHRAE 1989, 1999 and 2001), is the industry guideline for determining outdoor air requirements and its various editions have been adopted as part of many major building codes. The code is under "continuous maintenance," meaning it is frequently revised by amendment. In the 1989 and 1999 editions, using the Ventilation Rate Procedure, ASHRAE 62 required outdoor airflow on a cfm-per-person basis. Requirements for selected applications are shown in Table A.

Addendum 62n to ASHRAE Standard 62-2001 changed the Ventilation Rate Procedure. Rather than a simple airflow rate per person, the revised standard recognizes that indoor air pollutants are generated by both occupants and the building itself. As such, the standard defines ventilation requirements as the additive sum of two components: a people component and an area component.

Table B shows the revised requirements for several applications. Default values shown on an airflow-per-person basis are based on the indicated occupancy density. Note that the application names and

Table A
Outdoor Air Requirements for Ventilation
From ASHRAE Standard 62-1999 (Table 2)

Application	Est. Maximum Occupancy (P/1000 ft² or P/100 m²)	Outdoor Air Requirements	
		CFM/person	L/sec-person
Classroom	50	15	8
Laboratories	30	20	10
Training shop	30	20	10
Music Rooms	50	15	8
Libraries	20	15	8
Patient Rooms	10	25	13
Medical procedure	20	15	8
Physical therapy	20	15	8
Conference rooms	50	20	10
Assembly rooms	120	15	8
Dormitory sleep areas	20	15	8

Table B
Minimum Ventilation Rates In Breating Zone
From ASHRAE Standard 62-2001 Addendum 62n (Table 6.1)

Application	People Outdoor Air Rate		Area Outdoor Air Rate		Default Values		
					Est. Maximum Occupancy (P/1000 ft² or P/100 m²)	Combined Outdoor Air Rate (Note 1)	
	CFM/person	L/sec-person	CFM/person	L/sec-person		CFM/person	L/sec-person
Daycare (thru age 4)	10	5	0.18	0.9	25	17	8.6
Classroom (ages 5 thru 8)	10	5	0.12	0.6	25	15	7.4
Classroom (age 9 plus)	10	5	0.12	0.6	35	13	6.7
Lecture classroom	7.5	3.8	0.06	0.3	65	8	4.3
Art classroom	10	5	0.18	0.9	20	19	9.5
Science laboratories	10	5	0.18	0.9	25	17	8.6
Wood / metal shop	10	5	0.18	0.9	20	19	9.5
Computer lab	10	5	0.12	0.6	25	15	7.4
Multi-use assembly	7.5	3.8	0.06	0.3	100	8	4.1
Conference / meeting	5	2.5	0.06	0.3	50	6	3.1
Libraries	5	2.5	0.12	0.6	10	17	8.5

Notes:
1. Based on default estimated maximum occupancy shown.

Table C
Outdoor Air Enthalpy Calculations for Select Cities at Dry-Bulb and Dew-Point Design Conditions

City	Dry-Bulb Design Conditions				Dew-Point Design Conditions				
	Fdb (Note 1)	MWB (Note 1)	W	h	DP (Note 1)	HR (Note 1)	MDB (Note 1)	W	h
Charleston	92	77	0.016637	40.429	77	139	83	0.020170	**42.077**
Denver	90	59	0.003627	25.617	58	90	68	0.010309	**27.583**
Kansas City	93	75	0.014613	38.446	74	130	84	0.018194	**40.157**
Minneapolis	88	71	0.012433	34.816	71	116	81	0.016395	**37.439**
Mobile	92	76	0.015730	39.430	76	139	83	0.019491	**41.332**
Philadelphia	89	74	0.014679	37.532	74	126	81	0.018194	**39.412**
Portland	86	66	0.009129	**30.694**	60	78	72	0.011087	29.413
San Antonio	96	73	0.012211	36.538	75	135	81	0.018833	**40.113**
San Francisco	78	62	0.008236	**27.761**	58	73	66	0.010309	27.093

Notes:
1. Source: ASHRAE, *2001 Fundamentals Handbook*, Chapter 6.
2. Worst case outdoor air enthalpy condition is shown in bold type.
3. Variables:

DP Dew-point temperature (deg. F)
Fdb Dry-bulb temperature (deg. Fdb)
h Specific enthalpy of moist air (Btu/lb-deg F)
HR Humidity ratio (grains of moisture / lb of dry air)
MDB Mean coincident dry-bulb temperature (deg. Fdb)
MWB Mean coincident wet-bulb temperature (deg. Fwb)
W Humidity ratio (lb of water / lb of dry air)

default ventilation rates for 2001 differ somewhat from those indicated in the 1999 edition.

HUMIDITY

A U.S. Department of Energy study of humidity and ventilation control in 10 Georgia schools (Bayer 2000) concluded that: "most IAQ [indoor air quality] problems in school facilities can be avoided by providing adequate outdoor air ventilation on a continuous basis (15 cfm/student), controlling the indoor relative humidity between 30% and 60% and providing effective particulate filtration of the outdoor air."

In rooms (or facilities) with relatively dense occupancies, such as classrooms, outdoor air is the main source of moisture. As such, humidity control and outdoor air control are inseparably linked. Other moisture contributors are people, infiltration and internal sources (e.g., cooking equipment and coffee makers); but they are often secondary in size to outdoor air. The challenge of controlling relative humidity in the occupied space is largely a challenge of controlling moisture in the outdoor air stream.

ASHRAE Standard 62-2001 addresses moisture in Addendum 62x by limiting the allowable relative humidity in the occupied space to 65% or less at either of two design conditions:

- At the peak outdoor dew-point design conditions and peak indoor design latent load, or
- At the lowest space sensible heat ratio expected to occur and the concurrent (simultaneous) outdoor condition.

ASHRAE Standard 62 also notes that the outdoor air dry bulb temperature, solar load and sensible heat ratio in the space may be substantially different at outdoor dew-point design conditions than when calculated at outdoor dry-bulb conditions.

Consider the following examples. The 1% ASHRAE design data for several cities, both dry-bulb conditions and dew-point conditions, are shown in Table C. The humidity ratio and specific enthalpy for moist air at each condition have been calculated and are also shown. In seven of nine example cities the 1% dew-point design conditions present the worse case scenario for outdoor air design. The 1% dry-bulb design conditions are the worst case in only two. Note also that dew-point data governs outdoor air design conditions in a variety of geographical locations and is not a situation limited to southern, humid climates.

To demonstrate enthalpy differences graphically, the 1% dew-point and 1% dry-bulb conditions for Kansas City are plotted on a psychrometric chart in Fig. 4.

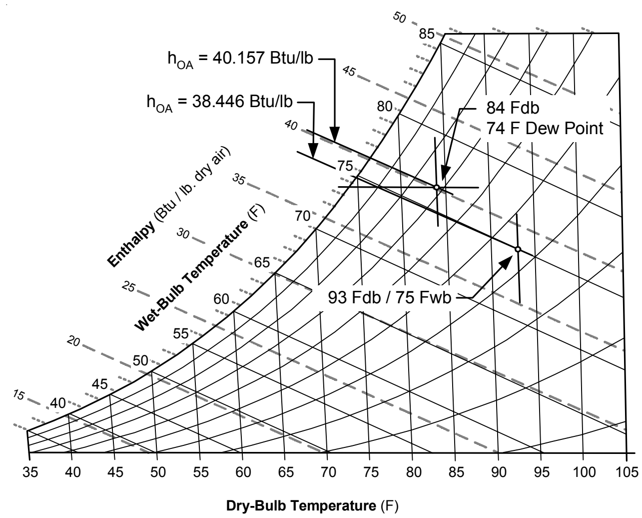

Fig. 4. ASHRAE 1% outdoor air dry-bulb and dew-point design conditions for Kansas City, Misouri

Addendum 62x to ASHRAE Standard 62-2001 provides an additional humidity control feature by requiring the design minimum outdoor air intake for a complete building be greater than the design maximum exhaust flow. In other words, the standard now requires buildings to be pressurized. The standard recognizes that certain individual spaces or zones may have specific, purposeful requirements for a neutral or negative pressure condition; however, the overall building condition should be maintained at a positive pressure.

ACOUSTICAL CONSIDERATIONS

Although sound and noise considerations are customary for most commercial and institutional HVAC systems, they are especially important in a classroom setting. There is mounting evidence that children are inefficient, immature listeners whose speech and perception skills are not fully developed until reaching adolescence. Their lack of development is compounded by several factors (Nelson 2003):

- The ability to understand words in a reverberant, noisy environment is a developed skill that does not reach adult levels until the teen years.
- Up to 20% of a school population may have a hearing loss as a result of illness (e.g., ear infections) or other congenital, genetic and environmental causes that neither the students nor their parents may be aware of.
- A large number of children learn in a language different than the one spoken in their home as a result of changing societal demographics.
- Many children have difficulty focusing their attention on speech sounds in the presence of background noise.

Today, acoustics and noise control in classrooms receive increased attention and may become an increasingly important consideration for HVAC system designers. A recently issued classroom acoustics standard, ANSI Standard S12.60, *Acoustical Performance Criteria, Design Requirements and Guidelines for Schools* (ANSI 2002), describes requirements for both ducted and unducted HVAC systems. It is not, at this time, a mandatory standard. An ARI position paper discussing ANSI S12.60 (Darbeau 2003) indicated that the International Building Code (IBC) recently rejected a proposal to include substantial extracts of the new standard into the 2003 IBC. The important message for system designers is not that ANSI S12.60 is a voluntary standard or that IBC rejected its inclusion into the 2003 IBC, it is to be aware of trends in the industry. The existence of voluntary standards has an effect on system design, and the prudent engineer will acknowledge the standards (rather than ignoring them) and consider their merit in current projects.

ANSI Standard S12.60, as now written, specifies maximum background sound level limits of 35 dBA and 55 dBC. (This would apply to all sound sources within the classroom, including unit ventilators.) To comply with the new ANSI requirements, a unit ventilator must have maximum A-weighted and C-weighted sound power levels of L_{wa}=45 and L_{wc}=65 respectively (Schaffer 2003). As a generic group, unit ventilators have traditionally had A-weighted sound power levels in the range of L_{wa}=54 to 67. ARI Standard 350 (ARI 2000) is the industry guideline used to determine a unit ventilator's sound power level (L_w).

ARI STANDARD 840

The Air-Conditioning and Refrigeration Institute, as a manufacturer's trade association, established ARI Standard 840-98, *Unit Ventilators*. The standard prescribes: definitions, classifications, testing and rating requirements, minimum data requirements for published ratings, performance requirements, operating requirements, marking and nameplate data, and conformance conditions.

ARI 840 defines a unit ventilator as, "[a] factory-made assembly, equipped with outside air ventilation and return air dampers capable of introducing ventilation air of at least 80% of rated Standard Air Flow." The definition also stipulates that the capability to introduce ventilation air must occur while providing any combination of heating, cooling, humidity control and filtering air. Unit ventilators are designed for free delivery of air into a room but may operate with minimal ductwork having a static resistance not exceeding 0.5 inches w.g. The standard also defines the upper airflow limit for unit ventilators as 3,000 cfm.

ARI 840 describes two tests. The *Room Air Test* measures airflow capacity with all air passing through the return air path (outdoor air opening sealed). The *Ventilation Air Test* measures airflow capacity with all air passing through the ventilation air path (return air opening sealed) and with an external static pressure of -0.05 inches w.g. at the inlet to simulate losses through intake louvers. The *Standard Ventilation Rate* is defined as the ratio of the two measured airflows expressed as a percentage (ventilation air ÷ room air). The tested Standard Ventilation Rate must be no less than 80%, and it must be at least 95% of the rated (published) value.

Standard Ventilation Rate is the only performance measure directly defined by ARI 840. Standard performance ratings for cooling and heating capacity, power input, standard airflow, fluid flow and pressure drop are governed by other ARI standards included in ARI 840 by reference:

- *Self-Contained Unit Ventilators:* ARI Standard 310/380.
- *Water-Source Self-Contained Unit Ventilators:* ARI Standards 320, 325 and 330.
- *Fan-Coil Unit Ventilators:* ARI Standards 440 or 210/240.

UNIT VENTILATOR SELECTION

The manual selection process described below outlines the basic steps to use in sizing a single unit ventilator for a specific room. Some steps can be accomplished more quickly and more accurately using the Carrier E-Cat equipment selection programs (available from Carrier sales representatives). The important point to remember is that computerized equipment selection is not a blind process. The system designer will determine the correct selection only if adequate thought is given to providing correct and proper input information to the software program. A good selection is one that is neither undersized not oversized. The repercussions of undersizing are easily predictable. The effects of oversizing are subtler and become evident over time as problems in humidity control, uneconomical operation, space temperature control and noise.

The steps below examine selection of a single unit ventilator. It does not address facility block loads and selection of central heating and cooling equipment.

Step 1: Determine Room Loads

Room loads should be calculated for the dry-bulb design conditions, the dew-point design conditions and for the buildings overall block load. Room loads must include heat gains (and losses) for: thermal transmission, solar, internal heat gains, people and infiltration. In buildings with large areas of fenestration, solar loads may influence not only the time of day when peak loads are realized, but also the time of year. Ventilation loads must also be calculated but treated separately. Initially, ventilation loads should not be included in the room peak load calculations.

Most unit ventilator control schemes have a morning warm-up cycle. Contingencies to provide supplemental capacity for raising the room temperature from the setback level to the normal level are not usually necessary. During the warm-up cycle, outdoor airflow is stopped which permits the unit's heating or cooling capacity to affect the room temperature alone.

Step 2: Determine the Correct Size Unit

Unit ventilator sizes are distinguished by airflow capacity. Nominal sizes are: 500, 750, 1,000, 1,250, 1,500, and 2,000 cfm. The designer should be aware that there may be a difference between a manufacturer's nominal rating and the actual airflow delivered. The *standard airflow rating* determined in accordance with ARI 840 is a measured rate that is not required by the standard to be within a given tolerance of the nominal rating. All Carrier models deliver a standard airflow rate within 5% of the nominal rating (when configured with a 3-row coil).

Choosing the *correct* unit ventilator size depends on three things:

1. Providing proper air circulation in the room
2. Providing the correct amount of outdoor air
3. Providing a unit with sufficient capacity to meet peak heating and cooling loads

Size Based On Air Circulation

Adequate airflow volume discharging from a unit ventilator assures good room air mixing and helps to avoid hot and cold spots in the room. The necessary flow rate is often defined by the number of air changes per hour in the occupied space. Some local codes specify the minimum air change rate. Typically, the airflow from a unit ventilator should provide 5 to 9 air changes per hour.

Unit supply airflow is defined by the equation:

$$\text{Unit Airflow cfm} = V_{SA}$$

$$V_{SA} = (\text{Air Changes/hr}) \frac{(\text{Room volume ft}^3)}{60 \text{ min/hr}} \qquad (1)$$

Complete the above calculation and make a preliminary selection by picking the nearest size unit.

Example:

Room dimensions: 24 feet by 30 feet

Ceiling height: 10 feet

Air change rate: 8 ac/hr

$$V_{SA} = (8 \text{ ac/hr}) \frac{(30 \text{ ft x } 24 \text{ ft x } 10 \text{ ft})}{60 \text{ min/hr}} = 960 \text{ cfm}$$

Select a unit delivering 1,000-cfm.

As a comparison, most ducted air-conditioning systems provide supply airflow at about 1 cfm per square foot (or slightly more) of floor area. In a room with a 10-foot ceiling, a supply airflow rate of 1 cfm/ft^2 is equal to 6 air changes per hour. Table D converts the supply airflow rate from air changes per hour to cubic feet per minute per square foot of floor area for a number of different ceiling heights.

Size Based On Ventilation Airflow

Proper outdoor airflow is necessary to acceptable indoor air quality. Outdoor air dilutes odors, carbon monoxide, volatile organic compounds and other contaminants. The required outdoor airflow is frequently specified by local code, which usually does so by referencing the appropriate edition of ASHRAE Standard 62 (e.g., 62-1989, 62-1999 or 62-2001).

Example:

Code requirement: ASHRAE Standard 62-1999.

Room occupancy: 25 people

Room type: Education / classroom

From ASHRAE 62-1999 Table 2 (part 2.2 Institutional Facilities) the minimum outdoor air requirement is 15 cfm per person.

$$\text{Ventilation Airflow} = V_{OA}$$

$$V_{OA} = 15 \text{ cfm/person x } 25 \text{ people} = 375 \text{ cfm} \qquad (2)$$

Check the outdoor air ratio:

$$\text{Outdoor Air Ratio} = \frac{V_{OA}}{V_{SA}} = \frac{375 \text{ cfm}}{1{,}000 \text{ cfm}} = 0.375 \qquad (3)$$

Ventilation loads above 40% of the unit's airflow capacity (outdoor air ratios greater than 0.40) are not recommended. A unit operating with a high percentage of ventilation air may be inadequate to control room humidity levels. In the above example, a unit delivering 1,000-cfm is acceptable.

Size Based On Meeting Total Heating and Cooling Loads

After determining the outdoor airflow, calculate the total heating and cooling loads including ventilation air. The total heating load is found by the equation:

$$\text{Total Heating Load} = \text{Ventilation Load} + \text{Room Load}$$

or, rewritten with variables:

$$Q_T = Q_{OA} + Q_R \qquad (4)$$

Table D
Supply Airflow Rate on a CFM / Ft² Basis Relative to Ceiling Height and Air Change Rate

Ceiling Height (ft)	Air Changes per Hour												
	4	4.5	5	5.5	6	6.5	7	7.5	8	8.5	9	9.5	10
15.0	1.000	1.125	1.250	1.375	1.500	1.625	1.750	1.875	2.000	2.125	2.250	2.375	2.500
14.5	0.967	1.088	1.208	1.329	1.450	1.571	1.692	1.813	1.933	2.054	2.175	2.296	2.417
14.0	0.933	1.050	1.167	1.283	1.400	1.517	1.633	1.750	1.867	1.983	2.100	2.217	2.333
13.5	0.900	1.013	1.125	1.238	1.350	1.463	1.575	1.688	1.800	1.913	2.025	2.138	2.250
13.0	0.867	0.975	1.083	1.192	1.300	1.408	1.517	1.625	1.733	1.842	1.950	2.058	2.167
12.5	0.833	0.938	1.042	1.146	1.250	1.354	1.458	1.563	1.667	1.771	1.875	1.979	2.083
12.0	0.800	0.900	1.000	1.100	1.200	1.300	1.400	1.500	1.600	1.700	1.800	1.900	2.000
11.5	0.767	0.863	0.958	1.054	1.150	1.246	1.342	1.438	1.533	1.629	1.725	1.821	1.917
11.0	0.733	0.825	0.917	1.008	1.100	1.192	1.283	1.375	1.467	1.558	1.650	1.742	1.833
10.5	0.700	0.788	0.875	0.963	1.050	1.138	1.225	1.313	1.400	1.488	1.575	1.663	1.750
10.0	0.667	0.750	0.833	0.917	1.000	1.083	1.167	1.250	1.333	1.417	1.500	1.583	1.667
9.5	0.633	0.713	0.792	0.871	0.950	1.029	1.108	1.188	1.267	1.346	1.425	1.504	1.583
9.0	0.600	0.675	0.750	0.825	0.900	0.975	1.050	1.125	1.200	1.275	1.350	1.425	1.500
8.5	0.567	0.638	0.708	0.779	0.850	0.921	0.992	1.063	1.133	1.204	1.275	1.346	1.417
8.0	0.533	0.600	0.667	0.733	0.800	0.867	0.933	1.000	1.067	1.133	1.200	1.267	1.333

Example:
A room with a 10.0 ft ceiling with a supply air rate of 7 air changes per hour has an equivalent supply rate of 1.167 cfm/ft².

where

$$Q_{OA} = V_{OA} \times 1.08 \times (T_R - T_{OA}) \qquad (5)$$

thus,

$$Q_T = [V_{OA} \times 1.08 \times (T_R - T_{OA})] + Q_R \qquad (6)$$

Where T_R and T_{OA} are the room and outdoor air dry-bulb temperatures respectively at design conditions.

Example:

Winter room temperature (T_R): *70°Fdb*

Outdoor air temperature (T_{OA}): *0°Fdb*

Calculated room heating load: 15,840 Btu/hr

The total heating load (Q_{TH}) is:

$$Q_{TH} = [375 \text{ cfm} \times 1.08 \times (70 - 0^\circ\text{F})] + 15{,}840$$

$$Q_{TH} = 44{,}190 \text{ Btu/hr}$$

The total cooling load (Q_{TC}) must be found in two components: total sensible cooling load (Q_{SC}) and total latent cooling load (Q_{LC}) using the following equations:

$$Q_{TC} = Q_{SC} + Q_{LC} \qquad (7)$$

$$Q_{SC} = [V_{OA} \times 1.08 \times (T_{OA} - T_R)] + Q_{RS} \qquad (8)$$

$$Q_{LC} = [V_{OA} \times 4840 \times (W_{OA} - W_R)] + Q_{RL} \qquad (9)$$

Where W_{OA} and W_R are the humidity ratios of the outdoor air and room air respectively (in units of pounds of moisture per pound of dry air), and Q_{RS} and Q_{RL} are the room sensible and room latent cooling loads respectively.

An alternative equation uses the outdoor air enthalpy (h_{OA}) and room air enthalpy (h_R) to calculate the total cooling load:

$$Q_{TC} = [V_{OA} \times 4.5 \times (h_{OA} - h_R)] + Q_{RS} + Q_{RL} \qquad (10)$$

Example:

Calculated room sensible cooling load (Q_{RS}): *25,400 Btu/hr*

Calculated room latent cooling load (Q_{RL}): *6,100 Btu/hr*

Outdoor air design conditions: 95°Fdb, 75°Fwb (67°F DP)

Outdoor air humidity ratio at design conditions (W_{OA}): *0.014148 lb. H_2O / lb. dry air*

Outdoor air enthalpy at design conditions (h_{OA}): *38.427 Btu/lb*

Indoor air design conditions: 75°Fdb, 50% RH (63°Fwb, 55°F DP)

Indoor air humidity ratio at design conditions (W_R): *0.009320 lb. H_2O / lb. dry air*

Indoor air enthalpy at design conditions (h_R): *28.215 Btu/lb*

Using Equations (9) and (10), the total sensible and total latent loads are:

$$Q_{SC} = [375 \text{ cfm} \times 1.08 \times (95 - 75^\circ\text{F})] + 25{,}400$$

$$Q_{SC} = 33{,}500 \text{ Btu/hr}$$

$$Q_{LC} = [375 \text{ cfm} \times 4840 \times (0.014148 - 0.009320)] + 6{,}100 \text{ Btu/hr}$$

$$Q_{LC} = 14{,}860 \text{ Btu/hr}$$

$$Q_{TC} = 33{,}500 + 14{,}860 = 48{,}360 \text{ Btu/hr}$$

For the sake of example, the total cooling load calculated using Equation (10) is:

$$Q_{TC} = [375 \text{ cfm} \times 4.5 \times (38.427 - 28.215)] + 25{,}400 + 6{,}100 = 48{,}730 \text{ Btu/hr}$$

The difference in results between equations is attributable to round-off and the approximate nature of the conversion factors.

Proper cooling operation requires an airflow rate of about 350 to 450 cfm per ton of room sensible cooling load. For this example, the range of required cooling airflow would be:

$$\text{Low} = \frac{33{,}500\ \text{Btu/hr}}{12{,}000\ \text{Btu/hr}\cdot\text{ton}} \times 350\ \text{cfm/ton} = 977\ \text{cfm}$$

$$\text{High} = \frac{33{,}500\ \text{Btu/hr}}{12{,}000\ \text{Btu/hr}\cdot\text{ton}} \times 450\ \text{cfm/ton} = 1{,}256\ \text{cfm}$$

Based on this evaluation, a unit delivering between 977 and 1,256 cfm may be acceptable.

At this point check the actual cooling capacity available from 1,000-cfm and 1,250-cfm units. In this example a 1,000-cfm unit ventilator is not available with adequate cooling capacity; thus, a 1,250-cfm unit will be required.

Step 3: Determine the Coil Entering Air Conditions

The entering dry-bulb temperature for cooling and heating are found by simple ratio using the equation:

$$\text{EAT} = \frac{(T_R \times V_{RA}) + (T_{OA} \times V_{OA})}{(V_{SA})} \tag{11}$$

where V_{RA}, V_{OA} and V_{SA} are the return air, outdoor air and supply airflow rates respectively. The supply airflow should be the actual airflow for the unit ventilator as configured (i.e., with coils, filters and accessories), and $V_{SA} = V_{RA} + V_{OA}$.

Example:

For a unit delivering 1,250 cfm of supply air, the cooling EAT will be:

$$\text{EAT} = \frac{(75^\circ\text{Fdb} \times 875\ \text{cfm}) + (95^\circ\text{Fdb} \times 375\ \text{cfm})}{(1{,}250\ \text{cfm})}$$

$$\text{Cooling EAT} = 81^\circ\text{Fdb}$$

and the heating EAT will be:

$$\text{EAT} = \frac{(70^\circ\text{Fdb} \times 875\ \text{cfm}) + (0^\circ\text{Fdb} \times 375\ \text{cfm})}{(1{,}250\ \text{cfm})}$$

$$\text{Heating EAT} = 49^\circ\text{Fdb}$$

The entering wet-bulb temperature for cooling is found using the same equation and substituting the outdoor and indoor wet-bulb conditions:

$$\text{EAT} = \frac{(63^\circ\text{Fwb} \times 875\ \text{cfm}) + (75^\circ\text{Fwb} \times 375\ \text{cfm})}{(1{,}250\ \text{cfm})}$$

$$\text{Cooling Wet Bulb EAT} = 66.6^\circ\text{Fwb}$$

Step 4: Select Cooling and Heating Coils

With the information determined in the above steps, use the Carrier E-Cat (Equipment Selection Program) to determine the heating and cooling coil selections.

Step 5: Determine the Free Cooling Capacity

When the outdoor air temperature is sufficiently low, a unit ventilator's outdoor air damper can be controlled as an air-side economizer to provide cooling. The amount of cooling available is determined as a function of the supply air rate (using 100% outdoor air) and the defined changeover temperature. At the changeover condition, which will be at an outdoor air temperature lower than the room temperature, the transmission losses will be equal to zero. In considering the ability of a unit ventilator to meet the cooling load in a free cooling mode, the only applicable loads are: solar, people and other internal loads. Infiltration and ventilation will also be zero.

The available free cooling capacity (Q_{FC}) is determined by the equation:

$$Q_{FC} = [V_{SA}\ \text{x}\ 1.08\ \text{x}\ (T_R - T_C)] \times \text{SVR} \tag{12}$$

where T_C is the changeover temperature and SVR is the Standard Ventilation Rating as defined in ARI 840.

Before establishing the changeover temperature, it is necessary to isolate the solar, people and internal sensible loads and use that information to determine the maximum supply air temperature that will satisfy those loads. Once the loads are known, the maximum changeover temperature is found with the following equation:

$$T_{C\ MAX} = T_R - [\frac{solar + people + internal}{V_{SA} \times 1.08 \times SVR}] \qquad (13)$$

Example:

Solar, people and internals sensible loads: 18,800 Btu/hr

Standard Ventilation Ratio: 98% (based on ARI 840)

$$T_{C\ MAX} = 75^\circ F - [\frac{18{,}800\ Btu/hr}{1{,}250\ cfm \times 1.08 \times 0.98}] = 60.8^\circ F$$

Set the control system changeover set point at temperature less than 60.8°F. It is prudent to allow a margin of 3°F to 5°F between the calculated temperature and the actual control set point. With the changeover set point defined at 56°F, the unit ventilator free cooling capacity is:

$$\begin{aligned} Q_{FC} &= [1{,}250\ cfm \times 1.08 \times (75 - 56^\circ F)] \times 0.98 \\ &= 25{,}140\ Btu/hr \end{aligned}$$

The designer should be aware that to meet ARI 840, the Standard Ventilation Ratio, which describes ventilation airflow as a percentage of the *standard* airflow rating (not the *nominal* rating) must at least 80%. For instance, if a nominal 1,000-cfm unit has a standard airflow rating of 900 cfm (actual airflow), the maximum outdoor airflow may be as low as 720 cfm and still meet ARI 840. Carrier unit ventilators have standard airflow ratings that are within 5% of the nominal airflow ratings.

CONTROLS

Unit ventilator designs have kept pace with advances in control technology and many different control cycles are available for ventilation, heating and cooling. Control complexity varies from a single unit ventilator controlled by a room thermostat, to multiple units monitored and controlled by a central building automation system.

There are three basic control modes:

- Warm-up
- Heating and ventilating
- Cooling and ventilating

Warm-Up Mode

Facilities operating with night setback will need a sequence to "warm-up" the room from its setback condition to its occupied condition. This may be true for both heating and cooling seasons. The winter warm-up mode begins at a defined time prior to the arrival of room occupants. If the room temperature is below the occupied set point, the outdoor air damper remains shut when the unit starts and the return air damper is fully open as shown in Fig. 5. The heating coil provides full heating until the room temperature is at the occupied set point. Summer operation (for cool-down) is similar. When the room temperature is above the occupied set point, the outdoor air damper remains shut when the unit starts and the return air damper is fully open. The cooling coil provides full cooling until the room temperature is at the occupied set point.

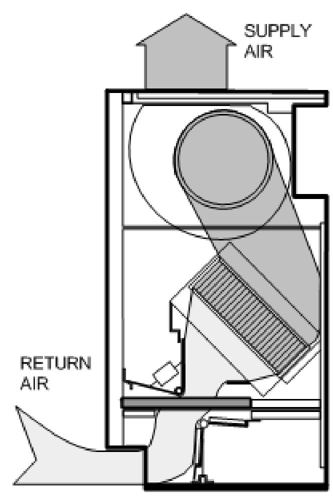

Fig. 5. Airflow during warm-up cycle

Heating and Ventilating Mode

During the heating season, after the initial warm-up period, the unit ventilator will changeover to the heating and ventilating mode. The outdoor air damper will open to its minimum position necessary to satisfy the room's outdoor air requirement. On 100% outdoor air units (i.e., ASHRAE Cycle I), the damper opens fully. The unit ventilator's heating coil (steam, hot water, electric) throttles in response to a local thermostat to maintain the occupied heating set point. The fan operates continuously when the room is occupied.

A unit ventilator operating with a hot water coil and a face and bypass damper would operate as follows:

- The unit operates at the lowest speed setting necessary to satisfy the room set point. (This is done to minimize noise and energy use.)
- The outdoor air damper is open to the minimum position.
- The two-position hot water control valve opens.
- Responding to the local thermostat, the face and bypass damper adjusts position to maintain a mixed leaving air temperature adequate to satisfy the room set point. A high-limit thermostat mounted in the unit prevents the discharge air temperature from rising too high.
- If the room temperature falls below the occupied heating set point, and the face and bypass damper is in the full-face position (no bypass), the fan speed increases to the next faster speed.
- After the fan motor changes speed, the outdoor air damper must adjust position to maintain the minimum outdoor air flow (otherwise the outdoor airflow volume will increase with increasing fan speed).
- During periods of decreasing heating load, if the room temperature is above the set point, the fan speed reduces to the next slower speed. The outdoor air damper must also adjust position, as mentioned above, to maintain the correct minimum ventilation airflow.
- If the fan is at the slowest speed, the face and bypass damper again throttles to maintain the room space temperature.
- If the room temperature rises above the occupied heating set point and the face and bypass damper is in the full bypass position, the two-position hot water control valve closes.

Cooling and Ventilating Mode

ASHRAE has defined four standard ventilation cycles: Cycle I, II, III and W. In most cases, unit ventilators can be adapted to any of the four cycles:

ASHRAE Cycle I

Except during warm-up, a unit ventilator operating with Cycle I admits 100% outdoor air at all times.

- During warm-up the outdoor air damper is closed, the return air damper is open, and the heating coil operates at full heat.
- When the room temperature rises to the occupied heating set point (completion of the warm-up cycle) the outdoor air damper opens fully and the return air damper closes.
- The unit ventilator heating coil modulates (with either a control valve or face and bypass dampers) to satisfy the local thermostat.

Figure 6 graphically represents ASHRAE Cycle I.

ASHRAE Cycle II

Cycle II is the most common control cycle. After the warm-up period, the unit ventilator outdoor air damper opens to a minimum position. The out-

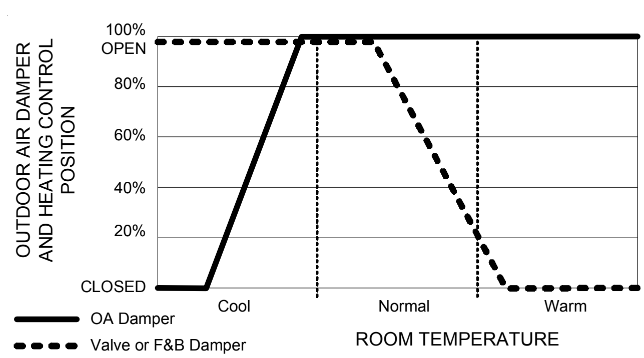

Fig. 6. ASHRAE Cycle I

door airflow stays fixed at the minimum position during normal operation. If room loads are transitioning from heating to cooling, the heating control will modulate down to a no heat condition (e.g., face and bypass damper in the full bypass position or hot water valve closed). At this point, the minimum outdoor air mixes with return air and is delivered to the space. If the room temperature rises above the occupied heating set point, the outdoor air damper modulates open to admit a greater percentage of outdoor air. Economizer operation using outdoor air for cooling continues until the unit is unable to maintain the room temperature at or below the occupied cooling set point. A unit ventilator with chilled water and hot water coils would provide economizer cooling as follows:

- Both the heating and cooling coils are off and the room temperature is between the occupied heating set point (lower) and the occupied cooling set point (higher).
- If the room temperature is greater than the mid-point between the occupied heating and cooling set points, and the outdoor air temperature is less than the defined free cooling changeover temperature, the outdoor air damper will modulate open, above the minimum position, to satisfy the local thermostat. See Fig. 7.
- A low-limit thermostat mounted in the unit prevents the discharge air temperature from being too low and freezing the hydronic coils.
- If modulating the outdoor air damper is inadequate to satisfy the local thermostat, the fan speed shall increase to the next faster speed.

Economizer operation may continue as long as the outdoor air temperature is below the changeover temperature (e.g., 56°F, see Unit Ventilator Selection, Step 5: Determine the Free Cooling Capacity on pages 13 to 14). Above the changeover temperature, if the room temperature rises above the occupied cooling set point, the units transitions to the cooling mode. The description below is for a unit with a chilled water cooling coil:

- The outdoor air dampers closes to the minimum position.
- The two-position chilled water control valve opens.
- The unit fan operates at the lowest speed setting necessary to satisfy the room set point.
- Responding to the local thermostat, the face and bypass damper adjusts position to maintain a mixed leaving air temperature adequate to satisfy the room set point as shown in Fig. 8.
- If the room temperature rises above the occu-

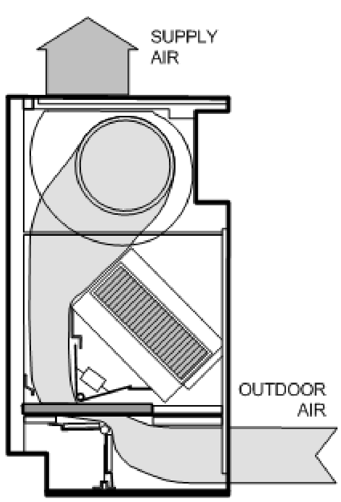

Fig. 7. Airflow during economizer cooling

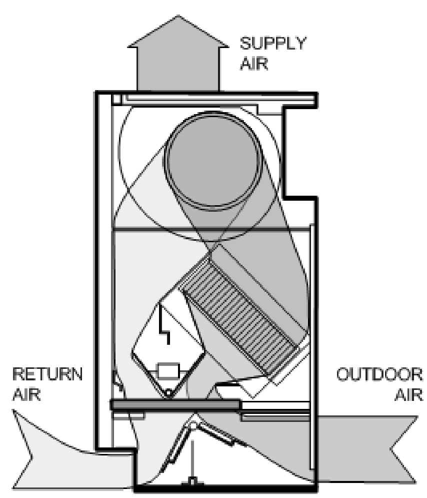

Fig. 8. Airflow during normal operation with outdoor air damper set at the minimum position and the face and bypass damper modulating

pied cooling set point and the face and bypass damper is in the full-face position (no bypass) the fan speed increases to the next faster speed.

- After the fan motor changes speed, the outdoor air damper must adjust position to maintain the minimum outdoor air flow (otherwise the outdoor airflow volume will increase with increasing fan speed).
- During periods of decreasing cooling load, if the room temperature is below the occupied cooling set point, the fan speed reduces to the next slower speed. The outdoor air damper must also adjust position, as mentioned above, to maintain the correct minimum ventilation airflow.
- If the fan is at the slowest speed, the face and bypass damper again throttles to maintain the room space temperature.
- If the room temperature drops below the occupied cooling set point and the face and bypass damper is in the full bypass position, the two-position chilled water control valve closes.

Figure 9 graphically represents ASHRAE Cycle II.

ASHRAE Cycle III

Cycle III is a variable outdoor air cycle. The unit ventilator controls modulate the outdoor air damper and the return air damper to maintain a constant heating coil entering temperature (e.g., 55°F). During the room warm-up cycle, the outdoor air damper is closed and the return air damper is open (as described above for Cycles I and II). A unit with a hot water coil would operate as follows:

- The unit operates at the lowest speed setting necessary to satisfy the room set point.
- The modulating hot water control valve is initially closed.
- Responding to a mixed air thermostat, the return and outdoor air damper modulate to maintain a constant coil entering air temperature.
- The modulating hot water control valve adjusts position to satisfy the room thermostat.
- If the room temperature falls below the occupied heating set point and the hot water control valve is in the full open position the fan speed increases to the next faster setting.

Figure 10 graphically represents ASHRAE Cycle III.

ASHRAE Cycle W

ASHRAE Cycle W is very similar to Cycle II. In Cycle W, the room thermostat controls the heating valve, and the low-limit thermostat controls the outdoor air and return air dampers.

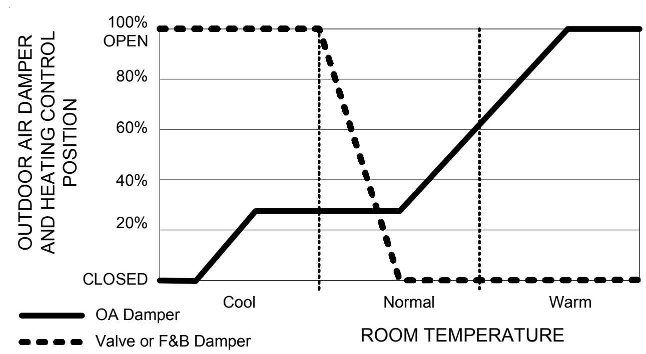

Fig. 9. ASHRAE Cycle II

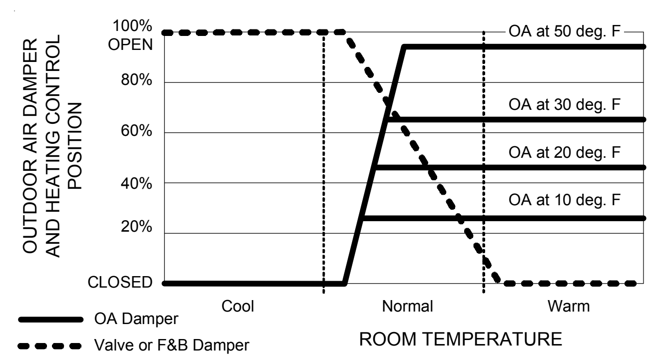

Fig. 10. ASHRAE Cycle III

Unoccupied Mode Operation

When a unit ventilator is in the "unoccupied" mode, the unit fan is stopped, the outdoor air damper is closed and the return air damper is open.

Unoccupied Fan Cycling

The unit ventilator fan starts periodically and operates for a short period of time to circulate air in the space. The frequency and duration may be adjusted; however, starting the fan once per hour and operating for one minute is a typical sequence.

Unoccupied Heating

If the room temperature drops below the unoccupied heating set point, the fan starts at the slow speed, and the heating system is enabled to maintain the room temperature. Heating control is similar to the occupied mode. When the unoccupied set point is satisfied, the heating coil is disabled and the fan stops.

Unoccupied Cooling

If the room temperature rises above the unoccupied cooling set point, the fan starts at the slow speed, and the cooling system is enabled to maintain the room temperature. Cooling control is similar to the occupied mode. When the unoccupied set point is satisfied, the cooling coil is disabled. The fan continues operation for 5 minutes (to evaporate condensate from the cooling coil) and then stops.

Unoccupied Economizer Cooling

If the outdoor air temperature is below the economizer changeover temperature (e.g., 56°F) and the room temperature rises above the unoccupied cooling set point, the fan starts and the return air damper opens. The unit operates until the local thermostat is satisfied. At that point the fan stops and the outdoor air damper closes. If the low-limit thermostat detects the supply air temperature is below its set point, the outdoor air damper closes and the fan stops.

Demand Controlled Ventilation

Rather than follow the outdoor air sequences described above under the standard ASHRAE cycles, the outdoor air damper modulates in response to a locally mounted IAQ (carbon dioxide) sensor. The minimum outdoor damper position is determined based on the minimum ventilation needed for the space alone (unoccupied). When the room is occupied, the amount of outdoor air supplied to the space is adequate to maintain the room at or below the IAQ set point. The outdoor air flow rate is limited only by the mixed air control that prevents the mixed air temperature entering the coil from falling below a defined level (e.g., 50°F).

Other Control Strategies

The above sequences are typical, but not all inclusive. Heating and cooling sequences will vary depending on the heating and cooling sources (e.g., steam, electric or hot water heating, and DX or chilled water cooling). Although face and bypass damper operation is described, hydronic coil modulation can be achieved using two-way control valves. Your Carrier representative can provide additional guidance on other control strategies.

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