403 MECHANICAL SYSTEMS & EQUIPMENT

Overview

DESIGN CONSIDERATIONS

Mechanical systems are one of the most significant users of energy in the types of buildings covered by the 90.1 Code. In typical nonresidential buildings (exclusive of process loads) HVAC energy consumption is second only to lighting. In typical residential buildings, HVAC and domestic water heating are the two largest consumers of energy. But not all mechanical systems perform equally efficiently: a poorly designed system can easily have twice the yearly energy costs of an energy conserving design.

Rather than require certain system types or designs, Section 403 of the 90.1 Code regulates mechanical system design by controlling the fundamental factors that make systems efficient or inefficient. This leaves the designer more flexibility in system selection while still ensuring good performance. However, it also makes the code more complicated to understand and apply. The intent of this manual is to make the code more usable to the designer and the code enforcement agency by explaining the purpose of each section of the code and giving examples of how it may be applied to real buildings and real systems.

Although compliance with Section 403 of the 90.1 Code assures a minimum level of mechanical system performance, the designer is encouraged to view the requirements as a starting point and investigate designs that exceed these requirements. Application of heat recovery, solar energy, or high efficiency equipment can create a system that is both more efficient than required by the code and exhibits an excellent return on investment.

SCOPE

The mechanical system requirements apply only to the mechanical systems serving buildings that fall within the overall scope of the code. In general, mechanical systems that serve an industrial process, such as a chilled water system cooling an injection mold or a boiler providing steam for an industrial process, do not come under the scope of this code. There are, however, several gray areas where a certain functions may be considered process or not, depending on the point of view. To help decide if the equipment is covered by the code, consider the following questions:

- Is the primary purpose of the mechanical system to provide human comfort or the proper environment for machinery or material located within the space? In a computer room, for instance, the HVAC system is fundamental to the equipment;

the computer equipment must be kept within certain temperature and humidity ranges to function properly. Therefore, the HVAC systems and equipment serving the space do not fall within the scope of this code. On the other hand, a cooling system in a commercial laundry that is used to spot cool the people operating the laundry equipment must comply since the primary purpose of the equipment is human comfort, not the process load.

- If answering the question above fails to resolve the ambiguity, ask whether there is any reason why the application should not fall under the code. For instance, is there some special space or fluid temperature that must be maintained that is very different from normal comfort conditions? If not, the application should meet the requirements of the 90.1 Code.

Many of the requirements of Section 403 apply only to larger, multiple zone systems. The breadth of the section may seem overwhelming to designers of simpler, single zone systems. To simplify matters, HVAC systems are categorized in this manual between unitary and hydronic systems and between single and multiple zone systems. Unitary equipment is self-contained equipment from a single manufacturer, while hydronic systems consist of separate components such as a chiller, cooling tower, and air handlers. Single zone systems are those that serve just one zone and are controlled by a single thermostat. Multiple zone systems are capable of simultaneously satisfying the comfort needs of a number of zones. The Reference section of this chapter has a general description of these system types. Table 403A shows how the HVAC requirements apply to these classifications. The white areas of the table are cases when the requirements clearly apply. The black areas show when the requirements do not apply, and the gray areas of the table show cases when the requirements may apply depending on the special circumstances of the system. Additional commentary is provided as appropriate.

Table 403A Applicable Section 403 Requirements for Different HVAC Systems

Requirement	Unitary		Hydronic	
	Single Zone	**Multiple Zone**	**Single Zone**	**Multiple Zone**
	Packaged rooftop or split system single zone > 7-1/2 tons	Packaged rooftop VAV	Chilled water 2-pipe or 4-pipe fan-coils	Chilled water VAV
Mechanical Equipment Efficiency (403.1)	Required	Required	Required	Required
Load Calculations (403.2.1)	Required	Required	Required	Required
Equipment/System Sizing (403.2.2)	Required	Required	Required	Required
Separate Air Distribution System (403.2.3)	Required	Required	Required	Required
Ventilation Capability (403.2.4)	Required	Required	Required	Required
Fan System Design (403.2.4)	**Not required**	Required	Not usually required	Required
Pumping System Design (403.2.5)	**Not required**	**Not required**	Required	Required
System Controls (403.2.6.1)	**Systems comply inherently**	Systems generally comply inherently	**Systems comply inherently**	Systems generally comply inherently
Zone Controls (403.2.6.2)	**Systems comply inherently**	Required	**Systems comply inherently**	Required
Zone Thermostatic Control Capability (403.2.6.3)	Required	Required	Required	Required
Heat Pump Auxiliary Heat (403.2.6.4)	Heat pump systems only	**Not Required**	**Not Required**	**Not Required**
Humidistats (403.2.6.5)	All humidification or active dehumidification systems	All humidification or active dehumidification systems	All humidification or active dehumidification systems	All humidification or active dehumidification systems
Simultaneous Heating and Cooling (403.2.6.6)	Not required except if reheat for dehumidification is used.	Required	Not required except if reheat for dehumidification is used.	Required
Air Temperature Reset (403.2.6.7)	**Not required**	Required	**Not required**	Required
Hydronic Temperature Reset (403.2.6.8)	**Not required**	**Not required**	Required	Required
Automatic Setback or Shutdown (403.2.7.1)	Required. Met with timeclock thermostat	Required	Required	Required
Shutoff Dampers (403.2.7.2)	Required if outside air intake exceeds 3000 cfm	Required if outside air intake exceeds 3000 cfm	Required if outside air intake exceeds 3000 cfm	Required if outside air intake exceeds 3000 cfm
Zone Isolation (403.2.7.3)	**Systems comply inherently**	Required	**Systems comply inherently**	Required
Economizers (403.2.8)	Not required for systems less than 3,000 cfm or 7.5 tons	Required in most applications	Required in most applications	Required in most applications
Piping Insulation (403.2.9.1)	Required for split systems.	Required for split systems.	Required	Required
Duct and Plenum Insulation (403.2.9.2)	Required	Required	Required	Required
Duct and Plenum Construction (403.2.9.3)	Required	Required	Required	Required
Manuals (403.2.10.1)	Required	Required	Required	Required
Air System Balancing (403.2.10.2)	Required	Required	Required	Required
Hydronic System Balancing (403.2.10.3)	**Not required**	**Not required**	Required	Required
Control System Testing (403.2.10.4)	Required	Required	Required	Required

Figure 403A Annotated Mechanical Summary Form

Building department may require applicant to certify here that plans comply with the code

ALL SYSTEMS (Complete Form)

Project Information	Project Address:	Date: For Building Department Use
	Applicant Name:	
	Applicant Address:	
	Applicant Phone:	
Project Description		

Applicant completes this basic information

GENERAL: This form can be used for all projects. Complete both pages.
(For small simple systems, it may be possible to use a one page summary. See that summary for criteria.)

Section numbers and requirements follow the order of the 90.1 Code

Circle "yes" if the requirement has been met or if "no", circle "yes" to the exception that applies

Enter appropriate values in blank spaces

Write page number(s) or location on drawings where information is identified

Check box if compliance h documentation as been reviewed and is acceptable

Note any special features that should be checked in the field

Verify in the field during suggested time of inspeciton

Check box when relevant inspection has been completed

Code Section Number	Requirements	Enter value or circle "yes" or "no"	List reference to plans or enter "n.a."	Check if documentation is acceptable. Provide notes to inspector, when necessary	Inspection type	Check when inspection performed
403.1	Equipment Efficiency			❑	Rough	❑
403.2.1	Load Calculations					
403.2.2	Equipment/System Sizing			❑	Rough	❑
excep #1:	smallest size	yes / no				
excep #2:	standby w/ auto controls	yes / no		❑	Rough	❑
excep #3:	multiple units w/ controls	yes / no		❑	Rough	❑
403.2.3	Separate Air System	yes / no		❑	Rough	❑
exception:	<25% or 1000 ft2	yes / no				
403.2.4	Ventilation/Fan System					
	Outdoor air to minimum:	yes / no		❑	Rough	❑
	Constant volume W/cfm:			❑	Rough	❑
	VAV system W/cfm:			❑	Rough	❑
	Motor control if >75 hp:	yes / no		❑	Rough	❑
excep #1:	fan system <10 hp	yes / no				
excep #2:	fan included in effic. tables	yes / no				
403.2.5	Pumping System Control	yes / no		❑	Rough	❑
excep #1:	minimum flow >50% req.	yes / no				
excep #2:	system w/ 1 control valve	yes / no				
excep #3:	pump system <10 hp	yes / no				
excep #4:	hydronic w/ auto reset	yes / no		❑	Rough	❑
403.2.6	Temperature/Humidity Ctrl					
403.2.6.1	System Controls	yes / no		❑	Rough	❑
403.2.6.2	Zone Controls	yes / no		❑	Rough	❑
exception:	independent perimeter sys.	yes / no		❑		
403.2.6.3	Zone Control Capability	yes / no		❑	Rough	❑
excep #1:	special occupancy/usage	yes / no		❑		
excep #2:	manual changeover	yes / no		❑	Rough	❑
403.2.6.4	Heat Pump Auxiliary Heat	yes / no		❑	Rough	❑
403.2.6.5	Humidistats	yes / no		❑	Rough	❑
403.2.6.6	Simul. Heating & Cooling	yes / no		❑	Rough	❑
excep #1:	VAV w/ min. air supply	yes / no		❑	Rough	❑
excep #2:	special pressurization	yes / no		❑		
excep #3:	site-recovered or site-solar	yes / no		❑	Rough	❑
excep #4:	humidity requirements	yes / no		❑		
excep #5:	peak air supply <300 cfm	yes / no		❑	Rough	❑
403.2.6.7	Air Temperature Reset	yes / no		❑	Rough	❑
exception:	no reheating/recooling	yes / no		❑	Rough	❑
403.2.6.8	Hydronic Temp. Reset	yes / no		❑	Rough	❑
excep #1:	403.2.5 w/o exception	yes / no		❑	Rough	❑
excep #2:	cannot be implemented	yes / no				

Applicant provides information requested in these two columns

Plans examiner fills out this column

Field inspector verifies proper installation or system or equipment at inspection time indicated

CHAPTER ORGANIZATION

Form
403.2.1

The Mechanical Summary form, annotated in Figure 403A, provides an organizing element for this chapter. The form itemizes each requirement and provides a place to reference on the drawings where compliance with each requirement is documented. This form is filled out by the permit applicant and is then used by the plan's examiner and the field inspector to verify energy code compliance. The text of this chapter follows the order of the Summary form. As each requirement is addressed, an icon of the Summary form appears in the margin highlighting the appropriate 90.1 Code reference on the form.

The Compliance and Enforcement section of this chapter describes how to fill out the Summary form in more detail and introduces a short form version for unitary single-zone systems. It also introduces the Fan System worksheet which is provided for the applicant to calculate the fan system power for variable air volume and constant volume fan systems.. It will be helpful for the reader to refer to these forms as each requirement is addressed below. Blank copies of all forms are found in Appendix D.

Requirements

The following is a discussion of all mechanical system requirements in the order they appear in the code.

MECHANICAL EQUIPMENT EFFICIENCY

Form
403.1

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

HVAC equipment must meet or exceed the energy efficiencies shown in Tables 403.1a through 403.1f of the 90.1 Code if it is to be used in a building or an HVAC system falling within the scope of this code. Efficiencies must be measured in accordance with the rating procedures specified in the tables. Equipment ratings may be self-certified by the manufacturer or rated under a nationally recognized certification program. Field tests are not required. Where more than one requirement applies to a given piece of equipment, all must be met.

Some of the equipment efficiency requirements of this section are also requirements of the National Appliance Energy Conservation Act (NAECA). The NAECA requirements apply to new equipment installed in existing buildings as well as to new buildings. The other requirements of this section apply only to HVAC installations in new buildings.

Omissions

Tables 403.1a to 403.1f cover the majority of equipment used in air conditioning systems. Equipment types that are not covered, such as absorption chillers, electric heaters, cooling towers and pumps, among others, need not meet any efficiency level to comply with the code; omission from the tables does not imply that such equipment is precluded from usage.

One reason for the omission of efficiency requirements for heat-operated chillers, such as absorption chillers, is the rapid change in design of these systems in recent years. Comprehensive rating procedures for the newer products, such as double-effect absorption machines, have only recently been developed. There was little reason for having efficiency limits for single-effect machines because this technology has reached an efficiency limit in its present design. A significant redesign to increase efficiency would not be justified given the small market for the product and to do so might eliminate these chillers from production. Single-effect machines, while significantly less efficient than the newer double-effect chillers, are still the best suited for low temperature heat recovery applications.

Mixed Manufacturers

If products from more than one manufacturer are used to produce a covered product with a cooling capacity less than 135,000 Btu/h, the performance of the combined system must be determined from data supplied by each manufacturer, and must meet the efficiency requirements for the applicable end-product category.

For systems with capacities greater than or equal to 135,000 Btu/h, the individual refrigeration component (condensing unit) alone must comply with the requirements of Table 403.1a; the indoor unit is not regulated in this section. However, the indoor fan energy may be regulated by Section 403.2.4. This requirement would apply, for example, to an indoor air handler used with an outdoor condensing unit to form a split system air conditioner.

Overall System and Equipment Energy Usage

Take care when using the standard efficiency ratings established in this section to compare different types of HVAC equipment. The ratings are sometimes at different rating conditions, such as groundwater-source heat pumps which are rated at 70°F compared to water-source heat pumps which are rated at 85°F. More importantly, the ratings may not include the energy used to operate auxiliary systems that may be required for the primary equipment to function as a complete system. For example:

- The efficiency ratings for water-cooled equipment cannot be directly compared to those for air-cooled equipment. Water-cooled equipment ratings do not include the energy used by the condenser water pumps and cooling tower fans while air-cooled package ratings include condenser fan energy.
- The ratings for condensing units cannot be directly compared to ratings for packaged or split system air conditioners. Condensing unit ratings do not include the energy used by indoor air handling fans.
- Efficiency ratings for different types of furnaces account only for gas usage but do not include the energy used by combustion air fans and indoor air handler fans which vary from one furnace to another.
- The efficiency of a chilled-water system cannot be compared to a unitary direct-expansion system using standard ratings. Chilled-water system efficiency does not include the energy used by chilled water pumps and air handler fans.

Different types of equipment may be applied in many different ways and the manufacturer can only test the performance of equipment as it exists when it leaves the factory. A fair comparison between different types of equipment, such as water- versus air-cooled equipment, requires knowledge of the auxiliary equipment needed for a complete system and the energy they use both at full and part load. Often a computerized energy analysis is the only way to make an accurate comparison.

Example 403A Multiple Requirements – Unitary Heat Pump

Q What are the efficiency requirements for a 10-ton unitary heat pump? Hint, 10 tons is equal to 120,000 Btu/h.

A Table 403.1b in the code contains the requirements for unitary heat pumps. The heat pump must meet both the high and low temperature COP requirements for heating (COPs of 3.0 and 2.0 respectively). It must also meet both the full-load EER and the part-load IPLV requirements for cooling (8.9 EER and 8.3 IPLV respectively).

Example 403B Multifunction Equipment Requirements – Space and Water Heating Boiler

Q A 2,000,000 Btu/h input gas-fired water-tube boiler is to provide domestic water heating in an external storage tank and will also provide space heating. The boiler has two-stage gas valves. What efficiency requirements must be met?

A The boiler serves as a "non-storage" (instantaneous) water heater, and therefore must meet the Table 404.1 requirement for a full-load thermal efficiency Et of at least 80%. Because it provides space heating as well, the boiler must also meet Table 403.1f requirements for a combustion efficiency Ec of at least 80% at both full load (high fire) and at its minimum firing rate (low fire). In this case, the Table 404.1 requirement is more stringent at high-fire since thermal efficiency includes casing losses; the Table 403.1f requirement applies at low fire.

Example 403C Process Conditioning – Computer Room

Q A computer-room air conditioner serves a large central computer room. Does it have to meet any of the requirements of the 90.1 Code?

A There are two issues presented in this problem: the requirements for unlisted equipment and the exemption for equipment that serves process loads.

Computer-room air conditioners, air conditioners that are specifically designed for conditioning computer rooms, are not listed in the tables of Section 403.1, and therefore have no minimum efficiency requirements to meet. If the computer room were conditioned by a standard air conditioner, one that is covered by one of the tables in Section 403.1, the air conditioner would in general have to meet the requirements appropriate to that product type. However, the present scope of the 90.1 Code does not include areas of buildings intended primarily for commercial or industrial processes, so equipment and systems that serve a computer room in general need not comply with any requirements of the code. But, if the system also serves adjacent, non-process areas such as storage and offices, areas which are covered by the code, then it must meet applicable requirements of Sections 403. As a general rule, if 90% or more of the load for a specific piece of equipment is process conditioning, then that equipment is exempt from the scope of the code.

Even if the unit need not comply, it would be cost effective to select a unit having very high efficiencies because it will operate so many hours of the year.

Example 403D Comparison of Site Energy Usage – Condensing and Non-condensing Furnaces

Q A manufacturer's product lines include a condensing furnace with an AFUE of 93% as well as high efficiency non-condensing furnace with an AFUE of 80%. Will the condensing furnace have the lower energy costs?

A Not necessarily. The AFUE only includes gas energy usage, not electrical power usage. The condensing furnace will usually require higher indoor fan power and higher combustion air fan power than the non-condensing furnace due to the extended heat exchanger area required for condensing. Depending on local gas and electricity rates, this higher fan energy may offset the lower gas energy usage. To accurately compare between the two, both the gas and electricity costs, calculated using local utility rates, must be included.

Example 403E Comparison of Site Energy Usage – Air-Source and Water-Source Heat Pumps

Q A water-source heat pump has a 10.1 EER and a 4.0 COP at standard full-load rating conditions. An air-source heat pump has a 9.5 EER and a 3.1 COP at standard rating conditions. Will the water-source heat pump use less energy?

A Again, not necessarily. The water-source heat pump rating does not include the energy required by condenser water pumps, cooling-tower fans or auxiliary boilers. Consider the following:

- With all the units cooling, the water-source heat pump system will probably be more efficient than the air-source heat pumps even with the condenser pump and tower fan energy use accounted for.
- With all of the units heating, it is likely that the water-source system will be much less efficient. Whatever energy the water-source heat pumps extract from the condenser water loop must be made up by the auxiliary boiler. The heat extracted from outside air by the air-source heat pumps, on the other hand, is essentially free.
- With units simultaneously cooling and heating, the water-source system recovers heat whereas the air-source heat pumps do not.

To accurately compare the two, a computerized energy analysis per Chapter 13 of ASHRAE/IES 90.1-1989 should be performed.

LOAD CALCULATIONS

Form
403.2.1

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

The designer must make heating and cooling load calculations before selecting or sizing HVAC equipment. The purpose of this requirement is to ensure that equipment is neither oversized nor undersized for the intended application. Oversized equipment not only increases owner costs, but usually operates less efficiently than properly sized equipment. It can also result in reduced comfort due to, for example, a lack of humidity control in cooling systems and fluctuating temperatures from short-cycling. Undersizing will obviously result in poor temperature control in extreme weather but can also increase energy usage at other times. For example, an undersized heating system may have to be operated 24 hours per day because it has insufficient capacity to warm up the building each morning in a timely manner.

Accurate calculation of expected heating and cooling loads begins with a reliable calculation methodology. The 90.1 Code requires in Section 403.2.1 that calculation procedures be in accordance with the ASHRAE Fundamentals Handbook, (1989). It is also acceptable to use similar computation procedures, such as those developed by some of the major equipment manufacturers and other professional groups. There is no universal agreement among engineers on a single load calculation procedure, and the available procedures produce results that may vary by 50% or more. This is because building loads are so complicated that all calculation methods and computer software must have simplifying assumptions embedded within them to make them practical to use. Depending on application, these simplifications can result in inaccuracies and/or errors. The designer should be aware of the limitations of the calculation tool used and apply "reality checks" to the results, based on past "real life" experience, to avoid sizing errors.

Once a reliable calculation procedure is selected, accurate load calculations then depend on the use of accurate design parameters. Because of the very large number of parameters that affect the size of HVAC loads, and the even larger range of possible values of these parameters, the 90.1 Code does not specifically limit design parameters. For instance, there is no “correct” inside space temperature for all applications; what is appropriate for a nursing home may not be appropriate for a detention center. Likewise, there is no “correct” equipment loads; one office space may have PCs and printers on every desk in 10 ft by 10 ft cubicles, while another may have many private offices with only small office equipment loads. The sheer number of variables and the wide range of reasonable values for each of them makes regulation of the details of load calculations impractical. The proper selection of design parameters is left totally up to the professional judgment of the designer.

EQUIPMENT AND SYSTEM SIZING

Form
403.2.2

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Most HVAC system components are less efficient at part load than they are at full load. Section 403.2.2 of the 90.1 Code therefore requires that HVAC systems and equipment be sized to provide no more than the cooling and heating design loads calculated in accordance with Section 403.2.1.

The intent of this restriction is often misunderstood because the term "oversizing" is commonly used in two different ways. As it applies here, "oversizing" is selecting equipment that has a higher output capability than required by the design load. But the term is also used to describe the sizing of equipment using design parameters that are more conservative than is typical for the industry, such as sizing a duct for 0.05 in./100 ft pressure drop, using a 49 in. fan when a 44 in. fan would work; sizing a coil or filter bank for 300 fpm, or selecting a cooling tower to deliver 75°F condenser water. These designs are not intended to increase the capacity of the system and are not considered "oversizing" in the context of the 90.1 Code. In fact they will result in improved energy efficiency and should be encouraged. On the other hand, sizing a chiller for 200 tons when the load is 150 tons, or sizing a fan for 10,000 cfm when the load requires only 8,000 cfm, is considered oversizing. This type of oversizing will almost always result in reduced energy efficiency. The important distinction between the two is that "oversizing" in this context results in a system whose design capacity exceeds the required design load.

For a single piece of equipment that has both heating and cooling capability, only one function, either the heating or the cooling, need meet the oversizing restriction. Capacity for the other function must be the smallest size, within available equipment options, that meets the load requirement. This is primarily intended to apply to unitary equipment, such as heat pumps or gas/electric units. Such equipment is generally selected to meet the space cooling load, which often leaves them oversized for heating because of the nature of the equipment (such as heat pumps whose capacity in the heating mode will be determined by its capacity in the cooling mode since the same compressor and heat exchangers are used) or by limited heating options (such as gas/electric units for which there are typically only one or two gas furnace options, the smallest of which may be larger than needed.) Another example is direct gas-fired absorption chillers/heaters which have both heating and cooling capability.

There are three exceptions to the system sizing requirement:

Form
403.2.2
Exception
#1-#3

(1) Equipment capacity may exceed design loads provided the equipment is the smallest size that can meet the load within the available options of the desired equipment line. The "desired equipment line" is at the discretion of the designer. (See Example 403F.)

(2) Equipment capacity may exceed design loads if the equipment is intended for stand-by use only, and controls are provided that allow its operation only when the primary equipment is not in operation.

(3) Multiple pieces of the same equipment type, such as multiple chillers or boilers, may have a total capacity exceeding the design load if controls are provided to sequence or otherwise optimally control the equipment as a function of the operating load. This is an ideal design technique to allow for additional load for future expansion without reducing energy efficiency. It also provides redundancy to mitigate the impact of equipment failure.

Section 403.2.1 gives designers flexibility in the calculation of heating and cooling loads. This prevents strict enforcement of the sizing restriction in this section of the code. The designer may simply adjust the design parameters in load calculations until an oversized piece of equipment is justified by the calculation. Nevertheless, a sizing restriction remains in the code at least to ensure that load calculations are done for each major piece of equipment (e.g. chiller, boiler, and air conditioning unit) and that they are used to help in the selection of that equipment. The accuracy of the calculations and the extent to which they are used in the equipment selection process is left up to the professionalism of the designer.

Example 403F Selection of Equipment within a Manufacturer's Line

Q A designer wishes to install a certain brand of equipment because of price, support by a local vendor, and availability. But a competing manufacturer has equipment that more closely matches the required loads. Does the latter equipment have to be used to minimize oversizing?

A No. Exception (1) allows the designer to select the equipment that best meets the load "within available options of the desired equipment line." This may seem to be an overly generous exception, but it would be impractical for a designer to search all available equipment manufacturers to find the one that best meets the design loads. A building with multiple air conditioners might each end up being manufactured by different companies, which is obviously impractical, not to mention a long term maintenance nightmare. This concept could be carried to even further extremes; if a designer wished to use a gas/electric unit to serve a space, would he have to switch to a hydronic heat pump instead if it best fit the load requirement?

Example 403G Zoning With Process Humidity Control Requirements – Computer Center

Q A computer center has 10,000 ft^2 of data processing area that must be maintained within tight temperature and humidity ranges. Adjacent to this area are 5,000 ft^2 of offices, tape storage, and other support spaces. For simplicity, the designers are considering serving these support spaces with the same raised floor systems serving the computer room. Would this design comply?

A It would only comply if the total air flow required to condition the support spaces was less than 25% of the total air flow of systems serving both the computer area and support spaces. Otherwise, separate systems would have to be installed to serve the support spaces, controlled only for comfort.

SEPARATE AIR DISTRIBUTION SYSTEMS

Form
403.2.3

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Often spaces housing temperature or humidity sensitive equipment or processes are located adjacent to spaces that need only comfort conditioning. To avoid the energy that would be wasted by over-conditioning the non-process spaces, the 90.1 Code requires that they be served by separate air handling systems, one controlled for comfort purposes and the other controlled as required by the process. Alternatively, the two spaces may be served by a single system if it is controlled only as required for comfort, while supplementary equipment (such as humidifiers, auxiliary cooling equipment) is added to maintain the process requirements. These measures are not necessary if the spaces requiring comfort conditioning use no more than 25% of the total system supply air quantity or if they do not exceed 1,000 ft^2.

VENTILATION AND FAN SYSTEM DESIGN

This code section has two parts: an outside air ventilation limitation and a fan power limitation.

Form
403.2.4

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Outside Air Ventilation

HVAC systems must be capable of supplying outside air at the minimum rate required by the applicable ventilation code. (The applicable ventilation code will vary by location; see local building or mechanical codes and amendments.) Most ventilation codes will be based in one form or another on ASHRAE Standard 62-1989, Ventilation for Acceptable Indoor Air Quality. The minimum outside air intake may be higher where required to make up process exhaust systems or other special requirements, or if energy costs are subsequently reduced, as occurs with an outside air economizer.

The 90.1 Code does not limit outside air intake to code minimum levels; it simply requires that systems have the capability to operate at those levels. This seemingly loose requirement is because many experts feel higher rates are required for adequate indoor air quality. The emission and control of indoor air pollutants is extremely complicated and there are few comprehensive technical reports or reference studies to use as design guidelines at this time.

If designed to supply higher rates than the minimum prescribed by code, systems must be capable of easy adjustment to supply only the minimum rate at some future time. Acceptable means of adjustment include varying fan speed by changing sheaves or adjusting moveable dampers. This would allow the system designer to build in added ventilation capability for the system during the first few months of building occupancy when the off-gassing from furnishings is most severe. The system must, however, be able to be adjusted to supply only the minimum rate at some later time.

In many cases, to meet this requirement, systems will have to include a return fan. For instance, a designer may wish to pressurize a warehouse using a 100% outside air furnace. If the supply rate of this furnace exceeded the code minimum ventilation rate, a return air system would have to be added to reduce outside air while maintaining the overall supply rate needed to meet heating requirements.

While adequate indoor air quality should take precedence over energy conservation, one must not overlook the energy consequence of achieving this goal by dilution with outdoor air. Conditioning outside air is extremely energy intensive in many climates and the designer should minimize outside air intake for ventilation.

A detailed discussion of ways to achieve acceptable indoor air quality at minimal energy usage is beyond the scope of this manual. The amount of outside air required can be minimized by use of:

- Source control (eliminate or minimize internal pollutant sources)
- Source containment (use of local exhaust systems or isolation of contaminants into special rooms)
- Demand controls (use of sensors to adjust outside air intake as a function of indoor contaminant levels)
- Use of transfer air (diluting pollutants in one room by transferring air from a less polluted or over-ventilated space rather than using outside air)
- Proper design and selection of air distribution systems to improve ventilation effectiveness

Where outside air rates must be high, the use of heat recovery systems such as heat wheels and heat pipes can reduce the energy impact .

Form
403.2.4

Fan Power Limitation

	Single Zone	Multiple Zone
Unitary	**Not required**	Required
Hydronic	Not usually required	Required

Fans are generally the largest energy using components of HVAC systems. But regulating fan system design to improve performance is made difficult by the wide number of fan applications, from small fan-coils serving a single zone to large central fan systems serving entire buildings.

Some buildings are composed of many small fan systems that together consume a large amount of energy. These are not regulated by the 90.1 Code for a few reasons. First, it is simply impractical to require the designer to demonstrate compliance for a myriad of different fan systems in a building. More importantly, it is unlikely that regulations would have a significant impact on the energy used by such small systems. Most small air conditioners and fan-coils are very limited in the external pressure they can overcome anyway, so it is unlikely that designers are wasting significant amounts of fan energy by poorly designing air distribution systems. Lastly, it is difficult to establish fan power limits that are applicable to all small fan applications; in one case the limit may be overly stringent while in another, the limit may be easily met making compliance a wasteful exercise.

Maximum Peak Power

The fan power limits in Section 403.2.4 are, therefore, are limited to relatively large systems with significant fan system pressure drops (more than 10 hp):

- Fans that supply a constant volume of air whenever the fan is operating must have a total fan system power demand no higher than 0.8 W/cfm of supply air.
- Fan systems whose supply volume varies as a function of load (VAV systems) must have a total fan system power demand no higher than 1.25 W/cfm of peak supply air.

Fan power can be calculated from the following equation:

$$W = 746 \times \left(\frac{BHP}{\eta_m \times \eta_d}\right)$$

where

W = the fan power in watts

BHP = the fan brake horsepower, which is the power required at the fan shaft, measured in horsepower. The fan brake horsepower requirement can be found in manufacturer's catalogue data or generated by their proprietary computer programs. Where data are not available, assume that fan brake horsepower is equal to fan motor horsepower.

η_m = the fan motor efficiency. Motor efficiencies will vary by motor type, speed, and size. Table 403B below provides typical motor efficiencies which should be used if actual data are not known.

η_d = the drive efficiency, such as belt drives and, where applicable, variable speed drives. Belt drive efficiencies vary depending on the number and type of belts. If better data are not available, assume belt drives are 97% efficient. Variable speed drive efficiencies, which vary with drive type and size, can be obtained from the manufacturer.

In general, only large central systems will find the fan-power ratio requirements of the 90.1 Code to be limiting. These include systems with long, high pressure duct runs or poor fitting designs and those with return fans and/or series type fan-powered boxes. Smaller systems will generally comply without any special efforts on the part of the designer.

The fan power restrictions are based on the total fan system power demand. This is defined as the energy demand of all fans in a system that operate at design conditions to supply air from the heating or cooling source (such as coils) to the conditioned spaces and return it to the source or exhaust it to the outdoors. However, the designer does not need to include the additional fan power required by air treatment or filtering equipment with final pressure drops in excess of 1 in. w.g. The following guidelines should be used to determine fan system power (See examples for further clarification):

- Only fans that (1) supply air from the heating or cooling source to the conditioned space, (2) return the air from the space to the source, or (3) exhaust air from the conditioned space to the outdoors need be included. Fans that simply recirculate air locally (such as conference room exhaust fans) need not be included.
- The fan "supply cfm," the denominator in the fan power ratio, is the air supplied from the cooling or heating source. Air that is induced into the air stream at the space (such as in induction systems) is not included in the "supply cfm."
- One fan system is distinctly different from another if it has different heating or cooling sources. For instance, a large ballroom supplied by two air handlers, each with separate supply fans and heating and cooling coils.
- Fans that ventilate only do not qualify as a fan system, e.g. garage exhaust fans or equipment room ventilation fans that transfer only unconditioned outside air. Fan

systems must include a heating or cooling source. (In any case, fans that only ventilate are unlikely to have any problems meeting the design requirements of this section since their pressure drops are typically very low.)

- Only fans that operate at "design conditions" need be included. For a heating supply fan system, only fans that operate at design heating conditions are included; for cooling systems, only fans that operate at design cooling conditions are included. Fans in series fan-powered mixing boxes are included, but not those in parallel fan-powered mixing boxes. For systems that have both heating and cooling capability, the system is rated by the higher of the power required at design heating or design cooling conditions.

VAV Part Load Performance

Section 403.2.4 of the 90.1 Code also limits variable air volume (VAV) system performance at part load. For individual VAV fans with motors 75 hp or larger, the power demand at 50% flow must not exceed 50% of the full-load power requirement, based on manufacturer's test data.

Figure 403B shows generic part-load performance curves for several fan types and static pressure control systems. The curves indicate that only air-foil (AF) and backward-inclined (BI) centrifugal fans might have a problem meeting the 50% power at 50% cfm requirement. To meet the 90.1 Code, these fans, if over 75 hp, will either have to be fitted with variable speed drives or substituted with another fan type such as a vane-axial fan with variable pitch blades. (A forward curved fan that large is not available generally.)

While only an effective requirement for large AF and BI centrifugal fans, variable speed drives are likely to be cost effective for VAV systems with almost any fan type, except perhaps vane-axial fans for which variable pitch control provides excellent part-load performance. Variable speed drives will also reduce noise levels at part load, as opposed to inlet vanes and discharge dampers which increase noise at part loads, and they will allow even air-foil fans to be operated down to very low flow rates (see discussion below under Zone Isolation).

The fan design criteria of Section 403.2.4 of the 90.1 Code do not apply to the following:

Form
403.2.4
Exception

(1) Small fan systems, with fan motor horsepower totaling 10 hp or less.

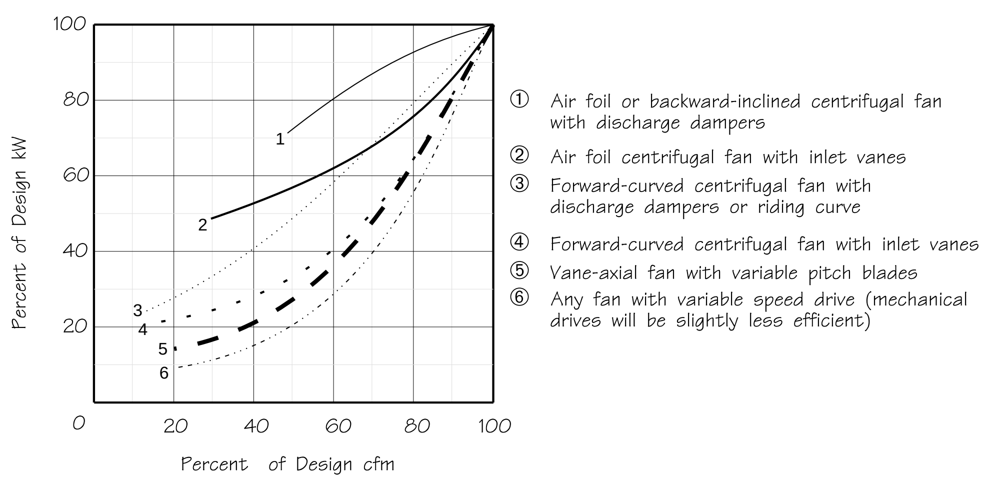
(2) Unitary equipment for which fan energy is included in the efficiency ratings of Section 403.1. This would include all unitary cooling equipment whose cooling performance is measured by EER or SEER, or whose heating performance is measured in HSPF.

Table 403B Typical Motor Efficiencies (%)

Nameplate Rating (hp)	90.1 Code Motor	High Efficiency Motor
1/20	35	--
1/10	35	--
1/8	35	--
1/6	35	--
1/4	54	--
1/3	56	--
1/2	60	--
3/4	72	--
1	75	82.5
1-1/2	77	84.0
2	79	84.0
3	81	86.5
5	82	87.5
7-1/2	84	88.5
10	85	89.5
15	86	91.0
20	87	91.0
25	88	91.7
30	89	92.4
40	89	93.0
50	89	93.0
60	89	93.6
75	90	94.1
100	90	94.1
125	90	94.5
150 and up	91	95.0

(Open Motors, 1,800 RPM synchronous speeds, nominal efficiencies. High efficiency motors from Energy Policy Act of 1992, effective 1997. Standard motor data are from the ASHRAE Fundamentals Handbook, (1989), page 26.8)

Figure 403B Part-Load Curves for a Variety of Fans



Example 403H Calculation of Fan Energy – Fan-Coil System

Q A building HVAC system consists of 40 fan-coils serving individual zones, each with 1/3-hp motors. Does this system need to comply with the fan power limit of Section 403.2.4?

A No. Each fan-coil is a separate fan system because each has a separate cooling and heating source. The total fan system power for each fan system is only 1/3 hp, well below the threshold of exception (1).

Example 403I Adjustment of Fan Energy – Excess Filter Pressure Drop

Q A supply fan rated is at 4 in. w.g. total static pressure including a filter assembly with a final (change-out) pressure drop of 1.25 in.wg. How is the supply fan energy calculated for compliance purposes?

A The supply fan energy need not include the energy required by the filter assembly above 1 in. w.g. So the fan power would be based on a total rating static pressure of:

SP = design SP – (filter SP - 1 in.) = 4 in. – (1.25 in. – 1 in.) = 3.75 in.

Fan energy would be determined from the supply fan curves or rating tables at the design air flow and 3.75 in. w.g. static pressure. As an approximation, fan power without the added filter pressure drop may be estimated to be (3.75/4.0) or 94% of the fan power at the 4 in. design condition.

Example 403J Fan System Design Requirements – VAV Change-over System

Q What are the fan system design requirements for a variable air volume change-over system (also called a variable volume and temperature system) which includes a bypass damper at the fan?

A This system is variable volume at the zone level, but the bypass damper maintains a relatively constant air flow through the fan. The system is therefore a constant volume system in this context, and it must meet the 0.8 W/cfm maximum power requirement.

Example 403K Fan Power Calculation – VAV System

Q Is the central VAV system described below in compliance with the fan power limits of Section 403.2.4?

Quantity	Fan Service	Design, cfm each	Brake Horsepower	Motor Horsepower
2	Supply fans with variable speed drives	75,000	70.5	75 high efficiency
4	Economizer relief fans	32,000	3.5	5
1	Toilet exhaust	6,750	2.7	3 high efficiency
1	Elevator machine room exhaust fan	5,000	unknown	3/4
2	Cooling tower exhaust fans	unknown	unknown	15
15	Conference room exhaust fans	500	240 watts	--
120	Series type fan-powered mixing boxes	1,300 (average)	unknown	1/3

A First, determine which fans to include in the fan wattage calculation.

- The supply fans are clearly included.
- The economizer relief fans are not included because they will not operate at peak cooling design conditions. Had return fans been used, they would have to be included in the calculation.
- The toilet exhaust fan is included since it exhausts conditioned air from the building rather than having it returned to the supply fan and it operates at peak cooling conditions.
- The elevator exhaust fan is not part of the system since, it is assumed in this case, the make-up air to the elevator room is from the outdoors rather than from the building. Had make-up air been transferred from the conditioned space, the fan would have been included.
- The cooling tower fans operate at design conditions but they are also not part of the system because they only circulate outside air.
- The conference room exhaust fans are assumed to be transfer fans; they simply exhaust air from the room and discharge it to the ceiling plenum. Since this air is not exhausted to the outdoors, the fans are not included.
- The series type fan-powered VAV boxes are included since they assist in supplying air to the conditioned space and operate at design cooling conditions. If the boxes were the parallel type, they would not be included since they would not operate at design cooling conditions.

The fans that are included and their power requirements are:

Fan Service	Quantity	Wattage Calculation	Power Each, watts	Total Power, watts
Supply fans	2	$746 \times 70.5 /(0.94 \times 0.97 \times 0.95)$	60,716	121,432
Toilet exhaust fan	1	$746 \times 2.7 /(0.86 \times 0.97)$	2,115	2115
Fan powered VAV boxes	120	$746 \times 1/3 /0.56$	444	53,280
Total				176,827

Example 403K Fan Power Calculation – VAV System (continued)

In these calculations, motor efficiencies are taken from Table 403B. The supply and toilet exhaust fan drive efficiencies are assumed to be 97% (belt drives). The fan-powered boxes are direct drive so they have no drive losses. Since the actual brake horsepower of these fans was not known, it is assumed to be equal to the motor nameplate rating. The supply fan variable speed drive efficiency was found from the manufacturer to be 95%. The final fan-power ratio is

The total supply air quantity in this formula is the air flow rate supplied through the heating or cooling source, which is equal to the total of the two supply fan air flow rates in this case. It is not the total of the fan-powered VAV box air flow rates; although this is the ultimate supply air rate to the conditioned space, this entire air flow does not flow through the heating or cooling source. Therefore,

The series fan-powered VAV boxes supply a constant flow of air to the conditioned space, but the primary air flow, the air flow through the cooling source, varies as a function of load, so this system meets the definition of a VAV system which is limited to 1.25 W/cfm. Therefore, the system complies.

Example 403L Calculation of Fan Power Energy – Floor-by-Floor System

Q A high-rise building has floor-by-floor supply fan systems but central toilet exhaust fans and minimum ventilation supply fans. How is the fan power limitation applied to this system?

A Each air handler counts as a fan system. The energy of the central toilet exhaust and ventilation fans must be allocated to each air handler on a cfm weighted basis. For instance, if one floor receives 2,000 cfm of outside air and the outside air fan supplies a total of 10,000 cfm at 3.5 kW, 20% (2,000/10,000) of the fan energy or 700 W is added to the fan power for the floor's fan system.

Example 403M Part Load VAV Fan System Efficiency

Q A VAV fan system includes a 60 hp supply fan and a 15 hp return fan. Does it have to meet the 50% kW at 50% cfm requirement?

A No. The 75 hp limit applies to each individual fan.

PUMPING SYSTEM DESIGN

Form
403.2.5
Exception
#1-#4

	Single Zone	Multiple Zone
Unitary	Not required	Not required
Hydronic	Required	Required

The 90.1 Code requires that pumping systems with modulating or two-position controls be designed for variable flow. The system must be able to operate down to 50% of design flow or lower.

The following exceptions apply:

(1) Systems for which flow rates greater than 50% of design flow are required for proper operation of the equipment, such as chillers and some types of boilers. While these systems are not required to be designed for variable flow, they can be by using primary/secondary pumping or using a bypass controlled to maintain minimum flow rates through the primary equipment. This is highly recommended for systems with large, high pressure drop distribution systems such as those serving college campuses and airports.

(2) Systems with only one control valve.

(3) Pumping systems with total system pump motor horsepower less than or equal to 10 hp. Pumping system total power is defined as the energy required at design conditions to supply fluid from the heating or cooling source (such as chiller or boiler) out to heat exchangers or coils and return it to the source.

(4) Systems that include supply temperature reset controls in accordance with Section 403.2.6.8 without using any of the exceptions listed in that section. The reset controls will cause water flow rates through control valves to stay high even at low loads: as water temperatures are raised, coil heat transfer effectiveness is reduced. It is possible to use both variable flow and supply temperature reset, although the energy savings together will not be as high as the savings of each feature individually.

The 90.1 Code does not require any specific type of pump flow or pressure control. Pumps that simply "ride their pump curves" – those that are not controlled at all – will still use less energy at low flows than at design flow. However, the higher pressures that occur at low flow may exceed control valve differential pressure ratings and cause flow rates to exceed those desired. This, in turn, causes temperature control problems. Efficiency can be improved and differential pressures can be better controlled by using multispeed motors, staged pumps, or variable speed drives.

Two-way control valves are less expensive than three-way valves and, because no special pump controls are required, meeting the requirements of this section can reduce system first costs over constant flow systems. Properly designed variable flow systems can almost be self balancing as well, reducing balancing costs in the future.

ZONE TEMPERATURE CONTROLS

Form
403.2.6.2

Unit temperature control (403.2.6.1)	Single Zone	Multiple Zone
Unitary	**Systems comply inherently**	Systems generally comply inherently
Hydronic	**Systems comply inherently**	Systems generally comply inherently

An HVAC thermostatic control zone is defined as a space or group of spaces whose load characteristics are sufficiently similar that the desired space conditions can be maintained throughout with a single controlling device. The 90.1 Code requires that the supply of heating and cooling to each such zone be controlled by an individual temperature controller that senses the temperature within the zone.

To meet this requirement, spaces must be grouped into proper control zones. For instance, spaces with exterior wall and glass exposures cannot be zoned with interior spaces. Similarly, spaces with windows facing one direction should not be zoned with windows facing another orientation unless the spaces are sufficiently open to one another that air may mix well between them to maintain uniform temperatures. For the purpose of complying with the code, a dwelling unit, such as an apartment or condominium, may be considered a single control zone. Thus the 90.1 Code does not require rooms of an apartment to be zoned by exposure or by function, although it is recommended that bedrooms be zoned separately from living areas to allow for different operating temperatures during the day and at night.

The independent perimeter system exception applies to zones that are served by two independent HVAC systems: the perimeter system and the interior system. The perimeter system is designed to offset only skin loads, those loads that result from energy transfer through the building envelope. Typically these systems are designed for heating only. The interior system is designed to handle cooling loads from lights and people. This system may also be designed to handle skin cooling loads if the perimeter system is heating-only.

Figure 403C shows an example of this HVAC system design. The perimeter system consists of a heating-only fan-coil, one for each building exposure. The interior system consists of a cooling-only VAV system serving the entire floor, including all exposures as well as interior zones.

This design would not strictly meet the thermostatic control requirements of Section 403.2.6.2 since the perimeter system supplies heating to several zones at once. The heating fan-coil shown in Figure 403D serves four zones of the VAV system. Therefore heating energy from the fan-coil is not controlled by individual thermostats in each zone as required. The interior system and the perimeter system are very likely to fight each other, with the perimeter system overheating some spaces and the interior system overcooling them to compensate. This is obviously energy wasteful. But the system can be designed to mitigate this inefficiency, and the exception allows this design only if:

- The perimeter system has at least one zone for each major exposure, defined as an exterior wall that faces 50 contiguous feet or more in one direction. For example, in Figure 403D, a zone must be provided for each of the exposures that exceed 50 feet in length, while the shorter exposures on the serrated side of the building need not have individual zones.
- Each perimeter system zone is controlled by one or more thermostats located in the zones served. In the past, perimeter systems were often controlled by outside air sensors that would reset the output of the system proportional to outside air temperature. But since solar loads can offset some of the heat loss from a space, this type of control inevitably causes overheating by the perimeter system when the sun is shining, and subsequent fighting with the cooling system. Even when this control is improved by solar-compensation, it can still result in wasteful fighting between interior and perimeter systems due to varying internal loads.

Therefore, only controls that respond to temperature within the zones served are allowed.

In Figure 403D, this requirement might be met by controlling the perimeter fan-coil off of one of the thermostats controlling one of the four interior system VAV zones on the exposure. Alternatively, all four thermostat signals could be monitored and the one requiring the most heat used to control the fan-coil. Finally, a completely independent thermostat could be installed in one of the rooms on the exposure to control the fan-coil, set to a setpoint that was below those controlling the VAV boxes.

Figure 403C Separate Cooling and Heating Systems within a Group of Zones

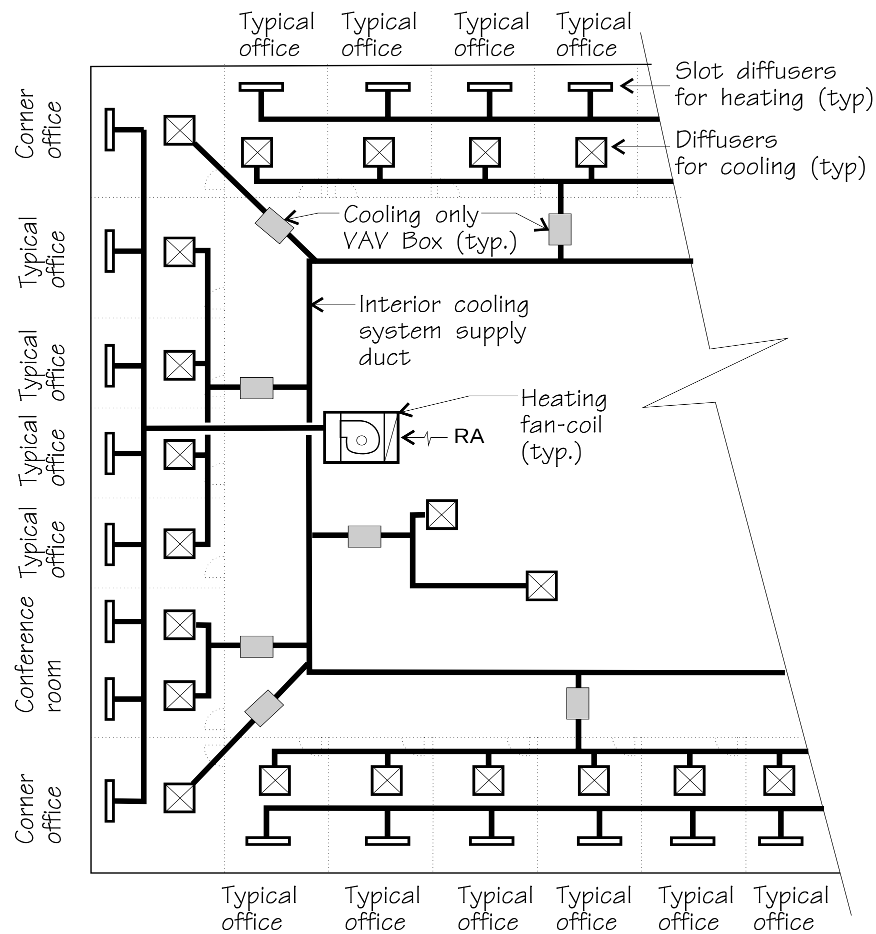
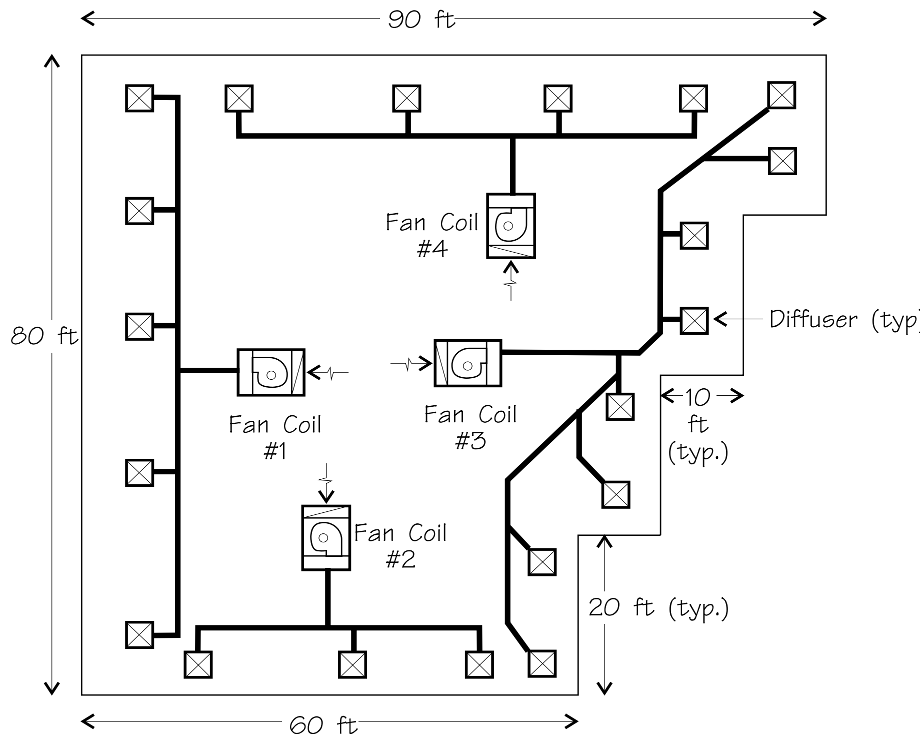


Figure 403D Independent Perimeter Zone Heating System



ZONE THERMOSTAT CONTROL CAPABILITIES

Form
403.2.6.3

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Setpoint Requirements

Zone thermostatic controls used to control space heating must be capable of being set down to 55°F or lower. Those controlling space cooling must be capable of being set up to 85°F or higher. A single thermostat controlling both heating and cooling must meet both requirements.

Note that the 90.1 Code does not require that a single thermostat be used to meet the entire range of setpoints. For instance, one thermostat with a range from 65°F to 80°F may be used to control heating and cooling during normal hours, while one or two other thermostats that can be set down to 55°F and up to 85°F can be installed for the purpose of night set-back and set-up.

Thermostatic controls may be set either locally (with adjustment buttons, switches, knobs, etc.) or remotely (such as by the front-end computer of a direct digital control system). Setpoints may also be changed by replacing elements for thermostats whose setpoint is a function of the sensing element. The 90.1 Code does not require or restrict the use of locking thermostat covers or other means to restrict adjustment by occupants.

Deadband Control

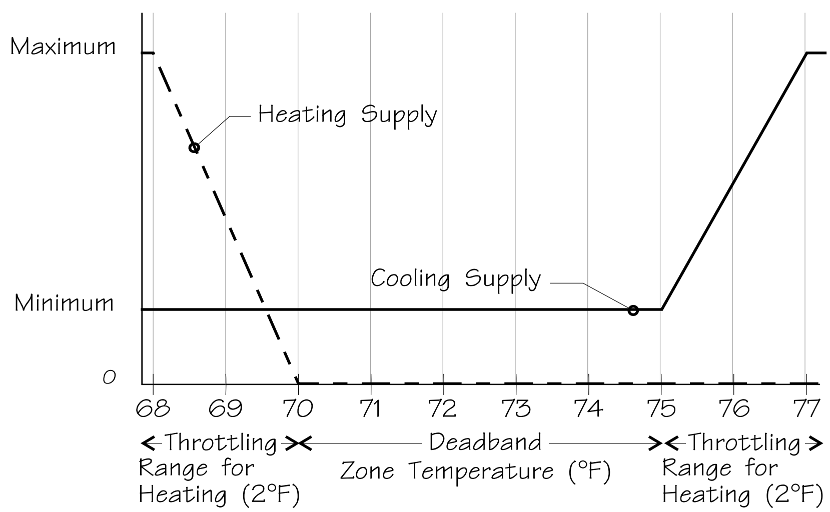
Zone thermostatic controls that control both space heating and cooling must be capable of providing a temperature range or deadband of at least 5°F within which the supply of heating and cooling to the space is shut off or reduced to a minimum.

Figure 403E shows a proportional control scheme that meets this requirement. This might apply to a VAV zone where the cooling source is cold supply air while heating is provided by reheat or perhaps an independent perimeter heating system. The point from where the cooling supply is shut off or reduced to its minimum position to where the heating is turned on is called the deadband and must be adjustable to at least 5°F.

The deadband requirement is typically met using dual setpoint thermostats, which are essentially two thermostats built into the same enclosure. One thermostat controls heating and one controls cooling. The deadband can be achieved by setting the two setpoints at least 5°F apart. (For proportional controls such as pneumatic controls that are calibrated so that the thermostat setpoint is at the midpoint of the control band, the setpoints would have to be set apart by at least 5°F plus one throttling range. For instance, in Figure 403E, the throttling range indicated is 2°F, so the deadband would be maintained by a heating setpoint of 69°F and a cooling setpoint of 76°F.)

Another type of thermostat that would meet the requirement is a deadband or hesitation thermostat. This thermostat is designed to provide a temperature range within which its output signal is neutral, calling for neither heating nor cooling.

Figure 403E Deadband Thermostatic Control



Form
403.2.6.3
Exception

The following exceptions apply:

(1) Thermostats in spaces that have special occupancies where precise space temperature control is required need not have deadband control. Examples include areas housing temperature sensitive equipment or processes, such as computer rooms and hospital operating rooms, or sensitive materials, such as a museum or art gallery. Other examples where deadband control may not be appropriate include homes for the aged, who may be sensitive to wide temperature swings. Buildings where deadband controls are appropriate include office buildings, retail stores, schools, and hotels.

(2) Deadband controls are not necessary for thermostats that require manual change-over between heating and cooling. This is typical of many residential thermostats. The reason for this exception is that occupants will generally allow the space temperature to swing considerably before changing the heating/cooling mode, thereby causing an effective deadband.

Example 403N Deadband Requirement − DDC Systems

Q A direct digital control (DDC) system using a space sensor and a "smart" controller is to be used to control a VAV box with hot water reheat. Does it have to meet the deadband requirement?

A Yes. This system qualifies as a "zone thermostat" although it uses a space sensor and computer rather than a conventional thermostat to control space temperature. The software in the "smart" controller would have to support two separate control loops with individual setpoints, one for heating and one for cooling, each with separate output signals connecting to the VAV damper and reheat control valve, respectively.

Example 403O Deadband Requirements − Single Setpoint Thermostat

Q A single setpoint thermostat is proposed to control a VAV box with hot water reheat. Since the thermostat can be adjusted in the winter to setpoints appropriate for heating, then changed in the summer to a setpoint 5 degrees higher, does it meet the deadband requirement?

A No. This does not meet the intent of this section. The deadband must be continuous and automatic.

Example 403P Deadband Requirement – Pneumatic Thermostats

Q A single setpoint pneumatic thermostat is proposed to control a fan-coil that has a cooling coil and an electric heating coil. The cooling coil control valve operates over a 2 to 7 psi range while the pressure switch for the heating coil is set to turn on the heat at 16 psi and off at 14.5 psi. Since there is a 7.5 psi range between the cooling and heating operating points, does this comply with the deadband requirement?

A No. Typically, pneumatic thermostat gains are calibrated in the range of 2 to 2.5 psi per degree. The 7.5 psi deadband would then correspond to about 3°F to 4°F, not the 5°F required. To meet the requirement, the thermostat gain would have to be 1.5 psi per degree which would cause about a 20°F swing between full cooling and full heating, which is not acceptable for comfort. Thus while this design could be adjusted to meet the 5°F deadband requirement, it would not maintain reasonable space comfort at the same time. Occupants would be forced to defeat the control to maintain comfort, reducing or eliminating the associated energy savings. This does not meet the intent of the 90.1 Code.

HEAT PUMP AUXILIARY HEAT

Form
403.2.6.4

	Single Zone	Multiple Zone
Unitary	Heat pump systems only	**Not required**
Hydronic	**Not required**	**Not required**

The heating capacity of air-source heat pumps will decrease as outside air temperatures fall. To make up for this deficiency, auxiliary heaters are often installed to augment the heat output from the heat pump. With an electric resistance heater (with a COP of 1), the efficiency of the system is significantly reduced compared to the heat pump operating alone (with a COP typically greater than 2). The 90.1 Code therefore requires that controls be provided that prevent auxiliary heater operation when the heating load can be met by the heat pump alone.

Of primary concern is morning warm-up when the space may be well below setpoint even during relatively mild weather. The heat pump could warm the space sufficiently quickly by itself, but typical thermostatic controls would cause the auxiliary heater to operate as well, wasting energy.

The best way to resolve this problem is to use an electronic thermostat designed specifically for use with heat pumps. This thermostat can sense if the heat pump is raising space temperature during warm-up at a sufficient rate, or maintaining space temperature during normal operation, and only energize the auxiliary heat if required. More traditional electric controls can also be used as demonstrated by Examples 403Q and 403R on two-stage thermostats.

Auxiliary heater operation is allowed during outdoor coil defrost cycles of less than 15 minutes duration.

Example 403Q Heat Pump Auxiliary Heat Control – Two-Stage Thermostat

Q Will a simple two-stage thermostat, wired to bring on the auxiliary heat as the second stage, meet the requirements of Section 403.2.6.4?

A No, because it will still cause auxiliary heat to be brought on during warm-up even when outdoor temperatures are mild and the heat pump has adequate capacity by itself. It can be used in conjunction with an outside air thermostat. See the following example.

Example 403R Heat Pump Auxiliary Heat Control – Two-Stage Thermostat with OSA Lock Out

Q Will an outdoor thermostat, wired to lock out auxiliary heat operation during mild weather, meet the requirements of this section?

A Yes, but only if used in conjunction with a two-stage thermostat and only if wired properly. Many manufacturer's installation diagrams show outdoor thermostats wired to provide an additional thermostat stage while using only a single-stage thermostat. It is wired so that electric heat operates with the heat pump when outdoor temperatures are cold (below the outdoor thermostat setpoint). This may cause the auxiliary heat to operate when it is not required since the heat pump may be able to meet the load even during cold weather. The suggested system using conventional controls is:

- A two-stage thermostat with the first stage wired to energize the heat pump and the second stage wired to bring on the auxiliary heat.
- An outdoor thermostat wired in series with the second stage so that the auxiliary heat will only operate if both the second stage of heat is required and the outdoor air is cold.

The outdoor thermostat setpoint must be set to the temperature at which heat pump capacity will be insufficient to warm up the space in a reasonable period of time during warm-up, e.g. 40°F.

HUMIDISTATS

Form
403.2.6.5

	Single Zone	Multiple Zone
Unitary	All humidification or active dehumidification systems	All humidification or active dehumidification systems
Hydronic	All humidification or active dehumidification systems	All humidification or active dehumidification systems

Systems with active means to provide humidification or dehumidification must be controlled by a humidistat.

- For humidifiers, the controller must be capable of being set down to 30% relative humidity.
- For dehumidifiers, the controller must be capable of being set up to 60%. See also Section 403.2.6.6 for limitations in the use of reheat for humidity control.

SIMULTANEOUS HEATING AND COOLING

Form
403.2.6.6

	Single Zone	Multiple Zone
Unitary	Not required except if reheat for dehumidification is used.	Required
Hydronic	Not required except if reheat for dehumidification is used.	Required

As air conditioning system designs were developed in the late 50s and early 60s, energy costs were a minor concern. The systems were designed primarily to provide precise temperature control with little regard for energy costs. Zone temperature control was achieved by reheating cold supply air (constant volume reheat system), recooling warm supply air (such as perimeter induction systems), or mixing hot and cold air (constant volume dual duct and multizone system). While these systems provided fine temperature control, they did so at great expense of energy. (See Reference Section).

To reduce this type of energy waste, Section 403.2.6.6 of the code requires that zone thermostatic and humidistatic controls must be capable of sequencing the supply of heating and cooling energy to each space. These controls must prevent:

- Reheating
- Recooling
- Mixing or simultaneous supply of air that has been previously mechanically heated and air that has been previously cooled, either mechanically or by economizer systems
- Other simultaneous operation of heating and cooling systems to the same zone

Single zone systems, provided their controls are capable of sequencing typical heating and cooling, will inherently meet these requirements. But most common multiple zone systems require the use of simultaneous heating and cooling for zone temperature control.

Form
403.2.6.6
Exception
#1-#5

There are five exceptions to the simultaneous heating and cooling requirements.

(1) Simultaneous heating and cooling is allowed if it is minimized by use of variable air volume (VAV) controls. These controls must reduce the air flow to each zone that is being reheated, recooled, or mixed to a minimum rate that does not exceed the larger of the following:

- 30% of the zone peak supply air volume. This criterion recognizes that typical air outlet performance drops off at low flows, reducing its ability to effectively mix supply air with room air. This is a particular problem with reheat boxes since the supply air temperature at the minimum volume will be warm and the buoyancy of the supply air will further decrease mixing unless the outlet velocities are maintained.
- The minimum required to meet code ventilation requirements. A conservative interpretation of some ventilation codes might lead to very high minimum flow rates in order to guarantee each zone receives minimum outside air regardless of the percentage of outside air brought in at the system level. Instead, to reduce energy costs and first costs, the designer is encouraged to take advantage of direct transfer of air from adjacent over-ventilated zones and indirect transfer of over-ventilation via the return air system (see Equation 6-1 of ASHRAE Standard 62-1989, and pages 9-20 to 9-22 of the ASHRAE/IES Standard 90.1-1989 User's Manual).
- 0.4 cfm/ft^2. Although there is little empirical evidence, many designers feel that a minimum circulation rate of supply air must be maintained for comfort. In fact ASHRAE 90.1 Code 55 states that there is no minimum air velocity required for comfort. Nevertheless, with little or no air movement, some designers feel occupants will complain of stuffiness.
- 300 cfm. This criterion was included specifically to address reheat systems and heating problems that can occur with zones that have low peak supply air volumes but relatively high heating loads, such as spaces with large glass areas facing north or shaded by overhangs and fins. These spaces can often require higher heating volumes than allowed by the above three criteria using reasonable supply air temperatures. In most cases 300 cfm will be sufficient for heating. If not, a fan-powered VAV box could be used.

(2) Zones where special pressurization relationships or cross-contamination requirements are such that VAV systems are not practical. This exception might apply to some areas of hospitals, such as operating rooms, and to laboratories which must be maintained at positive (or negative) pressures to prevent contaminants from entering (or escaping). VAV systems have been successfully used in these applications in an effort to reduce energy costs, but control is very complicated and requires precise air flow measuring and/or pressure measuring instruments. The risk of a failure of these controls, such as the possible release of dangerous chemicals or bacteria, may not be worth the potential energy savings.

(3) Systems where at least 75% of the energy for reheating or producing warm air supply in mixing systems is supplied by heat recovered from some process or equipment within the building (such as chiller condenser heat), or from a solar heating system installed at the site.

(4) Zones where specific humidity levels must be maintained for non-comfort purposes, such as some areas of museums where sensitive materials are displayed or stored, or some computer rooms where equipment must be maintained within precise humidity ranges. Note that much of the computer equipment manufactured today does not require this precise humidity control, such as common personal computers and most mini-computers, so this exception would not apply. Also note that Section 403.2.3 requires that systems serving these

types of spaces be separate from those serving zones that only require comfort conditioning.

(5) Zones with a peak supply air quantity of 300 cfm or less. This exception allows reheat to be used for small subzones of a larger zone. For instance, an air-conditioner might serve an office space, but more heat is needed at the front entry area to offset infiltration when doors are opened. If the supply to the entry is less than 300 cfm, then a reheat coil may be installed to meet this special heating requirement.

Example 403S Minimum Volume Settings – Series Fan-Powered Boxes

Q A VAV system has series type fan-powered boxes with electric heat serving perimeter zones. Zone minimum volume setpoints are set to about 15% of peak supply volumes. Exception (1) to 403.2.6.6 applies only to systems that "reduce the air supply to each zone to a minimum before reheating" takes place. Because of the series fans in the boxes, the supply air to the zone does not change. Does this system meet the 90.1 Code?

A Yes. Because of the non-zero minimum volume settings, the system does not meet the section proscribing simultaneous heating and cooling; the cold primary air is reheated when the electric heaters come on. But it complies with this section through exception (1) because the "air supply" referred to in the wording of the exception is the primary air supply, the supply air that is being reheated, not the total supply air to the space. While the fan maintains a constant overall supply volume to the space, the amount of air that is reheated is reduced as required by the exception.

Example 403T Zone Control Requirements – Packaged Gas/Electric Unit

Q A packaged gas/electric rooftop unit serves a single zone. What is required of this system to meet Section 403.2.6.6?

A The thermostat must be capable of precluding simultaneous operation of the furnace and air conditioner. The standard thermostat will of course do this, so the system meets this section without any added features required.

Example 403U Zone Dehumidification Controls – Open Freezer Cases

Q An air conditioner serving a grocery store with open freezer cases includes dehumidification controls to reduce frosting on the cases. The controls cause the supply air to be overcooled to remove moisture then reheated to satisfy space thermal requirements. Does this system comply with Section 403.2.6.6?

A Yes, it meets exception (4). But there may be other restrictions in the 90.1 Code: Section 403.3.2.3 may require that other areas of the grocery store that do not require humidity control, such as storerooms and adjacent offices, be served by separate systems.

Example 403V Reheat for Dehumidification

Q A constant volume single zone system serving a movie theater in Florida has, in addition to a thermostat which sequences heating and cooling, a humidistat that causes supply air first to be cooled to remove moisture then reheated to maintain space temperature. Does this system comply with Section 403.2.6.6?

A No, unless it complies with one of the exceptions. For instance, the system could be modified to include controls that first reduce air flow (such as inlet vanes or possibly a two-speed motor) before reheating takes place in order to meet the requirements of exception (1) of Section 403.2.6.6. Alternatively, reheat energy could be recovered from the cooling system's condenser (desuperheater or double bundle condenser) or wasteful reheat could be avoided completely by using a heat pipe or plate type heat exchanger that simultaneously precools the air entering the cooling coil as it reheats the supply air leaving of the coil.

AIR TEMPERATURE RESET

Form
403.2.6.7

	Single Zone	Multiple Zone
Unitary	**Not required**	**Required**
Hydronic	**Not required**	**Required**

Section 403.2.6.7 of the 90.1 Code requires that systems that supply heated or cooled air to more than one zone have controls that reset the supply air temperature at low loads: upwards for cooling systems and downwards for heating systems. Reset offers energy savings in three ways: losses due to zone level simultaneous heating and cooling are reduced; duct losses are reduced; and, for cooling systems, the number of hours the economizer can handle 100% of the cooling load is increased. This increased effectiveness of the economizer reduces the number of hours the mechanical cooling system operates at inefficient low load conditions. Reset also allows for more efficient cooling system operation by raising suction temperatures. This occurs directly with DX systems and indirectly, through chilled water reset, with chilled water systems. Reset also provides the secondary benefits of improved low load control at VAV boxes and increased overall circulation rates which can also increase outside air flow rates for systems with air economizers, possibly improving indoor air quality.

The reset control can be based on any of the following strategies:

- *Actual zone requirements, the zone that requires the coldest (cooling systems) or warmest (heating systems) supply air temperature.* In other words, supply air temperature setpoint is reset so that the zone whose supply air flow is closest to its design maximum is maintained near its maximum flow. With proportional controls, this is the zone with the highest (or lowest) thermostat signal. This strategy is theoretically the most energy efficient and most reliable at maintaining comfort. Reset can also be based on zone temperature: supply air setpoint is reset so that the "warmest" or "coolest" zone is maintained at a given high or low temperature limit. But since zones may not all have the same setpoints, this strategy will not ensure comfort and should be discouraged. Reset by zone demand is practical when zones have direct digital controls that can communicate with the fan system controls. But where zones are pneumatically or electronically controlled, the strategy seldom works well in practice due to poor zone thermostat calibration or poor signal selection and transmission.
- *Peak or representative zone requirements.* If supply air setpoints are limited within a fixed range (such as from 55°F to 60°F), this strategy can work well. The zones selected must be the zones that will usually require the lowest supply air temperatures (for cooling systems), such as south and west facing zones. A

possible flaw with this strategy is that the zone that may actually require the most heating or cooling is not one of those monitored.

- *Load indicators such as return air temperature or fan static pressure control signal.* These strategies should be used with caution, however, since they provide only an indication of average zone requirements. Some zones may be under heated or under cooled when the average zone is satisfied, or some zones may have lower or higher than average setpoints.
- *Outside air temperature.* This strategy works best for heating systems since heating loads in the most demanding zone will be almost proportional to the difference between outside and inside temperatures. It can also be used successfully for cooling systems if reset is limited to a small range (such as 55°F to 60°F) and the system does not serve zones that may have increased solar loads in the winter due to low sun angles.

The supply temperature must be reset by at least 25% of the design supply air to room air temperature difference. For instance a cooling system with a design supply air temperature of 55°F and a design space air temperature of 75°F must reset supply air temperature by up to 5°F (25% of the difference between 75°F and 55°F) or from 55°F up to 60°F.

Zones that experience relatively constant loads, such as interior zones, must be designed for the fully reset supply temperature. In other words, for a system with supply air temperatures reset from 55°F to 60°F, interior zone air outlets and terminal devices would have to be sized for 60°F supply air temperature, increasing their size above that required for the 55°F design temperature. If this were not done, no reset would be possible without causing discomfort in interior spaces that have constant loads. (Fan systems and central distribution ductwork would still be sized at the 55°F design condition.)

Form
403.2.6.7
Exception

Reset is not required for systems that comply with Section 403.2.6.6 of the 90.1 Code without exceptions (1) or (2). In other words, reset is not required for systems that either require no simultaneous heating or cooling for zone temperature control or simultaneous heating and cooling is only used in accordance with exceptions (3), (4), or (5). The reason for this exception is that systems that meet Section 403.2.6.6 will have no zone level reheat losses, so reset will offer little energy savings. Examples include three-deck and "Texas" multizone systems, and VAV systems that have zero minimum volume setpoints which is possible with VAV systems with fan-powered boxes or dual-duct VAV systems. Note that ventilation codes may not be met with zero minimum volumes.

While supply temperature reset reduces reheat energy and can improve chiller and economizer performance, it also will increase fan energy. For fan systems that are very efficient at part load, such as those having variable speed drives and very low minimum air flow rates at the zone level to minimize reheat losses, it is possible that reset will increase overall energy usage. Reset may still be desirable even if it increases energy costs because it also increases circulation rates which may improve indoor air quality and comfort. To eliminate the need for reset, the energy cost budget method of compliance with ASHRAE Standard 90.1 must be used. See Appendix A of this Manual and Section 102 of the 90.1 Code.

HYDRONIC TEMPERATURE RESET

Form
403.2.6.8

	Single Zone	Multiple Zone
Unitary	Not required	Not required
Hydronic	Required	Required

Resetting primary chilled water or hot water temperatures upward at part load improves the efficiency of the primary equipment and reduces energy losses through piping. In Section 403.2.6.8, the 90.1 Code requires that all chilled and hot water systems with a design capacity exceeding 600,000 Btu/h include controls that reset supply water temperatures upward (for cooling systems) and downward (for heating systems) at low loads.

Reset may be based on any of the following:

- *Actual system demand*, the cooling or heating coil that requires the coldest (cooling systems) or warmest (heating systems) water. In other words, supply water temperature is reset so that the coil control valve that is the furthest open is maintained near wide open. This strategy is both the most energy efficient and the most reliable at ensuring no coil is "starved." However, it is only practical if there are very few coils served by the system or if all coils are controlled by a direct digital control system that can communicate with the chiller or boiler control systems.

- *Building load indicators such as return water temperature.* This signal should be used with caution, however, since it provides only an indication of average system requirements. For instance, if one coil is at near design conditions while all others are at low load, this strategy would "starve" the first coil and comfort levels in the space it served would not be maintained. This strategy also does not work well if coils are used for dehumidification since colder supply water may be required even at low loads.

- *Outside air temperature.* This strategy works well for heating systems since space loads are almost proportional to the difference between inside and outside temperatures. This strategy will usually not be reliable for cooling systems because the majority of space cooling loads are independent of outside air temperature.

Supply temperature must be reset by at least 25% of the design supply to return water temperature difference. For instance, a chilled water system designed for 44°F supply water and 56°F return water at design conditions would have to reset temperatures by 3°F (25% of the difference between 56°F and 44°F) or from 44°F up to 47°F. In practice, much more aggressive reset is possible and systems should be able to reset supply temperatures by 100% of the design supply-to-return temperature difference or more. For instance, hot water heating systems designed for 180°F supply water temperature at design conditions should be able to operate down to 125°F or lower during mild weather regardless of design return water temperature.

Form
403.2.6.8
Exception

The following exceptions apply:

1. *Systems that are designed for variable flow in accordance with Section 403.2.5 without exception.* For such systems, the use of supply water temperature reset will reduce the pumping energy saved by the variable flow design. This exception, together with exception (4) to Section 403.2.5, means that hydronic heating and cooling systems either have to be variable flow or have supply temperature reset controls, but not both.

2. *Where the design engineer certifies to the building official that supply temperature reset will cause improper operation of heating, cooling, humidification or dehumidification systems.* This exception applies to systems for which supply water temperature cannot be reset by 25% or more of the design supply to return water temperature difference at any time without causing

improper or inadequate temperature or humidity control. Examples include systems requiring dehumidification capability at all times of the year. This exception should not be used for systems for which reset by actual coil demand (Strategy 1. described above) is practical, such as systems with direct-digital control of all valves, since this reset strategy will not allow any coil to be "starved."

Unlike supply air temperature reset which can significantly increase fan energy costs, supply water temperature reset will usually only have a small effect on flow rates. This is because reset only affects the water temperature difference (to which flow rate is inversely proportional) indirectly by reducing coil heat transfer effectiveness. It is therefore unlikely that reset will cause a pump energy increase large enough to offset the reduced piping losses and increased primary equipment efficiency.

Example 403W Reset Requirements – Boiler Reset on Outside Air

Q A gas-fired boiler includes a controller that resets the boiler hot water setpoint proportional to outside air temperature. In order to prevent flue gas condensation on the tubes and flue, hot water temperatures may not be reset as aggressively as they might be if a mixing valve were used. Does this design comply?

A Probably. The supply water need only be reset by 25% of the design supply to return water temperature difference. To prevent flue gas condensation, supply water temperatures should not fall below about 125°F or so. If the system were designed for 180°F supply and 140°F return, typical conditions, the 90.1 Code only requires reset down to 170°F (180 less 25% of the 40°F design temperature difference), well above the 125°F lower limit. (Note: designs that use a mixing valve to allow even lower supply water temperatures should also include this boiler reset controller to improve boiler efficiency by reducing stack and casing losses.)

AUTOMATIC SETBACK OR SHUTDOWN CONTROLS

Form
403.2.7.1

	Single Zone	Multiple Zone
Unitary	Required. Met with timeclock thermostat	Required
Hydronic	Required	Required

Most HVAC unitary systems serve spaces that are occupied on an intermittent basis, but in a fairly predictable manner. To reduce hydronic HVAC system energy usage during off-hours, the 90.1 Code requires that HVAC systems be equipped with automatic controls that will shut off the system or set back setpoints. Examples of acceptable automatic controls are timeclocks, programmable time switches, energy management systems, direct digital control systems, wind-up bypass timers, and occupancy sensors.

Since the term "HVAC system" applies to all equipment that provides any or all of the heating, ventilation, or cooling functions, this section requires time controls on systems ranging from simple ventilation fans to large chiller plants.

There are, however, two exceptions where time controls are not required:

Form
403.2.7.1
Exception
#1-#2

(1) Systems serving spaces that are expected to be in continuous operation. Examples include hospitals, police stations and detention facilities, central computer rooms, and some 24-hour retail establishments.

(2) Small equipment, those with full load demands of 2 kW (6,800 Btu/h) or less, may have readily accessible manual on/off controls in lieu of automatic controls. This exception is intended to apply to small independent systems such as conference room exhaust fans or small toilet room exhaust fans. The intent is that all energy associated with the operation of the equipment be included in the 2 kW. For instance, a fan-coil that would use chilled or hot water, requiring operation of a remote chiller or boiler, may use in excess of 2 kW of energy when it operates, even though the fan-coil fan itself may be less than 2 kW. In this case, the fan-coil would have to be automatically controlled, although many fan-coils may be interlocked to the same timeclock (see zone isolation below).

Historically, heat pump systems with electric resistance heat were considered to be less efficient when operated intermittently because of the increased use of the resistance heat during warm-up. But this increase is mitigated by the use of proper controls that lock out the auxiliary heat when the heat pump can handle the load (controls that are required by Section 403.2.6.4), and in most cases the savings resulting from reduced fan energy usage and reduced heat gains and losses during setback will offset the increased costs from the resistance heater.

Example 403X Time Controls – Hotel Guest Rooms

Q Does a hotel guest room fan-coil require individual time clock controls?

A No. Hotel guest rooms, while not actually continuously occupied, are continuously available for occupancy and thus fall under exception (1). However, systems that automatically set back or set up thermostat setpoints when rooms are not rented may be cost effective.

Example 403Y Time Controls – Fan-Powered Mixing Boxes

Q A VAV system has fan-powered mixing boxes on exterior zones. Do the fans have to be automatically shut off at night?

A In most cases, the fans in fan-powered boxes will be less than 2 kW and thus would not need automatic shut-off controls. However, they would still require readily accessible manual controls, which would be inconvenient and relatively expensive to install. It would be less expensive to interlock the fans with a primary air differential pressure switch so the fans would shut off when the main fan is shut off. For pneumatically controlled systems, interlock to a control air pressure switch so the fans would shut off when the control air is bled off at night. For digital control systems, automatic off-hour control can be done in software at essentially no cost.

If the VAV boxes had electric duct heaters interlocked to operate when the fan energizes, then automatic controls would be required if the sum of the fan and duct heater power requirement exceeds 2 kW.

Example 403Z Time Controls – Equipment Room Cooling Unit

Q An air conditioner serves an elevator equipment room in an office building and is controlled by a thermostat that cycles the indoor supply fan and the compressor on when there is a need for cooling. Does this unit need time controls so that it shuts off when the building is unoccupied at night?

A No. The equipment must be maintained at a given temperature at all times, and thus this unit qualifies for exception (1). Furthermore, when the elevators are inactive at night, the air conditioner will automatically shut off since there is no load in the space.

SHUT-OFF DAMPERS

Form
403.2.7.2

	Single Zone	Multiple Zone
Unitary	Required if outside air intake exceeds 3000 cfm	
Hydronic	Required if outside air intake exceeds 3000 cfm	

Fans that either introduce outside air into the building or exhaust air to outside the building must have dampers that automatically close when the fan is shut off. These dampers may be either the gravity type (barometric shutters) or motorized type, regardless of whether the fan is supplying or exhausting air.

The purpose of this requirement is to reduce infiltration into the building when ventilation systems are off. Infiltration will speed up the natural cooling or warming of the space during off-hours and thereby increase the energy required to maintain setback temperatures and possibly increase the energy use required to bring the space back to normal occupied temperatures.

There are four exceptions where shut-off dampers are not required:

(1) Systems serving spaces that are expected to be in use continuously, such as hospitals, police stations and detention facilities, hotel/motel guest rooms, and some 24-hour retail establishments.

(2) Individual fan systems supplying or exhausting 3,000 cfm or less. A small system, particularly a ducted system, will be only a minor source of infiltration during off-hours. In severe climates, the addition of dampers may be cost effective.

Form
403.2.7.2
Exceptions
#1-#4

(1) Gravity and other non-electrical ventilation systems may have dampers that are controlled by readily accessible manual controls.

(2) Dampers are not required where restricted by code such as at combustion air intakes or attic vents. Note that most building codes do not restrict the use of dampers in elevator shaft vents provided they are automatically opened in a fire situation. Where such dampers are not proscribed by code, they are required by this section and must be installed. Because of stack effect, such vents can be a significant source of infiltration.

In cold climates, dampers should be installed even where not required by the 90.1 Code for freeze protection and to reduce the amount of time it takes to warm up the building in the morning.

The 90.1 Code does not prescribe any damper construction characteristics. However, it is recommended that outside air dampers have blade and jamb seals when the associated fan system is used for building warm-up or cool-down. In other situations, outside air damper leakage may not be critical since the damper may be opened anyway for ventilation.

Example 403AA Automatic Damper for Outside Air Intake – Five-Ton Packaged Air Conditioner

Q A five-ton packaged air conditioner to be installed to serve an office space in Chicago, Illinois has a small outside air intake for minimum ventilation. Is an automatic damper required or is the manual (balancing) damper sufficient?

A The outside air intake would be designed to intake less than 3,000 cfm, so an automatic damper is not required.

ZONE ISOLATION

Form
403.2.7.3

	Single Zone	Multiple Zone
Unitary	**Systems comply inherently**	Required
Hydronic	**Systems comply inherently**	Required

Large central systems often serve zones that are occupied by different tenants and may be occupied at different times. When only a part of the building served by the system is occupied, energy is wasted if unoccupied spaces are also conditioned.

To minimize this waste, the 90.1 Code requires that systems serving zones that can be expected to operate non-simultaneously for 750 hours or more per year be equipped with isolation devices and controls that allow each zone to be shut off or set back individually.

Zones may be grouped into a single isolation area provided:

- The total conditioned floor area of the group does not exceed 25,000 ft^2
- All zones in the group are located on the same floor

Spaces that are expected to be unoccupied only when all other spaces are unoccupied need not be isolated. For example, isolation would not be required for the entry lobby of a multipurpose building since it is occupied when any of the building areas are in operation. This lobby would not benefit from isolation since it would need conditioning whenever the HVAC system is on.

In many cases, the building's eventual occupants are unknown when the HVAC system is designed, such as speculative buildings. In that case, isolation zones may be predesignated provided they do not violate either the 25,000 ft^2 or one floor rule established above.

If occupant schedules are unknown, assume that isolation will be required and make appropriate provisions in the HVAC system design.

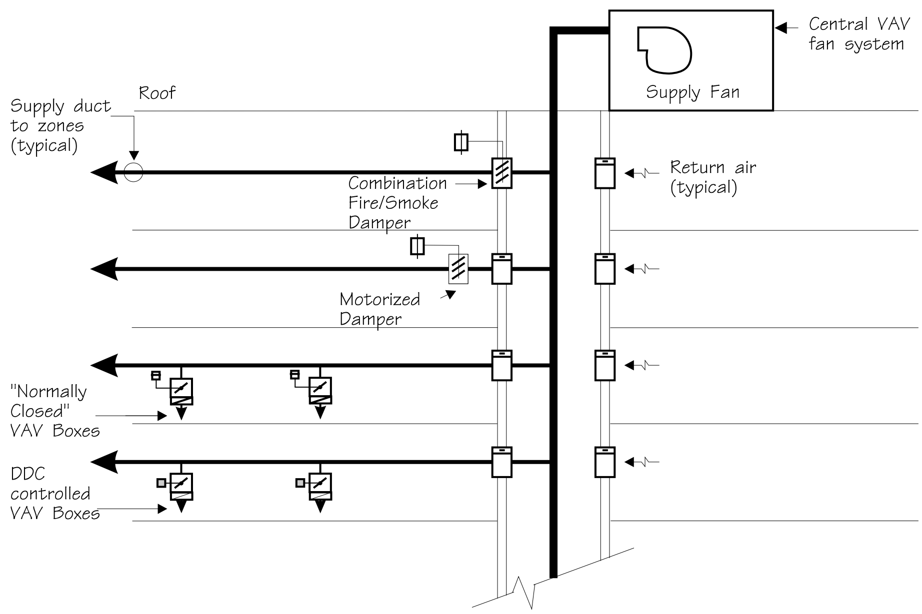
Each isolation area must include individual automatic time controls as if it were a separate HVAC system. This will allow each isolation zone automatically to operate on different time schedules.

Figure 403F shows a schematic riser diagram of a central VAV fan system serving several floors of a building, each assumed to be less than 25,000 ft^2. Isolation of each floor is required if they are to be occupied by tenants that can be expected to operate on different schedules, or if tenant schedules are unknown. Isolation of floors or zones may be easily accomplished by any one of the methods depicted schematically in Figure 403F:

- On the lowest floor, individual zones are controlled by direct digital controls (DDC). If the DDC software can be programmed with a separate occupancy time schedule for each zone or for a block of zones, isolation can be achieved without any additional hardware. The boxes are simply programmed to shut off or control to setback setpoints during unoccupied periods.
- On the next floor up, zone boxes are shown to be "normally closed," which means when control air or control power is removed, a spring in the box actuator causes the box damper to close. This feature can be used as an inexpensive means to isolate individual tenants or floors. The control source to each group of boxes is switched separately from other zones. When the space is unoccupied, the control source is shut off, automatically shutting off zone boxes. A separate sensor in the space can restore control to maintain setback or setup temperatures.
- On the next floor up in Figure 403F, isolation is achieved by simply inserting a motorized damper in the supply duct.

- On the top floor, the cost of this damper is saved or reduced by using a combination fire/smoke damper at the shaft wall penetration. Smoke dampers are often required by life safety codes to control floor air flow for pressurization. These dampers may serve as isolation devices at virtually no extra cost, provided they are wired so that life safety controls take precedence over off-hour controls. (Local fire officials generally allow this dual usage of smoke dampers and often encourage it since it increases the likelihood that the dampers will be in good working order when a real life safety emergency occurs.)

Figure 403F Isolation Methods for a Central VAV System



Note that on all floors in Figure 403F, shut-off is not shown on floor return openings. This is because the wording of the 90.1 Code requires only that "the supply of heating and cooling" be shut off. In addition, with a plenum return system, the amount of air drawn off an unconditioned floor will be negligible compared to the occupied floors that have positive air supply since the latter will be pressurized.

Note also that a positive means of zone shut off or setback is required. Shut-off VAV boxes (boxes with no minimum volume setting) cannot be assumed to close automatically during unoccupied periods due to low loads since there may be 24-hour internal loads (such as PCs, idling copy machines, or emergency lighting) and envelope loads are continuous.

Simply providing means for central system zone isolation does not end the design task. Central equipment must be capable of operating at the very low loads that can be expected when only one isolation zone is operating.

Experience has shown that almost any fan with a variable speed drive for static pressure control can operate stably to near zero flow. This is true even for large centrifugal fans which will eventually pass into the surge region of their fan curves as load reduces, provided this occurs when the fan is operating below about 50% speed and static pressure setpoints are less than about 2 in. w.g. Under these conditions, fan power is reduced to the point where surge pulsations will generally have too little energy to cause damage.

Large axial fans with variable pitch blades may also be able to operate at low flows without over-pressurizing ductwork. Fan curves at minimum blade pitch should be reviewed to be sure shut-off pressures are below duct design pressures.

Where fans cannot be selected to operate safely at low loads, large fans can be broken into smaller fans in parallel with operation, staged so only one fan operates at low loads.

The same considerations can apply to central chiller plants. The plant must be able to operate at low loads for extended periods. If frequent chiller cycling is not acceptable, either multiple or staged chillers can be used. As a last resort, hot-gas-bypass can be used to maintain stable low-load operation, but this can significantly increase energy costs and should only be used if installation cost budgets prohibit the use of multiple staged chillers.

Example 403BB Off-Hour Isolation Controls – Floor-by-Floor System

Q A speculative office building is designed to have an air handling system on each floor. What off-hour isolation provisions are required?

A If the floors are less than 25,000 ft^2 of conditioned area, then each floor may be considered an isolation zone. Each fan system, and associated zone level heating systems, must be able to operate on a different time schedule.

If the floors are larger than 25,000 ft^2 and expected to be occupied by different tenants operating on different schedules, the system will have to be broken into more than one isolation zone.

Example 403CC Off-Hour Isolation Controls – Water-Loop Heat Pump System

Q A speculative office building is served by an HVAC system consisting of individual hydronic heat pumps for each zone connected to a central condenser water pump, cooling tower, and boiler. A central outside air fan provides ventilation air to each heat pump. What off-hour isolation devices are required?

A The heat pumps must be grouped into isolation areas, ideally one area for each tenant. Unless they cover only one tenant or tenants that operate on similar schedules, isolation areas may be no larger than 25,000 ft^2 each and may include zones only from one floor. Each isolation area must include an individual time control to control the heat pumps within that area. This might be an individual timeclock thermostat for each zone, or for only one of the zones in the isolation area with interlocks to the other heat pumps in the area. Each isolation area control would need to be interlocked to start the central equipment as required.

The ventilation outside air fan and toilet exhaust fan need not include shut-off controls for each isolation area although this is desirable; the wording of the 90.1 Code states that only the "heating and cooling" be isolated, not ventilation or exhaust air.

Similarly, condenser water isolation valves are not required for each isolation area since water flow by itself does not supply "heating or cooling" to the space. Water could simply continue to flow through inactive heat pumps. While not required, automatic isolation valves at each heat pump, interlocked with its compressor, are recommended to reduce pump energy costs.

Example 403DD Off-Hour Isolation Controls – Fan-Coil System

Q A system consists of fan-coils for each zone piped back to a central chiller and boiler. What isolation devices are required?

A Assuming the fan-coils serve spaces that operated on different time schedules, they would have to be grouped into isolation zones similar to the heat pumps in the previous example. Each isolation area would need a timeclock interlocked to control the fan-coils. Because the supply of heating and cooling to each space will stop when the supply fan is shut off, the flow of chilled water and hot water to each fan-coil or to each isolation area need not be shut off independently of other areas, although this is desirable. The central chiller and boiler and associated pumps would be interlocked in parallel with each isolation area control so they would operate when any area is occupied.

ECONOMIZERS

Form
403.2.8

	Single Zone	Multiple Zone
Unitary	Not required for systems less than 3,000 cfm or 7.5 tons	Required in most applications
Hydronic	Required in most applications	Required in most applications

Commercial buildings generally have a cooling requirement even during cool and cold weather. Interior zones, zones not adjacent to the exterior window wall, require cooling year round. Some exterior zones with large expanses of glass, particularly if facing south or west, will require cooling during cool, sunny weather because of high solar loads increased by low wintertime sun angles. Other exterior zones can require cooling during cold weather because of high internal cooling loads from lights, people, and office equipment (computers, copiers).

To take advantage of this characteristic of nonresidential buildings, Section 403.2.8 of the 90.1 Code requires that cooling systems have either an air or water economizer.

Air Economizers

Air economizers (also called air-side economizers) use controllable dampers to increase the amount of outside air drawn into the building when the outside air is cool or cold and the system requires cooling (see Reference section). To meet the 90.1 Code, economizer systems must be able to supply 85% of the design supply air quantity as outside air. The 85% limit is provided rather than 100% to allow VAV system design to be optimized; during cool weather when the economizer is in operation, the system will not be at peak load so supply air quantities will be less than 100%. The economizer (such as outside air dampers) need only be designed for this reduced flow rate.

As the outside air warms up, there will be a point where outside air intake will increase energy usage. At this point, the economizer must be shut off and the system operated at the minimum outside air volume required for ventilation. The controller that causes this to occur is called the economizer high limit control or high limit switch.

There are several common high limit controllers. The simplest, called a fixed dry-bulb high limit, is simply a thermostat located in the outside air intake. At outside air temperatures below the thermostat's setpoint, the economizer is enabled, and at temperatures above the setpoint, the economizer is disabled. A variation of this concept is the differential dry-bulb controller which enables the economizer to operate whenever the outside air temperature is below the return air temperature.

Dry-bulb controllers can inadvertently cause the economizer to increase energy costs if the outside air is cool but moist. In these conditions, the enthalpy (the energy content of the air and water vapor mixture) of the outside air may exceed the enthalpy of the return air because of the high humidity even though its dry-bulb temperature may be lower. This can increase cooling energy by increasing the latent load. Common enthalpy controllers include a fixed enthalpy switch which enables the economizer to operate when outside air enthalpy is below a fixed setpoint, and differential enthalpy controls that enable the economizer to operate whenever outside air enthalpy is less than return air enthalpy.

Enthalpy controllers, particularly fixed enthalpy controllers, can also cause the economizer to inadvertently increase energy usage if the outside air is warm but dry. Under these conditions, the cooling coil may be dry, so no latent cooling is done. Even though its enthalpy may be lower, cooling outside air may take more energy than cooling return air if its dry-bulb temperature is higher and the coil is dry.

On packaged equipment, the most common enthalpy switch is an electronic device that is a type of fixed enthalpy controller but the "setpoint" is a curve that changes as a function of outside air temperature and humidity. In dry weather, the setpoint is essentially a fixed dry-bulb temperature, while in moist weather the setpoint becomes essentially a fixed enthalpy setpoint. In this way, the control will overcome both the "mistakes" made by dry-bulb controllers during humid weather and the "mistakes" made by enthalpy controllers during dry weather.

The 90.1 Code allows any of these high limit controls to be used, both those that measure enthalpy and those that measure dry-bulb temperature. The type selected is up to the designer. In dry climates, the dry-bulb controllers will be the least expensive and most reliable. In humid climates, the added cost and complication of the enthalpy controller will generally be justified by the resulting energy savings.

Because the economizer allows the introduction of large quantities of outside air, some provision must be made to relieve excess air to the outside to prevent over-pressurizing the building which can cause doors to stand open and, in extreme cases, elevator doors to jamb. The simplest, lowest cost, and most energy efficient design is to use barometric relief dampers. The dampers open to allow excess air to exfiltrate via building pressure. If the return air path from the space to the relief damper is too large, barometric relief will not be able to prevent over-pressurization and some form of powered relief must be used. The simplest and most efficient powered relief design is the relief/exhaust fan controlled directly off building pressure. If return air path pressure drop is very high, then return fans should be installed and designed to track supply fan air flow rate less any air flow required for mild building pressurization. The 90.1 Code does not specify which relief design to use, but one of the three must be used in order to ensure proper economizer operation.

Water Economizers

While air-side economizers use cool air directly to reduce cooling load, a water economizer (also called a water-side economizer) uses cool outside air indirectly first to cool water which then cools supply air through a cooling coil, thus reducing the mechanical cooling load.

The water economizer is essentially an indirect evaporative cooler. Water is circulated through a cooling tower where it is evaporatively cooled then circulated through cooling coils to cool supply air indirectly.

To meet the 90.1 Code, water economizers must be able to satisfy the system's entire expected cooling load when outside air temperatures are 50°F dry-bulb and 45°F wet-bulb and below. This design criterion is specified because, unlike air economizers which use cold outside air directly for cooling, the performance of water economizers depends greatly on the selection of components such as cooling towers and heat exchangers. (See Example 403GG.)

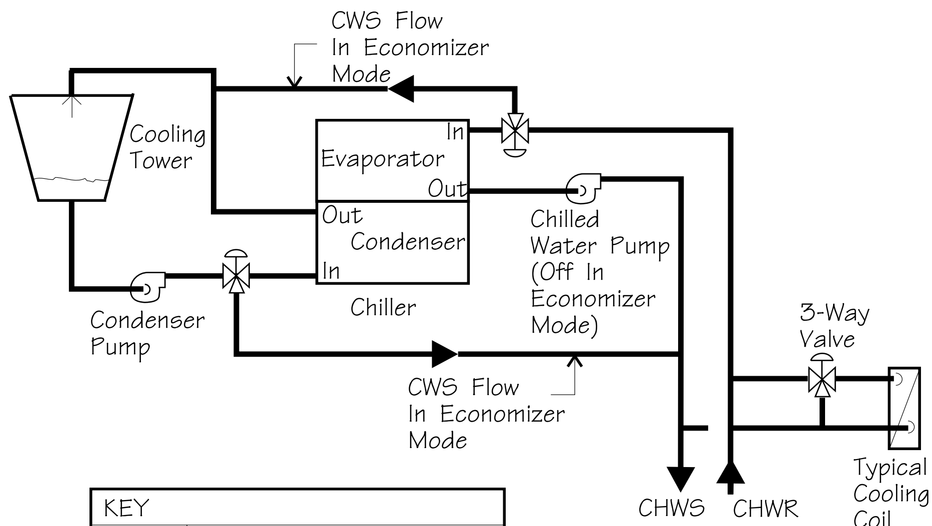
There are three common types of water economizers:

"Strainer-Cycle" or Chiller-Bypass Water Economizer: This type of economizer, shown in Figure 403G, has control valves that can divert condenser water from the cooling tower and run it directly into the normal chilled water piping loop, bypassing the chiller. This bypass configuration will occur as long as the tower can cool the condenser water sufficiently to handle the cooling load, usually around 45°F to 50°F. The term "strainer-cycle," as this type of economizer is commonly called, started as a trade name of a type of in-line water filter intended to clean the dirty open-circuit tower water before it flows into the clean (normally closed circuit) chilled water circuit. Because chilled water control valves and coils easily can become clogged, it is essential to install good water treatment systems with this type of economizer. To resolve this problem, a heat exchanger can be used to isolate the tower and chilled water circuits, at both considerable first cost expense as well as reduced energy savings due to higher pump heads and non-zero heat exchanger approach. Note that

the chiller-bypass water economizer is "non-integrated," meaning the chiller cannot operate when the condenser water is in the bypass arrangement, so the economizer must provide all of the cooling load or it can provide none of it. Because of this characteristic, this economizer design may not meet the integration requirement of this section. (See the discussion below).

Water Precooling Water Economizer: This type of economizer, shown in Figure 403H, uses cold tower water when it is available to precool chilled water return (through a heat exchanger) before it enters the chiller. One advantage of this type of economizer over the "strainer-cycle" is that it is "integrated," meaning it can provide "free" cooling even when the chillers are operating by reducing chilled water return temperatures. It also isolates the open circuit tower system from the chilled water system with the heat exchanger, reducing fouling problems caused by the poor water quality of the open circuit. But the heat exchanger reduces the cooling energy savings because the water leaving the heat exchanger cannot be as cold as the tower water, and it increases pump energy during economizer operation because of the pressure drop of the heat exchanger.

Figure 403G "Strainer Cycle" Water Economizer



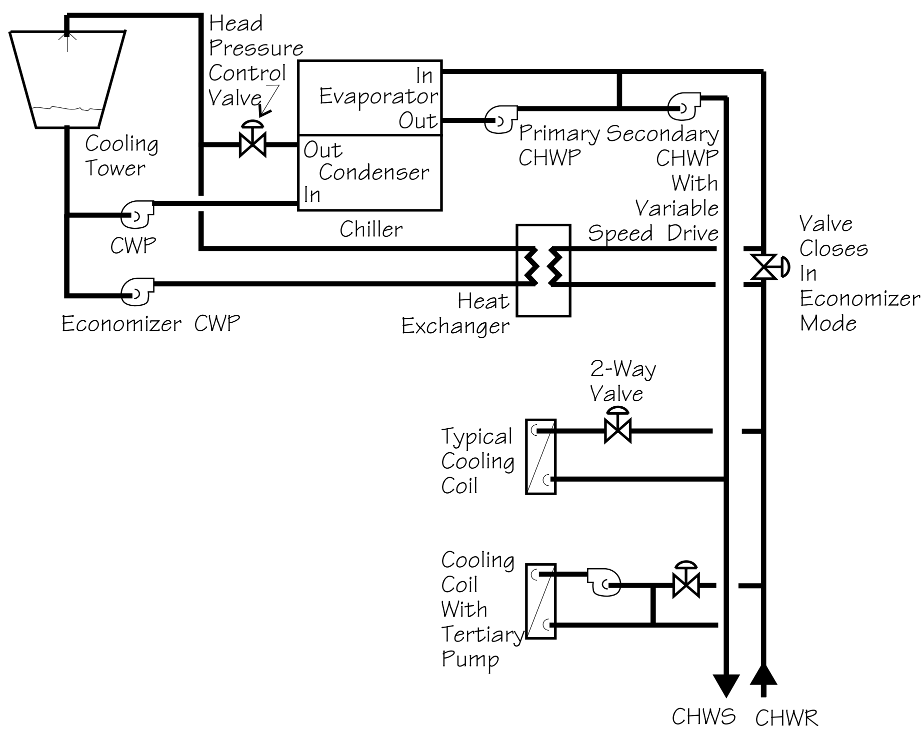
KEY	
CWR	Condenser Water Return
CWS	Condenser Water Supply
CHWS	Chilled Water Supply
CHWR	Chilled Water Return

This type of economizer works best when chilled water return temperatures are kept high, which improves the heat exchanger effectiveness and allows pre-cooling at warmer tower water temperatures. This can be achieved by using two-way valves at cooling coils. With two-way valves, return water temperatures will actually rise above design levels at part load due to the characteristics of cooling coil heat transfer, until the flow drops so low through coils that it becomes laminar and heat transfer effectiveness falls off.

Air Precooling Water Economizer: This type of water economizer requires an additional cooling coil upstream of the normal, mechanical cooling coil, as shown in Figure 403I. Water from the cooling tower first passes through the economizer coil, pre-cooling or fully cooling supply air, then goes on to remove condenser heat from the mechanical cooling system, with water flow modulated or bypassed to maintain head pressures, a control required due to the cold water temperatures. The three-way control valve shown in Figure 403I operates so that, if the tower water is warmer than the return air, the water bypasses the coil to avoid warming the air and increasing the cooling load. This is like the high limit control used with air economizers. Since the economizer and mechanical cooling can operate concurrently with this type of economizer, it is "integrated" and meets the integration requirements of this section (discussed below). This scheme is very popular when water cooled air conditioners are being used since the condenser water must be piped to the units anyway, so the only expense of the water economizer is the added coil and controls.

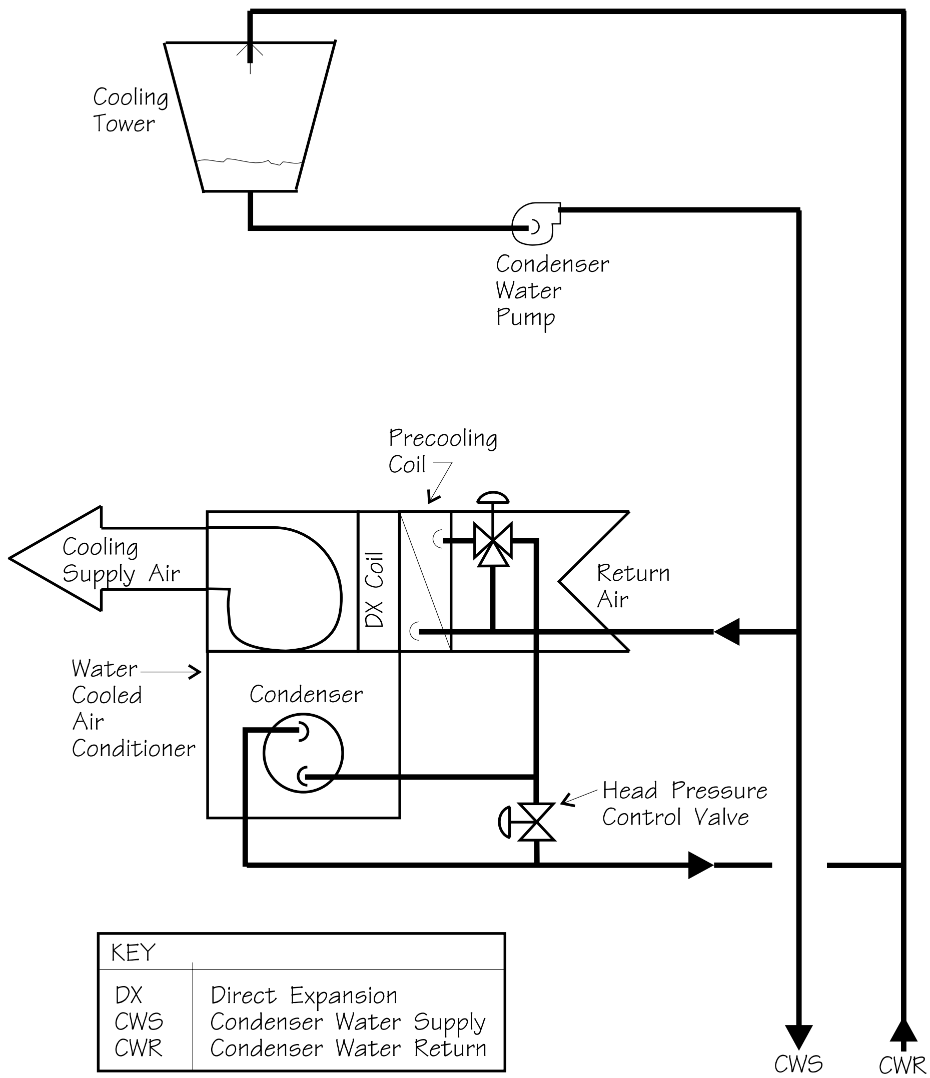
While either air or water economizers will meet the requirements of Section 403.2.8 of the 90.1 Code, air economizers will outperform water economizers except in very dry climates (such as Nevada, Arizona) or very cold climates where

Figure 403H Water Precooling Water Economizer with Two-way Valves



humidification must be added to offset the dryness caused by the introduction of cold, dry air. Water economizers require cooling tower fans and tower pumps to operate during economizer operation, plus added fan and pump pressure drops for economizers like that shown in Figure 403I that will increase fan and pump energy even during non-economizer operation. Air economizers do not have such parasitic loads except where supplemental humidification is required in cold climates. Air economizers also can improve indoor air quality because of the many hours where outside air intake exceeds minimum levels. Thus, air economizers are preferred over water economizers for most applications if the architecture of the building and the HVAC system type is suitable.

Figure 403I Air Precooling Water Economizer



Form 403.2.8 Exception #1-#7

The cost effectiveness of economizers depends on the load characteristics of the building, the type and size of the HVAC system and the local climate. Because there are many cases where economizers will not be cost effective, the 90.1 Code provides the following exceptions to the economizer requirement:

(1) Small cooling units that have a fan capacity of less than 3,000 cfm or a total (sensible plus latent) cooling capacity of less than 90,000 Btu/h (7.5 tons) at design conditions. For such small systems, the added cost of economizers will be significantly higher than the annual energy savings even in mild climates. Note that this exception applies to each individual unit, not to the sum of the capacities of every air conditioner in a building. In other words, if a building has five air conditioners each with five tons capacity, none of them are required to have economizers even though the total capacity for the building exceeds 3,000 cfm or 90,000 Btu/h.

(2) Systems with air-cooled or evaporatively-cooled condensers (those that do not have water-cooled condensers for which a water economizer is practical) need not have economizers if the system includes extensive filtering equipment provided in order to reduce outside air pollutants below EPA National Ambient Air Quality Standards.[1] This exception may apply in some urban areas with gaseous contaminants such as NOx, SOx, and CO that are difficult to filter. Areas with high particulate levels generally need not have “extensive” filtering requirements to meet EPA limits (standard HVAC filters will work) and those with high ozone levels, except in extreme cases, also will not require additional filters since ozone quickly oxidizes within the system before being supplied to the space.

(3) Systems with air-cooled or evaporatively-cooled condensers (those that do not have water-cooled condensers for which a water economizer is practical) need

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1 From EPA Report No. EPA-450/4-90-002, "National Air Quality and Emission Trends" Office of Air Quality Planning and Standards, Research Triangle Park, N.C. March 1990

not have economizers if the design engineer certifies to the building official that the use of an air economizer affects the operation of other systems, such as supermarket refrigeration systems, such that overall building energy costs are increased. This exception is commonly applied to supermarkets with open freezer cases for which poorly controlled economizers can increase space humidity levels which increases display case frosting and hence increases refrigeration system energy usage.

(4) Economizers are not required for systems that serve envelope-load dominated spaces, defined as spaces whose space sensible cooling load at design conditions, not including transmission or infiltration loads, is less than or equal to transmission plus infiltration loads calculated at 60°F outside air temperature. For such spaces, economizers will not be significant energy savers because cooling loads will not occur in cold weather. To demonstrate the applicability of this exception, simply recalculate space cooling loads at 60°F outside air temperature with all other design conditions unchanged. If solar and internal loads are offset by the heat losses through the envelope and by infiltration, then the system serving the space need not have an economizer.

(5) Economizers are not required for systems serving residential occupancies or hotel guest rooms. These spaces, which will also usually be applicable for exception (4), are envelope-load dominated and thus will not experience cooling loads during cold weather.

(6) Economizers are not required for systems for which at least 75% of the annual energy used for mechanical cooling is provided by energy recovery systems or solar cooling systems located on the building site. Examples are systems using absorption chillers powered by solar panel heated water or water heated by a heat recovery system.

(7) Economizers are not required for systems serving naturally ventilated spaces, defined as spaces that have operable openings such as windows or doors whose openable area exceeds 5% of the space conditioned floor area. The entire space must be open to and within 20 feet of the operable openings. To ensure that air conditioning systems are shut off when natural ventilation is available for cooling, automatic controls must be provided that shut off mechanical cooling systems when outside air temperatures are below 60°F.

Internal-to-external zone heat recovery systems, such as a system using a double-bundle chiller that rejects heat to the space heating system, can be more efficient than economizer systems in some climates and for some buildings, but their effectiveness is complicated by the interaction of the building with the climate and the HVAC systems. For this reason, heat recovery systems are not specifically recognized as alternatives to systems with economizers. If a heat recovery system is desired, compliance must be shown using the energy cost budget procedure of ASHRAE Standard 90.1-1989 in accordance with Section 102 of the 90.1 Code.

Economizer Integration

The 90.1 Code requires that economizers be integrated. Integrated economizers can reduce the cooling load while the remainder of the load is met by mechanical cooling. Economizers that cannot operate simultaneously with the mechanical cooling system are called non-integrated economizers. Integration can greatly extend economizer operation in mild climates which reduces cooling energy costs. For instance, a non-integrated air economizer will only be able to reduce cooling energy when outside air temperatures are below 55°F to 60°F, depending on required supply air temperatures. Above those temperatures, mechanical cooling is required so the economizer is shut off. If the economizer were integrated, it could continue to operate, reducing mechanical cooling energy usage even though it cannot provide the entire cooling load, until the high limit setpoint is reached, around 65°F to 75°F depending on the climate. In some climates, there are hundreds or even thousands of operating hours when the outside air temperature is in this range. Air economizers are usually integrated except for some that are applied to small packaged air conditioners, usually third party or after-market products. The controls are wired so that the compressor cannot operate until the economizer has been locked out by its high limit switch, or the economizer is interlocked to shut off when the compressor comes on.

Examples of a non-integrated water economizer are shown in Figure 403G, while those shown in Figures 403H and 403I are integrated economizers since the economizer and mechanical cooling may operate concurrently.

Form
403.2.8
Exception
#1-#2

(1) Small DX systems (those less than 15 tons) may be non-integrated to prevent coil frosting and possibly refrigerant liquid slugging that might occur at the low loads when the economizer is handling most of the load. Small DX units have limited unloading capability so compressor operation under these low-load conditions can quickly lower supply temperatures to the point where condensate on the coil can freeze. For larger DX systems, the economizer may be temporarily shut off or outside air flow reduced in order to prevent coil frosting at the lowest stage of compressor unloading. This maintains a minimum stable load on the compressor in much the same way that hot gas bypass does.

(2) Integration is not required if the system is located in a climate where there are less than 750 hours during the typical weather year when the dry-bulb temperature is between 55°F and 69°F inclusive between 8 a.m. and 4 p.m. For these climates, the energy saved by integrated controls is small since this is the temperature range within which integrated controls extend economizer operation. Weather data for many cities can be found in Section 301.1 of the code.

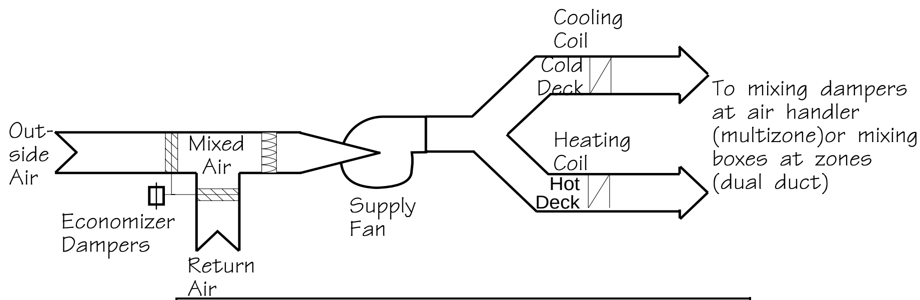
Economizers and Heating Systems

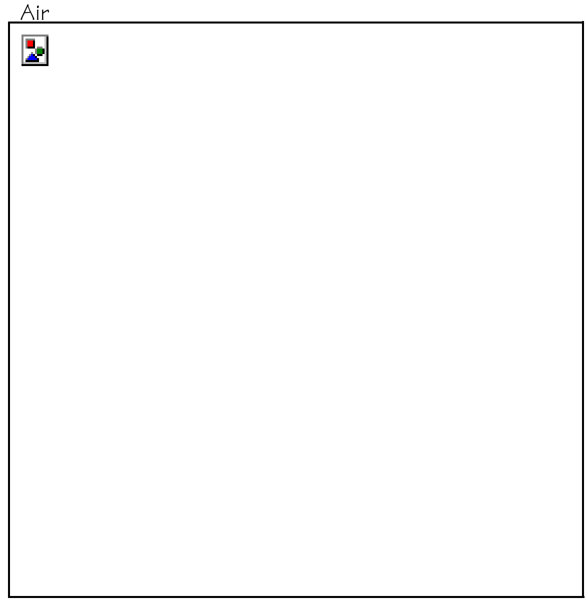
The 90.1 Code requires that the HVAC system and economizer design and controls be such that operation of the economizer does not increase building heating energy costs during normal operation. Systems for which at least 75% of the annual heating energy is provided by site-solar or site-recovered energy are exempt.

This requirement has many implications that can significantly limit HVAC system selection and design. For instance, the single fan dual duct or multizone system shown in Figure 403J lowers the temperature of the air entering the hot deck heating coil, increasing its energy usage. In order to use this type of system, a water economizer must be used, or the system must meet one of the economizer exceptions and have neither type of economizer. (Another resolution is to use a dual fan-dual duct system where the hot deck fan supplies only return air or return air plus minimum ventilation air. This system is often less expensive and easier to control than a single fan-dual duct system.) This requirement will not affect three-deck multizone or "Texas" multizone systems since they cannot work with an air economizer in any case (it would make the neutral deck a cold deck).

This requirement may also impact control system designs for some HVAC systems. For instance, for a VAV system that maintains non-zero minimum volume settings at zones that also have heating capability, the economizer controls must not supply air at a temperature cooler than the supply temperature provided by the mechanical cooling system. Otherwise, reheating energy costs will increase. To ensure this requirement is met, the economizer should be controlled using the same supply air controller that regulates the mechanical cooling system at the same setpoint.

Figure 403J Dual Duct or Multizone System





Example 403EE Economizer Controls with Multiple Compressors

Q A 20-ton packaged air conditioner with an air economizer and two steps of unloading (dual compressors) is controlled by a two-stage cooling thermostat. The economizer is wired to the first stage contact and one of the compressors is wired to the second stage contact. The second compressor can only operate if the economizer has shut off on its high limit switch. Does this meet the requirement for integrated economizer control?

A Yes. Even though the second compressor cannot operate concurrently with the economizer, the first compressor can, so the economizer will be able to meet part of the load while the mechanical cooling provides the remainder.

Example 403FF Strainer-Cycle Water Economizer

Q When can the "strainer-cycle" water economizer shown in Figure 403G be used?

A This economizer is non-integrated since the chiller cannot operate at the same time as the economizer, so it does not meet the integration requirement (second paragraph of Section 403.2.8 of the 90.1 Code.) It is only allowed if the system is located in a climate where there are less than 750 hours during the typical weather year when the ambient dry-bulb temperature is between 55°F and 69°F inclusive between 8 a.m. and 4 p.m. There are many areas of the country with these climate conditions where this type of water economizer may be used. The system may also be used if compliance can be shown via the energy cost budget method of ASHRAE/IES 90.1-1989 per Section 102. (See Appendix A)

Example 403GG Water-Side Economizer – Performance Verification

Q A system is designed to use the water economizer depicted in Figure 403G. How is compliance demonstrated with the 90.1 Code requirement that the economizer provide 100% of the expected cooling load at outside air temperatures of 55°F dry-bulb/45°F wet-bulb and below?

A Because it requires knowledge of the system's performance at off-design conditions, the calculations required to demonstrate compliance are rather complicated. The following approach is suggested:

Heating and Cooling Loads. Recalculate heating and cooling loads just as they were done for design loads except change the outside air temperature to 55°F dry-bulb and 45°F wet-bulb. All other parameters must remain at design conditions. The economizer must be able to meet the cooling load calculated in this manner without supplemental chiller operation.

Supply Air Temperature. Determine the supply air temperature of air handlers at the load calculated above. For VAV systems, supply air temperature should be reset upwards; reset is a requirement for most VAV systems (see Supply Air Reset below) but is usually desirable to enhance economizer performance in any case.

Coil Air Flow Rates. Determine the coil air flow rates using the reset supply air temperature.

Chilled Water Supply Temperature. Using manufacturer's coil selection charts or programs, determine the highest chilled water supply temperature that will meet these supply air conditions assuming design water flow rates.

Chilled Water Return Temperature. The coil selection chart or program will also determine the chilled water return temperature. If there are many cooling coils, either assume conservatively that all coils will operate as required by the "worst case" coil (the one requiring the lowest chilled water temperature), or redetermine the water flow rate required and leaving chilled water temperature of all other coils assuming the chilled water supply temperature of the "worst case" coil and determine the actual return water temperature based on the GPM weighted average of each coil's return water temperature.

Condenser Water Supply and Return Temperatures. Have the heat exchanger manufacturer determine the required cooling tower supply and return water temperatures using the following information: the chilled water supply and return temperature; the chilled water flow rate; and the design tower water flow rate.

Cooling Tower Performance. Verify that the cooling tower can meet the tower water flow rate and supply and return water temperatures determined above at a wet-bulb temperature of 45°F. Do this either by using manufacturer's catalogue data (if available at low wet-bulb temperatures) or by having the manufacturer check performance using factory data.

If the tower can meet these conditions, then the water economizer design complies with the 90.1 Code. If not, change the tower, heat exchanger, cooling coil or air-side designs and repeat the process.

PIPING INSULATION

Form
403.2.9.1

	Single Zone	Multiple Zone
Unitary	Required for split systems.	
Hydronic	Required	

All piping associated with HVAC systems must be thermally insulated in accordance with Table 403.2.9.1 of the 90.1 Code.

The values in Table 403.2.9.1 are minimum thicknesses of insulation having a conductivity falling in the range listed, when tested at the mean rating temperature listed, for each fluid design temperature range category. These conductivities are typical of fiberglass and most elastomeric foam insulations.

If a less common insulation product is to be used, such as cellular glass or calcium silicate, then the thicknesses listed in Table 403.2.9.1 must be adjusted by the following equation:

where

T = the minimum insulation thickness, in inches, for alternative material with a conductivity K

t = the insulation thickness, in inches, from Table 403.2.9.1

PR = the actual pipe outside radius, inches. This is generally not equal to half of the nominal pipe diameter; except for piping 14 in. and larger, actual outside diameter (OD) will be larger than the nominal diameter and depends on the piping material selected. Actual ODs can be found in standard piping tables. An abridged version for copper and steel is shown in Table 403C.

K = the conductivity of alternative material, in Btu-in/(h-ft^2-°F), when measured at the mean temperature indicated in Table 403.2.9.1 for the applicable fluid design temperature range

k = the lower value of the conductivity range listed in Table 403.2.9.1 for the applicable fluid design temperature range

Form
403.2.9.1
Exception
#1-#3

Insulation is not regulated in the following cases:

(1) Piping that is factory installed within equipment that is tested and rated in accordance with Section 403.1. The intent here is to exempt piping within equipment whose energy performance is tested, and piping losses are ostensibly accounted for in the ratings.

(2) Piping conveying fluids that have design operating temperatures between 55°F and 105°F, such as typical condenser water piping

(3) Piping that conveys fluids that have not been heated or cooled through the use of fossil fuels or electricity. This exception is intended to cover gas piping, cold domestic water piping, waste and vent piping, rain water piping, etc. which may carry fluids with operating temperatures outside the 55°F to 105°F range but for which no energy was consumed to bring the fluids to these temperatures. While insulating such piping will have no energy impact and is not required, it may be

desirable in some cases, and possibly required by building codes, to prevent condensation.

Insulation is also not required where heat gain or heat loss to or from the piping will not increase building energy costs. Examples of piping falling into this exception are condensate drains, liquid and hot gas refrigerant lines on AC units, and liquid lines on heat pumps.

Table 403C Copper and Steel Pipe Sizes

	Copper (all wall thicknesses)		Steel (all wall thicknesses)	
Nominal Pipe Size (in.)	**Actual Diameter (in.)**	**Actual Radius, PR (in.)**	**Actual Diameter (in.)**	**Actual Radius, PR, (in.)**
1/4	0.375	0.188	0.54	0.270
3/8	0.500	0.250		
1/2	0.625	0.313	0.84	0.420
5/8	0.750	0.375		
3/4	0.875	0.438	1.05	0.525
1	1.125	0.563	1.32	0.658
1-1/4	1.375	0.6875	1.66	0.830
1-1/2	1.625	0.813	1.90	0.950
2	2.125	1.063	2.375	1.188
2-1/2	2.625	1.313	2.875	1.438
3	3.125	1.563	3.50	1.750
4	4.125	2.063	4.50	2.250
5			5.56	2.782
6			6.625	3.313
8			8.625	4.313
10			10.75	5.375
12			12.75	6.375

Example 403HH Insulation – Chilled Water Return Piping

Q A chilled water system is designed for a chilled water supply temperature of 44°F with a 14°F range. Is insulation required on the return piping?

A No. Chilled water return temperature will be 58°F at design conditions, so this piping would fall under exception (2) and no insulation is required by the 90.1 Code. However, return water temperatures will often be lower at part load, and will often be lower than ambient dew point temperatures as well. Therefore, while the 90.1 Code does not require insulation, it usually requires minimal insulation from a practical standpoint to prevent condensation, and it may be cost effective since it reduces chiller load.

Example 403II Piping Insulation – Fan-Coils

Q A hotel is to have vertical fan-coils serving guest rooms. Chilled and hot water piping risers are to be installed within the fan-coils at the factory. Because of the space required, the standard factory insulation is only 1/2 in. regardless of pipe size. Does this piping fall under exception (1) and therefore not have to meet Table 403.2.9.1 requirements?

A No. The first exception only applies to equipment whose energy performance is rated in accordance with Section 403.1. Fan-coil performance may be rated as far as capacity is concerned, but they are not regulated by any energy performance criterion. Piping within the fan-coils, therefore, must be insulated in accordance with Table 403.2.9.1.

Example 403JJ Piping Insulation – Condenser Water System with Water-Side Economizer

Q A high-rise office building has a water-side economizer. Under cooling conditions, the condenser water operates in the range of 65°F to 95°F depending on outside conditions and cooling load. But during the winter, the cooling tower is controlled to cool water evaporatively down to 45°F. Does this piping require insulation?

A The 90.1 Code regulates piping insulation based on fluid "design" operating conditions, which refers to the fluid state at peak cooling or peak heating design conditions. In this case, it could be argued that the condenser water is only of concern in the cooling mode since at design heating conditions, which will occur during building morning warm-up, the water economizer will be inactive. Therefore, the condenser water piping would not have to be insulated. (In this case, insulation would have very little if any energy impact anyway. However, insulation may be desirable in some locations to prevent condensation.)

Example 403KK Calculation of Pipe Insulation Thickness – Cellular Glass

Q Cellular glass piping insulation is proposed for 10 in. chilled water lines running outdoors. (This insulation material is often preferred for outdoor installations since it is very durable and will not absorb water like fiberglass, which effectively destroys its insulating properties. There is, then, less concern about the quality of insulation weather- and vapor-proofing.) The design chilled water supply temperature is 44°F to 54°F. What thicknesses are required?

A From the manufacturer's catalogue, cellular glass has a conductivity of 0.33 Btu-in/(h-ft^2-°F) at 75°F mean temperature. The minimum insulation thickness is:

where

PR = 5.375 (from Table 403C above for steel pipe)
t = 1.0 in. (from Table 403.2.9.1)
K = 0.33 (from manufacturer's catalogue)
k = 0.23 (from Table 403.2.9.1)

The next largest 90.1 Code size is 1-1/2 in. insulation which is the thickness specified in this application.

DUCT AND PLENUM INSULATION

Form
403.2.9.2

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Supply and return ductwork and air plenums must be thermally insulated in accordance with Table 403.2.9.2.

Insulation requirements are broken into two columns, one for ducts used to convey air for space heating and one for cooling ducts, including both supplies and returns. Ducts that are part of both a cooling system and a heating system at different times of the year (such as ducts connected to a packaged rooftop unit that has both heating and cooling capability) must meet the more stringent requirement of both columns.

Requirements are further divided into two primary categories, one for ducts that are outdoors exposed to weather, and one for ducts that are either in unconditioned spaces (enclosed but not within the building envelope) or in conditioned spaces (within the building envelope). This second category is extended (by footnote f) to Table 403.2.9.2) to include semi-enclosed spaces such as vented attics and crawlspaces.

Insulation requirements for exposed ducts depend on the local weather as determined by cooling degree days (base 65°F) for cooling ducts and heating degree days (base 65°F) for heating ducts. Local weather data can be found in Section 301.1 of the code.

The insulation requirements for enclosed ducts depend on a value of "TD,"[2] defined as the temperature difference between the air conveyed in the duct and the ambient air surrounding the duct, both at design conditions. In many cases, determining these temperatures will require some judgment on the part of the designer, particularly for ambient temperatures of unconditioned spaces. A rough estimate of supply air, return air, and unconditioned space temperatures will usually suffice since each requirement is for a broad range of temperature differences.

The R-values listed in Table 403.2.9.2 are for the insulation material as installed, excluding air film resistances, but including the effect of the compression of duct wraps as they are typically installed. Duct wraps are generally assumed to be compressed to 75% of their nominal thickness when they are installed. Material conductivity is as measured in accordance with ASTM C518-85 at 75°F mean temperature.

Common duct insulation materials that meet the R-value levels in Table 403.2.9.2 are shown in Table 403D.

The following applications need not meet the requirements of the Table 403.2.9.2:

Form
403.2.9.2
Exception
#1-#2

(1) Plenums and casings that are factory installed and an integral part of equipment rated in accordance with Section 403.1. Like factory installed piping insulation, this exception is intended to apply to casings around tested equipment, the energy losses through which are accounted for in the energy performance tests and ratings. Although they are not always included in performance testing, optional casings such as economizer sections should also be exempted from insulation requirements if they are insulated to the same extent as the equipment to which they are attached. This is a practical requirement since designers seldom have the option to specify casing insulation levels.

(2) Ducts within the conditioned space they serve.

2 See footnote "d" for Table 403.2.9.2 Minimum Duct Insulation in the 90.1 Code.

Simplified Duct Insulation Table

Appendix D of the 90.1 Code provides an alternative simplified duct insulation table which is based on typical design supply air temperatures and typical ambient temperatures that vary as a function of outside weather and duct location. The table obviates the need to determine these temperatures, greatly simplifying compliance for both the user and code official. An example of a simplified table for Oakland, CA is shown in Table 403E.

Specific Application

Figure 403K identifies the primary duct configurations that have requirements from Table 403.2.9.2. The insulation requirements for each duct location identified in Figure 403K are described in the following, numbered as shown in the figure. In each section, the default insulation requirement is identified. This is the requirement that would result if the simplified procedure outlined in Appendix D to the 90.1 Code were followed:

Unit Casings and Plenums

Factory installed insulation that is part of the HVAC equipment is covered through the minimum equipment efficiencies of Section 403.1 and does not have requirements in this section (exception (1)). This includes equipment casings and plenums. This exception does not apply to built-up systems with field-fabricated plenums. These systems must be insulated per the requirements for ductwork in this section. Perimeter mechanical rooms that are used as mixing plenums must have exterior walls insulated to meet the requirements for return ductwork on the exterior of the building or the requirements of Section 402, whichever are more stringent.

Exhaust Ductwork

Exhaust ductwork also need not meet the requirements of this section since the section only applies to supply and return ductwork.

Supply and Return Ducts in Vented Attic

These ducts are located in an attic which is vented to the outside. Footnote f to Table 403.2.9.2 specifies that crawl spaces and attics, vented or otherwise, are to be considered unconditioned spaces for the purpose of determining duct insulation requirements. A vented attic will generally run 5°F to 20°F warmer than the outside air temperature during peak cooling, and be close to the outside air temperature during peak heating.

Default Supply Insulation. In most cases, supply ducts must be insulated to a minimum of R-5.0. They may be insulated to R-3.3 where they serve a cooling-only system and the design cooling outdoor dry-bulb temperature is less than 85°F.

Default Return Insulation. The required insulation for return ducts varies as follows: no insulation is required for cooling-only systems with design cooling dry-bulb temperatures less than 82°F; R-3.3 is required for heating systems with a design heating dry-bulb temperature greater than 33°F or cooling systems with a design temperature between 82°F and 107°F; R-5.0 is required for heating systems where the design temperature is less than 33°F or cooling systems with the design temperature greater than 107°F.

Supply and Return Ducts on Exterior of the Building

Exposed ductwork insulation requirements are determined by local weather conditions. In all cases the minimum insulation R-value is determined by the cooling degree days base 65°F (CDD65) for ducts serving cooling systems (supply and return) and the heating degree days base 65°F (HDD65) for ducts serving heating systems (supply and return). These weather parameters can be found in Section 301.1 of the 90.1 Code.

Table 403D R-values for Common Duct Insulation Materials

R-Value (hr-°F-ft^2)/Btu	Nominal Thickness, (in.)	Typical materials
3.3	1-1/2	1/2 to 1-1/2 lb./ft^3 fiberglass duct wrap
	1	3/4 to 3 lb./ft^3 fiberglass duct liner
	1-1/2	1/2 lb./ft^3 fiberglass duct liner
	1	fibrous glass duct board
	1	insulated flexible duct
5.0	3	1/2 lb./ft^3 fiberglass duct wrap
	2	3/4 to 1-1/2 lb./ft^3 fiberglass duct wrap
	1-1/2	3/4 to 2 lb./ft^3 fiberglass duct liner
	1	3 lb./ft^3 fiberglass duct liner
6.5	3	1/2 to 1-1/2 lb./ft^3 fiberglass duct wrap
	2	3/4 to 1-1/2 lb./ft^3 fiberglass duct liner
	1-1/2	2 to 3 lb./ft^3 fiberglass duct liner
8.0	4	1/2 to 3/4 lb./ft^3 fiberglass duct wrap
	3	1 to 1-1/2 lb./ft^3 fiberglass duct wrap
	3	3/4 to 1 lb./ft^3 fiberglass duct liner
	2	1-1/2 to 3 lb./ft^3 fiberglass duct liner

Table 403E Duct Insulation R-value (Oakland, CA)[a,b]

Duct Location	Duct Type		
	Cooling Supply	Heating Supply	All Return Ducts
Exterior of Building	R-3.3	R-5.0	R-5.0
Ventilated Attic	R-3.3	R-5.0	R-3.3
Unvented Attic	R-5.0	R-5.0	R-3.3
Other Unconditioned Spaces[c]	R-3.3	R-5.0	R-3.3
Indirectly Conditioned Spaces[d]	R-3.3	R-3.3	none
Buried	none	R-5.0	R-3.3

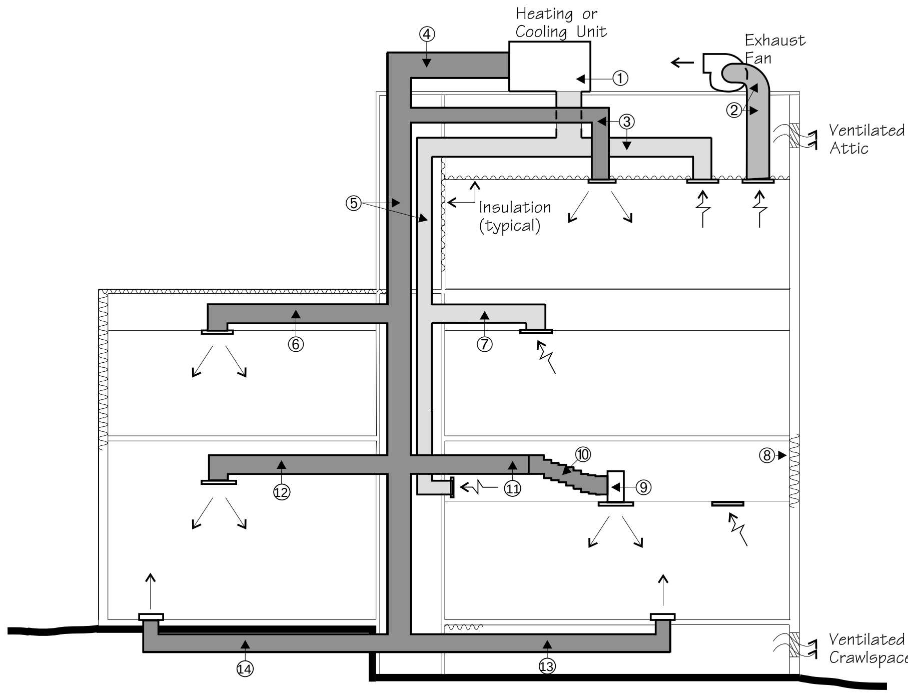
a. Insulation R-values are for the insulation as installed and do not include film resistance. The required minimum thicknesses do not consider water vapor transmission and condensation. For ducts which are designed to convey both heated and cooled air, duct insulation shall be as required by the most restrictive condition. Where exterior walls are used as plenum walls, wall insulation shall be as required by the most restrictive condition of this section or Section 402. Insulation resistance measured on a horizontal plane in accordance with RS-8 at a mean temperature of 75°F at the installed thickness.

b. Insulation resistance for runouts to terminal devices less than 10 ft in length need not exceed 3.3 (h.ft^2·°F)/Btu.

c. Includes crawlspaces, both ventilated and non-ventilated

d. Includes return air plenums, with and without exposed roofs above

Figure 403K Duct Insulation



KEY

1. Insulation of unit casing
2. Exhaust
3. Supply & return in vented attic
4. Supply & return on exterior of building
5. Supply & return in shaft
6. Supply & return in unvented attic
7. Return in indirectly conditioned ceiling space
8. Exterior wall of return plenum
9. Supply outlet in plenum
10. Supply runout in return plenum
11. Supply in return plenum
12. Supply & return in conditioned space
13. Supply & return in vented crawlspace
14. Buried supply

5. Supply and Return Ducts in Exposed Shaft

These ducts are located in unconditioned space similar to case 3 above. However, this wall cavity will not have the extreme temperatures of the attic since the space is not vented to the outdoors and has a minimal roof load. Where design temperatures for the shaft space are not available from load calculations, one can assume that at design conditions, the shaft temperature is near outside conditions at both cooling and heating peaks.

Default Supply Insulation. In most cases, supply ducts must be insulated to a minimum of R-5.0. They may be insulated to R-3.3 where they serve a cooling-only system and the design cooling dry-bulb temperature is less than 95°F.

Default Return Insulation. The required insulation for return ducts varies as follows: no insulation is required for cooling-only systems with design cooling dry-bulb temperatures less than 92°F; R-3.3 is required for heating systems with a design heating dry-bulb temperature greater than 33°F or cooling systems with a design temperature between 92°F and 117°F; R-5.0 is required with heating systems where the design temperature is less than 33°F or cooling systems with the design temperature greater than 117°F.

Supply (and Return) Ducts in Unvented Attic

Here the ducts are located within an unventilated attic. The temperature within the attic is often determined in load calculations. If not, it must be estimated. The attic temperature will be higher than space temperatures due to solar gains on the roof and heat from recessed light fixtures. An unvented attic will generally run 15°F to 30°F warmer than the outside air temperature during peak cooling, and be close to the outside air temperature during peak heating.

Default Supply Insulation. In most cases, supply ducts must be insulated to a minimum of R-5.0. They may be insulated to R-3.3 where they serve a cooling-only system and the design cooling dry-bulb temperature is less than 80°F.

Default Return Insulation. The required insulation for return ducts varies as follows: no insulation is required for cooling-only systems with design cooling dry-bulb temperatures less than 77°F; R-3.3 is required for heating systems with a design heating dry-bulb temperature greater than 33°F or cooling systems with a design temperature between 77°F and 102°F; R-5.0 is required with heating systems where the design temperature is less than 33°F or cooling systems with the design temperature greater than 102°F.

Return Ducts in Indirectly Conditioned Ceiling Space

This case is similar to case 6 but the ceiling space temperatures will be essentially equal to space temperatures for heating and slightly higher than space temperatures for cooling due to recessed light fixture heat gains. In general no insulation will be required on return ducts. See case 11 below for supply ducts in indirectly conditioned spaces.

Exterior Wall of Return Plenum

In this case the ceiling space is being used as a return plenum. The exterior walls of the space are effectively return duct walls exposed to the outside. This wall must be insulated to the more restrictive requirement of either Section 5 of SMACNA[3] HVAC Duct Leakage Test Manual, 1985, or the requirements for ducts located in the exterior of the building from Table 403.2.9.2 (403F). These latter requirements are discussed in case 4 above.

Supply Outlet in Return Plenum

The sheet metal plenum surrounding the air outlet is part of the supply duct system and therefore must be insulated the same as the supply ducts (case 11 below).

Default Supply Insulation. From Table 403H, supply ducts in indirectly conditioned spaces such as ceiling plenums must be insulated to a minimum of R-3.3. In practice, outlet plenums are generally internally lined with 1/2 in. insulation (R-2). To meet the R-3.3 requirement, 1 in. liner would be required, but plenums are seldom sized large enough to fit the thicker lining and it is simply not available from most manufacturers. Technically the 1/2 in. liner is not in compliance, but enforcement bodies are encouraged to waive the requirements of this additional insulation for practical considerations.

Supply Runout in Return Plenum

Per footnote f to Table 403.2.9.2 (403E), a runout of up to 10 ft to a terminal device (supply outlet or VAV box) need only be insulated to R-3.3. This is intended to allow 90.1 Code flexible duct with 1 in. insulation (about R-4.0) to be used. Flexible duct with 2 in. insulation is not commonly available. This exception holds even if the supply ducts are required to have R-5.0 insulation.

Supply Ducts in Return Plenum

For supply ducts located in a return plenum, the ΔT is the difference between the supply and return temperatures. In general the temperature in a return plenum will be 1°F to 2°F warmer than the space temperature due to heat from the lights.

Default Supply Insulation. From Table 403?, supply ducts in indirectly conditioned spaces such as ceiling plenums must be insulated to a minimum of R-3.3 for both heating and cooling systems.

Supply and Return Ducts in Conditioned Space

Per exception b to Table 403.2.9.2 (403E), supply and return ducts located in the conditioned space do not require insulation. From a practical standpoint, insulation may be desirable on cooling ducts to prevent condensation if the duct passes near local areas of high humidity as might occur in a kitchen. For typical spaces, condensation will generally not occur even at very low supply temperatures since the space relative humidity will be lowered correspondingly by the dry air supply.

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3 Sheet Metal and Air Conditioning Contractors National Association, Inc. (as referenced by RS-35 from Chapter 5 of the 90.1 Code).

Supply and Return Ducts in Vented Crawlspace

This is similar to the case of ducts in a vented attic (case 3 above). Because the space is vented and unlikely to experience strong solar loads, crawlspace temperatures will be very close to the outdoor temperature.

Default Supply Insulation. In most cases, supply ducts must be insulated to a minimum of R-5.0. They may be insulated to R-3.3 where they serve a cooling-only system and the design cooling dry-bulb temperature is less than 95°F.

Default Return Insulation. The required insulation for return ducts varies as follows: no insulation is required for cooling-only systems with design cooling dry-bulb temperatures less than 92°F; R-3.3 is required for heating systems with a design heating dry-bulb temperature greater than 33°F or cooling systems with a design temperature between 92°F and 117°F; R-5.0 is required with heating systems where the design temperature is less than 33°F or cooling systems with the design temperature greater than 117°F.

Supply and Return Ducts Below Grade

The amount of insulation required for ducts located below grade depends on the ground temperature. Data on design ground temperatures can be found in the ASHRAE 1987 HVAC Handbook, Table 3, Chapter 12. Although the ground does not meet the requirement of an unconditioned space and is technically located outside of the building envelope, the ductwork is not exposed to the same solar and temperature loads of exposed ductwork. Practically speaking this situation is much closer to ductwork in an unconditioned space than ductwork on the roof.

Default Supply Insulation. From Table 403.2.9.2 of the code, the required insulation for supply ducts varies as follows: no insulation is required for cooling-only systems with design ground temperatures less than 70°F; R-3.3 is required for cooling ducts with design ground temperatures over 70°F or heating ducts with the design ground temperatures over 75°F; R-5.0 is required for heating ducts with design ground temperatures under 75°F.

Default Return Insulation. Insulation is only required on heating return ducts where the design ground temperature is less than 58°F. In these cases, R-3.3 is required.

DUCT AND PLENUM CONSTRUCTION

Form
403.2.9.3

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Ducts and plenums must be constructed in accordance with SMACNA HVAC Duct Construction Standards - Metal and Flexible, 1985, and SMACNA Fibrous Glass Duct Construction Standards, 1979[4], or equivalent.

In addition to the requirements of these SMACNA Standards, two more stringent measures are required for compliance with the 90.1 Code:

Sealing Requirements for Low Pressure Supply Ductwork

Supply ductwork designed to operate at static pressures from 1/4 inches of water gage (in. w.g., also called inches of water column, in. wc) to 2-in. w.g. inclusive that are located outside the conditioned space or in return air plenums must be sealed in accordance with Seal Class C as defined in the SMACNA manuals referenced above (either the HVAC Fibrous Glass Duct Construction Standard (RS-36) or the HVAC Duct Leakage Test Manual (RS-35) and reprinted in Table 403F. SMACNA manuals only require Seal Class C for ducts designed for 2 in. static pressure, plus ducts designed for 1 in. or 1/2 in. pressure classes if they are part of a VAV system. The 90.1 Code extends this requirement to ducts designed for pressures as low as 1/4 in. In other words, virtually all HVAC supply ducts located outside the conditioned space or in return plenums must be sealed to Seal Class C.

Longitudinal seams are those parallel to the direction of air flow; spiral joints do not require sealing in any seal class. All other duct connections are considered transverse joints.

SMACNA does not limit the type of material used for sealing, except that exterior sealants must be specifically marketed for that application. But the 90.1 Code does limit the use of pressure sensitive tape, which generally tends to peel with time and is therefore not a reliable sealant in the long run. Such tape may not be used as the primary sealant for ductwork designed to operate at static pressures 1 in. w.g. and higher.

Table 403F SMACNA Sealing Classes and Requirements

Seal Class	Sealing Requirements	Required for SMACNA Static Pressure Construction Class
A	All transverse joints, longitudinal seams, and at penetrations of the duct wall	4 in. w.g. and higher
B	All transverse joints and longitudinal seams	3 in. w.g.
C	Transverse joints only	2 in. w.g. and VAV systems down to 1/2 in. pressure class

4 As referenced by RS-34 and RS-36, respectively, from Chapter 5 of the 90.1 Code.

Ductwork Operating in Excess of 3 in w.g.

Ductwork designed to operate at static pressures in excess of 3-in. w.g. must meet leakage limitations as follows:

- Ducts must be tested in accordance with the procedures outlined in Section 5 of the "SMACNA HVAC Duct Leakage Test Manual," 1985 (RS-35), with tests reported using forms equivalent to those outlined in Section 6 of that manual.
- To reduce costs, the entire duct system need not be tested. Tests may be made for only representative sections provided these sections represent at least 25% of the total installed duct area for the tested pressure class.
- Tested duct leakage at a test pressure equal to the design duct pressure must meet Leakage Class 6. Leakage class, as defined in the SMACNA Test Manual referenced above, is:

where
CL = the leakage class
F = the measured leakage rate in cfm/100 ft^2 of duct surface
P = the static pressure used in the test

The leakage class according to this formula must be less than six when the test pressure is equal to the design duct pressure class.

Example 403LL Duct Insulation Requirements

Q If the HVAC unit in Figure 403K were located in a building in Atlanta, Georgia, what would the default duct insulation values be assuming the simplified method of Appendix D to the 90.1 Code were used? Assume that the HVAC unit is a packaged rooftop heating and cooling system.

A The default insulation values for each of the duct conditions identified in Figure 403K are summarized in the table below. To determine the insulation requirements, the following design data and the default Table D-1 in the 90.1 Code (Appendix D) are required:

- Tosa-db (cooling) = 92°F (ASHRAE 1989 Fundamentals Handbook, Table 1, Chapter 24, 2.5% value)
- Tosa-db (heating) = 17°F (ASHRAE 1989 Fundamentals Handbook, Table 1, Chapter 24, 2.5% value)
- HDD65 = 3,070 (Appendix A of the 90.1 Code)
- CDD65 = 1,566 (Appendix A of the 90.1 Code)
- Tground (summer) = 70°F (ASHRAE 1987 HVAC Handbook, Table 3, Chapter 12)
- Tground (winter) = 54°F (ASHRAE 1987 HVAC Handbook, Table 3, Chapter 12)

Location	Supply	Return	Comment
1. HVAC unit casing	NA	NA	None, per exception (1)
2. Exhaust	NA	NA	None
3. Supply and return in vented attic	R-5.0	R-5.0	Heating requirement is more restrictive
4. Supply and return on roof	R-6.5	R-6.5	Cooling requirement is more restrictive
5. Supply and return in exposed shaft	R-5.0	R-5.0	Heating requirement is more restrictive
6. Supply and return in unvented attic	R-5.0	R-5.0	Heating requirement is more restrictive
7. Return in indirectly conditioned ceiling space	NA	none	
8. Exterior wall at return plenum	R-6.5	R-6.5	See Section 402 for possibly more restrictive requirement
9. Supply outlet in return plenum	R-3.3	NA	
10. Supply runout in return plenum	R-3.3	NA	See footnote e, Table 403.2.9.2
11. Supply duct in return plenum	R-3.3	NA	
12. Supply and return in conditioned space	none	none	Exception (2)
13. Supply and return in vented crawl space	R-5.0	R-5.0	Heating requirement is more restrictive
14. Buried supply and return	R-5.0	R-3.3	Heating requirement is more restrictive

Example 403MM Insulation Required for Pipes and Ducts in Conditioned Spaces

Q A VAV system supplies cooling to exterior zones and interior zones in a building that has no ceilings. The supply ducts run exposed through the interior spaces. Exterior zone VAV boxes with hot water reheat are located over interior corridors then ducted out to exterior rooms. Do hot water pipes and ducts require insulation?

A Yes and no. The VAV duct mains supply cooled air all year round and are located over interior spaces that also require cooling all year round. Heat gains or losses from the ductwork will therefore not affect energy usage and thus fall under exception (2). Zone ducts also need not be insulated since they are located in the spaces served. Hot water piping will require insulation in all zones.

Example 403NN Leakage Testing of Ducts – 3 in. w.g.

Q A duct system is designed to operate at a maximum operating pressure of 3 in. w.g., but to reduce radiated noise levels, the engineer has specified that ducts be constructed to the SMACNA requirements for 6 in. operating pressure. What are the testing requirements for this ductwork?

A The 90.1 Code only requires testing based on the actual design operating pressure, not the pressure that the duct might actually be able to withstand. In this case, no testing is required since the design static pressure does not exceed 3 in..

Example 403OO Leakage Testing of Ducts – 4 in. w.g.

Q If the previous example were changed so that design operating pressures were 4 in. instead of 3 in., at what pressure would the leakage tests be conducted, 4 in. or 6 in.?

A The ductwork would be tested at 4 in. since this is the actual design operating pressure.

Example 403PP Duct Sealing Requirements – Flex Connections at VAV Box

Q VAV boxes are connected to the medium pressure duct main (3 in. pressure class) with flexible duct. What are the sealing requirements at the flexible duct connections to the main and to the VAV box?

A Since the operating pressure is higher than 1 in. w.g., pressure sensitive tape may not be used alone to secure either end of the flexible duct. But tape may be used as a secondary sealant, with a draw-band or panduit strap used to provide the primary seal.

PROJECT COMPLETION (403.2.10)

HVAC systems must be tested and balanced to assure proper operation. Control systems must be tested, calibrated, and adjusted to proper operating setpoints. ASHRAE's Guideline 1-1989 Commissioning of HVAC Systems (Code 86801) is recommended, but not required, for use in developing a comprehensive commissioning plan.

Air and hydronic system balancing should be in accordance with the procedures published in ASHRAE Standard 111-1988 or those by the National Environmental Balancing Bureau (NEBB) or the Associated Air Balance Council (AABC).

Form
403.2.10.1

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Operating and Maintenance Manuals

An operating and maintenance (O&M) manual that includes complete operating instructions and routine maintenance requirements must be provided to the building owner upon completion of construction.

The wording of this section is necessarily general because of the wide range of complexity of HVAC systems, from window air conditioners at one end to large central plants and air handling systems built up from several different types of equipment on the other. The intent of this section is to provide to the owner all information necessary to properly operate the system as a whole, not just each piece of equipment. If the system is very complex, requiring coordination among several pieces of equipment, the proper operating instructions, setpoints, etc. must be specified and included in O&M documents. This is usually done in control system design documents, such as system schematics and control sequence descriptions. For simple systems, such as small individual unitary equipment, a detailed system description and control schematic is not necessary to operate the system properly and thus need not be included in the O&M manual.

Form
403.2.10.2
Exception
#1-#2

	Single Zone	Multiple Zone
Unitary	Required	Required
Hydronic	Required	Required

Air System Balancing

Air systems must be balanced by first adjusting fan speed to meet design flow. Damper throttling alone may be used for balancing fans only under two conditions:

(1) With motors 1 hp or smaller.

(2) Where throttling will increase fan power by no more than 1/3 hp above that required if the fan speed were adjusted. For instance, if the fan would require 5 hp if throttling were used to balance the system, but the energy demand would drop to 4.5 hp if operating at the correct speed for the required pressure drop, the fan speed would have to be adjusted since the difference exceeds 1/3 hp. The required horsepower can be estimated using the fan laws based on the actual system pressure drop and air flow at the original fan speed, measured at the initial system balance.

Hydronic System Balancing

Form
403.2.10.3
Exception
#1-#3

	Single Zone	Multiple Zone
Unitary	**Not required**	**Not required**
Hydronic	Required	Required

Hydronic systems must be balanced by adjusting pump speed or by trimming the pump impeller to meet design flow requirements. Valve throttling alone may be used for final balancing only in the following cases:

(1) Pumps with motors 10 hp or less.

(2) If valve throttling results in an increase in pump power of no more than 3 hp above that required if the impeller were trimmed. (See discussion under exception (2) above for air system balancing).

(3) To reserve pump capacity to overcome future fouling in open circuit piping systems (cooling tower systems). Throttling losses may not exceed the pressure drop expected for future fouling.

Pump balancing requirements are less stringent than those for fans. The cost to adjust fan speed (which requires adjusting or changing a motor sheave) is relatively inexpensive while adjusting pump motor speed is generally impractical; removing the pump impeller, trimming it as required, then replacing it, is an expensive procedure cost justified only if energy savings are substantial.

Control System Testing

Form
403.2.10.4

	Single Zone	**Multiple Zone**
Unitary	Required	Required
Hydronic	Required	Required

The standard requires that HVAC control systems be carefully calebraded and tested as part of the building commissioning process.

Example 403QQ Balancing Requirements – VAV Fan with VSD

Q

A fan serving a VAV system has a variable speed drive for static pressure control. During balancing its full-load fan speed is found to be 20% faster than required. Do fan sheaves need to be adjusted or changed?

A

No. The variable speed drive will automatically reduce the fan speed as required to meet system static pressure requirements. In general, fan and pumping systems with variable speed drives are self-balancing provided their pressure setpoints are correct.

Compliance and Enforcement

SUMMARY FORM AND WORKSHEET

One summary form and one worksheet are provided to assist with the calculations and documentation necessary in showing compliance with the mechanical systems and equipment requirements of the 90.1 Code. Blank copies of these forms are included in Appendix D of this manual. The Case Study section of this chapter provides examples of completed forms. A general description of the forms is provided below.

Summary Form and Checklist

The front page of this form contains basic project information; the back(s) lists information to be put on drawings submitted for a mechanical permit. The applicant completes the top and left hand side of the front page(s) and refers to the appropriate plan page on the drawings for each item on the checklist or, for the Summary, by circling a response indicating that a particular exception is applicable to this project. Note that there are two versions of the Summary form and Checklist: a complete version, and a short form for small simple systems. Make sure that the short form is only used when appropriate.

For systems with total fan system motor horsepower of greater than 10, it is likely that the additional worksheet will need to be submitted. Unless included on the plans, the Fan System Worksheet will need to be submitted to demonstrate the overall W/cfm for each system. The plans examiner then compares this information with the drawings, requests corrections from the applicant, notes any features which merit special attention by the inspector, and then forwards it to the inspector. The inspector verifies all categories where there is an inspection check box, unless the applicant has written in “NA” and the plans examiner has concurred. After the inspections, the final version of the form can be filed with the building permit drawings as a record of construction.

Fan System Worksheet

This worksheet is used to demonstrate that the fan system W/cfm (both for variable air volume and for constant volume systems) meets the requirements of Section 403.2.4. (The shaded areas of the form indicate information that is to be taken from the plans.) As indicated in the instructions on the worksheet, this calculation must be done separately and compliance demonstrated individually for each fan system. Extra lines for additional systems are provided on the back.

PERMIT APPLICANT'S RESPONSIBILITIES

At permit application, the goal of the applicant is to provide all the necessary information to show compliance with the 90.1 Code. If the plans examiner is able to verify compliance with one review, then the permit can be issued and construction started without delay. To assist in submitting the permit application, the applicant should review not only the following information specific to the applicant but also the subsequent two sections that review responsibilities of the plans examiner and the inspector. The following section addresses the two common problems with permit applications: (1) missing information, or (2) incorrect information.

Information may be missing because the applicant is not aware of all of the code requirements or because the required information is located on the specifications but not on the plans. Note that building departments generally approve plans, but not specifications. The Checklist on the back of the Mechanical System and Equipment Summary Form provides a detailed list of the type of information that needs to be on the plans. This information can then be provided in a number of ways:

- *On the drawings.* Provide HVAC layout with equipment location, air distribution ductwork and sizes, air intake and exhaust locations, piping layout, fan and pump type and location, control diagrams indicating type of HVAC control and the units that it controls.
- I*n schedules.* For instance, list heating and cooling equipment capacity and efficiency, fan horsepower and cfm, outside air cfm, duct insulation R-values, pipe insulation thickness.
- *Through notes and call outs.* Note that building owner to be given operation and maintenance literature, that control systems to be tested to assure that elements are calibrated and in good working order.
- *Through supplementary worksheets or calculations.* Provide calculations such as for fan system w/cfm. You may include these calculations on the drawings or incorporate as additional columns in the schedule or submit completed worksheets provided with this manual.

Incorrect information may be due to a lack of understanding of the code. More likely, it indicates that the code has changed since the last project. The applicant can use a correction list as a reminder to update the office specifications to avoid receiving this same correction again in the future. Some features to note are:

- Separate systems generally required for uses with different operating hours (i.e. office vs. retail), and for zones having special process temperature and/or humidity requirements.
- Constant volume reheat systems not allowed except for some areas of hospitals and laboratories.
- Equipment efficiency is specified at both peak conditions and at part load for many units.
- Economizers typically required for HVAC equipment of 3,000 cfm and larger and 90,000 Btu/h cooling capacity and larger. Economizer to be capable of partial cooling even when additional mechanical cooling is required. VAV fan motors 75 hp and larger will need to have variable frequency drive or equivalent.
- Pumping systems with more than 10 hp will need to have variable speed drive or equivalent.
- Automatic setback controls required.
- Ductwork designed to operate at static pressures in excess of 3 in. wc to have leak testing for at least 25% of the system ductwork. Air and water system balancing required.

PLANS EXAMINER'S RESPONSIBILITIES

The plans examiner must review each permit application for 90.1 Code compliance before a permit is issued. By letting the designer and contractor know what’s expected of them early in the process, the building department can help assure that the approved drawings comply with the code. This helps the inspector to avoid the headache of correcting a contractor who is following drawings that do not meet the code requirements.

The biggest challenge for the plans examiner is often determining where the necessary information is and whether the drawings are complete. The plans examiner should make sure that the applicant includes the Mechanical Summary and Checklist forms in this manual as part of the submittal package. The information provided on these forms makes the job easier and reduces plan review time.

A complete building mechanical systems and equipment plan review covers all of the requirements in Section 403 (service water heating is included in Section 404). For Section 403, first review the comments for the applicant above for a general sense of key requirements, then:

- Check that there's a heating and cooling equipment schedule with the correct efficiencies for both peak load and for part load (as applicable). Directories from nationally recognized certification programs such as ARI are a good resource for verification. Remember that compliance is required to be demonstrated at the standard rating conditions, not the proposed operating conditions.
- Check that heating and cooling load calculations have been submitted to support the equipment sizes selected.
- Check that the calculations have been done correctly and are based on the design conditions specified in Chapter 3 of the Code. Review the calculated loads and compare them against the peak output of the equipment specified. If the exception for smallest size is claimed, ask for copy of manufacturer's catalog to verify. If the exceptions for standby or multiple equipment are claimed, check that appropriate controls are installed.
- Check that separate systems are installed for zones with special process temperature and/or humidity requirements.
- Check that fan systems over 10 hp are capable of operating at minimum outside air supply levels.
- Check that the W/cfm for fan systems over 10 hp from the Fan System Worksheet complies with the code and matches the drawings. Make sure that all fans required to operate at peak conditions are included. This includes exhaust and relief fans, and any downstream series fans such as those in zone fan powered terminal units. For VAV systems, if brake horsepower is used rather than nameplate horsepower, ask for justification of value (generally, brake horsepower should not be less than 80% of nameplate horsepower).
- Check that VAV fans with motors 75 hp and larger have variable frequency drive or equivalent. Ask to see fan curves from manufacturers specifications.
- Check that pump systems over 10 hp have variable speed drive or equivalent. Ask to see pump curves from manufacturers specifications. If exceptions are claimed, have a note put on the drawing to document.
- Check that each heating and cooling system has a thermostat and that each zone is controlled by a thermostat in the zone. If exception is claimed, check that controls are interlocked to prevent simultaneous heating and cooling.
- Check that thermostats are capable of being set from 55° to 85° F with at least a 5° deadband.

- Check that heat pumps have automatic controls to prevent supplementary electric resistance heater operation when the load can be met by the heat pump.
- Check that humidistats are capable of being set to maintain a humidity from 30-60%.
- Check that there is no simultaneous heating and cooling, reheating, recooling by any HVAC system. If the VAV exception is claimed, check that the air supply to zone boxes can be reduced to the greater of 30% of peak supply volume, the minimum for outside air supply, 0.4 cfm/ft^2 or 300 cfm. If the one of the other exceptions is claimed, ask for supporting documentation and a note on the drawings.
- Check that air systems have controls for automatic supply temperature reset.
- Check that hydronic systems of at least 600,000 Btu/h design capacity have controls for automatic supply temperature reset. If the cannot-be-implemented-without-causing-improper-operation exception is claimed, ask for supporting documentation and a note on the drawings.
- Check that each system is equipped with automatic setback controls.
- Check that outdoor air supply and exhaust systems have automatic volume shutoff or reduction.
- Check that zones with different operating hours (i.e. office vs. retail) are either served by separate systems or have isolation devices to shut off or set back each zone independently.
- Check that each fan system of 3,000 cfm or greater and 90,000 Btu/h or greater has an air or water economizer. If any of the other exceptions are claimed (except for residential spaces), ask for supporting documentation.
- Check that economizer can provide partial cooling even when additional mechanical cooling is required to meet the remainder of the cooling load.
- Check that piping insulation thickness is on the drawings and complies with code.
- Check that duct insulation R-value is on the drawings and complies with code.
- Check that there is a note indicating that ducts are to be constructed and sealed in accordance with the applicable standard.
- Check that there is a note indicating that an operating and maintenance manual will be provided to the owner, that the air and hydronic systems will be balanced, and that the control system will be tested and calibrated.

Remember that good plan review is important. It is much easier to change a number on a drawing than to remove equipment after it has already been installed.

FIELD INSPECTOR'S RESPONSIBILITIES

The inspector's task is to make sure that the project is constructed in accordance with the approved plans. Be aware that a number of requirements will vary from project to project. Consequently, while some requirements may be learned once, others will necessitate on-site checking of the approved plans.

The primary challenge for the inspector may be educating the contractors about any changes in the code requirements so that installations are performed correctly, not simply the way they may have been routinely done in the past.

For this code, some of the most important items are listed below. As a start, review the responsibilities for the applicant and plans examiner in the previous two sub-sections to get a general sense of key requirements.

For the rough-in (okay to cover) inspection:

- Verify heating and cooling equipment efficiency.
- Verify heating and cooling equipment size. If exception claimed, verify that required controls are installed.
- Verify that separate air distribution systems are installed.
- Verify fan hp and cfm, and fan motor efficiency for systems over 10 hp.
- Verify variable frequency drive or other control type as indicated on drawings for VAV fan motors 75 hp and larger.
- Verify variable speed drive or other control type as indicated on drawings for pumping systems over 10 hp.
- Verify that each heating and cooling system has a temperature control device.
- Verify that the heating and cooling supply to each zone is controlled by a thermostat in that zone.
- Verify that heat pumps have controls to prevent electric resistance supplementary heater operation when the load can be met by the heat pump.
- Verify that simultaneous heating and cooling does not exceed that allowed by the drawings. Expect VAV systems in most cases, with constant volume reheat limited to certain sections of hospitals and laboratories.
- Verify that automatic setback controls are installed.
- Verify that outdoor air supply and exhaust systems have motorized or gravity dampers for automatic volume shutoff or reduction.
- Verify that zones with different operating hours (i.e. office vs. retail) are either served by separate systems or have isolation devices to shut off or set back each zone independently as indicated by the plans.
- Verify that each fan system of 3,000 cfm or greater and 90,000 Btu/h or greater has an air or water economizer unless exempted on the plans.
- Verify that ducts are constructed and sealed in the appropriate manner. Note that pressure sensitive tape cannot be used as the primary sealant for ducts designed to operate at static pressures of 1 in. wc or greater.
- Inform contractor of any missing items or corrections to be made.

For the final inspection:

- Verify piping insulation thicknesses.

- Verify duct insulation R-value.
- Verify that problems noted at the rough-in inspection have been addressed.

An inspector's ongoing challenge is responding to change orders during construction. The call is easy if a more efficient piece of equipment is being substituted for a less efficient one. For instance, it would be acceptable to substitute R-11 duct insulation if the drawings specify R-8 insulation. Also, a 90,000 Btu/h air cooled heat pump with a 3.6 COP @47°F for cooling would be an improvement over one with a 3.4 COP, provided that the equipment wasn't oversized and had the necessary controls to stage the use of electric resistance heat.

A more difficult change order is one that reduces efficiency. For example, if the proposed substitute heat pump has a COP of only 3.0 for cooling but meets the code minimum for this size heat pump, the inspector should check with the plans examiner to make sure that no tradeoffs were made that resulted in a requirement for a higher efficiency heat pump. Similarly, if the proposed substitute for the main supply fan has a lower motor efficiency, the inspector must definitely check with the plans examiner. In this case, compliance is based on a calculation of the total system W/cfm which will vary based on the fan hp, rated cfm and the motor efficiency. Whenever there are significant changes, the inspector is expected to request that the applicant submit revised plans so that the plans examiner can verify compliance and assure that there is a correct record on file in the building department.

An even tougher case is when the contractor has already installed non-complying equipment without checking with the inspector. For instance, a heat pump with a cooling COP of 2.8 may have been installed. The inspector should be quite strict for several reasons. First, since most contracts are awarded on a cost-competitive basis, the low bid company might win the job and then make its profit by installing non-complying equipment. This would be unfair to the higher-bid contractors. Second, a lenient inspector's job will be more difficult in the future. If a non-complying contractor skates by this time, that contractor will most likely have additional requests for future projects. In addition, other contractors will also begin to ask for special treatment. Self-policing, which works well if everyone is being treated fairly, will begin to decline.

Finally, there is the situation when the approved plans do not contain all of the code requirements. If information or notes are missing from the plans, the inspector can, for instance, simply direct the contractor to make the necessary changes in the field; i.e. direct the contractor to install an automatic setback thermostat, or to use some sealant for the ductwork other than pressure sensitive tape, for instance. The inspector's job is more difficult, however, if the drawings contain information which is wrong. Perhaps the inspector notices that the efficiency of the installed heat pump is too low and informs the contractor, but the contractor responds that they are following the approved plans and indeed they are. The inspector, as the representative of the building official, is clearly authorized to require that the contractor build the project to code. (If necessary the inspector can show the contractor the building department note which says "approved subject to errors and omissions.) In this case, it seems appropriate for the inspector to inform the plans examiner of the problem and ask the plans examiner to help solve the problem. The plans examiner may be able to suggest improvements in other areas that would compensate for this shortfall. It is important for the plans examiner and inspector to appreciate the challenges of each others' work and the benefits of a team effort.

Case Study – Office Building

The following case study demonstrates the recommended procedure for documenting compliance with the mechanical systems and equipment requirements given in Section 403 of the 90.1 Code. Requirements that must be documented include: equipment efficiency and sizing, outside air ventilation, fan system power consumption, pumping controls, temperature/humidity controls, off hour-controls, economizer controls, insulation levels, and project completion (testing and balancing).

This case study includes a completed version of the Mechanical Summary (complete) form, and a description of the fan system worksheet. A complete set of building plans would include: an HVAC floor plan, an HVAC roof plan and elevations, detail drawings of roof curbs and openings, a condensate piping schematic, a controls schematic and sequence description, and schedules for all equipment and terminal devices. For this case study, schedules and equipment descriptions are given below; however, no mechanical drawings have been included.

BUILDING DESCRIPTION

The building used in this case study is a new office to be constructed in Chattanooga, Tennessee. The single story building is served by two packaged air conditioning systems and a packaged heating system. The smaller of the two air conditioning units is used to condition a 145 ft^2 computer room, and operates 24 hours a day, seven days a week. The remaining office space is divided into 18 zones, each controlled by an individual thermostat. The perimeter zones are supplied by dual duct variable air volume (VAV) boxes that allow hot or cold air to be supplied to each zone without reheat or recooling, and without mixing airstreams. The interior zones are supplied by single duct VAV boxes which operate in cooling mode only. All ductwork in the building is located in the return plenum above the suspended ceiling. The air conditioning equipment and all local controls are linked through an energy management system. Envelope and lighting descriptions for this building are included in the previous two sections.

HVAC Equipment Schedules

The following schedules were taken from the plans for this building. Example heating, cooling, and fan equipment schedules are provided with the blank forms included in this manual. A complete set of plans should contain schedules with at least the information shown below.

VAV Box Schedule

In addition to heating, cooling, and fan equipment schedules, terminal unit and VAV box schedules should be provided. There are no requirements for terminal devices in the 90.1 Code, so the terminal unit schedule for the case study building has not been reproduced. The VAV box schedule for the case study building is helpful for showing compliance with the 90.1 Code, and is shown below.

Table 403G Heating Equipment Schedule

ID	Make & Model	Description	Input (Btu/h)	Output (Btu/h)	Controls	Efficiency @ Max. Capacity	Efficiency @ Min. Capacity
HU-1	Reznor HCRPB	Rooftop Gas	270,000	216,000	Gas Furnace Modulation from 50-	AFUE 80%	N/A

300	Furnace	100%; Electronic Ignition

Table 403H Cooling Equipment Schedule

ID	Make & Model	Description	Capacity (Btu/h)	Total CFM	Outside Air Range (cfm)	Efficiency (SEER/EER)	Part Load Efficiency (IPLV)
AC-1	Governaire TLE10-2536-E	Customized Packaged Rooftop DX	370,000	11,500	2,150-11,500	EER=9.6	IPLV=9.6
AC-2	Trane TCC018F100A	Packaged Rooftop DX	18,000	600	50-600	SEER=10.0	N/A

Table 403I Fan Equipment Schedule

ID	Make & Model	Description	Motor Hp	Fan Bhp	Flow Control	CFM	Motor Efficiency
HF-1	Included in HU-1	Heating Supply Fan	5	3.2	Constant	5,000	87.5%
EF-1	Cook 120CPV	Toilet Exhaust Fan	0.25	0.14	Constant	920	54%

Table 403J VAV Box Schedule

ID	Make & Model	Cooling Size	Cooling Design cfm	Cooling Max. cfm	Cooling Min. cfm	Heating Size	Heating Design cfm	Heating Max. cfm	Heating Min. cfm	Zone Area, ft^2
1	Trane VDDE	6”	430	430	170	5”	230	230	130	324
2	Trane VDDE	6”	430	430	170	5”	230	230	130	324
3	Trane VDDE	6”	500	500	200	6”	300	300	160	390
4	Trane VDDE	8”	680	680	230	6”	450	450	250	455
5	Trane VDDE	10”	870	870	490	8”	690	690	390	406
6	Trane VDDE	10”	900	900	260	8”	550	550	290	462
7	Trane VDDE	10”	960	960	380	8”	550	550	260	763
8	Trane VDDE	5”	180	180	90	5”	150	150	70	180
9	Trane VDDE	5”	200	200	120	5”	150	150	80	240
10	Trane VDDE	6”	500	500	100	6”	300	300	180	196
11	Trane VDDE	8”	780	780	380	6”	430	430	250	766
12	Trane VDDE	12”	1740	1740	610	10”	1120	1120	640	921
13	Trane VCCE	10”	1140	1480	260	X	X	X	X	1318
14	Trane VCCE	10”	1070	1390	260	X	X	X	X	1318
15	Trane VCCE	10”	880	1140	260	X	X	X	X	1318
16	Trane VCCE	10”	1070	1390	260	X	X	X	X	1318
17	Trane VCCE	8”	780	1010	170	X	X	X	X	1318
18	Trane VCCE	8”	780	1010	170	X	X	X	X	1318

Equipment and Control System Notes

In addition to the above schedules, the following information was taken from the plans for this case study building:

- Air conditioning equipment shall provide a minimum outside air flow of 2,150 cfm (25 per person, per local standards), and can provide up to 100% outside air.
- Each space conditioning zone shall be controlled by an individual thermostatic control device that responds to the temperature within the zone. When used to control heating, the control shall be adjustable down to 55°F or lower. For cooling, the control shall be adjustable up to 85°F or higher. When used to control both heating and cooling, the device shall be capable of providing a deadband of at least 5°F, within which the supply of heating and cooling is shut off or reduced to a minimum.
- During unoccupied hours, the rooftop heating unit shall be controlled by the energy management system to maintain a user-defined nighttime heating setpoint. All VAV boxes shall be controlled to their unoccupied heating setpoints.
- Gravity or automatic dampers interlocked and closed on fan shutdown shall be provided on the outside air intakes of all air conditioning units.
- A momentary contact push-button shall be located on each zone temperature sensor. When the HVAC equipment is scheduled off, an occupant may push the button. This puts the local zone and associated equipment into “occupied” mode for an owner predetermined length of time.
- Single Duct VAV - The damper shall be closed in all modes except the following: during occupied hours, the damper shall modulate to provide cold air flow at or below maximum cold air flow. Air flow shall be determined from space temperature and cooling setpoint control loop. On a rise in space temperature, cold air flow shall modulate up to maximum air flow. On a drop in space temperature, cold air flow shall decrease to the minimum cold air flow setpoint.
- Dual Duct VAV - Cold air damper shall be closed in all modes except the following: during occupied hours, the damper shall modulate to provide cold air flow at or below the maximum cold air flow. Air flow shall be determined from the space temperature and cooling setpoint control loop. On a rise in space temperature, cold air flow shall modulate up to the maximum air flow. On a drop in space temperature, cold air flow shall decrease to the minimum cold air flow setpoint. Hot air dampers shall be closed in all modes but the following: during occupied hours, the dampers shall modulate to provide hot air flow at or below the maximum hot air flow. Air flow setpoint shall be determined from the space temperature and heating setpoint control loop. On a drop in space temperature, hot air flow shall increase. On a rise in space temperature, hot air flow shall decrease to the minimum hot air flow setpoint.
- AC-1 shall be equipped with an air economizer with DDC controls. When in occupied mode, the energy management system shall operate the economizer and stages of mechanical cooling to maintain the supply air temperature setpoint.
- The supply air temperature setpoint shall be reset to avoid reheat but maintain the zone with the highest cooling or heating demand within tolerance. Setpoints may also be reset manually through the energy management system operator. A

variable frequency drive shall modulate to maintain the static pressure setpoint in supply air ductwork. Building pressure shall be maintained during economizer operation through barometric relief dampers in the building envelope.

- The heating unit shall be started based on a user entered time-of-day schedule. Optimum start shall be calculated based on the average space temperature and the historical time required per degree of deviation. The unit shall be locked out when the building does not require heating. Dedicated controls shall modulate the gas burner valve to maintain the supply air temperature setpoint.
- AC-2 can be controlled for optimum start, night setback, etc.; however, this system is currently scheduled to run 24 hours a day, seven days a week.
- All duct systems shall be constructed, installed, and sealed in accordance with SMACNA HVAC Duct Construction Standards - Metal and Flexible - 1985. Supply ducts or plenums that are designed to operate at static pressures from 0.25 in. w.g. to 2 in. w.g. and are located outside of conditioned space or in return plenums shall be sealed in accordance with seal class C as defined in the above SMACNA standard. Ductwork shall be insulated with a minimum of R-4.2 insulation.
- The air systems in this building shall be tested and balanced in accordance with the National Environmental Balancing Bureau Procedural Standards (1983), or in accordance with ASHRAE Standard 111-1988.
- This building shall be commissioned following the recommendations of ASHRAE Guideline 1-1989 Commissioning of HVAC Systems.
- The building owner shall be provided with operation and maintenance manuals for each piece of HVAC equipment, including the energy management system, AC-1, AC-2, HU-1, and EF-1. The building owner shall also be provided with control schematic diagrams and sequence descriptions.
- The exhaust fan EF-1 shall include a gravity backdraft damper that opens only when the system is in use. This fan only functions when the toilet area is occupied. AC-1 and AC-2 shall be equipped with automatic dampers that close the outside air intake when the supply fan is not in use.

References are made on the Mechanical Summary form to load calculations and to HVAC and control system diagrams. These have not been included in the case study; however, they should be included in any plans submitted to a building department for approval.

COMPLIANCE DOCUMENTATION

The compliance forms that address mechanical systems are not difficult to complete, provided that an organized methodology is used to collect and document the required information. This text discusses the procedure that was used to fill out the mechanical compliance forms for the case study building.

The Mechanical Summary form and Fan System Worksheet are the forms that must be completed to demonstrate compliance with the requirements of Section 403. There are two different Mechanical Summary forms. The first is a simplified, single page form that can be used if the building of interest is served by a single zone packaged system. For multizone packaged systems, and for hydronic systems, the second, two page form must be used.

The Fan System Worksheet can be used to determine the compliance of fan systems with fan motor horsepower totaling more than 10 hp, if the effect of these fans is not already accounted for in packaged system efficiency ratings. In most buildings, including the one given in this case study, this form will not be needed.

Mechanical Summary Form (All Systems)

The Mechanical Summary form is organized to follow the sequence of the requirements in the 90.1 Code. It is used by the permit applicant to demonstrate compliance, by the building department plan checker to verify compliance, and by the building inspector as a checklist for building inspection. The Compliance and Enforcement section of this chapter describes the responsibilities of each of these parties. The summary form includes a checklist that describes the information that should be included on the plans and specifications.

The mechanical system serving this case study building is described above. Although the building is served by unitary equipment, it is a multizone configuration; therefore, the complete, two page Mechanical Summary form is used to show compliance with the 90.1 Code.

403.1 Equipment Efficiency

The first item on the summary form is the mechanical equipment efficiency. HVAC equipment must meet the efficiency requirements specified in Tables 403.1a through 403.1f. The HVAC system in this building consists of an 18,000 Btu/h single package air conditioner, a 370,000 Btu/h single package air conditioner, and a 216,000 Btu/h gas-fired warm air furnace. In order to comply with the 90.1 Code, the 18,000 Btu/h unit must have a SEER of 9.7 or greater, the 370,000 Btu/h unit must have an EER of 9.6 or greater *and* an IPLV of 9.0 or greater, and the 216,000 Btu/h furnace must have an AFUE of 78% or greater. Efficiency ratings for these units were obtained from the equipment manufacturers and are listed on the equipment schedules above. "See schedule" is entered in the Mechanical Summary form. The plans examiner must compare the specified equipment efficiencies to those listed in the 90.1 Code. The building inspector must verify that equipment with the specified efficiencies is installed during the rough inspection.

403.2.1 Load Calculations

The designer of this building performed load calculations using one of the popular load calculation computer programs available from equipment manufacturers. The results of these calculations were submitted to the building department along with the building plans and specifications. Because these load calculations are not included on the building plans, "n.a." is entered on the summary form. The plans examiner needs to verify that load calculations are submitted with plans and specifications.

403.2.2 Equipment/System Sizing

The 90.1 Code requires that HVAC equipment not be over or undersized. In other words, a designer must perform load calculations and size HVAC equipment accordingly. There are three exceptions to the sizing requirement. The first exception applies when a designer specifies the smallest sized product from a given manufacturer that can meet the design load. A manufacturer's catalog must be supplied to the building department to verify this. The second exception applies when equipment is intended for standby use only. If this is the case, the equipment can be oversized as long as controls are provided that allow operation of the equipment only when the primary equipment is not functioning. Control schematics and sequence descriptions must be provided to the building department to verify this. The third exception applies when multiple pieces of the same equipment type with a total capacity greater than the design load are used. This set up can be utilized as long as control schematics and sequence descriptions are provided to the building department that show that the equipment is sequenced or otherwise optimally controlled to meet the design load.

The HVAC systems in this building were optimally sized based on the load calculations, and none of the exceptions apply. The equipment sizes are listed in the above schedules. "See schedule" is entered on the Mechanical Summary form and "no" is circled for the three exceptions. The plans examiner can verify that the equipment has been appropriately sized by reviewing the load calculations and comparing these results to the equipment sizes given in the equipment schedules. The building inspector must check that equipment of the specified sizes is installed during the rough inspection.

403.2.3 Separate Air System

The 90.1 Code requires that separate equipment must be used to satisfy process and comfort loads, unless comfort loads comprise less than 25% of the total load, or unless the spaces requiring comfort conditioning total less than 1,000 ft^2. The building in this case study includes a data processing room which contains large computer equipment with distinct conditioning requirements. This room is served by a separate, 18,000 Btu/h air conditioning unit. "Yes" is circled on the Mechanical Summary form to indicate that the requirements of the Code are met. "M2, " the page of the plans containing the HVAC floor plan, is entered as a reference, and "no" is circled for the exception entry. The plans examiner must verify that the main air conditioning unit in the building does not serve the data room, and that the data room air conditioner is not used for comfort conditioning. The building inspector must verify that the air distribution system for the two units are separate during the rough inspection.

403.2.4 Ventilation/Fan System

The 90.1 Code requires that HVAC systems are capable of reducing outside air ventilation to the minimum rate required by the local ventilation code. The outside air ventilation actually employed by a given HVAC system can exceed this minimum value; however, the system must be adjustable to a level at least as low as the minimum requirement. For this case study building, a minimum of 25 cfm of outside air must be supplied per person during operating hours. Since the building is designed for 86 people, this means that the minimum outside air ventilation requirement is 2,150 cfm. Similarly, there may be as many as two people in the data room during operating hours. Thus, 50 cfm of outside air must be supplied by the data room air conditioner. As mentioned above, the data room air conditioner is serving process loads only; however, since people may be in the room on occasion, outside air must be supplied. "Yes" is circled on the Mechanical Summary form and "see notes" is entered as a reference to indicate that the minimum outside air requirement has been met and that proof is given in the system notes above.

The fan power limits of the 90.1 Code are as follows: constant volume fan systems must have a total fan system power demand of no greater than 0.8 W/cfm, and VAV fan systems must have a total fan system power demand of no greater than 1.25 W/cfm. This requirement does not apply if the power demand of a given fan system is included in the efficiency rating of packaged equipment, or if the total fan system motor horsepower is 10 hp or less. In order to qualify as a "fan system," fans must be linked to a heating or cooling source. The fans in the two air conditioning units in this building are accounted for in the equipment efficiency ratings, and therefore are exempt from this requirement. The toilet ventilation fan in the building is not linked to a heating or cooling source, so it too is exempt. The fan in the heating equipment is not included in the AFUE rating; however, it is less than ten horsepower, so it is also exempt. "Yes" has been circled for the two fan system exemptions, and "See Schedules" has been entered.

The plans examiner must verify that the HVAC system notes specify that the minimum outside air requirements will be met and that the equipment schedules indicate that the furnace fan motor horsepower is less than 10 hp. The building inspector must verify that space conditioning equipment has adjustable outside air dampers that can limit outside air ventilation to the local minimum rate. He must also verify that there are no fans installed in the building with motor horsepower ratings greater than 10 hp that do not either meet the requirements of the 90.1 Code, or that are accounted for by the packaged equipment efficiency requirement.

403.2.5 Pumping System Control

The pumping system control requirement of the 90.1 Code is similar to the fan power requirement. Since there are no hydronic systems in this case study building, this section of the Mechanical Summary form has been left blank.

403.2.6 Temperature/Humidity Control

Each of the temperature/humidity control requirements in the 90.1 Code are listed on the Mechanical Summary form. The first requirement is that all HVAC systems include at least one temperature control device. Proof that the case study building meets this requirement is given on the HVAC floor plan, page M2 of the building plans. “Yes” is circled for system controls, and “M2” is entered on the summary form.

The next requirement is that each zone of a given building be controlled by individual thermostats. Perimeter systems that offset envelope loads and that serve zones also served by an interior conditioning system may be exempt from this requirement provided that individual temperature controls are provided for every 50 contiguous feet of exterior wall surface in a given orientation served by the perimeter system. An example of this exemption is given in Figure 403D. The case study building is divided in to 18 zones, each equipped with its own thermostat. This is shown on the HVAC floor plan. “Yes” is circled for zone controls, and “M2” is entered as a reference. “No” is circled to indicate that the independent perimeter system exception does not apply.

Zone level thermostats used to control heating must be capable of being set locally or remotely from an energy management system down to a setpoint of 55°F or lower. Similarly, thermostats used to control cooling must be capable being set locally or remotely up to a setpoint of 85°F or higher. When a thermostat is used to control both heating and cooling, it must be capable of providing a deadband of 5°F within which the supply of hot and cold air is either shut off or reduced to a minimum. Thermostats that require a manual change over from heating to cooling do not need to meet this requirement. Similarly, thermostats that are used to control space temperatures in special occupancies, such as hospitals or museums, are exempt from this requirement. The HVAC system notes above indicate that the thermostats in this case study building have heating setpoints adjustable to 55°F or lower and cooling setpoints adjustable to 85°F or higher. These thermostats are also capable of providing a 5°F deadband. “Yes” is circled on the summary form to indicate that the thermostats in this case study building meet the zone control capability requirements of the 90.1 Code. Similarly, “no” is circled to indicate that the thermostats do not meet the exemption criteria. “See notes” is entered on the summary form as a reference.

The next item pertains to heat pump controls. The case study building does not include any heat pumps, so this section of the form has been left blank. Similarly, there is no active humidification or dehumidification in this building, so humidity controls are not required, and the humidistat section of the form has been left blank.

Section 403.2.6.6 of the 90.1 Code pertains to simultaneous heating and cooling. The Code requires that zone level thermostats and, if appropriate, humidistats, must be capable of sequencing the supply of hot and cold air to a given space to prevent reheating, recooling, or mixing of previously mechanically heated or cooled air. There are several exceptions to this requirement, which are listed on the Mechanical Summary form and described in this chapter. The interior zones of this case study building have no heating requirements and therefore are served only by the air conditioning unit. The VAV boxes in these zones do not contain reheat coils, and are therefore incapable of reheating previously mechanically cooled air. The perimeter zones are served by dual duct VAV boxes. When a zone requires cooling, the VAV box hot deck damper is closed completely, and the cold deck damper modulates from its minimum position to fully open to serve the load. When a zone requires heating, the VAV box cold deck damper is closed completely, and the hot deck damper modulates from its minimum position to fully open to serve the load. As a result, cold and hot air are never mixed, and no reheating or recooling takes place. “No” is circled on the summary form to indicate that the system in this building has no simultaneous heating and cooling, and “M2” and “see schedule” are entered as references to the HVAC floor plan and the VAV box schedule. Similarly, “no” has been entered for each of the exceptions listed.

The next temperature/humidity control requirement of the 90.1 Code pertains to supply air temperature reset. HVAC systems must be capable of resetting the supply air temperature to the warmest (cooling) or coldest (heating) level that will still meet the building loads. Several different strategies for accomplishing this are listed in this chapter. The controls in this case study building reset the cold air supply to the warmest possible temperature that will still maintain conditions in the zone with the greatest cooling demand. Similarly, the energy management system in this building modulates the gas burner in the furnace to adjust the hot air supply to the lowest temperature that will maintain the zone with the greatest heating demand. An HVAC system is exempt from the supply air temperature reset requirement as long as it complies with the simultaneous heating and cooling requirement without invoking exceptions number one or two listed under section number 403.2.6.6 on the Mechanical Summary Form. Although this building complies with Section 403.2.6.6 without invoking any of the exceptions, supply air temperature reset has been specified in the design as an additional means of saving energy. “Yes” is circled on the summary form to indicate that all of the HVAC equipment in this building has supply air temperature reset, and “see notes” is entered as a reference.

The next item on the Mechanical Summary form pertains to the supply water temperature reset requirements of the 90.1 Code. Since this case study building has no hydronic systems, this section of the summary form has been left blank.

A plans examiner must verify that the references listed on the Mechanical Summary form support the existence of thermostats and humidistats that meet the requirements of the 90.1 Code. It is particularly important to verify that simultaneous heating and cooling is not taking place anywhere in the given building, unless the exemption criteria of Section 403.2.6.6 are met. Control schematics and sequence descriptions should be inspected to verify that supply air temperature reset is included in the design, unless the appropriate exemptions are met. During the rough inspection, the building inspector should verify that the thermostats installed meet the requirements of the Code, and that the building either is exempt from the simultaneous heating and cooling requirement, or that VAV boxes which limit or eliminate simultaneous heating and cooling are installed. The existence of supply air temperature reset controls must also be verified.

403.2.7 Off Hour Controls

HVAC systems must include controls that shut off the system or setback the temperature during unoccupied hours, unless the systems serve spaces that are in continuous operation, such as computer rooms, or unless the systems have full load demands of less than 2 kW, such as toilet exhausts. The main air conditioning unit and the furnace in the case study building are controlled through the energy management system to reset to nighttime conditions during unoccupied hours. The second air conditioner in the building is exempt from this requirement because it serves a space that is in continuous operation. Similarly, the toilet exhaust fan is exempt because its total power demand is less than 2 kW. "Yes" is circled on the form to indicate that automatic setback controls exist on the main HVAC units. Similarly, "yes" is circled for both exemption criteria to indicate that the data room air conditioner and the toilet exhaust fan are exempt. "M5," the plan page containing the control system schematic is entered, as is "see notes," because the above HVAC system notes contain a description of the setback controls. "See schedules" is also listed as proof that the toilet fan is less than 2 kW.

Fans which introduce outside air into the building or which exhaust air from the building must include dampers which automatically close when the fan is shut off. The main air conditioning unit in this case study building has automatic dampers on the outside air inlet which close when the supply fan is off. The furnace in this building does not introduce outside air into the building, and, since dampers are not required on combustion air intakes, it is not required by the 90.1 Code to have dampers. The second air conditioning unit in the building operates continuously, so it is exempt from this requirement; however, it has been equipped with dampers in the event that it does shut down. The toilet exhaust fan in this building has a gravity controlled damper that is open only when the fan is in use. This fan is not required to have a damper, since it exhausts less than 3,000 cfm; however, since the damper decreases the infiltration load of the building, it has been included. "Yes" is circled on the Mechanical Summary form to indicate that shutoff dampers are installed as necessary. Similarly, "yes" is circled to indicate that the continuous operation exemption applies to the second air conditioning unit. "No" is circled for the remaining three exceptions, since they do not apply. "M5," the control system schematic, and "see notes" are entered of the form as references to proof that automatic dampers are included in the design.

The final off-hour control required by the 90.1 Code is zone isolation. The Code requires that systems that serve zones that can be expected to operate nonsimultaneously for more than 750 hours a year must include isolation devices that allow the supply of heating and cooling to these zones to be shut off or setback independently. The thermostats in this case study building include a timed override switch. Pressing this switch changes the setpoints in that zone from "unoccupied" to "occupied" for a predetermined length of time. Zones in which the override button has not been pressed remain in the unoccupied mode. "Yes" is circled for zone isolation controls on the Mechanical Summary form, and "M5" and "see notes" are entered as references to the appropriate controls schematic and descriptions.

The plans examiner must verify that the appropriate off-hour controls have been included in the HVAC system design. It is important that control system schematics and descriptions are included in the plans. The building inspector should check for off-hour controls during the rough inspection. Thermostats should be inspected for override switches, such as the ones in this case study building, and fans which bring in or exhaust outside air must be inspected for automatic dampers as appropriate.

403.2.8 Economizer Controls

The 90.1 Code requires that each cooling system in a given building include an air or water economizer. Air economizers must be either temperature or enthalpy controlled, and must be able to automatically modulate the outside air and return dampers to provide up to 85% of the design supply air quantity as outside air. When air economizers are used, an automatic exhaust system must be included which prevents the interior of the building from becoming pressurized. Water economizers must be designed to satisfy 100% of the expected system cooling load when the outside air temperature is at or below 50°F dry bulb and 45°F wet bulb. Both types of economizer systems must be integrated with the mechanical cooling equipment. Integrated economizers are capable of providing partial cooling while mechanical cooling satisfies the remainder of the load. There are several exemptions to the economizer controls requirement that are listed on the Mechanical Summary form and are described in detail in this chapter.

The main air conditioning unit in this case study building has a temperature controlled air economizer. The energy management system modulates the outside air flow from the minimum required to meet ventilation requirements up to 100% of the design air flow based on the suitability of the outside air for meeting the system's cooling loads. In addition, the economizer and mechanical cooling are integrated in this unit. The energy management system stages the amount of mechanical cooling provided based on the ability of the outside air to meet the cooling demand. Barometric dampers are included in the building envelope that open when the pressure inside the building is greater than the pressure outside. This air conditioning unit complies with the 90.1 Code. The second air conditioning unit in the building serves only the data room and provides less than 3,000 cfm of supply air. Because the design air flow is less than 3,000 cfm, this system is exempt from the economizer controls requirement of the Code.

The 90.1 Code requires that economizer operation does not increase the building heating energy use during normal operation. The interior zones of this case study building have no heating demand. The perimeter zones are served by dual duct VAV boxes. The cooling side of these boxes close completely during the heating mode. Thus, economizer operation does not affect heating energy use, and the system complies with the 90.1 Code.

"Yes" is circled on the Mechanical Summary form to indicate that the main cooling system in this building has the appropriate economizer controls. "Yes" is also circled for the first exception of the economizer controls requirement because the data room air conditioning unit in this building does not have an automatically controlled economizer, and is not required to because it supplies less than 3,000 cfm of supply air. "No" is circled for exceptions two through seven, because these exceptions do not apply to the case study building. "Yes" is again circled to indicate that the economizer on the main air conditioning unit is integrated. "No" is circled for the two integrated economizer exemptions. The integrated control requirements do not apply to the data room air conditioner, since this unit is exempt from having an automatically controlled economizer. Page M4 and M5, and "see notes" are entered as references for the plans examiner. These references include the control schematic drawings and the control descriptions.

The plans examiner must verify that an automatically controlled air economizer has been included in the design, and that it is sized to supply up to 85% of the design supply cfm. He must also verify that an automatic exhaust system, such as the barometric dampers in this building, has been specified. The controls schematic and descriptions must be reviewed to verify that economizer operation is integrated with the mechanical cooling, and that it does not increase the heating energy use of the building. The building inspector must check to see that economizers with automatic controls, and an automatic exhaust system, have been installed during the rough inspection. The control system should be inspected to verify that the economizer is integrated. For this building, the inspector would also have to verify that the data room air conditioner supplies less than 3,000 cfm, and that the dual duct VAV boxes installed in the perimeter zones include cold deck dampers which close completely during heating mode.

403.2.9.1 Piping Insulation

This section applies only to hydronic systems, since refrigerant and condensate line losses do not contribute to building energy use. Since the case study building does not include any hydronic systems, this section has been left blank.

403.2.9.2 Duct and Plenum Insulation

Table 403.2.9.2 describes the duct insulation requirements of the 90.1 Code. Simplified duct insulation requirements are given in Appendix D of the Code. The ducts in this case study building are located in the return air plenum above the ceiling of the office space. Ducts located in a return air plenum air considered to be in “indirectly conditioned space” by the 90.1 Code. Table D-1 of the Code indicates that all ducts in “indirectly conditioned space” must be insulated with at least R-3.3 insulation. All of the ducts in this building are insulated with R-4.2 insulation. “Yes” is circled to indicate that the ductwork in this building complies with the Code. “No” is circled for each of the exemption criteria. “See notes” and page M2 are entered as references for the plans examiner. Page M2 is the HVAC floor plan, which indicates that all of the ducts will be located in the return air plenum. R-4.2 insulation is specified in the HVAC system notes given above.

The plans examiner must verify that all of the ductwork will indeed be located in the return air plenum, and that insulation of at least R-3.3 has been specified. Exterior ducts and located in unconditioned space have different insulation requirements, so it is important to verify that such ducts do not exist. The building inspector must verify duct insulation during his final inspection. Again, it is important to check that insulation of the correct R-value has been used and that there are no ducts located outside of the return air plenum.

403.2.9.3 Duct/Plenum Construction

Ducts and plenums must be constructed in accordance with SMACNA HVAC Duct Construction Standards - Metal and Flexible, 1985, and SMACNA Fibrous Glass Duct Construction Standards, 1979, or equivalent. In addition, ducts designed for static pressures ranging from 0.25 in. w.g. to 2 in. w.g. must be sealed in accordance with SMACNA Seal Class C. Pressure sensitive tape cannot be used at static pressures of 1 in. w.g. or higher. Ductwork designed to operate at 3 in. w.g. or higher must be tested in accordance with Section 5 of the “SMACNA HVAC Air Duct Leakage Test Manual, 1985,” and must meet Leakage Class 6 as defined in the SMACNA manual.

All of the ductwork in this case study building is designed to operate at static pressures less than 2 in. w.g.. All ducts are to be constructed in accordance with the SMACNA Standards and will be sealed in accordance with Seal Class C. This information is included in the HVAC system notes given above. “Yes” is circled on the Mechanical Summary form to indicate that the requirements of the Code have been met. “See notes” is entered as a reference.

The plans examiner must verify that the duct construction requirements have been addressed. The building inspector must inspect the ducts during the initial inspection, before they are insulated, and during the final inspection, to verify that the requirements of the Code have been met.

403.2.10.1 Completion Manuals

The 90.1 Code requires that all information required to properly operate and maintain HVAC equipment must be provided to the building owner upon completion of construction. This generally means that the building owner will be given copies of the operation and maintenance (O&M) manual for each piece of HVAC equipment. The building owner should also be provided with schematic diagrams and sequence descriptions of the control system.

The HVAC system notes above indicate that the designer intends to supply the building owner with the required information. “Yes” is circled on the Mechanical Summary form, and “see notes” is listed as a reference. The plans examiner should check that the designer intends to supply the appropriate material to the building owner. The building inspector must verify that the building owner has all of the necessary information during the final inspection.

403.2.10.2 Air System Balancing

The 90.1 Code requires that all air systems are balanced in a manner which first minimizes throttling losses through damper adjustment, and then adjusts fan speed to meet design air flow. Damper throttling alone may be used for air system balancing if the system includes only fan motors of 1 hp or less, or if throttling increases fan power no more than 1/3 hp above the power draw that would be obtained by adjusting the fan speed.

The air systems in this case study building were balanced according to ASHRAE Standard 111-1988 and the National Environmental Balancing Bureau Procedural Standards, 1983. These standards meet the requirements of the 90.1 Code. “Yes” is circled on the Mechanical Summary form to indicate that air system balancing has been addressed. “Yes” is also circled for the first exception, because the toilet exhaust system does not need to be balanced. “No” is circled for the remaining exception, since it does not apply. “See notes” is entered as a reference because the specification of air system balancing is given in the above HVAC system notes. “See schedules” is also entered as a reference, so that the plans examiner can verify that the toilet exhaust system is exempt. The plans examiner must verify that the designer intends to have all air systems, except the toilet exhaust, tested and balanced. The building inspector must verify that testing and balancing has taken place during the final inspection, and that the toilet exhaust fan motor is less than 1 hp.

403.2.10.3 Hydronic System Balancing

There are no hydronic systems in this case study building, so this section of the Mechanical Summary form has been left blank.

403.2.10.4 Control System Testing

Control systems must be tested, calibrated, and adjusted to assure proper operation. It is recommended, but not required, that ASHRAE’s Guideline 1-1989 Commissioning of HVAC Systems (Code 86801) be followed to implement a commissioning plan. The building in this case study has been commissioned based on the ASHRAE procedure, and a commissioning report has been given to the building owner. “Yes” is circled on the Mechanical Summary form to indicate that the designer has addressed control system testing, and “see notes” is entered as a reference to the above HVAC system notes. The plans examiner must verify that the designer intends to have the control system tested. The building inspector must check that the control system has been tested in the final inspection.

Fan System Worksheet

The Fan System Worksheet was not completed for this case study building because all of the fan systems that were not accounted for by packaged system efficiency ratings met the fan power exemption requirements of the 90.1 Code. If the fan systems were not exempt, the Fan System Worksheet could have been used to show compliance with the requirements of the Code. The completion of the worksheet is straightforward.

The worksheet is divided into tables. Each table corresponds to a fan system. The overall watts per cfm for constant volume fan systems must be less than 0.8 W/cfm. For VAV systems, it must be less than 1.25 W/cfm. To complete the worksheet, each fan system should be assigned a number. Each fan in the system should then be assigned an ID which links it to the fan equipment schedule. The rated flow rate for the fan should be entered in the cfm column. Similarly, the rated motor horsepower and fan brake horsepower should be entered in the nameplate HP and

Brake HP columns, respectively. If the brake horsepower is unknown, the rated horsepower should be entered in this column. Next, the motor efficiency is entered. If the motor efficiency is unknown, Table 403B can be used to obtain a default efficiency for a given horsepower. The power consumption of each fan in the system is then determined according to the formula shown in the Watts column.

In order to determine the overall system watts per cfm, the flow rates for each fan in the system must be added, and the sum entered in the Total Fan cfm field. Next, the total system power consumption is determined by adding all of the values in the Watts column, and entering this sum in the Total Watts field. The Total Watts are then divided by the Total cfm, and this quotient is entered in the Overall System W/cfm field. If this value is less than the appropriate requirement listed above, the fan system complies with the code. This procedure must be repeated for each fan system in the building.

Table 403K Mechanical Summary

Case **Study – Restaurant**

The following information is provided as an exercise. The HVAC system details are given for a new building to be constructed in the midwestern United States. Enough information is provided to complete the required mechanical system compliance forms.

BUILDING DESCRIPTION

The building used in this example is a new, single story restaurant to be constructed in Columbus, Ohio. The restaurant is divided into five zones, each served by a separate packaged rooftop air conditioning/furnace unit. Each zone is controlled by an individual thermostat. There are seven exhaust fans providing kitchen, dishwasher, and bathroom exhaust, and three make-up air fans. All air systems in the building are constant volume. All ductwork in the building is located in the return air plenum above the ceiling. Load calculations, and a complete set of building plans, including an HVAC floor plan, roof plan, and section diagrams, equipment schedules, and a controls description, have been submitted to the building department for approval. Envelope and lighting descriptions for this building are included in the previous two sections.

HVAC Equipment Schedules

The following schedules were taken from the plans for this building. Example heating, cooling, and fan equipment schedules are provided with the blank forms included in this manual. The HVAC units are labeled RTU-1 through RTU-5 on the plans. RTU-1 serves the a dining area along the perimeter of the building. Similarly, RTU-2 serves another perimeter dining area. RTU-3 serves the interior section of the building, which includes a dining area, the bar, restrooms, and the foyer. RTU-4 serves the office, the cashier, the majority of the kitchen, and the service area. RTU-5 serves the dry storage area, the receiving area, the employee dressing room, the remainder of the kitchen, and the dishwashing area.

Equipment and Control System Notes

In addition to the above schedules, the following information was taken from the building plans:

- Each HVAC system shall be equipped with at least one automatic device to setback or shut-off the system during periods of non-use or alternate-use of the building zone served by the system.
- Automatic temperature control devices for regulation of space temperature shall be capable of being set from 55°F to 85°F and shall have the ability to operate heating and cooling in sequence. The control shall be adjustable to provide a deadband of up to 10°F between heating and cooling.
- Automatic dampers interlocked and closed at fan shutdown shall be provided on the outside air intakes of all air conditioning units.
- Make-up air fan operation shall be controlled by exhaust fan operation. Make-up air fans shall not operate except when air is being exhausted through the exhaust

system. Make-up air fans shall be equipped with inlet dampers which automatically close during periods of non-operation.

- Exhaust fans shall be equipped with gravity controlled dampers which close during periods of non-operation.
- Packaged HVAC equipment shall be capable of providing a minimum outside air flow of 4,800 cfm (15 cfm of outside air per person at full occupancy).
- All packaged air conditioning units shall be equipped with air economizers that are capable of providing up to 100% outside air. Economizer operation shall be controlled by outside air temperature.
- All duct systems shall be constructed, installed, and sealed in accordance with SMACNA HVAC Duct Construction Standards - Metal and Flexible - 1985. All supply ducts and plenums shall be sealed in accordance with seal class C as defined in the above SMACNA standard. Pressure sensitive tape shall not be used on ducts designed for more than 1 in. w.g. of static pressure.
- All ductwork shall be wrapped with 2 in. thick 0.6 lb/ft^3 density fiberglass insulation.
- The air systems in this building shall be tested and balanced in accordance with ASHRAE Standard 111-1988.
- The building owner shall be provided with operation and maintenance manuals for all HVAC equipment.

Table 403M Heating Equipment Schedule

ID	Make & Model	Description	Input (Btu/h)	Output (Btu/h)	Controls	Efficiency @ Max. Capacity	Efficiency @ Min. Capacity
RTU-1	Carrier 48DJD008	Pkg. Rooftop Gas Furnace	120,000	96,000	Electronic Ignition	AFUE=80%	N/A
RTU-2	Carrier 48DJD008	Pkg. Rooftop Gas Furnace	120,000	96,000	Electronic Ignition	AFUE=80%	N/A
RTU-3	Carrier 48DJD006	Pkg. Rooftop Gas Furnace	74,000	58,460	Electronic Ignition	AFUE=79%	N/A
RTU-4	Carrier 48DJD008	Pkg. Rooftop Gas Furnace	120,000	96,000	Electronic Ignition	AFUE=80%	N/A
RTU-5	Carrier 48DJD006	Pkg. Rooftop Gas Furnace	74,000	58,460	Electronic Ignition	AFUE=79%	N/A

Table 403N Cooling Equipment Schedule

ID	Make & Model	Description	Capacity (Btu/h)	Total CFM	Outside Air Range (cfm)	Efficiency (SEER/EER)	Part Load Efficiency (IPLV)
RTU-1	Carrier 48DJD008	Pkg Rooftop Air Conditioner	90,000	6,000	1,100	EER=9.0	IPLV=8.5
RTU-2	Carrier 48DJD008	Pkg Rooftop Air Conditioner	90,000	6,000	1,100	EER=9.0	IPLV=8.5
RTU-3	Carrier 48DJD006	Pkg Rooftop Air Conditioner	60,000	5,000	750	EER=8.5	N/A
RTU-4	Carrier 48DJD008	Pkg Rooftop Air Conditioner	90,000	6,000	1,100	EER=9.0	IPLV=8.5
RTU-5	Carrier 48DJD006	Pkg Rooftop Air Conditioner	60,000	5,000	750	EER=8.5	N/A

Table 403O Fan Equipment Schedule

ID	Make & Model	Description	Motor Hp	Fan Bhp	Flow Control	CFM	Motor Efficiency
SF-1	Captiveaire SAU-10	Make-up air fan	0.75	0.75	Constant	2,100	72%
SF-2	Captiveaire SAU-10	Make-up air fan	0.75	0.75	Constant	2,100	72%
SF-3	Captiveaire SAU-12	Make-up air fan	1.0	1.0	Constant	2,450	82.5%
EF-1	Captiveaire CA-18	Exhaust Fan	0.75	0.75	Constant	3,000	72%
EF-2	Captiveaire CA-18	Exhaust Fan	0.75	0.75	Constant	3,500	72%
EF-3	Captiveaire CA-18	Exhaust Fan	0.75	0.75	Constant	3,000	72%
EF-4	Cook ACEB120C2B	Exhaust Fan	0.17	0.17	Constant	600	35%
EF-5	Cook ACEB100C2B	Exhaust Fan	0.17	0.17	Constant	300	35%
EF-6	Cook ACEB100C2B	Exhaust Fan	0.17	0.17	Constant	200	35%
EF-7	Captiveaire FMX-13B	Exhaust Fan	0.33	0.33	Constant	1,100	56%

Reference

MULTI-ZONE SYSTEMS

Most buildings have different spaces, called temperature zones, which require varying amounts of heating and cooling simultaneously. A zone facing east, for instance, will require most of its cooling capacity in the morning, while west facing zones will peak in the afternoon. Interior spaces, those with no outdoor exposures at all, will require cooling all year round, even when perimeter zones require heat.

Because of these varying load requirements, multi-zone HVAC systems must be able to individually vary the amount of energy added or extracted from each space, and often must be able to heat one zone and cool another simultaneously. Another complication is the requirement for ventilation and minimum air circulation. There are times when a space has no heating or cooling load, but the HVAC system must still provide ventilation air, hopefully without overheating or overcooling the space.

Theoretically, the most efficient HVAC systems are single zone systems, which are able to sequence the supply of cooling and heating to each space while maintaining outside air ventilation and air circulation rates. However, because of the small size of these systems, the fan and cooling equipment is not as efficient as larger equipment. Furthermore, economizer and other energy saving strategies are often not cost-effective for such small systems. Moreover, single systems are seldom practical on multistory buildings and in speculative buildings, where zoning patterns are not known at the time of construction.

Multiple zone systems have the advantage of handling the load requirements of many temperature zones simultaneously, but most are inherently inefficient at part load. The source of this inefficiency is the need to simultaneously supply heating and cooling energy to multiple zones, typically by reheating cold air, recooling warm air, or mixing warm and cold air streams.

Variable Air Volume Systems

VAV systems vary the amount of air supplied to a space, rather than leaving the supply volume constant and varying the temperature of the air, as with constant volume systems. For example, during heating conditions at the perimeter zones, a damper or air valve controlled by the space thermostat first reduces the air volume to the space to some present minimum, typically 30% to 40% of the peak volume, before the reheat coil is energized. Energy is saved three ways: first less cooling energy is used because the volume of air cooled is reduced (this is not a factor when the system is operating in economizer mode). less heating energy is used because less air needs to be reheated, and fan energy is reduced because the amount of air the fan needs to move is reduced.

Efficiency can be further improved with temperature reset. Temperature reset controls on VAV systems reset the supply air temperature so that the zone requiring the most cooling is minimally satisfied at full air volume. Such control reduces reheat energy, and for systems using economizer controls, temperature reset also reduces cooling energy by increasing the number of hours that the economizer is able to operate. For VAV systems, fan volume , and therefore fan energy, will increase with temperature reset. However, the savings in reheat energy will almost always offset this increase. Even when the saving are small, the temperature reset control has the advantage of increasing air circulation rates, which are often so low as to be a comfort problem in some VAV systems.

Figure 403 L Typical Variable Air Volume System

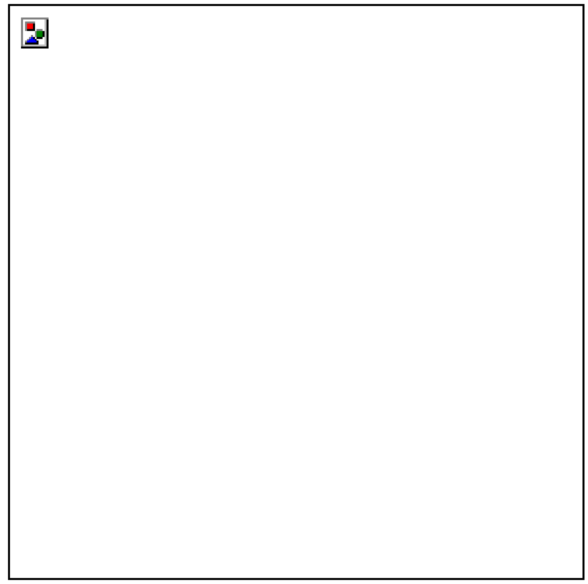
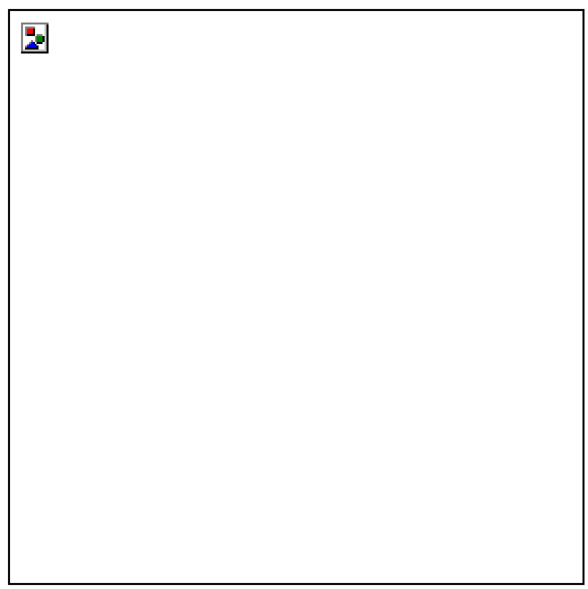


Figure 403 M Typical Constant Volume System



Constant Volume Reheat Systems

These systems provide a constant volume of cool air to several zones. Each zone has a reheat coil controlled by a space thermostat. This coil reheats the cool air as required to satisfy the thermostat setting. The inefficiencies of the system are obvious. For example, during peak heating conditions, a typical constant volume reheat system first cools return air from about 75°F to about 55°F, then each perimeter zone the heating coil must warm this 55°F air to about 110°F. Both heating and cooling energy is wasted.

Constant Volume Dual Duct and Multi-zone Systems

Dual duct systems are all-air distribution systems with two supply ducts or “decks”, one supplying cool air and one supplying warm air. Mixing boxes at each zone mix the right amount each supply air stream to satisfy the space thermostat. For instance, as the cooling load in a zone decreases, the mixing damper, controlled by the space thermostat, reduces the amount of cold air to the mixing box and simultaneously increases the amount of hot air. The total volume of air to each zone remains constant. Multizone systems are thermodynamically similar, except the mixing dampers are at a central location, and individual supply ducts are extended to each zone.

The inefficiencies of this system are due to the mixing of the hot and cold air streams when the system operates at part load.

Efficiency can be increased by using variable air volume. With VAV, control dampers with sequential, or slightly overlapping, control ranges for each of the cooling and heating supplies to the mixing box. As the cooling load decreases in the space, the cooling control damper reduces the cooling air volume to the mixing box, but the heating air damper remains closed. When the volume reaches a preset minimum, as required by ventilation codes or to maintain minimum circulation rates, the heating damper will start to open. As the zone requires heating, the cooling damper will close completely and the heating damper will open to satisfy the load.

Figure 403 N Typical Dual Duct System

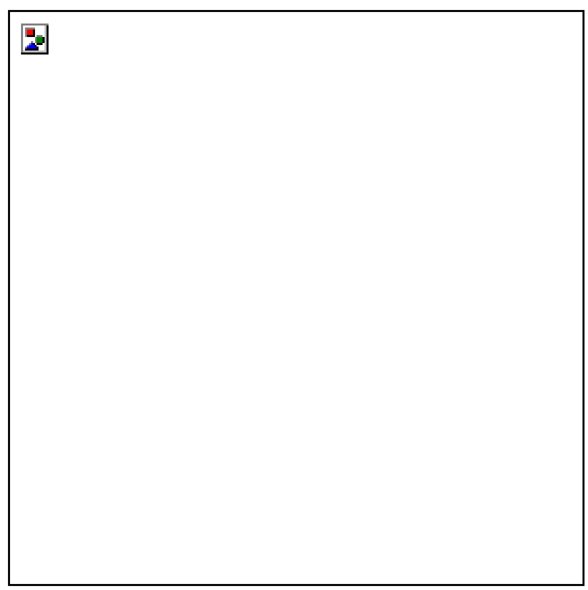
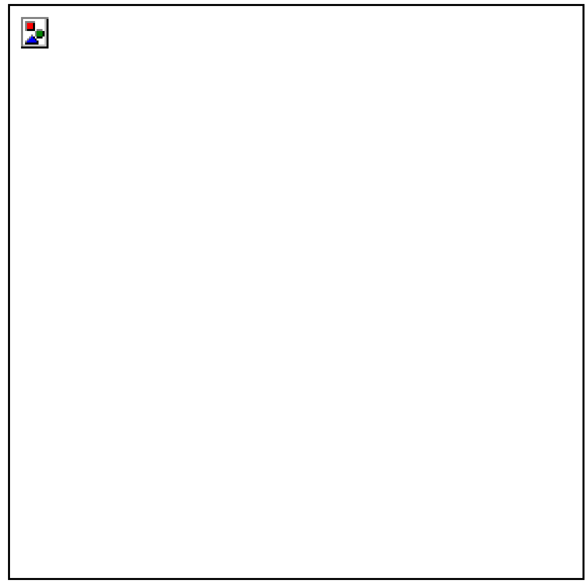


Figure 403 O Typical Multi-Zone System



Recooling Systems

Recooling systems are similar to reheat systems except the central fan system supplies warm air and cooling coils are provided at each zone to recool the air to satisfy the space load. These systems are very uncommon because they are no better at satisfying space comfort conditions than reheat systems, but are substantially more expensive due to additional piping costs, coil costs, and condensate removal problems at each zone coil.

Independent Systems

Independent systems are those with two separate systems supplying heated and/or cooled air to a single space. One example is a central cooling system, with an independent perimeter system supplying air just to perimeter zones. The perimeter system is usually heating-only, such a baseboard or radiant system, but may also have cooling capability as well. If the independent perimeter system has cooling capability, the interior VAV systems would be sized to handle non-transmission cooling loads, while the perimeter cooling system should be sized to supplement the interior system during warm weather. With the heating-only perimeter systems, the interior system would handle the entire cooling load, while the perimeter system would generally provide all of the heating capability.

The control challenge with independent systems is to prevent "fighting", where one system supplies too much cooling and the other system must compensate by overheating, or vice-versa.

The recommended control strategy is to provide sequential control of heating and cooling supplies. For example, on a call for heating with a VAV cooling system with perimeter baseboard heating, the signal from a single thermostat in each zone could be first close the VAV damper to reduce ventilation codes then begin to open the baseboard control valve, or cycle the electric heating coil.

ECONOMIZERS

Most commercial buildings experience cooling loads even in mild or cool weather. This is true for perimeter spaces because of high lighting, people and solar loads which are not a function of outside air temperature. Interior spaces, which have no exposures to the outside, require cooling on a year round basis, except perhaps for morning warm-up situations. Because these cooling loads occur during cool weather, it is possible to use the cool outdoor air to reduce mechanical cooling loads. This is called economizer control or "free cooling".

There are two basic types of economizer control, the air economizer and the water economizer. Each has many variations explained below.

Air Economizers

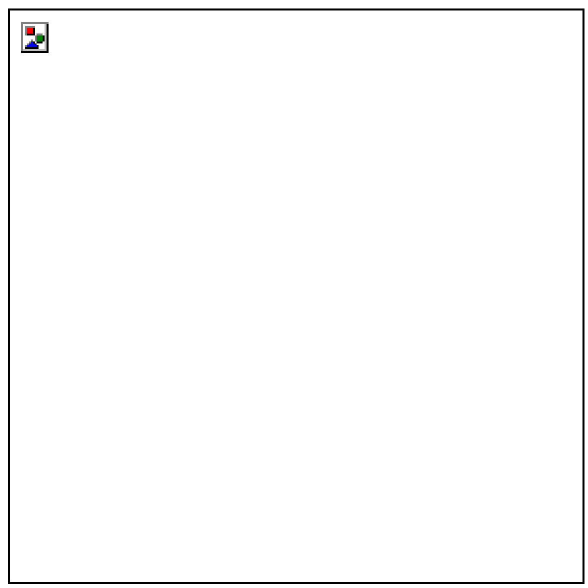
The basic air economizer consists of a set of dampers on the return air and outside air intakes and controls which position the damper blades to control the flow of air into the system. When outside air temperature is less than the return air temperature, but greater than the supply air temperature setpoint, the dampers are controlled to introduce 100% outside air (see Figure 403P). When the outside air temperature is less than the supply air temperature setpoint, the dampers are modulated to mix outside and return air in the correct proportions so that the temperature of the mixed air entering the supply system is equal to the supply air temperature setpoint. In either of the above situations, the economizer controls satisfy all parts of the cooling load. When the outside temperature is greater than the return air temperature, the dampers allow all return air to enter the fan system, except for code required ventilation air. At this point, the economizer is unable to provide and “free cooling”. This type of economizer is called an integrated return air temperature economizer. It is integrated because it is able to operate simultaneously with mechanical cooling.

Another type of air economizer is called an integrated enthalpy economizer, where the air enthalpy (heat content) is compared instead of temperatures. This control is theoretically more efficient than the temperature economizer for most climates, but it is less reliable due to the practical problems of measuring air moisture.

A high limit economizer is similar to the return air economizer but return air conditions are not measured. Instead a cut-off point, or high limit condition for outside air temperature is pre-selected (e.g. 78°F) above which the economizer dampers return to the minimum outside air mode.

A two position or non-integrated economizer is one which is not able to operate when mechanical cooling is required simultaneously. The dampers are often interlocked with the cooling system to return to the minimum outside air position whenever the cooling system operates. This is often required for direct expansion systems without hot gas bypass or other low load control.

Figure 403P Air Economizer Operation



Water Economizers

Water economizers are essentially indirect and/or direct evaporative coolers. In large commercial projects, they are usually indirect coolers, using condenser water and cooling towers to chill water during cool, dry weather. This systems has an advantage over air economizers in climates where wintertime humidification is required; the large amounts of dry air introduced with air economizers often increases humidification loads. However, in most climates, water economizers are generally not as energy efficient as air economizers because they require the operation of cooling tower fans and condenser pumps. Air economizers also have the advantage of introducing large amounts of outdoor air for a majority of the building operating hours, which can improve indoor air quality.

One type of water economizer uses a separate pre-cooling coil in series with the mechanical system cooling coil. Condenser water is run through tower and cooled as much as possible (typically to a minimum of about 50°F). The water then is run through the economizer coil, precooling the air before it enters the mechanical system cooling coil. this is an integrated economizer since mechanical cooling can be provided simultaneously with the economizer cooling.

Another popular type of water economizer, called a "strainer cycle", is a two-position economizer. This system runs condenser water through the cooling tower, then diverts the condenser water through the chilled water system whenever it is cold enough to provide 100% of the cooling load. When the load or outdoor temperatures grow larger so that the condenser water is no longer cold enough to do all the cooling, the system reverts back to the mechanical cooling mode and supplies mechanically cooled water to the chilled water system. In this way, the economizer system cannot provide partial cooling; it's all or nothing. These two-position economizers are not nearly as efficient as integrated economizers, especially in mild climates where there are many operating hours in the 50°F to 75°F range.

PART-LOAD PERFORMANCE

Historically, energy codes only limit the full-load equipment efficiency, the efficiency of the equipment at standard rating conditions representative of typical peak design or "full load" conditions. While equipment full-load efficiency is important, particularly in areas with high utility demand charges, performance at part load is usually more relevant in determining annual equipment energy costs. In recognition of this issue, the 90.1 Code requires that most equipment be tested and meet efficiency limits at part-load as well as at full-load conditions.

Actual equipment load profiles will vary by system type, building architecture, occupancy patterns, and climate. Therefore it is only possible to determine performance at representative part-load conditions. While the results will not predict actual performance precisely, they can be used to compare different manufacturers and different equipment types in much the same way that EPA mileage ratings allow different cars to be compared, even though actual mileage will vary.

There are three types of part-load descriptors used in the Tables 403.1a through 403.1f of the 90.1 Code:

1. *HSPF, AFUE, and SEER.* These are seasonal performance descriptors for DOE-covered heat pumps, gas furnaces, and air conditioners, respectively, determined in accordance with DOE Test Procedure 10 CFR Part 430, Subpart A. These ratings are based on a typical weather profile and include the effects of equipment cycling. All products covered by the National Appliance Energy Conservation Act (NAECA) must be rated using these descriptors.

2. *IPLV, or Integrated Part-Load Value.* IPLV is a measure of part-load performance for some ARI-rated equipment with unloading capability. The units of IPLV are the same as those for the corresponding full-load descriptor; for instance an IPLV for an air conditioner has the same units as the energy efficiency rating (EER). IPLV is a weighted average of the steadystate equipment performance at several load conditions. An IPLV may contain up to three part-load efficiencies in addition to the full-load efficiency. Because measurements are steady-state, IPLV does not include the effects of equipment

cycling; equipment is rated only at the part loading allowed by the equipment and its controls.

3. *Low or High Temperature Ratings.* Some equipment is rated at an off-design condition in addition to the standard full-load condition. While not actually a part-load condition, since the equipment is tested at full capacity, the off-design rating can give an indication of equipment performance at different temperature conditions. Examples include high (47°F) and low (17°F) outside air temperature COP ratings for heat pumps and high (85°F) and low (75°F) temperature ratings for hydronic heat pumps. The ratings are steady state and do not include the effect of equipment cycling.

Example 403RR Part-Load Performance Requirements – Air Conditioner with a Single Compressor

Q An 8-ton rooftop air conditioner has a single compressor with no unloading capability. Does this unit have to meet the IPLV requirement of Table 403.1a?

A No. IPLVs are determined by measuring performance at steady-state part-load conditions. If the equipment cannot operate at that condition without cycling, its steady-state performance cannot be measured. Thus for a single speed compressor with no cylinder unloading, IPLV requirements do not apply.

Example 403SS IPLV Calculation – Water-Cooled Reciprocating Chiller

Q How is the IPLV determined for a water-cooled reciprocating chiller with limited unloading capability?

A For this product category IPLV is determined in accordance with ARI Standard 590 which requires that chiller performance be determined using the following equation:

$$\text{IPLV} = 0.17 \times \text{EER}_{100} + 0.39 \times \text{EER}_{75} + 0.33 \times \text{EER}_{50} + 0.11 \times \text{EER}_{25}$$

where

EER_n = EER at n% of full load

EER is measured at standard rating conditions with condenser water "relief" (2.5°F drop in condenser water temperature for each 10% drop in load). The weighting factors in the equation are based on an HVAC system with an air-side economizer serving a typical office building located in Atlanta. (To approximate performance for an actual application, the "Applied Part-Load Value" or APLV can be determined with the above equation using weighting factors customized for the actual building type and climate.)

If the unit cannot be operated at the 75%, 50% or 25% capacity levels due to limitations in its capacity control system, then the unit operation must be evaluated at other load points and the standard rating efficiencies determined by straight line interpolation between actual operating efficiencies. Extrapolation is not allowed. If a unit cannot operate below a standard rating capacity point, then the unit must be run at its minimum step of capacity at the condenser conditions corresponding to the standard rating capacity condition. The efficiency is then determined using the following equation:

$$EER_L = EER_M \times CD$$

where

EER_L = efficiency at part-load ratio L, Btuh/W

EER_M = measured efficiency at minimum part-load ratio M determined using condenser relief conditions for
part-load ratio L, Btuh/W

CD = cycling degradation factor = 1.13 - 0.13 (L/M)

The chiller has three steps of unloading. Measured performance is indicated as follows:

Step	Capacity tons	Part-Load Ratio, %	Power kW	EER Btuh/W	CWS °F	CWR °F	CHWS °F	CHWR °F
3	123.0	100	92.3	13.0	85.0	94.7	44.0	54.0
2	85.5	69.5	59.6	14.0	77.4	84.1	44.0	51.0
1*	48.6	39.5	32.7	14.5	69.9	73.7	44.0	48.0
1	50.4	41.0	33.9	14.5	66.3	70.2	44.0	48.1

The point labeled 1* is the performance at the minimum step of unloading using the standard condenser conditions for 25% load condition.

The rating points 1, 2, and 3 are plotted with straight lines between to determine the unit's performance at the 75% and 50% conditions:

From the graph (or by mathematical interpolation), the equipment EER at 75% and 50% load can be determined as 13.8 and 14.3 respectively.

Since the unit cannot operate at or below 25% load, performance at 25% load must be calculated by first measuring the performance at the lowest step of unloading, at the condenser water conditions appropriate for the 25% point (step 1* in the table above). CD is determined as:

$$CD = 1.13 - 0.13 \times 0.25 / 0.41 = 1.05$$

The EER_{25} is then calculated to be:

$$EER_{25} = EER_{(M=41\%)} \times CD = 14.5 \times 1.05 = 15.2$$

Example 403SS IPLV Calculation – Water-Cooled Reciprocating Chiller (continued)

The results are summarized as follows:

Step	**Part-Load Ratio, %**	**Weight Factor**	**EER, Btu/W-h**
A	100%	0.17	13.0
B	75%	0.39	13.8
C	50%	0.33	14.3
D	25%	0.11	15.2
IPLV			14.0

The IPLV is 14.0, which is equivalent to a COP of 4.10 (14.0/3.413).