

providing insights for today's hvac system designer

Engineers Newsletter

volume 39-3

ASHRAE Standard 90.1-2010

Updates to mechanical system mandatory and prescriptive requirements

The 2010 version of ASHRAE Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*, will be published this fall. Significantly, the goal of ASHRAE Standard 90.1-2010 was to achieve 30 percent energy-cost savings compared to 90.1-2004 (aggregated across 16 building types in 17 climate zones).

This newsletter provides an overview of the mechanical changes that pertain to heating, ventilation and air-conditioning.

ASHRAE Standard 90.1-2010 is simply 90.1-2007, with the incorporation of more than 100 approved addenda, 52 of which affect mechanical systems and will be the focus of this article.

Other significant changes were made to the scope, lighting, building envelope and modeling sections of the standard, which will affect load calculations and equipment sizing.

The addenda that affect mechanical systems can be roughly grouped into the following areas:

- Equipment efficiency
- System design and control requirements
 - Waterside
 - Outdoor air
 - Airside

These changes are outlined in the following sections.

Equipment Efficiency Changes

The following is a summary of updates to the equipment efficiencies required by 90.1-2010.

Air conditioners and heat pumps.

Efficiencies for air conditioners and heat pumps have been improved substantially in recent years. In fact, when compared to the 1989 standard, there is approximately a 17 percent Energy Efficiency Ratio (EER) improvement in the larger sizes and 30 percent in the 5- to 20-ton category. For small, less than 5-ton, equipment the change is approximately 34 percent Seasonal Energy Efficiency Ratio (SEER) improvement.

Equipment schedules now incorporate a new term called Integrated Energy Efficiency Ratio (IEER). This new metric was developed for unitary products to replace Integrated Part Load Value (IPLV) and encourage designs that have better part-load performance. (Addendum S)

Single-zone VAV. As of January 1, 2012, unitary equipment 110,000 Btuh (9.2 tons) or larger serving a single zone must provide means to reduce fan speed to two-thirds or less at loads below 50 percent. This can be accomplished by two-speed or variable-speed fans.

Chilled-water air-handling units have a similar requirement (effective January 2010); see “VAV requirements for chilled water air-handling units serving a single-zone,” p. 7.

The low-speed airflow can be the larger of the limit stated above or the ASHRAE 62.1 ventilation requirement for the zone. Outdoor air measurement or a compensating damper will be necessary for proper ventilation at reduced speed. Discharge air temperature sensors or multiple stages of compression may be necessary for proper control of the unit. (Addendum N)

Water-cooled and evaporatively cooled air conditioners and heat pumps. New full-load efficiency requirements are 3 to 5 percent more stringent as of June 2011, with IEER increases of 5 to 13 percent. (Addendum CO)

In addition, two-position valves are now required on water-cooled air conditioners as well as on water-loop heat pumps. See further discussion in “Waterside,” p. 3. (Addendum AK)

Packaged terminal air conditioners and heat pumps (PTAC/PTHP). As of October 2012, PTACs and PTHPs will be required to meet new, aggressive efficiency levels. (Addendum BW)

The definition of non-standard size was also added so that standard-sized units in new construction must meet a different efficiency level than replacement units having to fit into existing wall sleeves. (Addendum T)

Water-to-water heat pumps. The water-to-water heat pump as a heating or cooling device now falls within the scope of the standard. The required

efficiency levels vary by application (such as water source, ground water source, ground source) to correspond with the rating temperatures typical for those applications. (Addendum BG)

Computer room air conditioners (CRAC). The scope of 90.1 has been expanded to include systems used in commercial process cooling applications. (Addendum AQ)

As a result, there is a new class of covered product: computer room air conditioners, or CRAC units. ASHRAE 127 is the referenced test procedure. The ASHRAE test procedure has defined a term called Sensible Coefficient of Performance (SCOP). SCOP-127 calculates efficiency at conditions more reflective of the mostly sensible cooling that occurs in data centers and computer rooms. (Addendum BU)

Variable-refrigerant-flow (VRF) multi-splits. While minisplits have been covered by 90.1 for many years and certified to AHRI 210/240, a similar class of equipment called a multi-split or VRF system was previously not covered. This type of split system has multiple indoor DX fan coils sharing a common compressor and condenser. The AHRI 1230 test procedure is now published and there are efficiency requirements for these systems. The IEER requirements get more stringent in July 2012.

VRF systems can include heat recovery through a refrigerant-to-water heat exchanger. There is a special category in each size for VRF systems with heat recovery. (Addendum CP)

Note: As we finalized this newsletter, AHRI was about to release a certification program for VRF multi-splits.

Chillers. Since January 2010, Addendum M has been in effect. The big change was the addition of an alternative compliance path for water-cooled chillers. One path has a more stringent IPLV (part load) requirement, while the other has a more stringent

full-load kW/ton requirement. Each path has both a full-load and a part-load metric to meet. (Addendum M)

Glycol or brine in chillers. Other changes to the chiller section targeted bringing more chillers under the scope of the standard. The first is how to consider the performance and compliance of chillers with glycol for freeze protection that operate at a “normal” chilled-water temperature. In the past, chillers with a freeze point of 27°F or lower had enough glycol to exempt them from meeting the standard. In reality, designers add glycol to air-cooled chillers in cold climates, even though there are no design changes to the chiller, only a performance degradation. The test procedure AHRI 550/590 uses water during the test. Efficiency levels must be calculated for these chillers with water as the test fluid, and meet the values in Table 6.8.1C.

This change is slightly more nuanced for centrifugal chillers. When a centrifugal chiller has glycol in it, heat transfer is degraded. More lift is required of the compressor which changes the design slightly, and its ability to meet the performance in Table 6.8.1C when tested with water. The additional efficiency needed to overcome losses from an oversized compressor operating at lower lift must be compensated for by the chiller design, which serves to discourage unnecessary use of glycol and improves the stringency of the standard. (Addendum BL)

Chiller test procedures. Related interpretations have been issued in recent years to discuss how to accommodate customer-requested deviations from standard test conditions, in particular tolerances and tube fouling. The actual, as built, chiller performance is to be calculated with standard tolerance and fouling, before comparing it with the 90.1 required efficiency level to determine compliance. (IC 90.1-2004-14, IC 90.1-2004-15)

Another interpretation affirmed that the energy metric, both full-load and part-load, are to be measured at the line side of the starter or variable-frequency drive (VFD). This means that starter and drive losses must be included in the chiller performance data. (IC 90.1-2004-6)

Chiller scope expansion. Along with changes for glycol use, the scope for positive displacement chillers, both air- and water-cooled, was expanded to include those with a leaving fluid temperature greater than 32°F. (Addendum BL)

The last change for chillers in 90.1-2010 is the non-standard centrifugal equation and the temperatures and flows that are within the scope. Centrifugal chillers are covered by 90.1 if they have:

- 36°F or higher evaporator leaving fluid temperature,
- 115°F or lower leaving condenser fluid temperature, and
- 20°F to 80°F of lift (leaving condenser minus leaving evaporator temperature).

Roughly 98 percent of centrifugal chillers sold will now be subject to meeting 90.1 requirements. For example, an 85.1°F entering condenser water temperature can no longer be used to avoid the 90.1 requirements. This loophole could not be closed until a new, non-standard adjustment equation was developed. The new adjustment can apply to more combinations of conditions. As chillers deviate from standard conditions, the 90.1 requirement becomes slightly more stringent than in the past version.

The adjustment is handled by a polynomial equation, sometimes referred to as “K-adjust” or K_{adj} . (Addendum BT)

All of these changes together made it difficult to keep the non-standard adjustment tables for centrifugals within the pages of the standard. Instead, a spreadsheet tool will be included on the User’s Manual CD and examples in the printed version of the User’s Manual.

Heat rejection. Heat rejection is another area that saw a significant change. There are now limits on the use of centrifugal fans in cooling towers, once the tower handles more than 1100 gpm. Towers above this flow rate now have to meet the more stringent axial fan power level of 38.2 gpm per horsepower. There are exceptions for site and acoustic restrictions. (Addendum U)

A new tower category was added in response to confusion over whether a fluid cooler, or ‘closed-circuit cooling tower’, was intended to be covered by the cooling tower requirements. Closed-circuit cooling towers have different test conditions more reflective of the range and approach. Closed-circuit cooling towers equipped with an axial fan have a more stringent level than those equipped with a centrifugal fan. (Addenda A, L)

Liquid-to-liquid heat exchangers. Liquid-to-liquid heat exchangers now have a referenced test procedure. This has encouraged heat exchanger manufacturers to submit their products to a certification program, which gives users a common reference for performance. No target efficiency levels have been set, but could be in the future. (Addendum AD)

Electrical equipment.
Motors. Other changes include federal requirements for integral horsepower general purpose motors to be premium efficiency, as defined by the National Electrical Manufacturers Association (NEMA). Special purpose motors are exempt from this requirement. Motor types are defined by NEMA, and an excerpt of the definition of a general purpose motor is included in 90.1 for ease of reference. (Addenda AJ, BK)

Transformers. Low-voltage, dry-type, distribution transformers are covered by 90.1-2010 to reflect 2005 federal requirements. (Addendum O)

System Design and Control Requirements

When it comes to energy savings, much of the impact of the mechanical section relates to system requirements. The general areas fall into three categories: *waterside*, *outdoor air*, and *airside* design and control requirements.

Waterside

Two-position valves in water-cooled unitary products. Condenser water flow for unitary systems can only be variable if (at least) a two-position valve is included at the unit, to shut off water flow when the unit is turned off. Since 2001, water-source heat pumps have been required to have these valves; now this is a requirement for water-cooled unitary air conditioners also. For water-cooled unitary systems of greater than 5 hp, a VFD is required on the pump motor, or it must have similar performance. (Addendum AK)

Variable-flow and variable-speed pumping. In hydronic systems other than water-cooled unitary systems, if the system power is greater than 10 hp, a VFD is required on the pump motor, or it must have similar performance. This is a dramatic change. In the past, at 10 hp for the system, the system had to have, at minimum, two-way valves and ride the pump curve. At 50 hp per pump, with at least 100' of head, a VFD was required. Now, when individual pumps are 5 hp or more, and the system power is at least 10 hp, VFD-like performance is required. (Addendum AK)

Service water booster systems. Pumps are often installed in service water systems (aka domestic hot water) to boost the pressure at a specific point in the system. Downstream of that booster system, 90.1-2010 prohibits installing a device solely for the purpose of reducing the pressure of the water flow, except for safety devices. Pressure must be measured and pump operation varied to better follow the load. And pump(s) must be shut off when no service water flow is required. (Addendum CV)

Pump pressure optimization. With all the new requirements for variable speed pumping, it follows that 90.1 now requires pump pressure optimization for systems with DDC controls. There are two main requirements: the differential pressure (DP) setpoint can be no more than the DP that corresponds to 110 percent of the design flow rate, and the DP setpoint has to be reset until one valve is nearly wide open. (Addendum AK)

Maximum flows in nominal pipe sizes. The allowances change based on the annual hours of operation of that system, and whether it is variable flow/variable speed. See Table 1.

This change limits the amount of frictional loss that the system pumps must overcome. System design delta T impacts which pipe sizes are permitted. For example, if you really want 6-inch and not 8-inch pipes, you may have to increase the delta T (reduce the flow rate).

For pipe sizes larger than 12", you adhere to a maximum velocity. (Addenda AF, CC)

Pump head calculations. Standard 90.1-2010 requires that pump head calculations be performed prior to sizing pumps. This is similar to the requirement for load calculations. (Addendum V)

Pipe insulation. Chilled-water piping insulation requirements have modest changes, but hot water and especially steam pipes have significant upgrades. There are exceptions: for example, when pipes are in the interior walls between conditioned spaces. And if non-metallic pipe is used, and it's greater than schedule 80 thickness, you may reduce insulation thickness to an equivalent heat transfer per linear foot. There are adjustments for buried pipe. (Addenda BA, BI)

Outdoor Air

There are four main topics for outdoor air system design and control: economizers, energy recovery ventilators, dampers, and ventilation controls.

Economizers. A sweeping addendum changed economizer requirements. That is, economizers are now economically justified in most climate zones. Two exceptions are climate zones 1a and 1b, which include the areas highlighted in Figure 1 and other very hot climates worldwide. (Addendum CY)

Smaller systems will need economizers as well. If the individual fan and coil is 54,000 Btu/h (4.5 tons) or greater, then an economizer is required. Some jurisdictions have interpreted this as applying if the *entire system* is greater than 4.5 tons, but

90.1 explicitly states that the threshold is based on the *individual* fan and coil. For a fan-coil, VRF or zone-based fan and coil system, 54,000 Btu/h is the largest terminal size before an economizer is required. This requirement could be satisfied by either a water or air economizer.

One way that more climates were justified is by dropping the exceptions for integrated economizing. This means that the economizer will be the first stage of cooling, followed by mechanical cooling, until the high limit shutoff point. The high limit changes by climate zone.

There are 12 exceptions to the economizer requirement (paraphrased):

- 1 climate zones 1a or 1b
- 2 fan + coil size < 54,000 Btu/h (computer rooms use 90.1-2007 size limits)
- 3 non-particulate air treatment is required by ASHRAE 62.1
- 4 percent of air that is humidified to over 35°F dewpoint (75 percent of design airflow for health care, 25 percent for process cooling, not applicable to computer rooms)
- 5 systems with condenser heat recovery of a specific size (see below discussion)
- 6 systems serving residential spaces with a size < 270,000 Btu/h
- 7 systems with cooling loads less than the skin loads at 60°F outdoor temperature
- 8 systems operating < 20 hours per week
- 9 systems where outdoor air for cooling would affect open refrigeration cases
- 10 efficiency trade-off

You may trade off the economizer with an improvement in equipment efficiency. The trade-off has been expanded to include more types of mechanical cooling equipment, including applied systems, and it applies to the part load metric (or the full load metric if a part load metric does not exist.)

Table 1. Piping System Design Maximum Flow Rate in GPM (IP) (ASHRAE Standard 90.1-2010 Table 6.5.4.5)

Operating hours/yr	≤2000 hours/yr		>2000 and ≤4400 hours/yr		>4400 hours/yr	
	Other	Variable flow/ Variable speed	Other	Variable flow/ Variable speed	Other	Variable flow/ Variable speed
Nominal pipe size (in.)						
2 1/2	120	180	85	130	68	110
3	180	270	140	210	110	170
4	350	530	260	400	210	320
5	410	620	310	470	250	370
6	740	1100	570	860	440	680
8	1200	1800	900	1400	700	1100
10	1800	2700	1300	2000	1000	1600
12	2500	3800	1900	2900	1500	2300
Maximum velocity for pipes over 12" size	8.5 fps	13.0 fps	6.5 fps	9.5 fps	5.0 fps	7.5 fps

Table 1. Eliminate Required Economizer for Comfort Cooling by Increasing Cooling Efficiency (ASHRAE Standard 90.1-2010 Table 6.3.2)

Climate Zone	Efficiency Improvement ^a
2a	17%
2b	21%
3a	27%
3b	32%
3c	65%
4a	42%
4b	49%
4c	64%
5a	49%
5b	59%
5c	74%
6a	56%
6b	65%
7	72%
8	77%

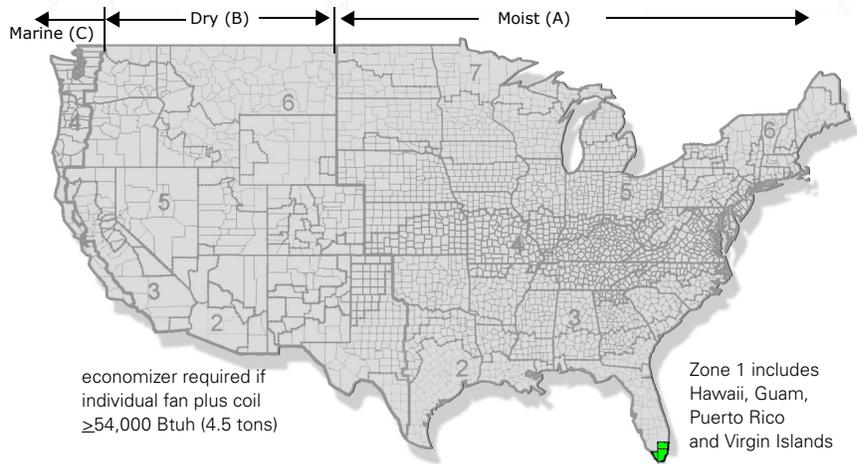
^a If a unit is rated with an IPLV, IEER or SEER then to eliminate the required air or water economizer, the minimum cooling efficiency of the HVAC unit must be increased by the percentage shown. If the HVAC unit is only rated with a full load metric like EER or COP cooling then these must be increased by the percentage shown

- 11 data center load threshold (see *Data center exceptions for economizers*)
- 12 data centers where 75 percent of the design load serves an essential facility, Tier IV design, Critical Operations Power Systems, or pending financial transactions (paraphrased)

Heat recovery economizer exception.

There are a number of exceptions for required economizers, one of which was expanded. Heat recovery is often at odds with economizer operation. This is because a number of hours when heating is needed happens when the cooling load is being shed by the economizer. You can't have all of both economizing and condenser heat recovery.

Figure 1. Economizer requirements



Systems with dedicated outdoor air, such as VRF, water-source heat pumps, large fan coils, etc., will have difficulty meeting the air economizer requirement, because the ductwork is not sized to deliver enough air to meet the full space cooling load. Most of these will fall under the 54,000 Btu/h threshold, but some will not. These systems can be designed to alternatively include heat recovery through water-to-water or refrigerant-to-water heat exchangers or water economizers.

Data center exceptions for economizers. If one of the following four situations is true, then an economizer is not required for the computer room(s) (data centers):

- 1 the total design cooling load of all computer rooms in the building is less than 3,000,000 Btu/h and the building is not served by a centralized chilled-water plant, or
- 2 the room total design cooling load is less than 600,000 Btu/h and the building in which they are located is served by a centralized chilled-water plant, or

- 3 the local water authority does not allow cooling towers, or
- 4 less than 600,000 Btu/h of computer room cooling equipment capacity is being added to an existing building. (Addendum BU)

Waterside economizer sizing for data centers. For data center designers who prefer water economizers, slightly different sizing requirements were made to reflect nearly constant year-round loads. When an economizer is required, the expected cooling load at 40°F dry bulb/ 35°F wetbulb must be met with evaporative water economizers or the expected cooling load at 35°F dry bulb must be met with dry cooler water economizers. (Addendum BU).

Table 3. Maximum Damper Leakage (ASHRAE Standard 90.1-2010 Table 6.4.3.4.4)

Climate Zone	Maximum Damper Leakage (cfm per ft ²) at 1.0 in. w.g.			
	Ventilation Air Intake		Exhaust Relief	
	non-motorized ¹	motorized	non-motorized ¹	motorized
1,2				
any height	20	4	20	4
3				
any height	20	10	20	10
4, 5b, 5c				
less than 3 stories	not allowed	10	20	10
3 or more stories	not allowed	10	not allowed	10
5a, 6, 7, 8				
less than 3 stories	not allowed	4	20	4
3 or more stories	not allowed	4	not allowed	4

¹ Dampers smaller than 24 in. in either dimension may have leakage of 40 cfm/ft²

Dampers. Outdoor air intake dampers in climates 5a, 6, 7, 8 must be motorized (Table 3), low-leak (AMCA Class 1) dampers. Gravity or backdraft dampers are permitted for all uses in climates 1 through 3, or for exhaust/relief dampers in one or two story buildings in climates 4 through 8. (Addendum CB)

Exhaust air energy recovery. Energy recovery ventilators (ERVs) are justifiable in a lot more situations, now that energy costs have risen and the component costs have come down. There are three intersecting thresholds: climate, a percentage of outdoor air, and the supply airflow rate (see Table 4). There is no ERV requirement, for any size system in any climate, if below 30 percent outdoor air at full design airflow.

There are 10 exceptions to the ERV requirement, related to situations where exhaust heat recovery is not economical or practical.

- (a) Laboratory systems meeting 6.5.7.2.
- (b) Systems serving spaces that are not cooled and that are heated to less than 60°F.
- (c) Systems exhausting toxic, flammable, paint, or corrosive fumes or dust.
- (d) Commercial kitchen hoods used for collecting and removing grease vapors and smoke.
- (e) Where more than 60 percent of the outdoor air heating energy is provided from site-recovered or site solar energy.

- (f) Heating energy recovery in climate zones 1 and 2.
- (g) Cooling energy recovery in climate zones 3c, 4c, 5b, 5c, 6b, 7, and 8.
- (h) Where the largest source of air exhausted at a single location at the building exterior is less than 75 percent of the design outdoor airflow rate.
- (i) Systems requiring dehumidification that employ energy recovery in series with the cooling coil.
- (j) Systems expected to operate less than 20 hrs per week at the outdoor air percentage covered by table 6.5.6.1.

Zone-level demand-controlled ventilation (DCV). As of 90.1-2007, zone-level DCV is required for zones with more than 40 people per 1000 square feet (100 in the 2004 version). As discussed in an earlier newsletter², several techniques are suitable for this control, depending on the space type and use.

Ventilation optimization (ventilation reset). In addition to zone-level ventilation control, there is a new requirement for system level ventilation adjustments in multiple-zone systems with DDC controls. One exception is that this control is not required if the system requires an ERV by section 6.5.6.1 (Table 4). What is *not* required, is dynamic recalculation of the critical zone, with the ability to “pop open the critical boxes” to reduce the outdoor-air (OA) fraction and reduce the overall system intake airflow. This technique is allowed under exception a2 in 6.5.3.2(a). This is sometimes referred to as limiting the “maximum Z,” based on the variable assigned to the zone OA fraction in Standard 62.1. So for those looking to exceed the minimum requirements, this is one place to do it.^{1,2} (Addendum CK)

Airside

In addition to single-zone VAV, there are several new or modified requirements for airside controls.

Supply air reset. A requirement for supply air temperature (SAT) reset was removed from 90.1 in the 1999 version, but has returned in the 2010 version. Part of the ambivalence 90.1 has shown for this requirement is the overall impact of increase in fan energy compared to the decrease in reheat energy.

The goal is to set up the SAT reset strategy so that it gradually resets to full reset at a point when most zones have minimal cooling loads, and the additional fan energy will not overwhelm the reheat savings. A simple approach for doing this was mentioned in a recent newsletter³. The outdoor air temperature based strategy begins resetting the supply air setpoint when it is 70°F outdoors, with 1°F of reset per 2°F additional reduction in outdoor air temperature until the maximum reset supply air setpoint is reached.

Table 4. Energy Recovery Requirement (IP) (ASHRAE Standard 90.1-2010 Table 6.5.6.1)

Zone	% Outdoor air at full design airflow rate					
	≥ 30% and < 40%	≥ 40% and < 50%	≥ 50% and < 60%	≥ 60% and < 70%	≥ 70% and < 80%	≥ 80%
	Design Supply Fan airflow rate (cfm)					
3B, 3C, 4B, 4C, 5B	NR	NR	NR	NR	≥ 5000	≥ 5000
1B, 2B, 5C	NR	NR	≥ 26000	≥ 12000	≥ 5000	≥ 4000
6B	≥ 11000	≥ 5500	≥ 4500	≥ 3500	≥ 2500	≥ 1500
1A, 2A, 3A, 4A, 5A, 6A	≥ 5500	≥ 4500	≥ 3500	≥ 2000	≥ 1000	≥ 0
7, 8	≥ 2500	≥ 1000	≥ 0	≥ 0	≥ 0	≥ 0

Above 70°F, the outdoor air provides little or no cooling benefit for economizing, and the cooling load in most zones is likely high enough that reheat is not required to prevent sub-cooling the space. Additionally, the colder supply-air temperature allows the system to provide sufficiently dry air to the zones, improving part-load dehumidification.

Dual maximum control on VAV boxes.

This change is aimed at avoiding the need to increase zone outdoor airflow due to temperature stratification when supplying hot air from ceiling-mounted diffusers and returning through ceiling-mounted grilles. A new option was added for the maximum amount of air that can be cooled and then reheated (Figure 2). The new control option requires that primary airflow be reduced to 20 percent of design cooling airflow, but then allows primary airflow to be modulated up to a maximum of 50 percent of design airflow for heating. This allows heating to be accomplished with a lower discharge-air temperature, which improves zone air-distribution effectiveness. (Addendum H)

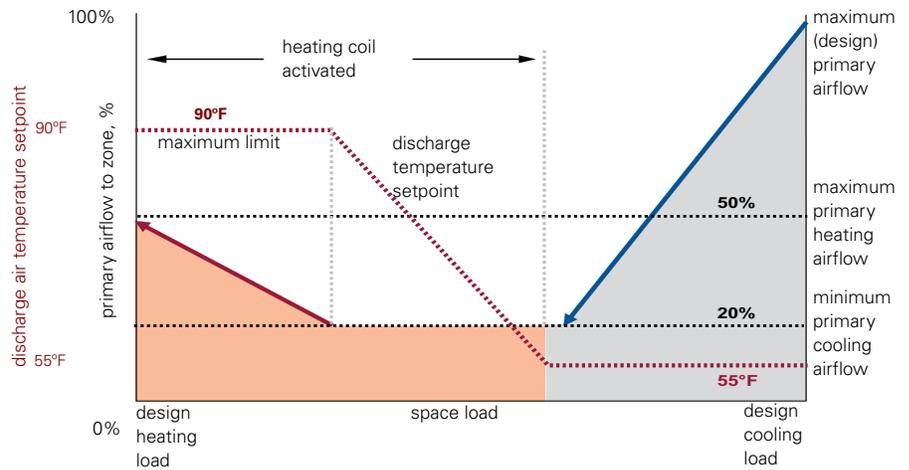
You may find the new, “dual maximum” control necessary for complying with the next requirement. A previous engineers newsletter¹ explains this control in detail.

Overhead heating temperature limit. This change limits the zone discharge-air temperature to no more than 20°F above the zone thermostat setpoint (with exceptions for morning warm-up) when supplying hot air from diffusers located above the breathing zone, and returning through grilles located above the breathing zone. This minimizes temperature stratification and short circuiting to increase zone air distribution effectiveness.

The new dual maximum control strategy may be used so that airflow can be increased in heating mode. (Addendum BX)

VAV control on lab exhaust systems. In the same section as the dual maximums, there used to be an exception for spaces like hospitals and labs where the pressure control requirements set the minimum stop on the VAV box (the maximum amount of

Figure 2. Dual maximum control option for zone heating



air that can be reheated). This blanket exception was removed, and a new section on laboratory exhaust specifies the effectiveness and control needed to make sure that these zones are not maintained at constant flow when the air is not needed for pressure control. (Addendum AS)

VAV requirements for chilled water air-handling units serving a single-zone. As previously mentioned on page 2 for packaged systems (effective January 2012), and as of January 2010 for chilled water systems, variable-air-volume fan control is required for single-zone systems. This applies to chilled-water air handlers with motors 5 hp or greater. This change requires either two-speed motors or variable-speed drives on the supply fan(s), and may invoke the need for discharge air temperature sensors and outdoor airflow measurement. (Addendum N)

Fan power limitation. When the single-zone VAV change occurred, the 90.1 committee realized it needed to revise the definition of which allowable fan horsepower applies to VAV systems that don't have zone dampers increasing the system pressure. Single-zone VAV systems are required to use the more stringent constant volume fan power limit.

Allowances for exhaust systems were previously poorly defined. This created a hardship for several types of buildings and specifically laboratories, hospitals, and vivariums. The blanket exception to the fan power limitation for fume hoods has been removed. Other changes relate to credits for exhaust and return systems, provided the system complies

with the requirements (see “VAV control on lab exhaust systems”).

Heat recovery pressure drop adjustments . Within the fan power limitation section there is a new pressure drop credit equation for ERVs that compensates for effectiveness. A higher pressure drop adjustment is allowed for more effective energy recovery, reflecting the balance between fan energy and heat recovered. Coil runaround loops have their own pressure drop adjustment on the system fan power limit:

- ERV pressure drop adjustment: (2.2 x Energy Recovery Effectiveness) – 0.5 in w.c. for each air stream
- Coil runaround loops are given a pressure drop adjustment of 0.6 in. w.g. per air stream

Other Changes

Several changes to the mechanical section were not explicitly covered in this newsletter. They relate to:

- Elevator lighting and ventilation (Addendum DF)
- Garage ventilation controls (Add. DI)
- Duct leakage —class A (Add. CQ)
- Kitchen exhaust hoods (Add. AX)
- Radiant panels (Add. AE)
- Heat pump pool heaters (Add. Y)
- Furnaces & water heating (Add. K, AO)

Summary

When the standard is published this fall, it is intended to be adopted directly by energy code jurisdictions, and by reference in IECC, NFPA, and other model codes.

Preliminary results of modeling performed by Pacific Northwest National Laboratories (PNNL) were shared at the June 2010 annual meeting. Not all approved addenda were included at that time.

In addition to the changes in 90.1, there were ventilation rate changes in ASHRAE Standard 62.1. These are included in the modeling.

From a whole building perspective, including receptacle loads (for example copiers, vending machines, etc.), the energy cost savings is over 23 percent and the energy savings is almost 25 percent, when compared to reference buildings meeting 90.1-2004.

Until the change to the Title, Purpose, and Scope made by Addendum AQ, the committee could not address loads from receptacles or processes. If receptacle thermal loads are included in the model, but not their electrical consumption, the savings estimates rise to almost 29 percent for cost and 31 percent for energy.

In addition, there are a significant number of addenda that have yet to be modeled, so the final saving estimates are expected to be even greater. Congratulations to all who worked on 90.1-2010.

By Susanna Hanson, applications engineer and Jeanne Harshaw, program manager, Trane. Susanna was a member of the 90.1 Mechanical Subcommittee throughout the development of 90.1-2010. You can find this and previous issues of the Engineers Newsletter at www.trane.com/engineersnewsletter. To comment, e-mail us at comfort@trane.com.



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Trane now offers online courses for LEED credential maintenance and AIA required learning units at no charge

Since becoming a U.S. Green Building Council (USGBC) Education Provider in February 2010, Trane has been working to make Leadership in Energy and Environmental Design (LEED) credential maintenance courses accessible online. The newly developed course pages allow LEED professionals to more easily complete the continuing education required to maintain credentials and stay competitive in the sustainable building industry.

The courses were developed and are offered free of charge to demonstrate Trane's commitment to sustainable design. LEED Accredited Professionals (APs) and AIA members can participate and earn an average of 1.5 Continuing Education (CE) hours per program.

Visit www.trane.com/continuingeducation to view current courses and details.

Trane believes the facts and suggestions presented here to be accurate. However, final design and application decisions are your responsibility. Trane disclaims any responsibility for actions taken on the material presented.

Engineers Newsletter **LIVE!**

To register, contact your
local Trane office.

ASHRAE Standard 90.1-2010

This program will discuss
major envelope, mechanical,
lighting, and modeling addenda
that will be incorporated into
90.1-2010.

October 2010

New application manuals now available

Central Geothermal Design and Control. (SYS-APM009-EN, April 2010)

Chilled-Water VAV Systems. (SYS-APM008-EN, August 2009)

Multiple Chiller System Design and Control. (SYS-APM001-EN, May 2009)

Visit www.trane.com/bookstore to order and view a complete list of resources.