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# 10. Covered Processes

## 10.1 Introduction

This chapter of the Nonresidential Compliance Manual addresses covered processes for the *2016 Building Energy Efficiency Standards* (Energy Standards) (§120.6 and §140.9).

### 10.1.1 Organization and Content

This chapter is organized as follows:

- 10.1 - Introduction to Covered Processes
- 10.2 - Enclosed Parking Garages
- 10.3 - Commercial Kitchens
- 10.4 - Computer Rooms
- 10.5 - Commercial Refrigeration
- 10.6 - Refrigerated Warehouses
- 10.7 - Laboratory Exhaust
- 10.8 - Compressed Air Systems
- 10.9 - Process Boilers
- 10.10 - Elevator Lighting and Ventilation Controls
- 10.11 - Escalators and Moving Walkways Speed Controls

### 10.1.2 Compliance Forms Checklist

Compliance documentation includes the Certificates of Compliance, reports and other information that are submitted to the enforcement agency with an application for a building permit. Compliance documentation also includes documentation completed by the installing contractor, engineer/architect of record, or owner's agent to verify that certain systems and equipment have been correctly installed and commissioned.

To assist in submitting proper compliance documentation, a checklist is available (Process Compliance Forms & Worksheets - Page 1 of NRCC-PRC-01-E). This checklist helps the enforcing agency to look for specific compliance documentation that is required for covered processes by the 2016 Energy Standards.

Under the prescriptive compliance approach, the project designer is responsible for completing the Process Compliance Forms & Worksheets. The project designer is required to check all applicable boxes on the NRCC-PRC-01-E on page 1 and furnishes all associated forms relevant to the checked features. This checklist is required on plans for all submittals with covered processes. For the performance compliance approach this form will automatically be completed by the approved computer compliance program. The following are the compliance forms:

- NRCC-PRC-01-E Certificate of Compliance - Process Compliance Forms & Worksheets (required on plans for all submittals with covered processes)
- NRCC-PRC-02-E Certificate of Compliance - Garage Exhaust

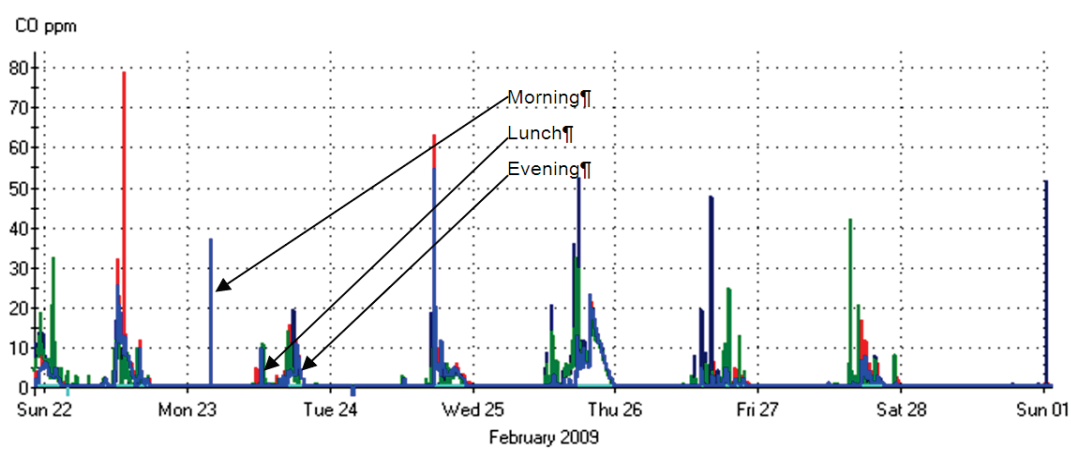
- NRCC-PRC-03-E Certificate of Compliance - Commercial Kitchen Requirements
- NRCC-PRC-04-E Certificate of Compliance - Computer Room Requirements
- NRCC-PRC-05-E Certificate of Compliance - Commercial Refrigeration
- NRCC-PRC-06-E Certificate of Compliance - Refrigerated Warehouse
- NRCC-PRC-07-E Certificate of Compliance - Refrigerated Warehouse ≥ 3,000 $ft^2$
- NRCC-PRC-08-E Certificate of Compliance - Refrigerated Warehouse ≥ 3,000 $ft^2$ Served by the Same Refrigeration System
- NRCC-PRC-09-E Certificate of Compliance - Laboratory Exhaust
- NRCC-PRC-10-E Certificate of Compliance - Compressed Air System
- NRCC-PRC-11-E Certificate of Compliance - Process Boilers Requirements
- NRCC-PRC-12-E Certificate of Compliance – Elevator Lighting and Ventilation Controls
- NRCC-PRC-13-E Certificate of Compliance - Escalators and Moving Walkways Speed Controls

## 10.2 Enclosed Parking Garages

### 10.2.1 Overview

Garages exhaust systems are sized to dilute the auto exhaust at peak conditions to an acceptable concentration for human health and safety. Energy management control system (EMCS) monitoring of garage carbon monoxide (CO) concentrations show that in a typical enclosed garage, there are three periods of concern: in the morning when cars enter the garage; during the lunch break when cars leave and reenter; and at the end of the day when cars leave. This mandatory measure requires modulating ventilation airflow in large enclosed parking garages based on pollutant concentrations. By modulating airflow based on need rather than running constant volume, the system will save energy and maintain a safe environment.

**Figure 10-1: Garage CO Trends**



### 10.2.2 Mandatory Measures

For garage exhaust systems with a total design exhaust rate ≥10,000 cubic feet per minute (cfm), §120.6(c) requires automatic controls to modulate airflow to ≤50% of design based on measurements of the contaminant concentrations. This includes:

- Minimum fan power reduction of the exhaust fan energy to ≤ 30% of design wattage at 50% of design flow. A two speed or variable speed motor can be used to meet this requirement.
- CO concentration shall be measured with at least one sensor per 5,000 $ft^2$ with each sensor located where the highest concentrations of CO are expected.
- CO concentration of 25 ppm or less as control set point at all times.
- A minimum ventilation of 0.15 $cfm/ft^2$ when the garage is "occupied".
- The system must maintain the garage at a neutral or negative pressure with respect to adjacent occupiable areas when the garage is scheduled to be occupied.
- CO sensors shall be certified to the minimum performance requirements listed under §120.6(c) of the Energy Standards.
- Acceptance testing of the ventilation system per NA7.12.

#### 10.2.2.1 Minimum Fan Power Reduction

*§120.6(c)2*

Where required, the fan control must be designed to provide ≤30% of the design fan wattage at 50% of the fan flow. This can be achieved by either a two speed motor or a variable speed drive.

#### 10.2.2.2 CO Sensor Number and Location

*§120.6(c)3*

CO sensors (or sampling points) must be located so that each unique sensor serves an area no more than 5,000 $ft^2$. Furthermore, the standard requires a minimum of two sensors per "*proximity zone*." A *proximity zone* is defined as areas that are separated by obstructions including floors or walls.

The typical design for garage exhaust is to have the exhaust pickups located on the other side of the parking areas from the source of makeup air. The ventilation air sweeps across the parking areas and towards the exhaust drops. Good practice dictates that you would locate sensors close to the exhaust registers or in dead zones where air is not between the supply and exhaust. Floors and rooms separated by walls should be treated as separate proximity zones.

#### 10.2.2.3 CO Sensor Minimum Requirements

*§120.6(c)7*

To comply, each sensor must meet all of the following requirements:

- Certified by the manufacturer to: be accurate to +/- 5%; have 5% or less drift per year; and require calibration no more than once per year.
- Be factory calibrated.

- The control system must monitor for sensor failure. If sensor failure is detected the control system must reset to design ventilation rates and transmit an alarm to the facility operators.

### 10.2.3 Prescriptive Measures

There are no prescriptive measures for enclosed garage exhaust.

### 10.2.4 Additions and Alterations

There are no separate requirements for additions and alterations.

## 10.3 Commercial Kitchens

### 10.3.1 Overview

There are four energy saving measures associated with commercial kitchen ventilation. These four prescriptive measures address:

- Direct Replacement of Exhaust Air Limitations.
- Type I Exhaust Hood Airflow Limitations.
- Makeup and Transfer Air Requirements.
- Commercial Kitchen System Efficiency Options.

### 10.3.2 Mandatory Measures

There are no mandatory measures specific to commercial kitchens. Installed appliances and equipment must meet the mandatory requirements of §110.1 and §110.2, respectively.

### 10.3.3 Prescriptive Measures

#### 10.3.3.1 Kitchen Exhaust Systems

*§140.9(b)1*

This section addresses kitchen exhaust systems. There are two requirements for kitchen exhaust:

- A limitation on use of short-circuit hoods §140.9(b)1A.
- Maximum exhaust ratings for Type I kitchen hoods §140.9(b)1B.

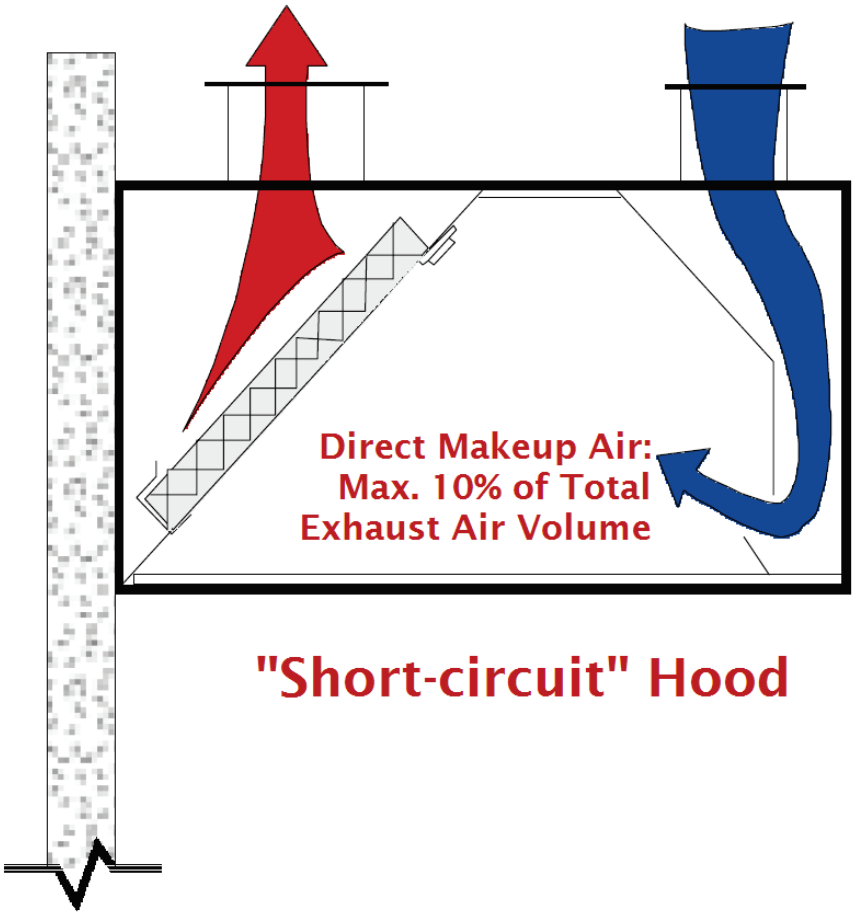
**A. Limitation of Short-Circuit Hoods**

*§140.9(b)1A*

Short-circuit hoods are limited to ≤10% replacement air as a percentage of hood exhaust airflow rate. The reasons for this include the following:

Studies by Pacific Gas & Electric (PG&E), American Gas Association (AGA) and the Energy Commission have shown that in short-circuit hoods, direct supply greater than 10% of hood exhaust significantly reduces capture and containment. This reduces the extraction of cooking heat and smoke from the kitchen, forcing facilities to increase the hood exhaust rate. This results in higher consumption of energy and conditioned makeup air.

**Figure 10-2: Short-Circuit Hood**




### B. Maximum Exhaust Ratings for Type I Kitchen Hoods

*§140.9(b)1B*

The Energy Standards also limit the amount of exhaust for Type I kitchen hoods based on Table 140.9-A (Table 10-1 below), when the total exhaust airflow for Type I and II hoods are greater than 5,000 cfm. Similar to the description regarding short-circuit hoods, excessive exhaust rates for Type I kitchen hoods increases energy consumption and increases energy use for conditioning of the makeup air.

There are two exceptions for this requirement:

1. Exception 1 to §140.9(b)1B: where ≥75% of the total Type I and II exhaust makeup air is transfer air that would otherwise have been exhausted. This exception could be used when you have a large dining area adjacent to the kitchen, which would be exhausting air for ventilation purposes even if the hoods were not running. The exception is satisfied if: the air that would otherwise have been exhausted from the dining area (to meet ventilation requirements); is greater than 75 percent of the hood exhaust rate; and is transferred to the kitchen for use as hood makeup air.
2. Exception 2 to §140.9(b)1B: for existing hoods that are not being replaced as part of an addition or alteration.

The values in Table 140.9-A are based on the type of hood (left column) and the rating of the equipment that it serves (columns 2 through 5). The values in this table are typically

less than the minimum airflow rates for unlisted hoods. These values are supported by ASHRAE research for use with listed hoods (RP-12002). To comply with this requirement, the facility will likely have to use listed hoods. The threshold of 5,000 cfm of total exhaust was included in the Energy Standards to exempt small restaurants.

The definitions for the types of hoods and the duty of cooking equipment are provided in ASHRAE Standard 154-2011.

**Table 10-1: Maximum Net Exhaust Flow Rate, cfm per Linear Foot of Hood Length**

| Type of Hood | Light Duty Equipment | Medium Duty Equipment | Heavy Duty Equipment | Extra Heavy Duty Equipment |
|---|---|---|---|---|
| Wall-mounted Canopy | 140 | 210 | 280 | 385 |
| Single Island | 280 | 350 | 420 | 490 |
| Double Island | 175 | 210 | 280 | 385 |
| Eyebrow | 175 | 175 | Not Allowed | Not Allowed |
| Backshelf/Pass-over | 210 | 210 | 280 | Not Allowed |

*Energy Standards Table 140.9-A*

### 10.3.3.2 Kitchen Ventilation

*§140.9(b)2*

This section covers two requirements:

- Limitations to the amount of mechanically heated or cooled airflow for kitchen hood makeup air §140.9(b)2A.
- Additional efficiency measures for large kitchens §140.9(b)2B.

For these requirements it is important to understand the definition of mechanical cooling and mechanical heating, which the Energy Standards define as:

- ***Mechanical cooling*** is lowering the temperature within a space using refrigerant compressors or absorbers, desiccant dehumidifiers, or other systems that require energy from depletable sources to directly condition the space. In nonresidential, high-rise residential, and hotel/motel buildings, cooling of a space by direct or indirect evaporation of water alone is not considered mechanical cooling.
- ***Mechanical heating*** is raising the temperature within a space using electric resistance heaters, fossil fuel burners, heat pumps, or other systems that require energy from depletable sources to directly condition the space.

It is important to note that direct and indirect evaporation of water alone is not considered mechanical cooling. Therefore, air cooled by the evaporation of water can be used as kitchen hood makeup air with no restrictions.

#### A. Limitations to the Amount of Mechanically Heated or Cooled Airflow for Kitchens

*§140.9(b)2A*

This section limits the amount of mechanically cooled or heated airflow to any space with a kitchen hood. The amount of mechanically cooled or heated airflow must not exceed either:

- The supply flow required to meet the space heating or cooling load.
- The hood exhaust minus the available transfer air from adjacent spaces.

The supply flow required to meet the space heating or cooling loads can be documented by providing the load calculations.

To calculate the available transfer air:

1. Calculate the minimum outside air (OA) needed for the spaces that are adjacent to the kitchen.
2. From the amount calculated in 1, subtract the amount of air used by exhaust fans in the adjacent spaces. These include toilet exhaust and any hood exhaust in adjacent spaces.
3. From the amount calculated in 2, subtract the amount of air needed for space pressurization. The remaining air is available for transfer to the hoods.

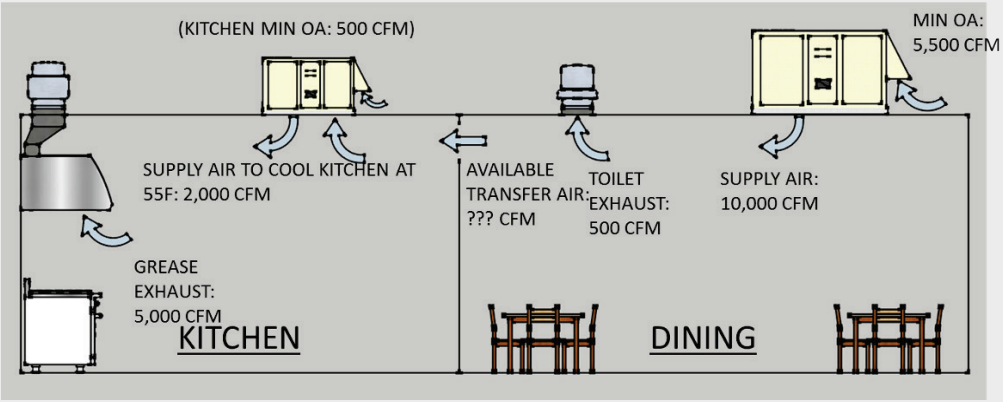
An exception is provided for existing kitchen makeup air units (MAU) that are not being replaced as part of an addition or alternation.

While the requirement to use available transfer air only refers to "adjacent spaces", available transfer air can come from any space in the same building as the kitchen. A kitchen on the ground floor of a large office building, for example, can draw transfer air from the return plenum and the return shaft. The entire minimum OA needed for the building, minus the other exhaust and pressurization needs, is available transfer air. If the return air path connecting the kitchen to the rest of the building is constricted, resulting in high transfer air velocities, then it may be necessary to install a transfer fan to assist the transfer air in making its way to the kitchen. The energy use of a transfer fan is small compared to the extra mechanical heating and cooling energy of an equivalent amount of OA.

Example 10-1

**Question**

What is the available transfer air for the kitchen make up in the scenario shown in the following figure?




**Answer**

5,000 cfm calculated as follows.

The OA supplied to the dining room is 5,500 cfm. From this we subtract 500 cfm for the toilet exhaust and 0 cfm for building pressurization.

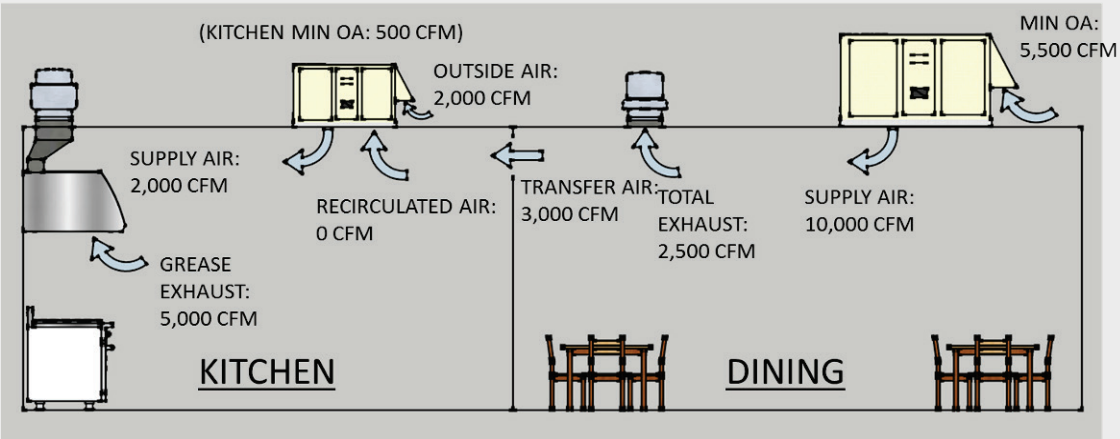
5,500 cfm – 500 cfm – 0 cfm = 5,000 cfm

The remaining 5,000 cfm of air is available transfer air.

Example 10-2

**Question**

Assuming that this kitchen needs 2,000 cfm of supply air to cool the kitchen with a design supply air temperature of 55°F, would the following design airflow meet the requirements of §140.9(b)2A?




**Answer**

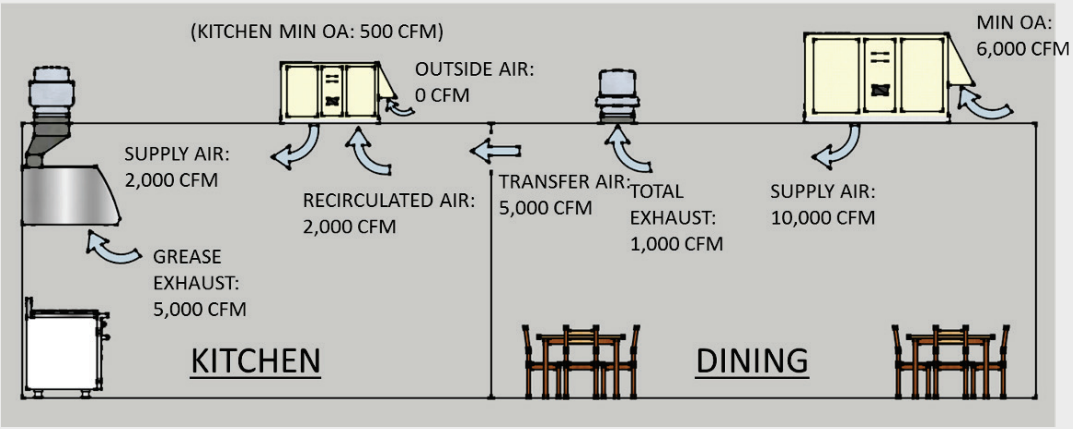
Yes. This example meets the first provision of §140.9(b)2A. The supply flow required to meet the cooling load is 2,000 cfm. Thus up to 2,000 cfm of mechanically conditioned make up air can be provided to the kitchen. Note that the supply from the MAU, 2,000 cfm, is not as large as the hood exhaust, 5,000 cfm. This means that the remainder of the makeup air, 3,000 cfm, must be transferred from the dining room space.

Although this is allowed under §140.9(b)2Ai, this is not the most efficient way to condition this kitchen as demonstrated in the next example.

Example 10-3

**Question**

Continuing with the same layout as the previous example, would the following design airflow meet the requirements of §140.9(b)2A?




**Answer**

Yes. In this example, 100% of the makeup air, 5,000 cfm, is provided by transfer air from the adjacent dining room. Note that the OA on the unit serving the dining room has been increased to 6,000 cfm to serve the ventilation for both the dining room and kitchen. Since the dining room has no sources of undesirable contaminants we can ventilate the kitchen with the transfer air.

Comparing this image to the previous example you will see that this design is more efficient for the following reasons:

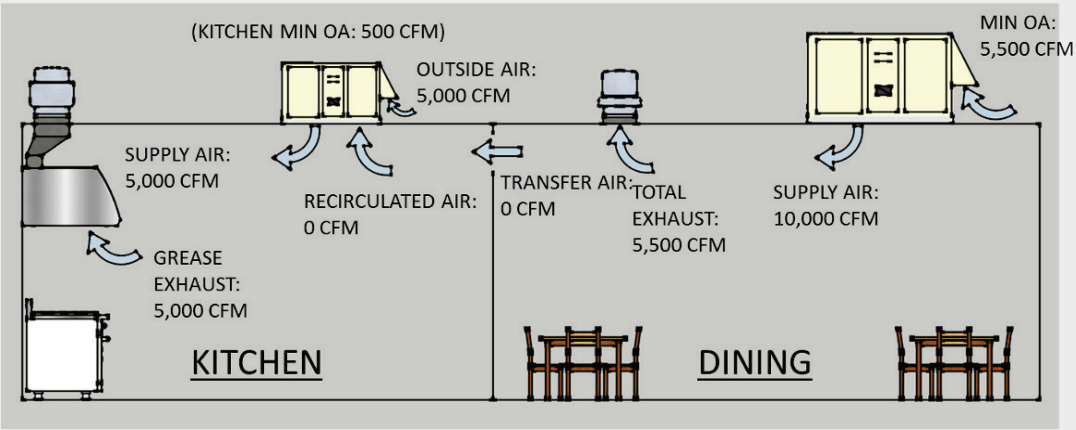
1. The total outside airflow to be conditioned has been reduced from 7,500 cfm in the previous example (2,000 cfm at the MAU and 5,500 cfm at the dining room unit) to 6,000 cfm; and
2. The dining room exhaust fan has dropped from 2,500 cfm to 1,000 cfm reducing both fan energy and first cost of the fan.

Note that an even more efficient design would be if the kitchen MAU had a modulating OA damper that allowed it to provide up to 5,000 cfm of outside air directly to the kitchen when OA temperature <kitchen space temperature. When OA temperature >kitchen space temperature then the OA damper on the MAU is shut and replacement/ventilation air is transferred from the dining area. This design requires a variable speed dining room exhaust fan controlled to maintain slight positive pressure in the dining area. This design is the baseline design modeled in the Alternative Calculation Methods (ACM) Reference Manual for performance compliance. The baseline model assumes that transfer air is available from the entire building, not just the adjacent spaces.

Example 10-4

**Question**

Continuing with the same layout as the previous examples, would the following design airflow meet the requirements of §140.9(b)2A?




**Answer**

Not if the kitchen unit is mechanically heated or cooled. Per §140.9(b)2A the maximum amount of air that can be mechanically heated or cooled must be less than either:

1. Per §140.9(b)2Ai: 2,000 cfm, the supply needed to cool the kitchen (from Example 10-2)
2. Per §140.9(b)2Aii: 0 cfm, the amount of hood exhaust (5,000 cfm) minus the available transfer air (also 5,000 cfm from Example 10-2).

### B. Additional Efficiency Measures for Large Kitchens

*§140.9(b)2B*

For kitchens or dining facilities that have more than 5,000 cfm of Type I or II hood exhaust, the mechanical system must meet one of the following requirements:

1. At least 50% of all replacement air is transfer air that would have been exhausted.
2. Demand ventilation control on at least 75% of the exhaust air.
3. Listed energy recovery devices with a sensible heat recovery effectiveness ≥40% on ≥50% of the total exhaust flow.

4. ≥75% of the makeup air volume that is:
   a. Unheated or heated to no more than 60°F.
   b. Uncooled or cooled without the use of mechanical cooling.

**Transfer Air:** There is an exception for existing hoods not being replaced as part of an addition or alteration.

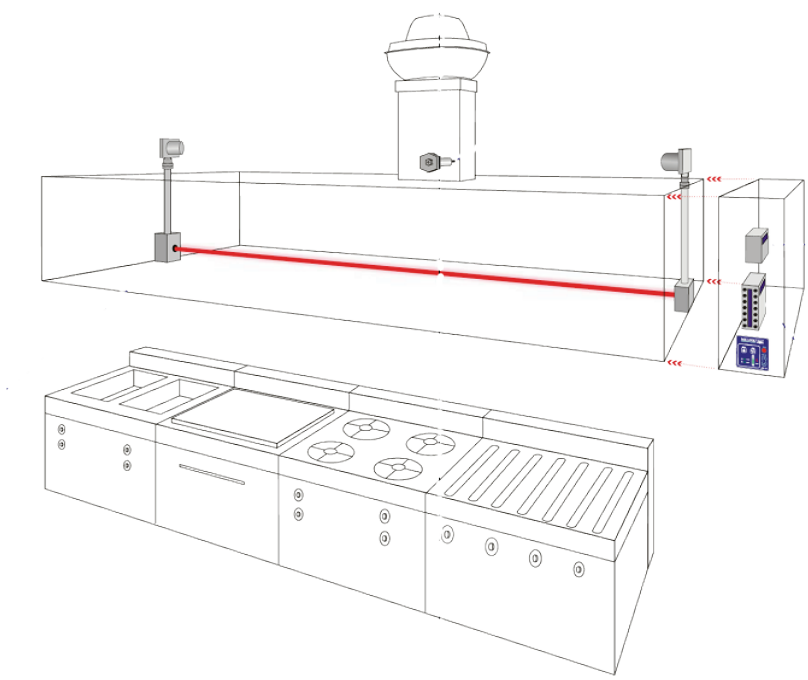
The concept of transfer air was addressed in the discussion of §140.9(b)2A above.

**Demand Ventilation Control:** Per §140.9(b)2Bii, demand ventilation controls must have all of the following characteristics:

- Include controls necessary to modulate airflow in response to appliance operation and to maintain full capture and containment of smoke, effluent and combustion products during cooking and idle.
- Include failsafe controls that result in full flow upon cooking sensor failure.
- Include an adjustable timed override to allow occupants the ability to temporarily override the system to full flow.
- Be capable of reducing exhaust and replacement air system airflow rates to the larger of:
  - 50% of the total design exhaust and replacement air system airflow rates.
  - The ventilation rate required per §120.1.

There are several off the shelf technologies that use smoke detectors that can comply with all of these requirements.

**Figure 10-3: Demand Control Ventilation Using a Beam Smoke Detector**



**Energy Recovery:** Energy recovery is provided using air to air heat exchangers between the unit providing makeup air and the hood exhaust. This option is most effective for extreme climates (either hot or cold). It is not commonly used in the mild climates of California.

**Tempered Air with Evaporative Cooling:** The final option is to control the heating (if there is heating) to a space by setting the temperature setpoint to 60°F and to use evaporative (non-compressor) cooling or no cooling at all for most of the makeup air.

#### 10.3.3.3 Kitchen Exhaust Acceptance

*§140.9(b)3*

Acceptance tests for these measures are detailed in NA7.11. See Chapter 13 of this manual.

### 10.3.4 Additions and Alterations

The application of these measures to additions and alterations was presented in the text from the previous section.

## 10.4 Computer Rooms

### 10.4.1 Overview

§140.9(a) provides minimum requirements for conditioning of *computer rooms*. A *computer room* is defined in §100.1 Definitions as " a room whose primary function is to house electronic equipment and that has a design equipment power density exceeding 20 watts/ft$^2$ of conditioned floor area." All of the requirements in §140.9(a) are prescriptive.

### 10.4.2 Mandatory Measures

There are no mandatory measures specific to computer rooms. The equipment efficiencies in §110.1 and §110.2 apply.

### 10.4.3 Prescriptive Measures

The following is a summary of the measures in this section:

- Air or water side economizer - §140.9(a)1
- Restriction on reheat or recool - §140.9(a)2
- Limitations on the type of humidification - §140.9(a)3
- Fan power limitations - §140.9(a)4
- Variable speed fan control - §140.9(a)5, and
- Containment - §140.9(a)6

Each of these requirements is elaborated on in the following sections.

#### 10.4.3.1 Economizers

*§140.9(a)1*

This section requires integrated air or water economizer. If an air economizer is used to meet this requirement, it must be designed to provide 100% of the expected system cooling load at outside conditions of 55°F Tdb with a coincident 50°F Twb. This is different from the non-computer room economizer regulations (§140.4(e)), which require that an air economizer must supply 100 percent of the supply-air as outside air. A computer room air economizer does not have to supply any outside air if it has an air-to-air heat exchanger that

can meet the expected load at the conditions specified and can be shown (through modeling) to consume no more energy that the standard air economizer.

If a water economizer is used to meet the is requirement it must be capable of providing 100% of the expected system cooling load at outside conditions of 40°F Tdb with a coincident 35°F Twb.

See chapter 4 for a description of integrated air- and water-economizers and implementation details.

There are several exceptions to this requirement:

1. **Exception 1 to §140.9(a)1**: Individual *computer rooms* with cooling capacity of <5 tons in a building that does not have any economizers. This exception is different from the 54,000 btu-h (4.5 ton) exception in 140.4(e) for non-computer rooms in two important ways. First, the computer room exception refers to the cooling capacity of all systems serving the computer room, whereas the non-computer room regulation refers to each cooling system serving the building. Second, the computer room exception only applies if none of the other cooling systems in the building includes an economizer. Even a 1 ton computer room would have to be served by an existing cooling system with an economizer (see exception 4 below). The analysis for this requirement was performed using a 5 ton AC unit with an air/air heat exchanger. Even with the added cost and efficiency loss of a heat exchanger the energy savings in all of the California climates justified this requirement.
2. **Exception 2 to §140.9(a)1**: New cooling systems serving an existing c*omputer room* in an existing building up to a total of 50 tons of new cooling equipment per building. This exception permits addition of new IT equipment to an existing facility that was originally built without any economizers.

   This exception recognizes that an existing space with capacity for future expansion may not have been sited or configured to accommodate access to outside air.

   Above 50 tons of capacity (~175kW of IT equipment load) you would be forced to either provide economizer cooling or offset the energy loss by going the performance approach. Ways to meet this requirement include:

   - Provide the new capacity using a new cooling system that has a complying air or water economizer, or
   - If the facility has a chilled water plant install an integrated water-side economizer with a minimum capacity equal to the new *computer room* cooling load. Water-side economizers can be added to both air and water cooled chilled water plants.
3. **Exception 3 to §140.9(a)1**: New cooling systems serving a new c*omputer room* up to a total of 20 tons of new cooling load in an existing building.

   This is similar to the previous exception but now you have the option to plan the new space in a location where you can employ a system with an integrated economizer.
4. **Exception 4 to §140.9(a)1**: Applies to *computer rooms* in a larger building with a central air handling system with a complying air-side economizer that can fully condition the c*omputer rooms* on weekends and evenings when the other building loads are off. This exception allows the *computer rooms* to be served by fan-coils or split system DX units as long as the following conditions are met:

   - The economizer system on the central air handling unit is sized sufficiently that all of the *computer rooms* are less than 50% of its total airflow capacity.

- The central air handling unit is configured to serve only the *computer rooms* if all of the other loads are unoccupied. And,
- The supplemental cooling systems for the *computer rooms* are locked out when the outside air drybulb temperature is below 60°F and the non-*computer room* zones are less than 50% of the design airflow.

Example 10-5

**Question**

A new data center is built with a total c*omputer room* load of 1,500 tons of capacity. If the *computer rooms* are all served using recirculating chilled water *computer room* air-handling units (CRAHs) in in-row air handling units (IRAHs) would this data center meet the requirements of §140.9(a)1 if the chilled water plant had a water side economizer that complied with the requirements of §140.4(e)?

**Answer**

Yes, if the economizer can meet 100 percent of the 1,500 ton load at 40°F dry-bulb and 35°F wet-bulb. The design conditions in §140.9(a) would require a different heat exchanger and cooling towers than the conditions in §140.4(e) for non-process spaces for a given expected load. Note that the load on the cooling towers, while in economizer-only mode, is lower that the design load even if the computer room load is constant because the towers do not have to reject the heat from the chillers. Note also that there are no redundancy requirements in the energy code. Many data centers have more cooling towers than needed to meet the design load so that the design load can be met even if one or more towers is not available. If the system is capable of running all cooling towers in economizer-only mode than all towers can be included in the calculation for determining compliance with this requirement.

Example 10-6

**Question**

A new data center is built with chilled water CRAH units sized to provide 100% of the cooling for the IT equipment. The building also has louvered walls that can open to bring in outside air and fans on the roof that can exhaust air. Does this design meet the requirements of §140.9(a)1?

**Answer**

Yes provided that all of the following are true:

- The economizer system moves sufficient air so that it can fully satisfy the design IT equipment loads with the CRAH units turned off and the outside air dry-bulb temperature at 55°F. And,
- The control system provides integrated operation so that the chilled water coils in the CRAH units are staged down when cool outside air is brought into the data center. And,
- The economizer system is provided with a high limit switch that complies with §140.4(e). Although fixed dry-bulb switches are allowed in §140.4(e) they are not recommend in this application as the setpoints were based on office occupancies. A differential dry-bulb switch would provide much larger energy savings.

Example 10-7

**Question**

A new office building has a house air system with an air-side economizer that complies with §140.4(3). This building has two IDF rooms with split system DX units one is 4 tons of capacity and the other is 7-1/2 tons of capacity. Do the IDF rooms meet the requirements of the Energy Standards?

**Answer**

Not necessarily. Both IDF rooms are required to have economizers. The 4 ton IDF room does not meet Exception 1 to §140.9(a)1 because it is a building with an economizer. Per Exception 4 to §140.9(a)1 the IDF rooms can also be served by split system DX units without economizers if if they are also served by VAV boxes from the VAV reheat system. The DX units must be off when the VAV reheat system has enough spare capacity to meet the IDF loads. The VAV reheat system must be at least twice the capacity of all the IDF rooms. When the office spaces are expected to be unoccupied (e.g., at night) their VAV boxes must be shut so that the VAV system can serve only the IDF rooms.

Example 10-8

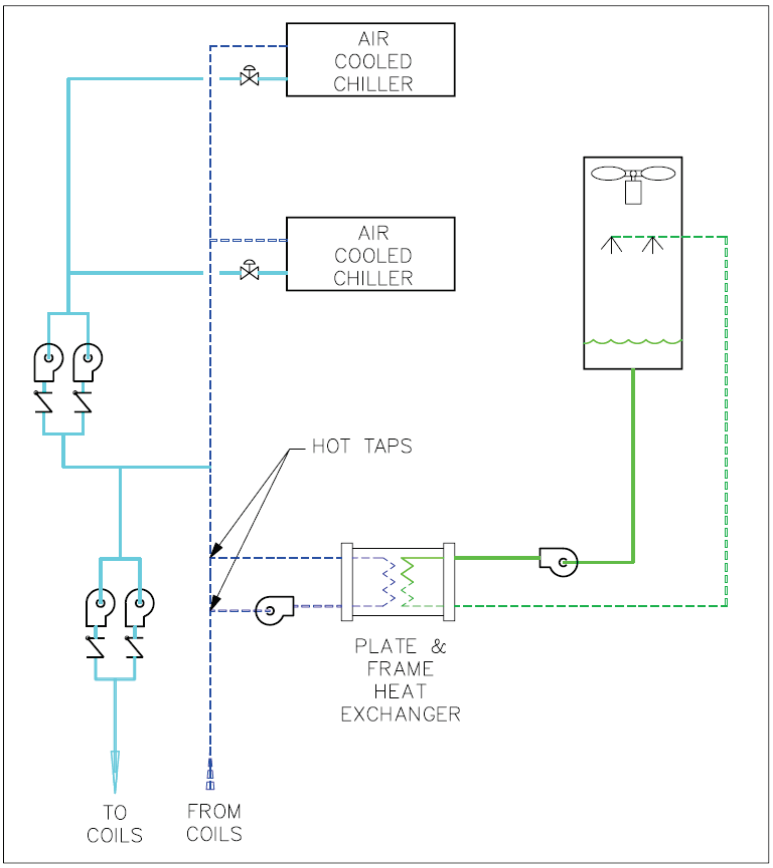
**Question**

A new data center employs rear door heat exchangers that are cooled entire with water that comes from a closed circuit fluid cooler. Does this design meet the economizer requirements of §140.9(a)1?

**Answer**

Yes. The standard definitions for e*conomizer* (both air and water) both have the phrase "to reduce or eliminate the need for m*echanical cooling*." In turn, the definition of *mechanical cooling* is "lowering the temperature within a space using refrigerant compressors or absorbers, desiccant dehumidifiers, or other systems that require energy from depletable sources to directly condition the space." Since this system does not use compressors, it complies.

**Figure 10-4: Example of water-side economizer retrofit on a chilled water plant with air-cooled chillers**



#### 10.4.3.2 **Reheat/Recool**

*§140.9(a)2*

§140.9(a)2 prohibits reheating, recooling or simultaneous heating or cooling in c*omputer rooms*. In addition the definition of cooling includes both *mechanical cooling* and e*conomizers.* This provision is to prohibit use of CRAC and CRAH units with humidity controls that include reheat coils.

#### 10.4.3.3 **Humidification**

*§140.9(a)3*

§140.9(a)3 prohibits the use of non-adiabatic humidification for c*omputer rooms*. The requirement of humidity control in *computer rooms* is controversial. On the low humidity side humidification was provided to reduce the risk of electrostatic discharge. On the high humidity side the concern has been CAF formation on the circuit boards. For both of these issues there is insufficient evidence that the risks either adequately address through the use of humidity controls. The telecommunications industry Standard for central office facilities has no restrictions on either the low or high humidity limits. Furthermore, the Electrostatic Discharge Association (ESDA) removed humidification as a primary control over electrostatic discharge in electronic manufacturing facilities (ANSI/ESDA Standard 20.20) as it wasn't effective and didn't supplant the need for personal grounding. The Energy Standards allows for humidification but prohibits the use of non-adiabatic humidifiers including the steam humidifiers and electric humidifiers that rely on boiling water as both of these add cooling load with the humidity. The technologies that meet the adiabatic requirement are direct evaporative cooling and ultrasonic humidifiers.

#### 10.4.3.4 **Fan Power and Control**

*§140.9(a)4 and 5*

In §140.9(a)4, fan power for equipment cooling computer rooms is limited to 27W/kBtuh of net sensible cooling capacity. Net sensible cooling capacity is the sensible cooling capacity of the coil minus the fan heat. Systems that are designed for a higher airside ΔT (e.g., 25°F) will have an easier time meeting this requirement than systems designed for lower ΔT (e.g., 15°F)

Fan controls (§140.9(a)5) requires that fans serving *computer rooms* must have either variable speed control or two speed motors that provide for a reduction in fan motor power to ≤50% of power at design airflow when the airflow is at 67% of design airflow. This applies to chilled water units of all sizes and DX units with a rated cooling capacity of ≥ 5 tons.

#### 10.4.3.5 **Containment**

*§140.9(a)6*

Computer rooms with a design IT equipment load exceeding 175 kW per room are required to have containment to separate the computer equipment inlets and outlets. The requirement can be met using hot-aisle containment, cold aisle containment or in-rack cooling. Exceptions are provided for:

- Expansions of existing *computer rooms* that don't already have containment.
- Computer racks with a design load of < 1 kW/rack (e.g. network racks).
- Equivalent energy performance demonstrated to the AHJ through use of CFD or other analysis tools.

**Figure 10-5: Example of aisle containment using chimney racks**

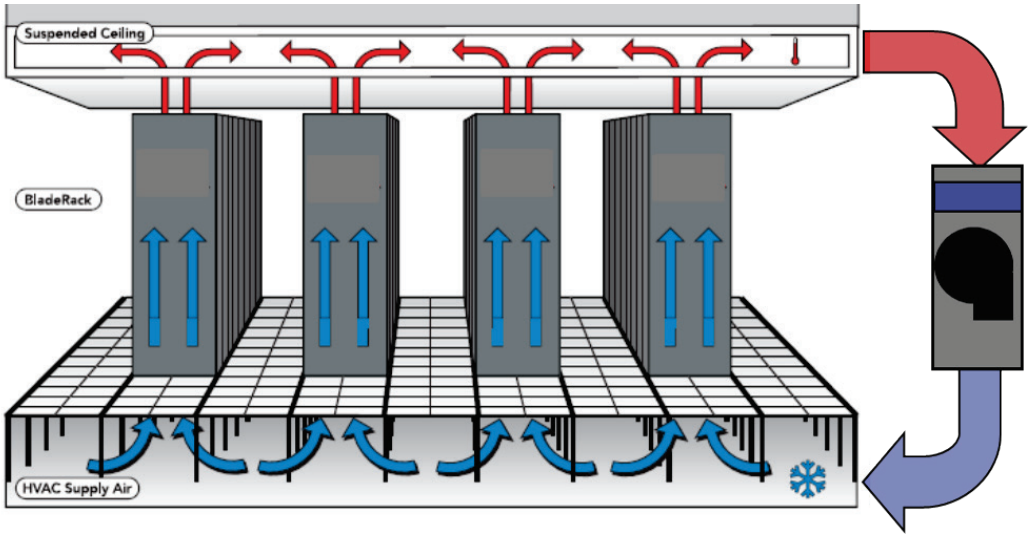



**Figure 10-6: Example of aisle containment using hard partitions and doors**

### 10.4.4 Additions and Alterations

The application to additions and alternations are covered under each measure.

## 10.5 Commercial Refrigeration

### 10.5.1 Overview

This section addresses §120.6(b) of the Energy Standards, which covers mandatory requirements for commercial refrigeration systems in retail food stores. This section explains the mandatory requirements for condensers, compressor systems, refrigerated display cases, and refrigeration heat recovery. All buildings under the Energy Standards must also comply with the General Provisions of the Energy Standards (§100.0 – §100.2, §110.0 – §110.10, §120.0 – §120.9, §130.0 – §130.5), and additions and alterations requirements (§141.1).

#### 10.5.1.1 Mandatory Measures and Compliance Approaches

The energy efficiency requirements for commercial refrigeration are all mandatory. There are no prescriptive requirements or performance compliance paths for commercial refrigeration. Since the provisions are all mandatory, there are no tradeoffs allowed between the various requirements. The application must demonstrate compliance with each of the mandatory measures. Exceptions to each mandatory requirement where provided are described in each of the mandatory measure sections below.

#### 10.5.1.2 What's New in the 2016 Energy Standards

In the 2016 Energy Standards, the exception to §120.6(b)3 is no longer available. This affects retail food stores with 8,000 square feet or more of conditioned area that use either: refrigerated display cases, or walk-in coolers or freezers connected to remote compressor units or condensing units, which are normally open for business for less than 140 hours per week. These stores must now comply with §120.6(b)3, without exception.

#### 10.5.1.3 Scope and Application

*§120.6(b)*

§120.6(b) of the Energy Standards applies to retail food stores that have 8,000 square feet or more of conditioned area, and utilize either refrigerated display cases or walk-in coolers or freezers, which are connected to remote compressor units or condensing units. The Energy Standards have minimum requirements for the condensers, compressor systems, refrigerated display cases, and refrigeration heat recovery systems associated with the refrigeration systems in these facilities.

The Energy Standards do not have minimum efficiency requirements for walk-ins, as these are deemed appliances and are covered by the California Appliance Efficiency Regulations (Title 20) and Federal Energy Independence and Security Act of 2007. Walk-ins are defined as refrigerated spaces with less than 3,000 square feet of floor area that are designed to operate below 55°F (13°C). Additionally, the Energy Standards do not have minimum equipment efficiency requirements for refrigerated display cases, as the minimum efficiency for these units is established by Federal law in the Commercial Refrigeration Equipment Final Rule but there are requirements for display cases that do result in reduced energy consumption.

Example 10-9

**Question**

The only refrigeration equipment in a retail food store with 10,000 square feet of conditioned area is self-contained refrigerated display cases. Does this store need to comply with the requirements for Commercial Refrigeration?

**Answer**

No. Since the refrigerated display cases are not connected to remote compressor units or condensing units, the store does not need to comply with the Energy Standards.

Example 10-10

**Question**

A new retail store with 25,000 square feet conditioned area has two self-contained display cases. The store also has several display case line-ups and walk-in boxes connected to remote compressors systems. Do all the refrigeration systems need to comply with the requirements for Commercial Refrigeration?

**Answer**

There are no provisions in the Energy Standards for the two self-contained display cases. The refrigeration systems serving the other fixtures must comply with the Energy Standards.

## 10.5.2 Condensers Mandatory Requirements

*§120.6(b)1*

This section addresses the mandatory requirements for condensers serving commercial refrigeration systems. These requirements only apply to stand-alone refrigeration condensers and do not apply to condensers that are part of a unitary condensing unit.

If the work includes a new condenser replacing an existing condenser, the condenser requirements do not apply if all of the following conditions apply:

1. The Total Heat of Rejection of the compressor system attached to the condenser or condenser system does not increase, and
2. Less than 25% of the attached refrigeration system compressors (based on compressor capacity at design conditions) are new, and
3. Less than 25% of the display cases (based on display case design load at applied conditions) that the condenser serves, are new. Since the compressor system loads commonly include walk-ins (both for storage and point-of-sale boxes with doors), the 25% “display case" should be calculated with walk-ins included.

Example 10-11

**Question**

A supermarket remodel includes a refrigeration system modification where some of the compressors will be replaced, some of the refrigerated display cases will be replaced, and the existing condenser will be replaced. The project does not include any new load and the design engineer has determined that the total system heat of rejection will not increase. The replacement compressors comprise 20% of the suction group capacity at design conditions, and the replacement display cases comprise 20% of the portion of the design load that comes from display cases. There are no changes in walk-ins. Does the condenser have to comply with the provisions of the Energy Standards?

**Answer**

No. This project meets all three criteria of the exception to the mandatory requirements for condensers:

1. The new condenser is replacing an existing condenser

2. The total heat of rejection of the subject refrigeration system does not increase

3a. The replacement compressors comprise less than 25% of the suction group design capacity at design conditions

3b. The replacement display cases comprise less than 25% of the portion of the design load that comes from display cases.

### 10.5.2.1 Condenser Fan Control

*§120.6(b)1A,B,& C*

Condenser fans for new air-cooled or evaporative condensers, or fans on air- or water-cooled fluid coolers or cooling towers used to reject heat on new refrigeration systems, must be continuously variable speed. Variable frequency drives are commonly used to provide continuously variable speed control of condenser fans and controllers designed to vary the speed of electronically commutated motors are increasingly being used for the same purpose. All fans serving a common high side, or indirect condenser water loop, shall be

controlled in unison. Thus, in normal operation, the fan speed of all fans within a single condenser or set of condensers serving a common high-side should modulate together, rather than running fans at different speeds or staging fans off. However, when fan speed is at the minimum practical level minimum level, usually no higher than 10-20%, the fans may be staged off to further reduce condenser capacity. As load increases, fans should be turned back on prior to significantly increasing fan speed, recognizing a control band is necessary to avoid without excessive fan cycling. Control of air-cooled condensers may also keep fans running and use a holdback valve on the condenser outlet to maintain the minimum condensing temperature once all fans have reached minimum speed; with the holdback valve set below the fan control minimum saturated condensing temperature setpoint.

To minimize overall system energy consumption, the condensing temperature control setpoint must be continuously reset in response to ambient temperatures, rather than using a fixed setpoint value. This strategy is also termed ambient-following control, ambient-reset, wetbulb following and drybulb following—all referring to control logic which changes the condensing temperature control setpoint in response to ambient conditions at the condenser. The control system calculates a control setpoint saturated condensing temperature that is higher than the ambient temperature by a predetermined temperature difference (i.e. the condenser control TD). Fan speed is then modulated so that the measured SCT (saturated condensing temperature) matches the calculated SCT control setpoint. The SCT control setpoint for evaporative condensers or water-cooled condensers (via cooling towers or fluid coolers) must be reset according to ambient wet bulb temperature, and the SCT control setpoint for air-cooled condensers must be reset according to ambient dry bulb temperature.

The condenser control TD is not specified in the Standard. The nominal control value is often equal to the condenser design TD; however the value for a particular system is left up to the system designer. Since the intent is to utilize as much condenser capacity as possible without excessive fan power, common practice is to optimize the control TD over a period of time such that the fan speed is in a range of approximately 60-80% during normal operation (i.e. when not at minimum SCT and not in heat recovery).

The minimum saturated condensing temperature setpoint must be 70°F (21°C) or less. For systems utilizing halocarbon refrigerants with glide, the SCT setpoint shall correlate with a midpoint temperature (between the refrigerant bubble-point and dew point temperatures) of 70°F (21°C) or less. As a practical matter, a maximum SCT setpoint is also commonly employed to set an upper bound on the control setpoint in the event of a sensor failure and to force full condenser operation during peak ambient conditions. This value should be set high enough that it does not interfere with normal operation.

Split air-cooled condensers are sometimes used for separate refrigeration systems, with two circuits and two rows of condenser fans. Each condenser half would be controlled as a separate condenser. If a condenser has multiple circuits served by a common fan or set of fans, the control strategy may utilize the average condensing temperature or the highest condensing temperature of the individual circuits as the control variable for controlling fan speed.

Alternative control strategies are permitted to the condensing temperature reset control required in §120.6(b)1C. The alternative control strategy must be demonstrated to provide equal or better performance, as approved by the Executive Director.

Hybrid condensers, manufactured with integral capability to operate with either air-cooled or evaporative-cooled operation, are not covered. Air-cooled condensers with separately installed evaporative precoolers added to the condenser are not considered hybrid

condensers for the purpose of this Standard. Air cooled condensers with an added evaporative precooling must meet the requirements for air cooled equipment, including specific efficiency and ambient-following control.

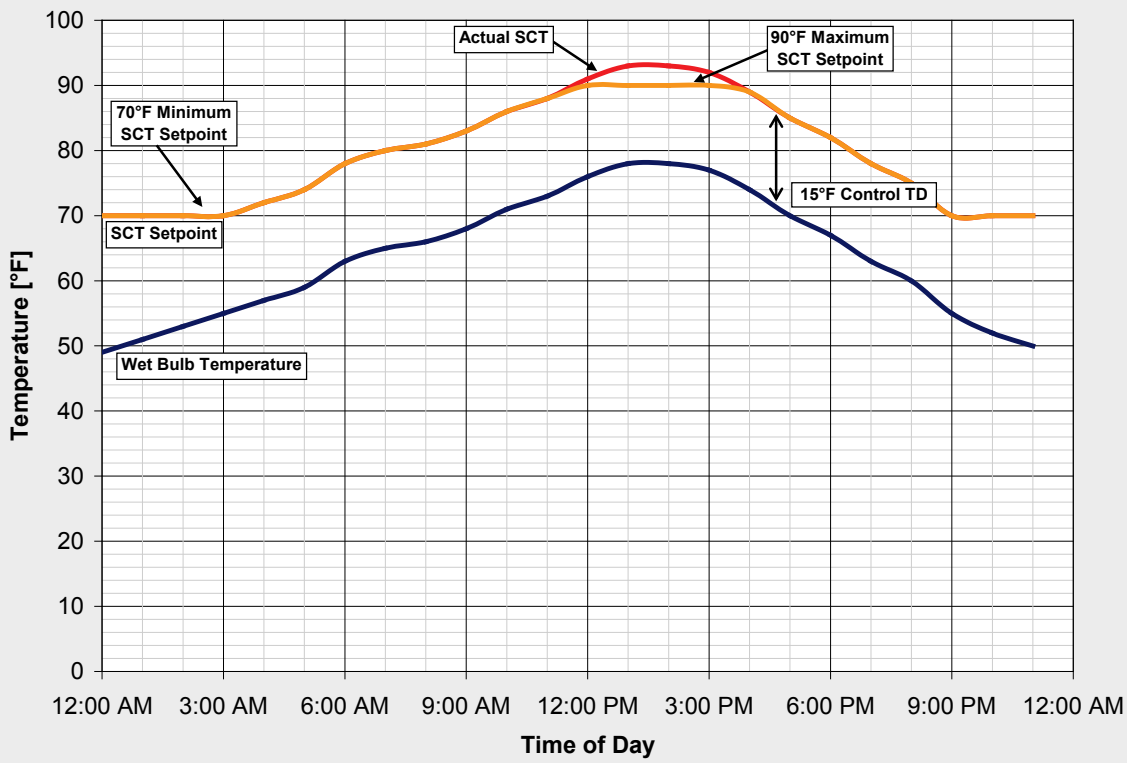
Example 10-12

**Question**

A new supermarket with an evaporative condenser is being commissioned. The control system designer has utilized a wet bulb-following control strategy to reset the system saturated condensing temperature (SCT) setpoint. The refrigeration engineer has calculated that adding a TD of 15°F (8.3°C) above the ambient wet bulb temperature should provide a saturated condensing temperature setpoint that minimizes the combined compressor and condenser fan power usage throughout the year. What might the system SCT and SCT setpoint trends look like over an example day?

**Answer**

The following figure illustrates what the actual saturated condensing temperature and SCT setpoints could be over an example day using the wet bulb-following control strategy with a 15°F (8.3°C) TD and also observing the 70°F (21°C) minimum condensing temperature requirement. As the figure shows, the SCT setpoint is continuously reset to 15°F (8.3°C) above the ambient wet bulb temperature until the minimum SCT setpoint of 70°F is reached. The figure also shows a maximum SCT setpoint (in this example, 90°F (32.2°C) which may be utilized to limit the maximum control setpoint, regardless of the ambient temperature value or TD
parameter.




### 10.5.2.2 Condenser Specific Efficiency

All newly installed evaporative condensers and air-cooled condensers with capacities greater than 150,000 Btuh (at the specific efficiency rating conditions) shall meet the minimum specific efficiency requirements shown in Table 10-2.

**Table 10-2: Fan-powered Condensers – Minimum Specific Efficiency Requirements**

| Condenser Type | Minimum Specific Efficiency | Rating Condition |
|---|---|---|
| Evaporative-Cooled | 160 Btuh/Watt | 100°F Saturated Condensing Temperature (SCT), 70°F Entering Wetbulb Temperature |
| Air-Cooled | 65 Btuh/Watt | 105°F Saturated Condensing Temperature (SCT), 95°F Entering Drybulb Temperature |

Condenser specific efficiency is defined as:

*Condenser Specific Efficiency = Total Heat Rejection (THR) Capacity / Input Power*

The total heat rejection capacity is defined at the rating conditions of 100°F Saturated Condensing Temperature (SCT) and 70°F outdoor wetbulb temperature for evaporative condensers, and 105°F SCT and 95°F outdoor drybulb temperature for air-cooled condensers. Input power is the electric input power draw of the condenser fan motors (at full speed), plus the electric input power of the spray pumps for evaporative condensers. The motor power is the manufacturer's published applied power for the subject equipment, which is not necessarily equal to the motor nameplate rating. Power input for secondary devices such as sump heaters shall not be included in the specific efficiency calculation.

The data published in the condenser manufacturer's published rating for capacity and power shall be used to calculate specific efficiency. For evaporative condensers, manufacturers typically provide nominal condenser capacity, and tables of correction factors that are used to convert the nominal condenser capacity to the capacity at various applied condensing temperatures and wetbulb temperatures. Usually the manufacturer publishes two sets of correction factors: one is a set of "heat rejection" capacity factors, while the others are "evaporator ton" capacity factors. Only the "heat rejection" capacity factors shall be used to calculate the condenser capacity at the efficiency rating conditions for the purpose of determining compliance with this section.

For air-cooled condensers, manufacturers typically provide the capacity at a given temperature difference (TD) between SCT and drybulb temperature. Manufacturers typically assume that air-cooled condenser capacity is linearly proportional to TD; the catalog capacity at 20°F TD is typically twice as much as at 10°F TD. The condenser capacity for air-cooled condensers at a TD of 10°F shall be used to calculate efficiency. If the capacity at 10°F TD is not provided, the capacity shall be scaled linearly.

Depending on the type of condenser, the actual manufacturer's rated motor power may vary from motor nameplate in different ways. Air cooled condensers with direct-drive OEM motors may use far greater input power than the nominal motor horsepower would indicate. On the other hand, evaporative condenser fans may have a degree of safety factor to allow for higher motor load in cold weather conditions (vs. the 100°F SCT/70°F WBT specific efficiency rating conditions). Thus, actual motor input power from the manufacturer must be used for direct-drive air-cooled condensers. For evaporative condensers and fluid coolers, the full load motor power, using the minimum allowable motor efficiencies published in the Nonresidential Appendix NA-3: Fan Motor Efficiencies is generally conservative but manufacturer's applied power should be used whenever possible to more accurately determine specific efficiency.

There are three exceptions to the condenser specific efficiency requirements.

1. If the store is located in Climate Zone 1 (the cool coastal region in northern California).
2. If an existing condenser is reused for an addition of alteration.
3. If the condenser capacity is less than 150 MBH at the specific efficiency rating conditions.

Example 10-13

**Question**

An air-cooled condenser is being designed for a new supermarket. The refrigerant is R-507. The condenser manufacturer's catalogue states that the subject condenser has a capacity of 500 MBH at 10°F TD between entering air and saturated condensing temperatures with R-507 refrigerant. Elsewhere in the catalog, it states that the condenser has ten ½ hp fan motors that draw 450 Watts each. Does this condenser meet the minimum efficiency requirements?

**Answer**

First, the condenser capacity must be calculated at the specific efficiency rating condition. From Table 10-6, we see that the rating conditions for an air-cooled condenser are 95°F entering drybulb temperature and 105°F SCT. The catalog capacity is at a 10°F temperature difference, which is deemed suitable for calculating the specific efficiency (105°F SCT - 95°F entering Drybulb = 10°F TD). Input power is equal to the number of motors multiplied by the input power per motor:

$$10 \text{ fan motors} \times \frac{450 \text{ Watts}}{\text{fan motor}} = 4{,}500 \text{ Watts}$$

The specific efficiency of the condenser is therefore:

$$500\text{MBH} \times \frac{1{,}000 \text{ Btu/hr}}{4{,}500 \text{ Watts}} / 4{,}500\text{Watts} = 111\text{Btu/hr/ Watts}$$

This condenser has a specific efficiency of 111 Btuh per Watt, which is higher than the 65 Btuh per Watt minimum requirement. This condenser meets the minimum specific efficiency requirements.

Example 10-14

**Question**

An evaporative condenser is being designed for a new supermarket. The manufacturer's catalog provides a capacity of 2,000 MBH at standard conditions of 105°F SCT and 78°F wetbulb temperature. The condenser manufacturer's catalog provides the following heat rejection capacity factors:

| Non-standard Conditions Heat Rejection Capacity | | | |
|---|---|---|---|
| Saturated Condensing Temperature (°F) | Wet Bulb Temperature (°F) | | |
| | 70 | 75 | 78 |
| 95 | 1.20 | 1.35 | 1.65 |
| 100 | 0.95 | 1.10 | 1.25 |
| 105 | 0.80 | 0.90 | 1.00 |

Elsewhere in the catalog, it states that the condenser model has one 10 HP fan motor and one 2 HP pump motor. Fan motor efficiencies and motor loading factors are not provided by the manufacturer. Does this condenser meet the minimum efficiency requirements?

**Answer**

First, the condenser capacity must be calculated at the specific efficiency rating condition. From Table 10-6, we see that the rating conditions for an evaporative condenser are 100°F SCT, 70°F WBT and a minimum specific efficiency requirement is 160 Btuh/Watt. From the Heat Rejection Capacity Factors table, we see that the correction factor at 100°F SCT and 70°F WBT is 0.95. The capacity of this model at the specific efficiency rating conditions is:

$$2{,}000 \text{ MBH} / 0.95 = 2{,}105 \text{ MBH}$$

To calculate input power, we will assume 100% fan and pump motor loading and minimum motor efficiencies since the manufacturer has not yet published actual motor specific efficiency at the specific efficiency rating conditions. We look up the minimum motor efficiency from Nonresidential Appendix NA-3: Fan Motor Efficiencies. For a 10 HP 6-pole open fan motor, the minimum efficiency is 91.7%. For a 2 HP 6-pole open pump motor, the minimum efficiency is 88.5%. The fan motor input power is calculated to be:

$$1\ \text{Motor} \times \frac{10\ \text{HP}}{\text{Motor}} \times 746\ \frac{\text{Watts}}{\text{HP}} \times \frac{100\%\ \text{assumed loading}}{91.7\%\ \text{efficiency}} = 8{,}135\ \text{Watts}$$

The pump motor input power is calculated to be:

$$1\ \text{Motor} \times \frac{2\ \text{HP}}{\text{Motor}} \times 746\ \frac{\text{Watts}}{\text{HP}} \times \frac{100\%\ \text{assumed loading}}{88.5\%\ \text{efficiency}} = 1{,}686\ \text{Watts}$$

The combined input power is therefore:

*8,135 Watts + 1,686 Watts = 9,821 Watts*

Note: Actual motor power should be used when available (see note in text).

Finally, the efficiency of the condenser is:

$$(2{,}105\ \text{MBH} \times \frac{1{,}000\ \text{Btuh}}{\text{MBH}}) / 9{,}821\ \text{Watts} = 214\ \text{Btuh/Watt}$$

214 Btuh per Watt is higher than the 160 Btuh per Watt requirement; this condenser meets the minimum efficiency requirements.

Air-cooled condensers shall have a fin density no greater than 10 fins per inch. Condensers with higher fin densities have a higher risk of fouling with airborne debris. This requirement does not apply for air-cooled condensers that utilize a micro-channel heat exchange surface, since this type of surface is not as susceptible to permanent fouling in the same manner as traditional tube-and-fin condensers with tight fin spacing.

The fin spacing requirement does not apply to condensers that are reused for an addition or alteration.

### 10.5.3 Compressor System Mandatory Requirements

*§120.6(b)2*

This section addresses mandatory requirements for remote compressor systems and condensing units used for refrigeration. In addition to the requirements described below, all the compressors and all associated components must be designed to operate at a minimum condensing temperature of 70°F (21°C) or less.

#### 10.5.3.1 Floating Suction Pressure Controls

*§120.6(b)2A*

Compressors and multiple-compressor suction groups must have floating suction pressure control to reset the saturated suction pressure control setpoint based on the temperature requirements of the attached refrigeration display cases or walk-ins.

Exceptions to the floating suction pressure requirements are:

1. Single compressor systems that do not have continuously variable capacity capability.
2. Suction groups that have a design saturated suction temperature of 30°F or higher.
3. Suction groups that comprise the high side of a two-stage or cascade system.
4. Suction groups that primarily serve chillers for secondary cooling fluids.
5. Existing compressor systems that are reused for an addition or alteration.

The examples of a two-stage system and a cascade system are shown in Figure 10-7 and Figure 10-8, respectively. Figure 10-9 shows a secondary fluid system.

**Figure 10-7: Two-stage System using a Two-Stage Compressor**

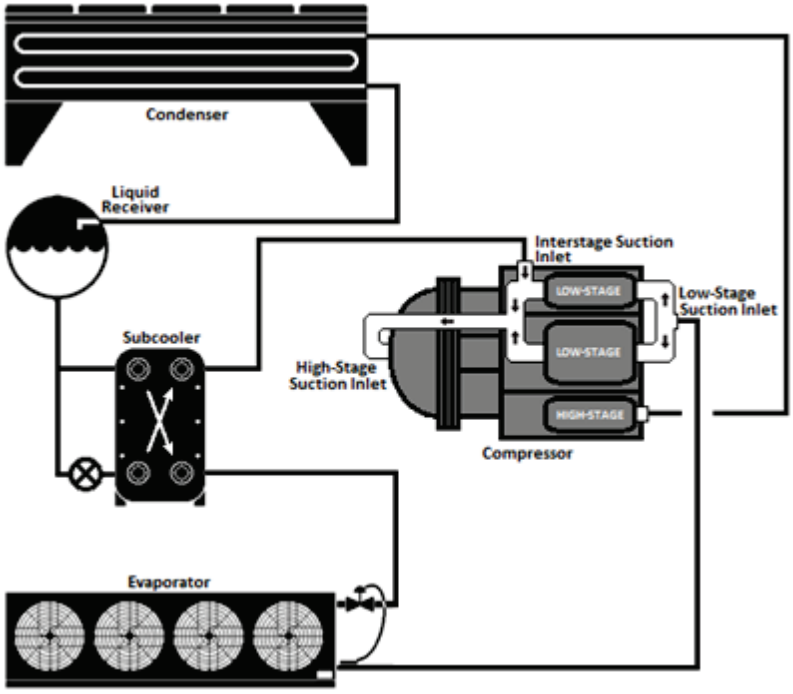



**Figure 10-8: Cascade System**

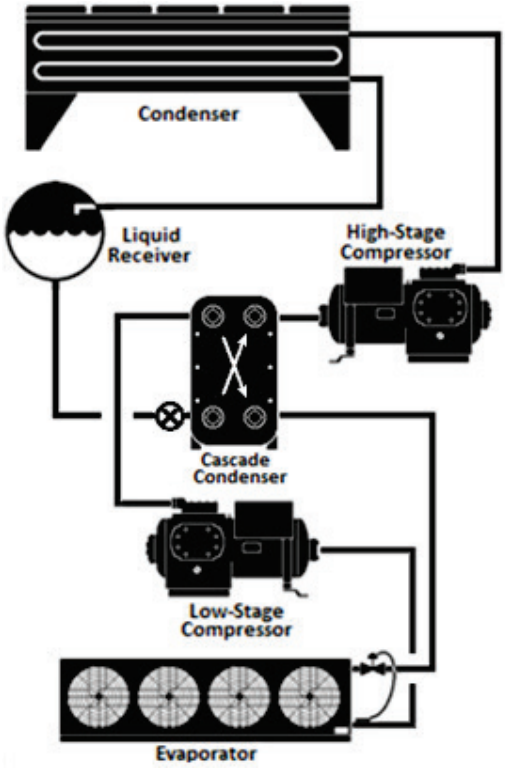
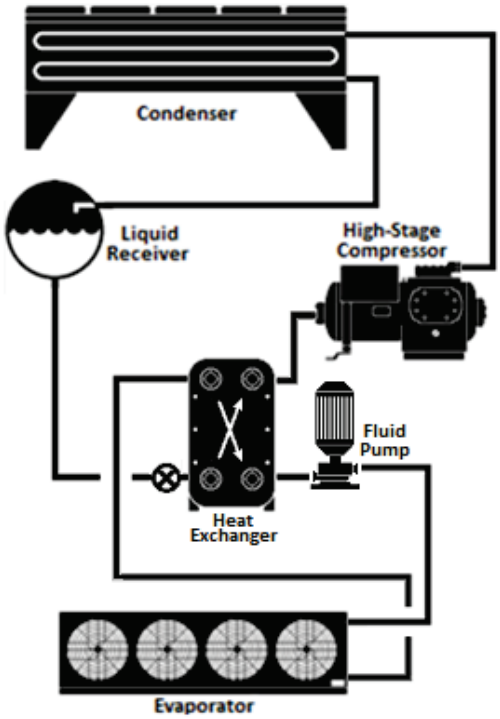


**Figure 10-9: Secondary Fluid System**




Example 10-15

**Question**

A retail food store has four suction groups, A, B1, B2 and C, with design saturated suction temperatures (SST) of -22°F, -13°F, 28°F and 35°F, respectively. System A is a condensing unit. The compressor in the condensing unit is equipped with two unloaders. Suction group B1 consists of a single compressor with no variable capacity capability. Suction group B2 has four compressors with no variable capacity capability and suction group C has three compressors with no variable capacity capability. Which of these suction groups are required to have floating suction pressure control?

**Answer**

Suction Groups A and B2 are required to have floating suction pressure control. The rationale is explained below.

Suction group A: Although the suction group has only one compressor, the compressor has variable capacity capability in the form of unloaders. Therefore, the suction group is required to have floating suction pressure control.

Suction group B1: The suction group has only one compressor with no variable capacity capability. Therefore, the suction group is not required to have floating suction pressure control.

Suction group B2: Although the suction group has compressors with no variable capacity capability, the suction group has multiple compressors which can be sequenced to provide variable capacity capability. Therefore, the suction group is required to have floating suction pressure control.

Suction group C: The design SST of the suction group is higher than 30°F. Therefore, the suction group is not required to have floating suction pressure control.

Example 10-16

**Question**

A retail food store has two suction groups, a low temperature suction group A (-22°F design SST) and medium temperature suction group B (18°F design SST). Suction group A consists of three compressors. Suction group B has four compressors that serve a glycol chiller working at 23°F. Which of these suction groups are required to have floating suction pressure control?

**Answer**

Suction group A: The suction group has multiple compressors. Therefore, the suction group is required to have floating suction pressure control.

Suction group B: Although the suction group has multiple compressors, it serves a chiller for secondary cooling fluid (glycol). Therefore, the suction group is not required to have floating suction pressure control.

Example 10-17

**Question**

A retail food store is undergoing an expansion and has two refrigeration systems: an existing system and a new $CO_2$ cascade system. The existing system consists of four compressors and a design SST of 18°F. The cascade refrigeration system consists of four low temperature compressors operating at -20°F SST and three medium temperature compressors operating at 26°F SST. Which of these systems are required to have floating suction pressure control?

**Answer**

Existing system: Although the system has multiple compressors, the compressor system is being reused, and the existing rack controller and sensors may not support floating suction pressure control. Therefore, the system is not required to have floating suction pressure control.

Cascade system: Only low temperature suction group of the system is required to have floating suction pressure control.

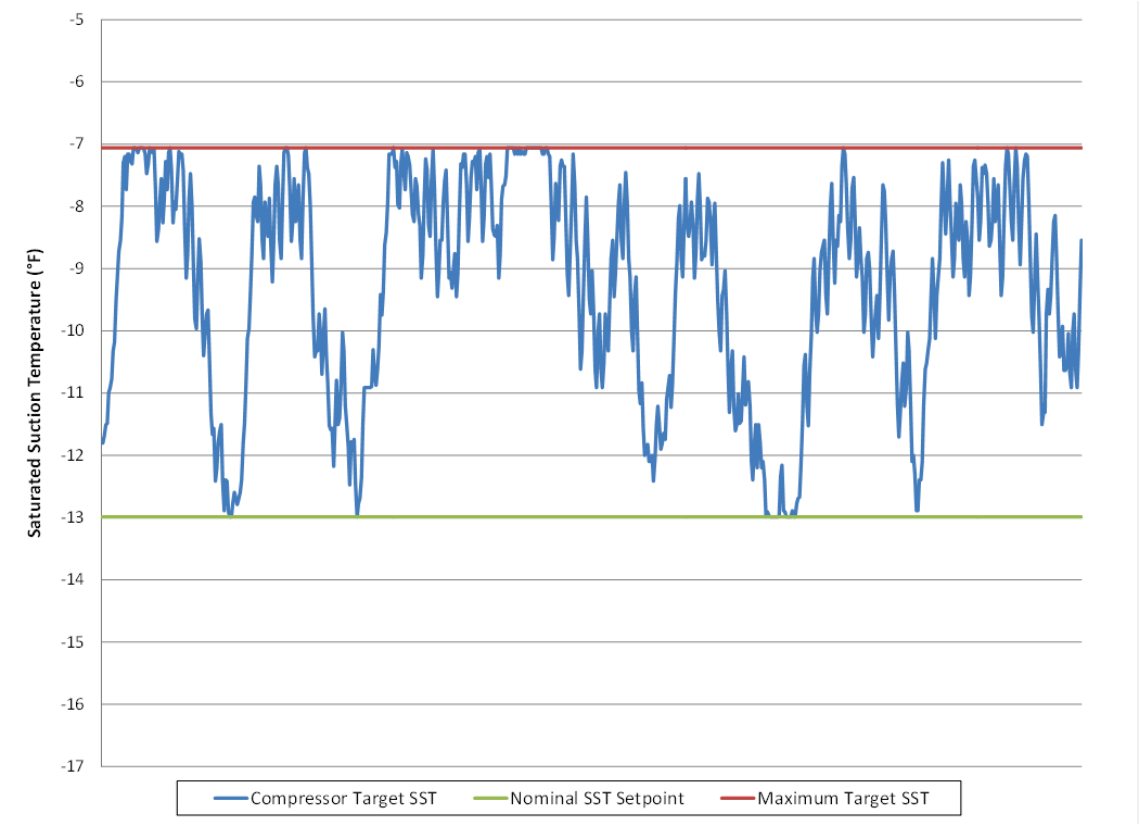
Evaporator coils are sized to maintain a design fixture temperature under design load conditions. Design loads are high enough to cover the highest expected load throughout the year, and inherently include safety factors. The actual load on evaporator coils varies throughout the day, month and year, and an evaporator coil operating at the design saturated evaporating temperature (SET) has excess capacity at most times. The SET can be safely raised during these times, reducing evaporator capacity and reducing the required "lift" of the suction group, saving energy at the compressor while maintaining proper fixture (and product) temperature.

In a floating suction pressure control strategy, the suction group target saturated suction pressure (SST) setpoint is allowed to vary depending on the actual requirements of the attached loads, rather than fixing the SST setpoint low enough to satisfy the highest expected yearly load. The target setpoint is adjusted so that it is just low enough to satisfy lowest current SET requirement of any attached refrigeration load while still maintaining target fixture temperatures, but not any higher. The controls are typically bound by low and high setpoints limits. The maximum float value should be established by the system designer, but a minimum value equal to the design SST (that is no negative float) and a positive float range of 4-6°F of saturation pressure equivalent have been used successfully.

Figure 10-10 shows hourly values for floating suction pressure control over a one week period, expressed in equivalent saturation temperature. The suction pressure control setpoint is adjusted to meet the temperature setpoint at the most demanding fixture or walk-in. The difference in SST between the floating suction pressure control and fixed suction

pressure control translates into reduced compressor work and thus energy savings for the floating suction control.

**Figure 10-10: Example of Floating Suction Pressure Control**




### A. Floating Suction Pressure Control with Mechanical Evaporator Pressure Regulators

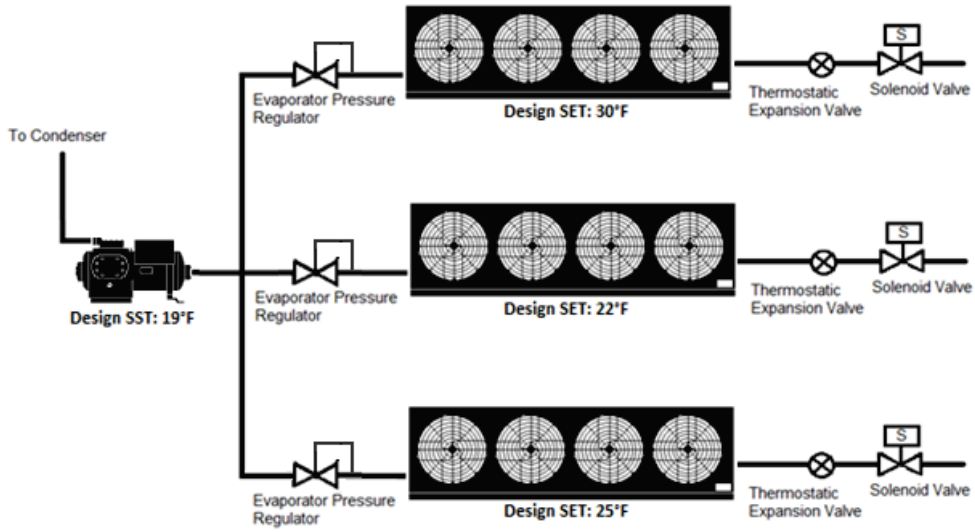
Mechanical evaporator pressure regulators (EPR valves) are often used on multiplex systems to maintain temperature by regulating the SET at each multiple evaporator connected to the common suction group, and often to also function as a suction-stop valve during defrost. EPR valves throttle to maintain the pressure at the valve inlet and thus indirectly control the temperature at the case or walk-in. The valves are manually adjusted to the pressure necessary to provide the desired fixture or walk-in air temperature. The load (circuit) with the lowest EPR pressure governs the required compressor suction pressure setpoint.

Floating suction pressure on a system with EPR valves requires special attention to valve settings on the circuit(s) used for floating suction pressure control. EPR valves on these circuit(s) must be adjusted “out of range”, meaning the EPR pressure must be set lower than what would otherwise be used to maintain temperature. This keeps the EPR valve from interfering with the floating suction control logic. In some control systems, two circuits are used to govern floating suction control; commonly designated as primary and secondary float circuits. EPR valves may also be equipped with electrically controlled wide-open solenoid pilots for more fully automatic control if desired.

Similar logic is applied on systems using on/off liquid line solenoid valves (LLSV) for temperature control, with the control of the solenoid adjusted slightly out of range to avoid interference with floating suction pressure.

These procedures have been employed to float suction on supermarket control systems since the mid-80's, however careful attention is still required during design, start-up and commissioning to insure control is effectively coordinated.

**Figure 10-11: Evaporators with Evaporative Pressure Regulator Valves**



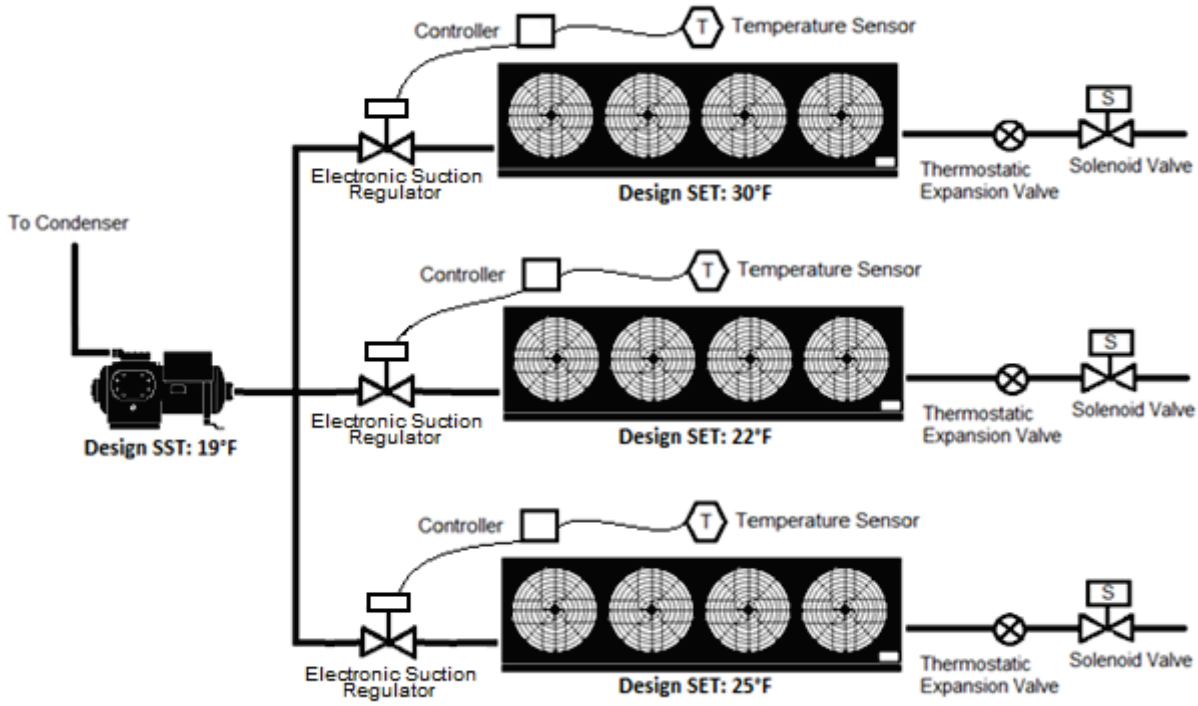

## B. Floating Suction Pressure Control with Electronic Suction Regulators

An electronic suction regulator (ESR) valve is an electronically controlled valve used in the place of a mechanical evaporator pressure regulator valve. ESRs are known in industry as electronic suction regulator (ESR) or electronic evaporator pressure regulator (EEPR). It is important to note that ESR valves are not pressure regulators; instead they control the flow through the evaporator based on a setpoint air temperature at the case or walk-in. ESR valves are modulated to maintain precise temperature. This provides more accurate temperature compared to an EPR which controls temperature indirectly through pressure and is subject to pressure drop in piping and heat load (and thus TD) on the evaporator coil.

Floating suction pressure strategies with ESR valves vary depending on the controls manufacturer, but will generally allow for more flexibility than systems with EPR valves. In general, the control system monitors how much each ESR valve is opened. If an ESR is fully open, indicating that the evaporator connected to the ESR requires more capacity, the control system will respond by decrementing the SST setpoint. If all ESR valves are less than fully open, the control system increments the suction pressure up until an ESR valve fully opens. At this point, the control system starts floating down the suction pressure again. This allows suction pressure to be no lower than necessary for the most demanding fixture.

Figure 10-12 shows multiple evaporators controlled by ESR valves connected to a common suction group.

**Figure 10-12: DX Evaporators with ESRs on a Multiplex System**




### 10.5.3.2 **Liquid Subcooling**

*§120.6(b)2B*

Liquid subcooling must be provided for all low temperature compressor systems with a design cooling capacity of 100,000 Btuh or greater and with a design saturated suction temperature of -10°F or lower. The subcooled liquid temperature of 50°F or less must be maintained continuously at the exit of the subcooler. Subcooling load may be handled by compressor economizer ports, or by using a suction group operating at a saturated suction temperature of 18°F or higher. Figure 10-13 and Figure 10-14 show example subcooling configurations.

Exceptions to the liquid subcooling requirements are:

1. Low temperature cascade systems that condense into another refrigeration system rather than condensing to ambient temperature.
2. Existing compressor systems that are reused for an addition or alteration.

**Figure 10-13: Liquid Subcooling Provided by Scroll Compressor Economizer Ports**

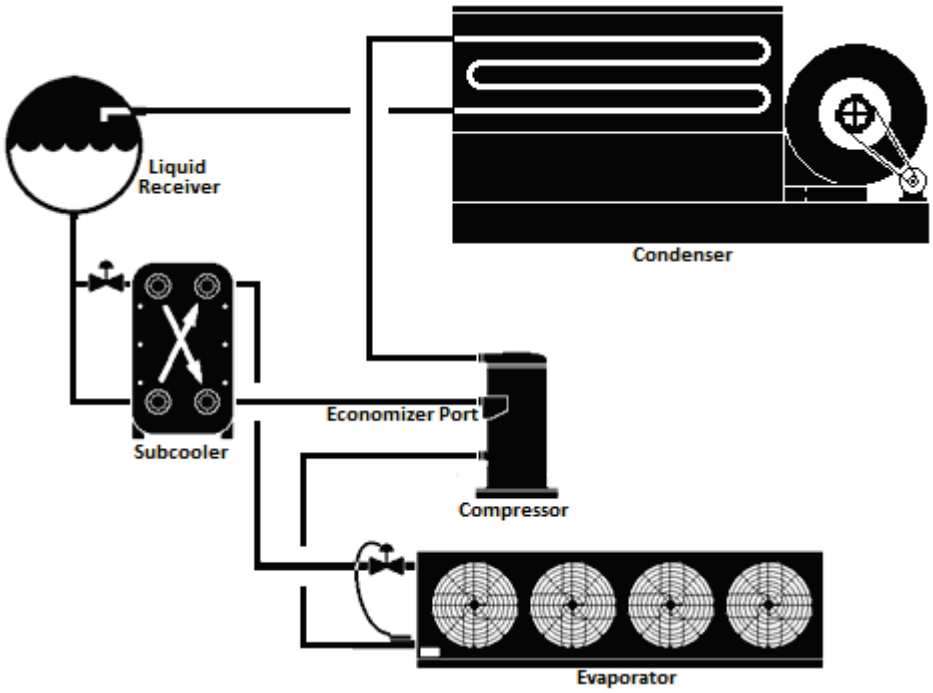
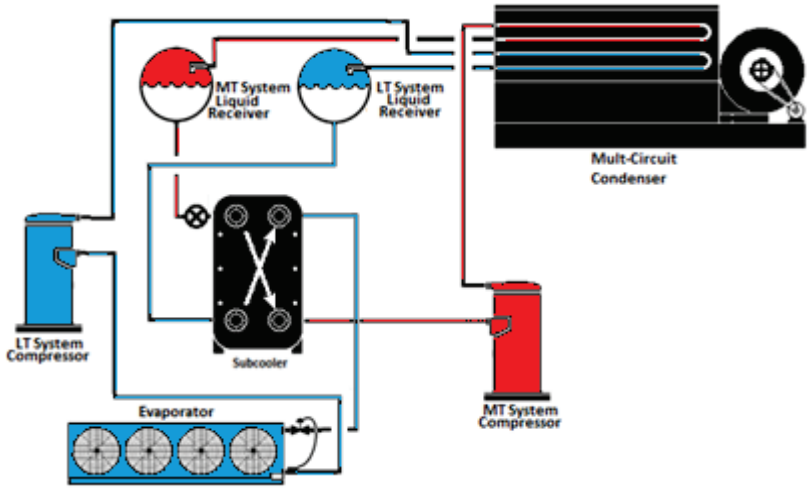



**Figure 10-14: Liquid Subcooling Provided By a Separate Medium-Temperature System**




## 10.5.4 Refrigerated Display Case Lighting Control Requirements

*§120.6(b)3*

All lighting in refrigerated display cases, and lights installed on glass doors of walk-in coolers and freezers shall be controlled by either automatic time switch controls and/or motion sensor control.

**A. Automatic Time Switch Control**

Automatic time switch controls shall turn off the lights during non-business hours.

Timed overrides for a display case line-up or walk-in case may be used to turn on the lights for stocking or non-standard business hours. The override must time-out and automatically turn the lights off again in one hour or less. The override control may be enabled manually (e.g. a push button input to the control system) or may be scheduled by the lighting control or energy management system.

**B. Motion Sensor**

Motion sensor control can be used to meet this requirement by either dimming or turning off the display case lights when space near the case is vacated. The lighting must dim so that the lighting power reduces to 50% or less. The maximum time delay for the motion sensor must be 30 minutes or less.

### 10.5.5 Refrigeration Heat Recovery

*§120.6(b)4*

This section addresses mandatory requirements for the use of heat recovery from refrigeration system(s) to HVAC system(s) for space heating and the charge limitations when implementing heat recovery, including an overview of configurations and design considerations for heat recovery systems. Heat rejected from a refrigeration system is the total of the cooling load taken from display cases and walk-ins in the store plus the electric energy used by the refrigeration compressors. Consequently, there is a natural relationship between the heat available and the heating needed; a store with greater refrigeration loads needs more heat to make up for the cases and walk-ins and also has more heat available.

The heat recovery requirements apply only to space heating.

There are many possible heat recovery design configurations due to the variety of refrigeration systems, HVAC systems and potential arrangement and locations of these systems. A number of examples are presented here but the Energy Standards do not require these configurations to be used. The heat recovery design must be consistent with the other requirements in the Energy Standards such as condenser floating head pressure.

At least 25 percent of the sum of the design Total Heat of Rejection (THR) of all refrigeration systems with individual design Total Heat of Rejection of 150,000 Btu/h or greater must be utilized for space heat recovery.

Exceptions to the above requirements for heat recovery are:

1. Stores located in Climate Zone 15, which is the area around Palm Springs, California. Weather and climate data are available in Joint Appendix JA2 – Reference Weather/Climate Data.
2. The above requirements for heat recovery do not apply to the HVAC and refrigeration systems that are reused for an addition or alteration.

The Energy Standards also limit the increase in hydrofluorocarbon (HFC) refrigerant charge associated with refrigeration heat recovery. The increase in HFC refrigerant charge associated with refrigeration heat recovery equipment and piping must not be greater than 0.35 lbs. per 1,000 Btuh of heat recovery heating capacity.

Example 10-18

**Question**

A store has three new distributed refrigeration systems, A, B and C, with design Total Heat of Rejection (THR) of 140,000 Btuh, 230,000 Btu/h and 410,000 Btuh, respectively. What is the minimum required amount of refrigeration heat recovery?

**Answer**

Refrigeration systems B and C have individual design THR of greater than 150,000 Btu/h, whereas refrigeration system A has design THR of less than 150,000 Btuh. Therefore, the store must have the minimum refrigeration heat recovery equal to 25% of the sum of THR of refrigeration systems B and C only. The minimum required heat recovery is therefore:

*25% x (230,000 Btuh + 410,000 Btuh) = 160,000 Btuh*

Example 10-19

**Question**

How should the Total Heat of Rejection be calculated for the purpose of this Section?

**Answer**

The THR value is equal to the total compressor capacity plus the compressor heat of compression.

Example 10-20

**Question**

A 35,000 ft2 food store is undergoing an expansion to add 20,000 square feet area. The store refrigeration designer plans to use two existing refrigeration systems with 600,000 Btu/h of design total heat rejection capacity and add a new refrigeration system with a design total heat rejection capacity of 320,000 Btu/h. The store mechanical engineer plans on replacing all the existing HVAC units. Is the store required to have refrigeration heat recovery for space heating?

**Answer**

Yes. The store must have the minimum required refrigeration heat recovery from the new refrigeration system. The new refrigeration system has a design THR of greater than 150,000 Btu/h threshold. The minimum amount of the refrigeration heat recovery is 25% of the new system THR. The existing refrigeration systems are not required to have the refrigeration heat recovery.

#### 10.5.5.1 Refrigeration Heat Recovery Design Configurations

The designer of heat recovery systems must consider the arrangement of piping, valves, coils, and heat exchangers as applicable to comply with the Energy Standards. Numerous refrigeration heat recovery systems configurations are possible depending upon the refrigeration system type, HVAC system type and the store size. Some possible configurations are:

1. Direct heat recovery.
2. Indirect heat recovery.
3. Water loop heat pump system.

These configurations are described in more detail with the following sections.

### A. Direct Heat Recovery

Figure 10-15 shows a series-connected direct condensing heat recovery configuration. In this configuration, the heat recovery coil is placed directly within the HVAC unit airstream (generally the unit serving the main sales area), and the discharge refrigerant vapor from the compressors is routed through the recovery coil and then to the outdoor refrigerant condenser when in heating mode. If two or more refrigeration systems are used for heat recovery, a multi-circuit heat recovery coil could be used.

This configuration is very suitable when the compressor racks are close to the air handling units used for heat recovery. If the distance is too far, an alternative design should be considered; the long piping runs may result in a refrigerant charge increase that exceeds the maximum defined in the Energy Standards, or there may be excessive pressure losses in the piping that could negatively affect compressor energy.

**Figure 10-15: Series Direct Heat Recovery Configuration**

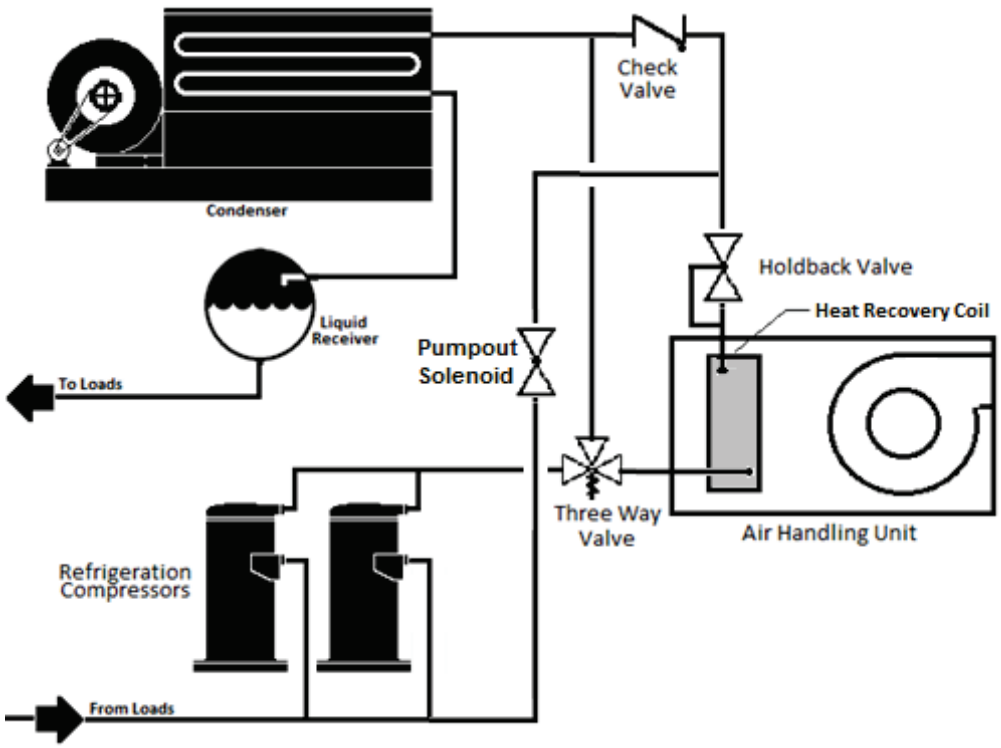
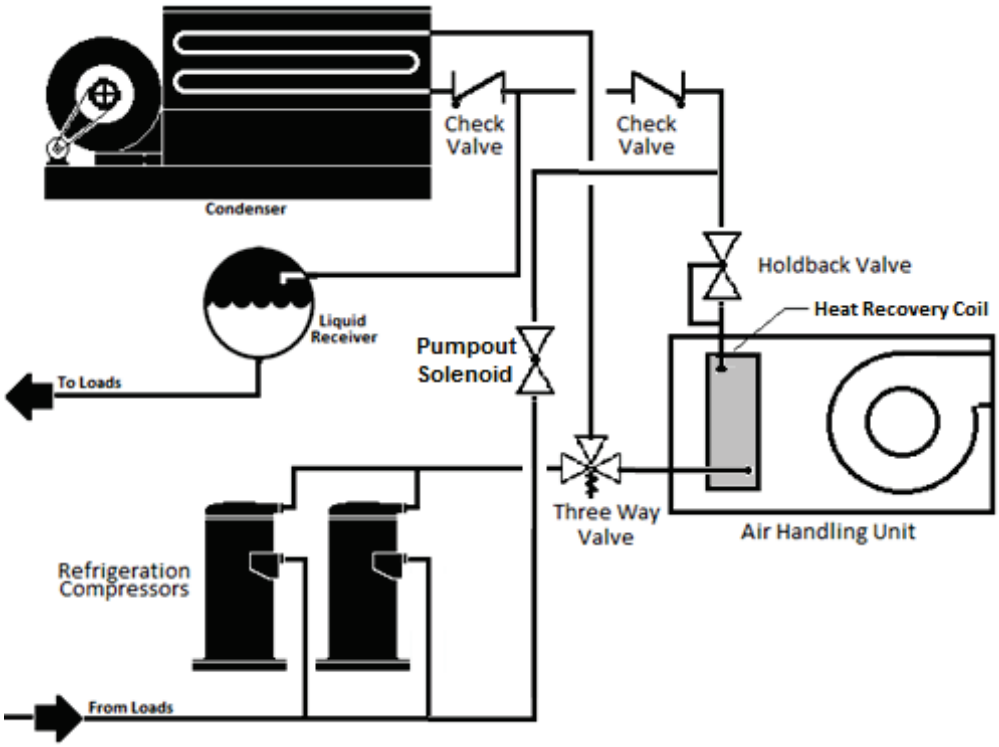



Figure 10-16 shows a parallel-connected direct-condensing configuration. In this configuration, the heat recovery coil handles the entire condensing load for the connected refrigeration system(s) when the air handling unit is in heating mode. Reduced refrigerant charge is the primary advantage of this configuration. Since the unused condenser (either the heat recovery condenser or the outdoor condenser) can be pumped out, there is no increase in refrigerant charge. A high degree of design expertise is required with this configuration in that the heat recovery condenser and associated HVAC system must take the entire heating load while operating at reasonable condensing temperatures—in any event, no higher than the system design SCT and in most instances with reasonable design no higher than 95°F-100°F condensing temperature in the heat recovery condenser. Ducting with under case or low return air design is essential in this type of system, in order to obtain cooler entering air and maintain reasonable condensing temperatures. Provision is required for practical factors such as dirty air filters.

Since the main condenser is not in use during heat recovery, the condenser floating head pressure requirements do not apply.

**Figure 10-16: Parallel Direct Condensing Heat Recovery Configuration**

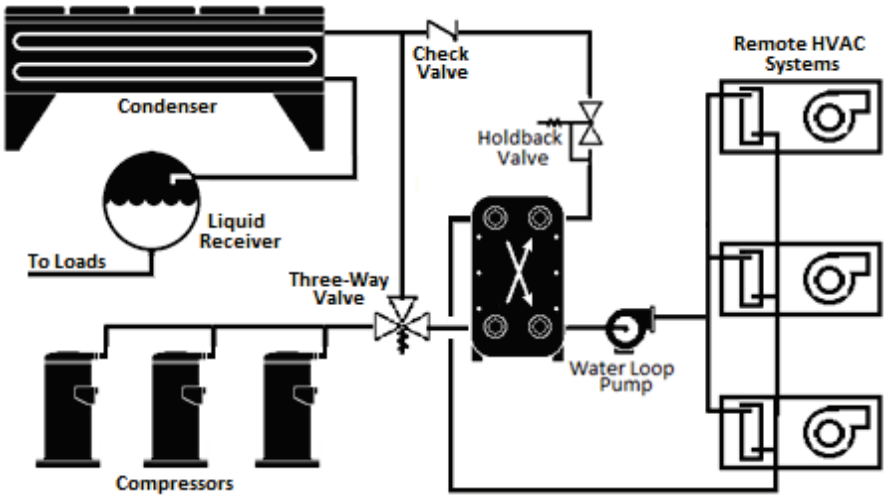



## B. Indirect Heat Recovery

Figure 10-17 shows an indirect heat recovery configuration with a fluid loop. In this configuration, the recovered heat is transferred from the refrigerant to an intermediate fluid, normally water or water-glycol, which is circulated through a fluid-to-air heat exchanger located in the air handling unit airstream. Like the direct condensing configuration, discharge refrigerant gas from the compressors is routed through the refrigerant-to-fluid heat exchanger and then to the outdoor refrigerant condenser when in heating mode.

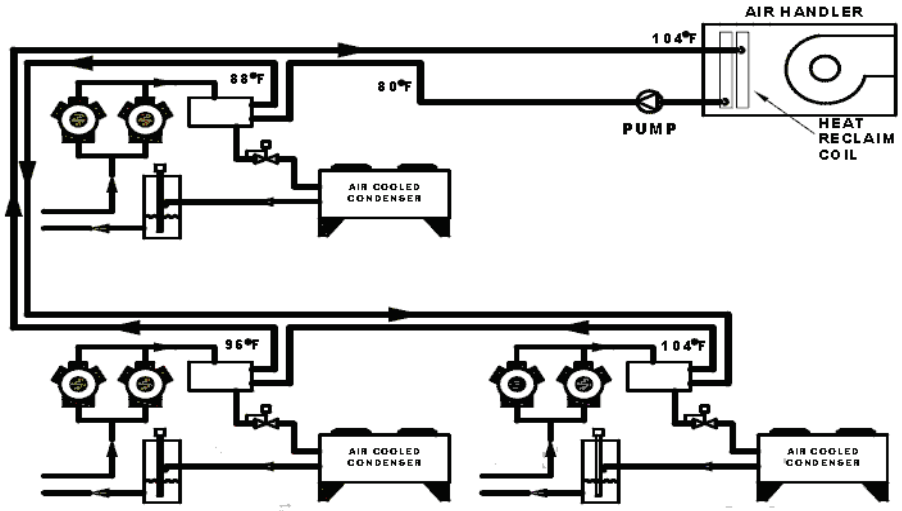
The refrigerant-to-fluid heat exchanger can be located close to the refrigeration system compressors, maximizing the available heat for recover while keeping the overall refrigerant charge increase low. This configuration is also suitable when multiple HVAC units are employed for the refrigeration heat recovery. Indirect systems must utilize a circulation pump to circulate the fluid between the HVAC unit and the recovery heat exchanger.

**Figure 10-17: Indirect Heat Recovery with an Indirect Loop**




Multiple refrigeration systems can also be connected in parallel or in series, using a common indirect fluid loop. Figure 10-18 shows three refrigeration systems connected in series by a common fluid loop. The temperatures shown are only examples.

**Figure 10-18: Series-Piped Indirect Water Recovery**




This configuration allows the refrigerant-to-water condenser temperature difference (TD) to be kept low at each refrigeration system (e.g. 8°-10°F is possible) while maintaining a sufficiently high water-side TD at the air handling unit (e.g. 20°-25°F depending on specifics) to allow an effective selection of the water-to-air heating coil vs. the available airflow. This method also minimizes both the required fluid flow and pump power.

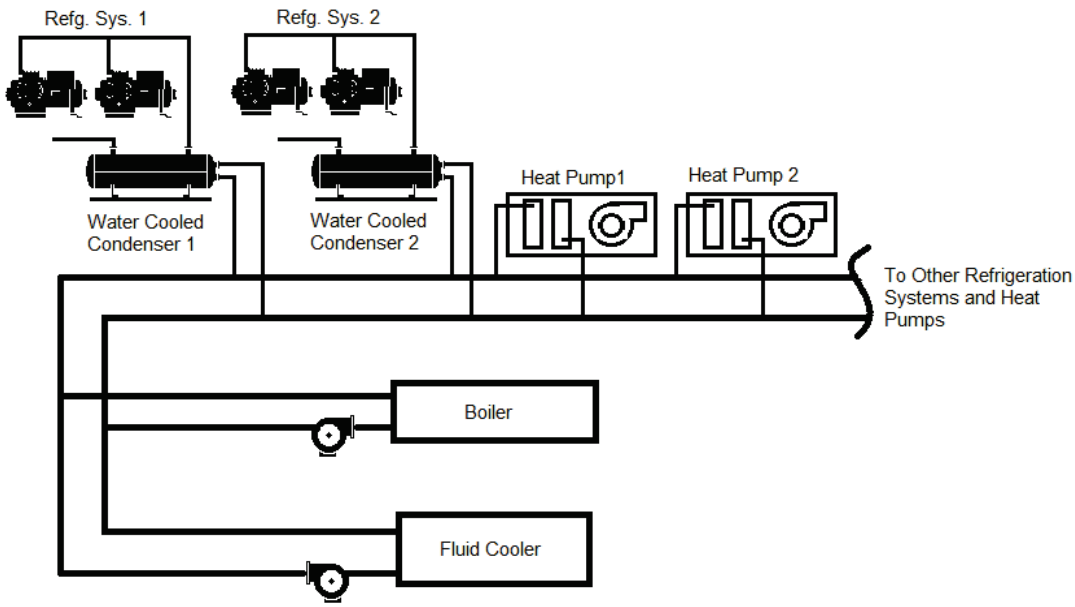
### C. Water Loop Heat Pump Heat Recovery

Water-source heat pumps (WLHP) can be used for in conjunction with water cooled refrigeration systems, connected to a common water loop as shown in Figure 10-19. Refrigeration systems heat pumps serving various zones of the store reject heat into a water loop, which in turn is rejected to ambient by an evaporative fluid cooler. When the

heat pumps are in heating mode, they extract the heat rejected by the refrigeration systems from the water loop. Additional heat, if required, is provided by a boiler connected to the water loop. A significant advantage of this design is low refrigerant charge, since the refrigeration systems use a compact water-cooled condenser, typically with less charge than an air-cooled condenser and no heat recovery condenser is required. Compared with other methods, however, the electric penalty is somewhat higher to utilize the available heat.

The floating pressure requirements in the standard would apply to the fluid coolers, i.e. controls to allow refrigeration systems to float to 70°F SCT and use of wetbulb following control logic.

**Figure 10-19: Water Loop Heat Pump Example**

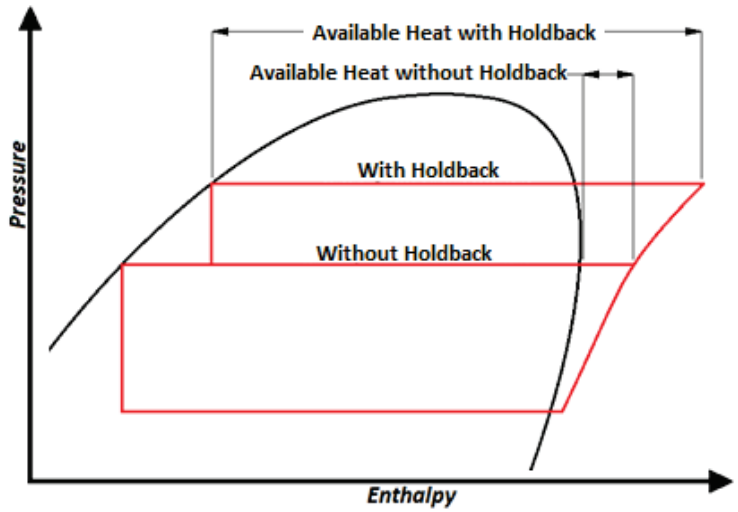



### 10.5.5.2 Control Considerations

#### A. Holdback Considerations

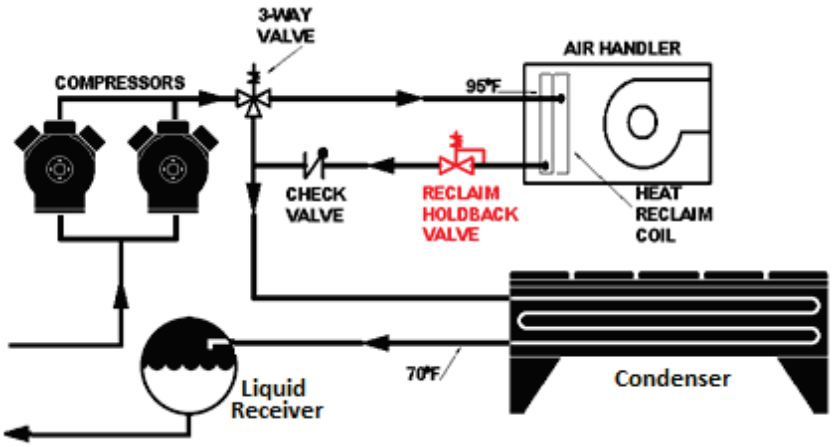
For direct and indirect systems, a holdback valve is required to control the refrigerant condensing temperature in the heat recovery coil (for direct systems) or the refrigerant-to-water condenser (for indirect systems) during heat recovery operation. Regulating the refrigerant pressure to achieve condensing recovers the latent heat from the refrigerant. Without condensing, only the sensible heat (i.e. superheat) is obtained, which is only a small fraction of the available heat. Figure 10-20 is a pressure-enthalpy diagram showing the difference in available recovery heat from a refrigeration system with and without a holdback valve.

**Figure 10-20: Pressure-Enthalpy Diagram with and without a Holdback Valve**




The holdback valve regulates pressure at its inlet, and is located at the exit of the recovery heat exchanger. Figure 10-21 shows a direct-condensing configuration with the proper location of the holdback valve.

**Figure 10-21: Direct-condensing configuration showing location of holdback valve**

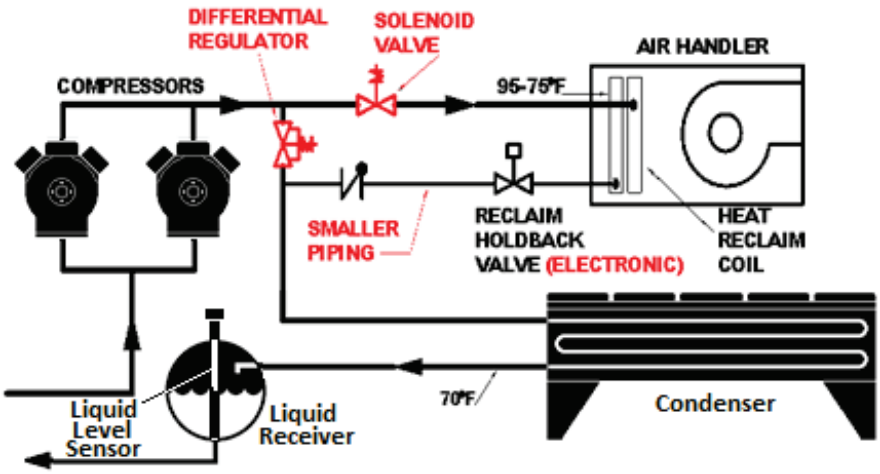



A more advanced design uses an electronic holdback valve controlled based on the temperature of the air entering the heat recovery coil. The electronic heat recovery holdback valve controls the valve inlet pressure and thus the heat recovery coil condensing temperature to maintain only the pressure necessary to achieve the required condensing TD (heat recovery SCT less entering air temperature) thereby minimizing compressor efficiency penalty. This is particularly useful when the volume outside air can significantly change the mixed air temperature entering the heat recovery coil. In colder climates, reducing the heat recovery holdback pressure can be important as a means to avoid over-condensing (i.e. subcooling). As shown in the pressure-enthalpy diagram above, there is additional flash gas handled by the condenser (even if the refrigerant fully

condenses in the heat recovery coil) which is necessary to maintain piping and condenser velocity and thus minimize the charge in the outdoor condenser.

Other designs can replace the three-way valve with a differential pressure regulator and solenoid valve. Figure 10-22 shows a direct-condensing configuration with an electronic heat recovery holdback valve, solenoid valve, and differential pressure regulator.

**Figure 10-22: Direct-condensing configuration showing differential regulator, solenoid valve, electronic holdback valve**

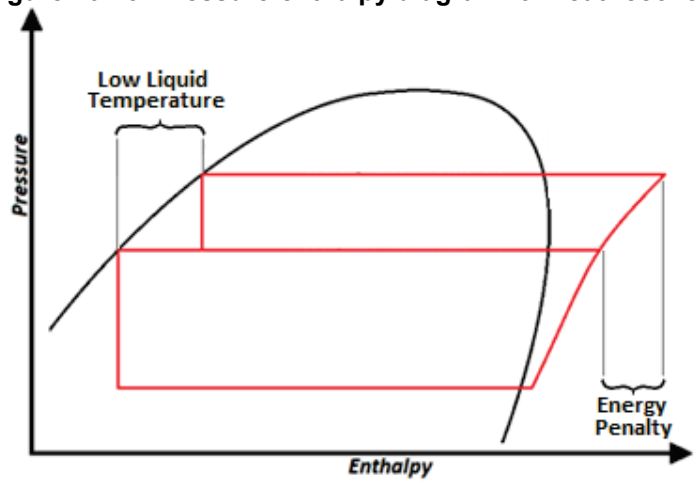



## B. Heat Recovery and Floating Head Pressure

There is typically a tradeoff between heat recovery and refrigeration system efficiency, in that compressor discharge pressure must be increased to provide condensing for heat recovery. If implemented properly, the electric penalty at the refrigeration system compressors is small compared to the heating energy savings.

The Energy Standards require that the minimum condensing temperature at the refrigeration condenser shall be 70°F or less. That means that (in the typical case of series-connected heat recovery) the refrigeration "cycle" still benefits from lower refrigerant liquid temperature, even if the compressor power is somewhat increased during heat recovery. The pressure-enthalpy diagram shown in Figure 10- 23 shows the incremental energy penalty at the refrigeration compressors due to the higher discharge pressure required for heat recovery, as well as the lower liquid temperature (and thus improved refrigerant cooling capacity) by floating head pressure at the outdoor condenser.

**Figure 10-23: Pressure-enthalpy diagram for heat recovery**



### 10.5.5.3 Recovery Coil Design Considerations

#### A. Recovery Coil Sizing Example

Selecting an appropriately-sized heat recovery coil is essential to proper heat recovery system operation. The following example details the process of selecting a right-sized heat recovery coil.

Example 10-21

**Question**

A supermarket is being constructed that will utilize heat recovery. The refrigeration system selected for recovery has the following parameters:

Design Refrigeration Load: 455.8 MBH
System design SST: 24°F
Representative Compressor Capacity at Design Conditions: 54.2 MBH
Representative Compressor Power at Design Conditions: 5.59 kW

The HVAC system serving the supermarket sales area is a central air handling unit. Heat recovery will be accomplished with a direct-condensing recovery coil inside the air handling unit, downstream of both the return air duct and the outside air damper. The air handling unit has the following design parameters:

Design Air Volume: 25,000 cfm
Design Coil Face Area: 41.7 sq.ft.

To avoid excessive pressure drop across the recovery coil, the designer will select a coil with a fin density of 10 fins per inch. The heat recovery circuit will use a holdback valve set at 95°F SCT.

What is the procedure for selecting a heat recovery coil?

**Answer**

To size a heat recovery system, the designer should first establish a design recovery coil capacity by analyzing the refrigeration system from which heat will be recovered. Best practice dictates that the recovery system should be sized to recovery most of the available system total heat of rejection at typical operating conditions, not peak conditions. Since we are designing for average operating conditions, the designer assumes the average refrigeration load is 70% of the design load. Therefore, the average system THR for heating design is:

*Average System THR = 70% x Design Refrigeration Load x THR Adjustment Factor*

where:

$$\text{THR Adjustments Factor} = \frac{\text{Representative Compressor THR}}{\text{Representative Compressor Capacity}}$$

and:

*Rep. Compressor THR = Rep. Compressor Capacity + Rep. Compressor Heat of Compression*

Using values from the example:

$$\text{Representative Compressor THR} = 54.2\text{MBH} + (5.50\ \text{kW} \times 3.415\ \frac{\text{MBH}}{\text{kW}})$$

Representative Compressor THR = 73.3 MBH

Therefore,

$$\text{THR Adjustment Factor} = \frac{73.3\ \text{MBH}}{54.2\ \text{MBH}}$$

*THR Adjustment Factor = 1.35*

Using the values in this example and the calculated THR Adjustment Factor, the average system THR is:

*Average system THR = 70% x 455.8 MBH x1.35*

*Average system THR = 430.1 MBH*

It is important to note that the recovery system will not be capable of extracting 100% of the total heat of rejection since the condenser operates at a lower pressure and will reject additional heat, even if the heat recovery coil achieves full condensing. In addition, the heat recovery coil performance may often be limited by the available airflow across the coil and the consequent temperature rise vs. the heat being transferred. This performance is determined through evaluation of coil performance, considering entering air temperature, and condensing temperature, as well as the coil design (e.g. rows, fins, air velocity and other factors). Airside pressure drop can be minimized by using a larger face area, requiring lower face velocity and fewer rows.

For in this example, it was assumed that after evaluating coil performance, 85% of the average THR could be recovered with a reasonable coil velocity and coil depth.

*Available Heat for Reclaim = 85% x Average System THR*

*Available Heat for Reclaim = 85% x 430.1 MBH*

*Available heat for Reclaim = 365.6 MBH*

The available heat for recovery is the design capacity of the recovery coil we will select for our air handling unit.

Next, the designer needs to know the face velocity of the airstream in the air handling unit. The face velocity is:

$$F.V. = \frac{\text{Design cfm}}{\text{AHU Face Area}}$$

$$F.V. = \frac{25{,}000\ \text{cfm}}{41.7\ \text{ft}^2}$$

*F.V. = 600 ft/min*

Finally, the designer needs to know the temperature difference between the condensing temperature (inside the recovery coil) and the temperature of the air entering the recovery coil. Since the coil will be installed in an air handling unit downstream of the outside air damper, the designer assumes that the air entering the coil is a mix of return air from the store and outside air. The designer must determine an appropriate design temperature for the air entering the recovery coil (Entering Air Temperature or EAT) during average heating hours, which is this instance was determined to be 65°F. From the example, the heat recovery system will have a holdback valve setting of 95°F SCT. Therefore, the temperature difference is:

*TD = SCT – EAT*

*TD = 95°F - 65°F*

*TD = 30°F*

Using the face velocity, design coil capacity, and temperature difference between condensing temperature and entering air temperature, the designer then refers to the air handling unit catalog to select a recovery coil. Then the designer uses the following two tables:

Heat reclaim correction factor for temperature difference between air and refrigerant.

| Temperature Difference (°F) | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 60 |
|---|---|---|---|---|---|---|---|---|
| Correction Factor | 0.4 | 0.5 | 0.6 | 0.7 | 0.8 | 0.9 | 1.0 | 1.2 |

Hot Gas Reclaim Heating Capacities
MBH per SQ FT of coil face area

| Rows | FPI | Face Velocity (ft/min) | | |
|---|---|---|---|---|
| | | 500 | 550 | 600 |
| 2 | 8 | 10.9 | 11.38 | 11.85 |
| | 10 | 12.15 | 12.73 | 13.18 |
| | 12 | 13.13 | 13.77 | 14.35 |
| 3 | 8 | 14.56 | 15.25 | 15.9 |
| | 10 | 15.93 | 16.8 | 17.63 |
| | 12 | 17.08 | 18.03 | 18.95 |
| 4 | 8 | 17.43 | 18.47 | 19.47 |
| | 10 | 18.75 | 19.92 | 21.07 |
| | 12 | 19.98 | 21.25 | 22.5 |

The designer enters the first table with the calculated TD of 30°F, finding a correction factor of 0.6. We enter the second table with the value:

$$\text{MBH per SQ FT} = \frac{\text{(Design Coil Capacity)}}{\text{Coil Face Area}} / \text{Correction Factor}$$

$$\text{MBH per SQ FT} = \frac{\text{(4184 MBH)}}{41.7\ ft^2} / 0.6$$

*MBH per SQ FT = 16.72*

Per design requirements, the designer will select a 10 fin-per-inch coil. From the second table, the designer selects the 3-row, 10 fin-per-inch coil for this application.

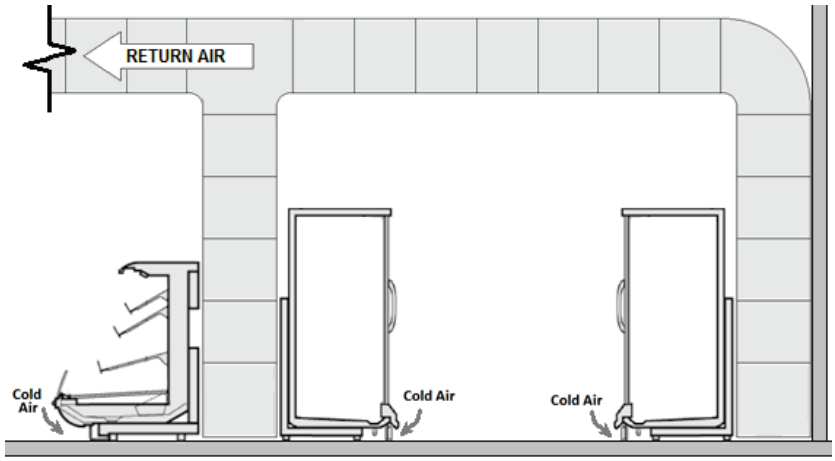
More commonly, computerized selection tools are used to select heat recovery coils, allowing vendors to provide multiple selections for comparison.

## B. Air-side Integration Considerations

### 1. Return Air Location

In supermarkets, ducting return air from behind display cases or near the floor is beneficial in improving comfort by removing the stagnant cool air that naturally occurs due to product refrigeration cases. This approach also increases the effectiveness of refrigeration heat recovery by increasing the temperature difference between the return air temperature and the refrigerant condensing temperature in the heat recovery coil. Figure 10-24 shows the location of an HVAC return air duct positioned to scavenge cool air from the floor level near refrigerated display cases.

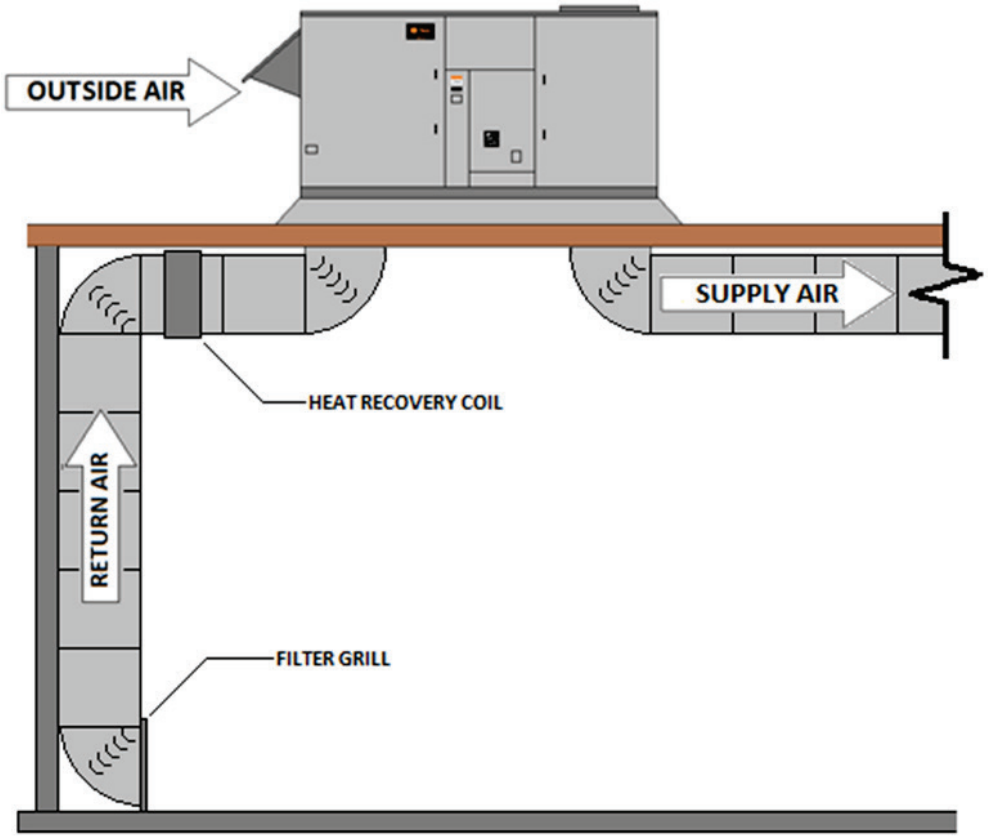
**Figure 10-24: Low Return Air Example**



### 2. Return Air Duct Configuration

Heat recovery can be incorporated into rooftop HVAC units (RTU) by installing the heat recovery coil inside the RTU cabinet or by installing in the return air duct upstream of the RTU, as shown in Figure 10-25. Location inside the RTU is preferable when outside air is a substantial part of the heating load, but location in the return air duct is reasonable and can provide greater flexibility in selecting the heat recovery coil (e.g. for low face velocity and pressure drop), particularly when coupled with low return air on units located in the refrigerated space, which predominantly provide heating. The fan design must allow for the additional ductwork and coil pressure drop.

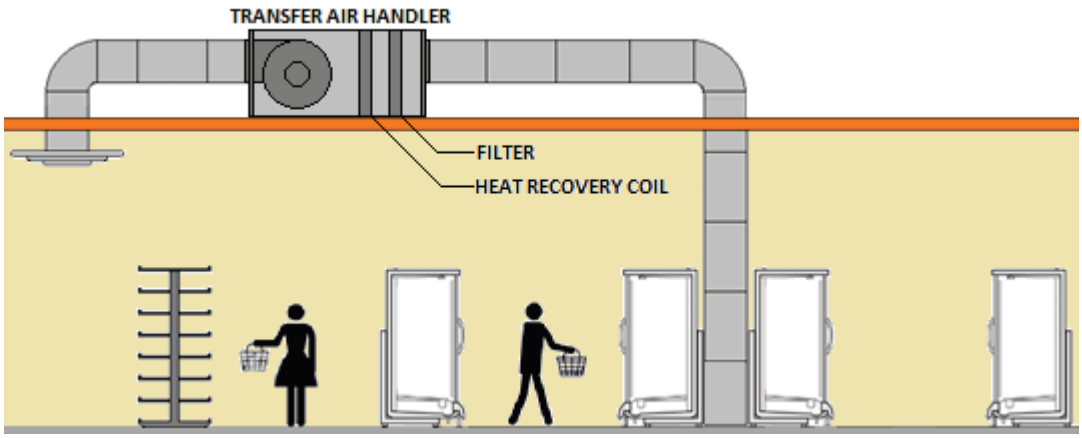
**Figure 10-25: Heat Recovery Coil in Return Air Duct**



### 3. Transfer Fan Configuration

A ducted transfer system is sometimes employed to remove cold air from aisles with refrigerated display cases (rather than blowing warm air into the refrigerated areas) and can be an easy and appropriate way to utilize heat recovery, particularly from smaller distributed systems. Figure 10-26 depicts a ducted transfer system.

**Figure 10-26: Ducted Transfer System**




### 4. Calculating Charge Increase

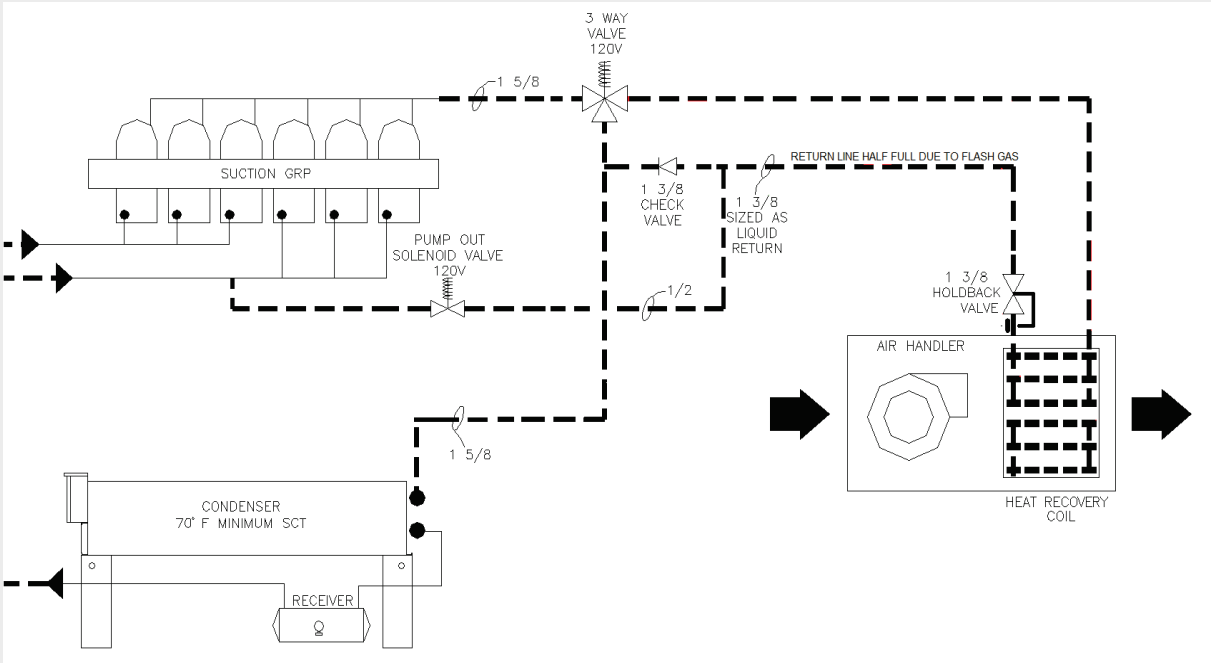
The Energy Standards require that the increase in HFC refrigerant charge from all equipment related to heat recovery for space heating shall be less than 0.35 lbs. for every 1,000 Btuh of heat recovery capacity at design conditions. Refrigerant charge may increase due to the addition of the recovery coil itself (either the refrigerant-to-air heat exchanger for direct configurations, or the refrigerant-to-water heat exchanger for indirect configurations), and the additional piping between the compressor group and the recovery coil. In addition, the refrigerant leaving the recovery coil and entering the refrigerant condenser will be mostly condensed, which increases the charge in the outdoor condenser compared with normal operation. Operating the outdoor condenser at lower pressure (i.e. the required floating heat pressure control) vs. the higher setting at the heat recovery coil holdback valve creates pressure drop, flashing of some liquid to vapor and an increase in velocity due to the much larger volume of a pound of vapor vs. a point of liquid refrigerant. Split condenser control, which is very common in cooler climates, can also be used to close-off and pump out half of the outdoor condenser.

It is the responsibility of the system designer to fully understand how the heat recovery system affects overall refrigerant charge.

Example 10-22

**Question**

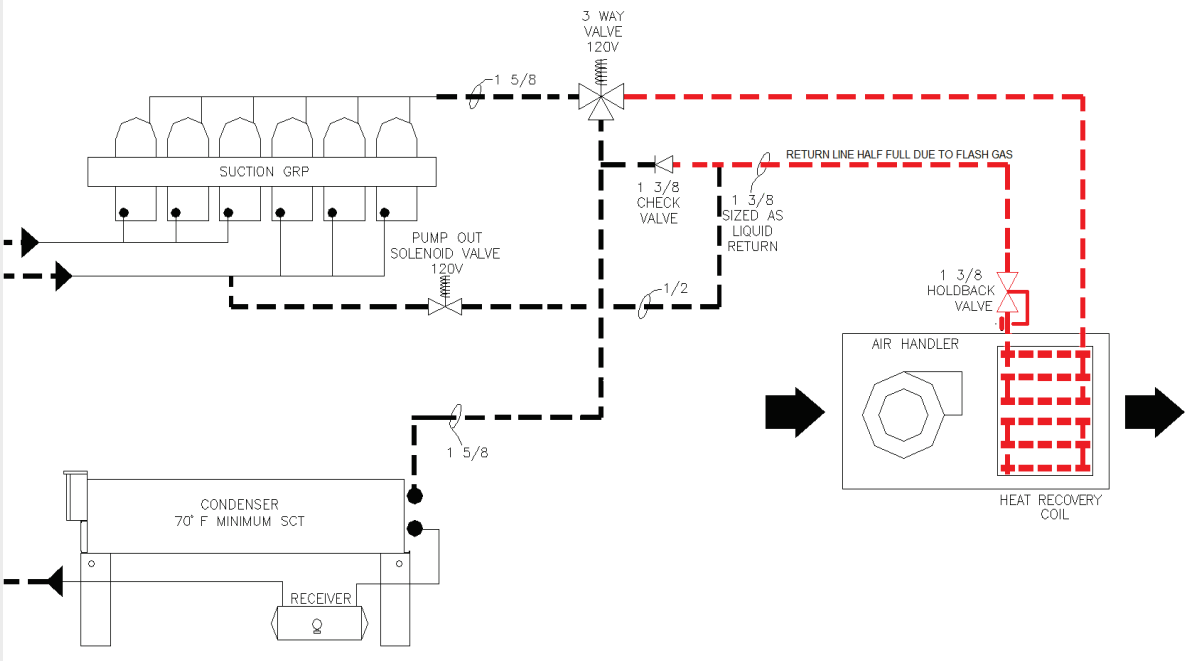
A heat recovery system is being designed for a new supermarket. The refrigerant is R-404A. The proposed design is shown below:




Which piping runs should be included in the calculation of refrigerant charge increase in the proposed design?

**Answer**

Only the additional piping required to route the refrigerant to the heat recovery coil needs to be considered in this calculation. The piping runs shown in red in the following figure should be included in the calculation of refrigerant charge increase from heat recovery.



Example 10-23

**Question**

What is the refrigerant charge size increase in the example described above?

**Answer**

The system designer prepares the following analysis to calculate the charge size in the refrigerant piping

| | Saturation Temperature (°F) | Pipe OD (in) | Pipe ID (in) | Pipe Length (ft) | Line Volume ($ft^3$) | % Vapor, Liquid by Mass | % Vapor, Liquid by Volume | Refrigerant Charge (lbs) |
|---|---|---|---|---|---|---|---|---|
| Discharge Line to Reclaim Coil | 95 | 1 5/8 | 1 1/2 | 100 | 1.2 | 100%, 0% | 100%, 0% | 6.7 |
| Liquid/Vapor Return Line | 80 | 1 3/8 | 1 1/4 | 100 | 0.9 | 9%, 91% | 59%, 41% | 25.5 |
| | | | | | | | Total Charge: | 32.2 |

The outdoor condenser has a capacity of 350 MBH at a TD of 10°F. Using manufacturers published data, the designer determines that the condenser normal operating charge (without heat recovery) is 26.9 lbs. To calculate the charge increase in the condenser due to heat reclaim, the designer estimates the condenser could be as much as 75% full of liquid, resulting in a condenser charge of 68.8 lbs. with heat recovery.

The heat recovery coil has a capacity of 320 MBH at a design TD of 20°F. The system designer uses manufacturer's documentation to determine that the heat recovery coil refrigerant charge is 14.1 lbs.

The total refrigerant charge with heat recovery is:

*32.2 lbs (piping) + 68.8 lbs (system condenser) + 14.1 lbs (recovery coil) = 115.1 lbs*

Therefore, the refrigerant charge increase with heat recovery is:

115.1 lbs – 26.9 lbs = 88.2 lbs

Example 10-24

**Question**

In the example above, does the recovery design comply with the requirement in the Energy Standards that the recovery design shall utilize at least 25% of the design Total Heat of Rejection (THR) of the refrigeration system?

**Answer**

The system designer determines that the total THR of all the refrigeration systems in the new supermarket is 800 MBH. From the previous example, the heat recovery capacity is 320 MBH.

$$100\ \% \times \frac{320\ MBH}{800\ MBH} = 40\%$$

Therefore, the design is in compliance with the Energy Standards.

Example 10-25

**Question**

In the example above, does the recovery design comply with the requirement in the Energy Standards that the recovery design shall not increase the refrigerant charge size by more than 0.35 lbs. of refrigerant per 1,000 Btuh of recovery capacity?

**Answer**

From the previous example, the recovery capacity is 320 MBH at design conditions, and the total refrigerant charge size increase is 88.2 lbs.

$$\frac{88.2\ lbs}{320\ MBH} = 0.28\ lbs/Btuh$$

Since the refrigerant charge increases by less than 0.35 lbs/MBH, this design is in compliance with the Energy Standards.

### 10.5.6 Additions and Alterations

*§141.1(b)*

Requirements related to commercial refrigeration additions and alterations to existing buildings are covered by the Energy Standards in §141.1(b). The specific requirements for additions and alterations for Commercial Refrigeration are included in §120.6(b).

### 10.5.7 Compliance Documentation

Compliance documentation includes the forms, reports and other information that are submitted to the enforcement agency with an application for a building permit (Certificate of Compliance). Compliance documentation also includes documentation completed by the installing contractor, engineer/architect of record, or owner's agent to verify that certain systems and equipment have been correctly installed and commissioned (Installation Certificate).

**Form NRCC-PRC-05-E for Commercial Refrigeration - Certificate of Compliance**

NRCC-PRC-05-E is the primary form for commercial refrigeration in retail food stores, which provides compliance information for the use of the enforcement agency's field inspectors. This form must be included on the plans. A copy of this form should also be submitted to the enforcement agency along with the rest of the compliance submittal at the time of building permit application. With enforcement agency approval, the applicant may use alternative formats of these forms (rather than the Energy Commission's forms), provided the information is the same and in similar format.

## 10.6 Refrigerated Warehouses

### 10.6.1 Overview

This section of the manual focuses on the Energy Standards provisions unique to refrigerated warehouses. The Energy Standards described in this chapter of the manual address refrigerated space insulation levels, underslab heating requirements in freezers, infiltration barriers, evaporator fan controls, condenser sizing and efficiency requirements, condenser fan controls, and screw compressor variable speed requirements.

All buildings regulated under Part 6 of the Energy Standards must also comply with the General Provisions of the Energy Standards (§100.0 – §100.2, §110.0 – §110.10, §120.0 – §120.9, §130.0 – §130.5), and additions and alterations requirements (§141.1). These topics are generally addressed in Chapter 3 of this manual.

#### 10.6.1.1 Mandatory Measures and Compliance Approaches

The energy efficiency requirements for refrigerated warehouses are all mandatory. There are no prescriptive requirements or performance compliance paths for refrigerated warehouses. Since the provisions are all mandatory, there are no trade-offs allowed

between the various requirements. The application must demonstrate compliance with each of the mandatory measures. Exceptions to each mandatory requirement, when applicable, are described in each of the mandatory measure sections below.

### 10.6.1.2 Scope and Application

*§120.6(a)*

§120.6(a) of the Energy Standards addresses the energy efficiency of refrigerated spaces within buildings, including coolers and freezers, as well as the refrigeration equipment that serves those spaces. Coolers are defined as refrigerated spaces designed to operate at or above 28°F (-2°C) and at or below 55°F (13°C). Freezers are defined as refrigerated spaces designed to operate below 28°F (-2°C). The subsections of §120.6(a) that cover refrigerated space requirements are 1, 2, 3, 6, and 7. The Energy Standards do not address walk-in coolers and freezers, as these are covered by the Appliance Efficiency Regulations (Title 20). A walk-in is defined as a refrigerated space that is less than 3,000 ft² in floor area. However, refrigeration systems (compressors and condensers that have a common refrigerant supply) that serve a sum total of 3,000 ft² or more <u>are</u> required to comply with the subsections of §120.6(a) that addresses those components specifically (subsections 4, 5, and 7). Also note that refrigeration systems and refrigerated display fixtures in grocery stores are covered in §120.6(b) and are described in Section 10.5 of this manual.

Additionally, areas within refrigerated warehouses designed solely for the purpose of quick chilling or quick freezing of products are exempt from the Energy Standards. Quick chilling and freezing spaces are defined as spaces with a design refrigeration evaporator load of greater than 240 Btu/hr-ft² of floor space, which is equivalent to 2 tons per 100 ft² of floor space. A space used for quick chilling or freezing and also used for refrigerated storage must still meet the requirements of §120.6(a).

The intent of the Energy Standards is to regulate storage space, not quick chilling or freezing space, or process equipment. Recognizing that there is often a variety of space types and equipment connected to a particular suction group in a refrigerated warehouse, it is not always possible to identify compressor plant equipment that serves the storage space only. It would be outside the intent of the Energy Standards to apply the compressor plant requirements to an industrial process that is not covered by the Energy Standards simply because a small storage space is also attached to the suction group. Similarly, it would be outside of the intent of the Energy Standards to exclude a compressor plant connected to a suction group serving a large storage space covered by the Energy Standards on the basis of a small process cooler or quick chill space also connected to the same suction group. For the purposes of compliance with the Energy Standards, the compressor plant requirements apply when 80 percent or more of the design refrigeration capacity connected to the suction group is from refrigerated storage space(s). A suction group refers to one or more compressors that are connected to one or more refrigeration loads, whose suction inlets share a common suction header or manifold.

A variety of space types and processes may be served by a compressor plant at different suction pressures. When all of these compressors share a common condensing loop, it is impossible to address only the equipment serving refrigerated storage spaces. For the purposes of compliance with the Energy Standards, the provisions addressing condensers, subsection 4, apply only to new condensers that are part of new refrigeration systems when the total design capacity of all refrigerated storage spaces served by compressors using a common condensing loop is greater than or equal to 80 percent of the total design capacity.

In addition to an all-new refrigerated facility, the Energy Standards cover expansions and modifications to an existing facility and an existing refrigeration plant. The Energy Standards

do not require that all existing equipment must all comply when a refrigerated warehouse is expanded or modified using existing refrigeration equipment. Exceptions are stated in the individual equipment requirements and an explanation of applicability to Additions and Alterations is included in Section 10.4.

#### 10.6.1.3 Ventilation

§120.1(a)1 of the Energy Standards, concerning ventilation requirements, includes an exception for "Refrigerated warehouses and other spaces or buildings that are not normally used for human occupancy and work". The definition of refrigerated warehouses covers all refrigerated spaces at or below 55°F (13°C) which will in some instances include spaces with occupancy levels or durations, effect of stored product on space conditions, or other factors which may require ventilation for one or more reasons. Accordingly, while the Energy Standards do not require ventilation for refrigerated warehouses, it is acknowledged that ventilation may be needed in some instances and is left to the determination of the owner and project engineer.

Example 10-26

**Question**

A space that is part of a refrigerated facility is used solely to freeze meat products and not for storage. The design evaporator load is 310 Btu/hr-ft² at the applied conditions. Does the space have to comply with the space requirements (subsections 1, 2, 3, 6, and 7) of the Energy Standards?

**Answer**

No. The design evaporator capacity is more than 240 Btu/hr-ft² and the space is not used for long-term storage. This space meets the definition of a quick chilling space. Therefore, the space does not have to comply with the space requirements (subsections 1, 2, 3, 6, and 7) of the Energy Standards.

Example 10-27

**Question**

A refrigerated warehouse space is used to cool and store melons received from the field. After the product temperature is pulled down, the product is stored in the same space for a few days until being shipped or sent to packaging. The design evaporator capacity is 300 Btu/hr-ft$^2$ at the applied conditions. Does the space have to comply with the space requirements (subsections 1, 2, 3, 6, and 7) of the Energy Standards?

**Answer**

Yes. While the design evaporator capacity is greater than 240 Btu/hr-ft² and the space is used for product pull down for part of the time, the space is also used for holding product after it has been cooled. Accordingly, the space has to comply with the space requirements (subsections 1, 2, 3, 6, and 7) of the Energy Standards.

Comment: The Standard does not define a specific time limit that a quick chill (which for clarity includes quick "freeze") space could operate as a holding space (i.e. at full speed and thus full fan power). The typical high fan power density in a quick chill space, particularly at full speed after the high cooling load has been removed, is very inefficient. Thus a reasonable expectation for a dedicated quick chill space is to allow no more time (at full speed) than is appropriate to remove the product in a normal business cycle of loading, cooling/freezing, and removing product once it has been reduced to temperature. If product is to be held any longer, variable speed is required to reduce fan power. Variable speed requirements are discussed in under mechanical system requirements (sub-section 10.6.3B) of this Chapter 10.

Example 10-28

**Question**

A new refrigeration system serves both storage and quick chilling space. The design refrigeration capacity of the storage space is 500 tons. The design capacity of the quick chilling space is 50 tons. Is the refrigeration system required to meet the requirements of the Energy Standards?

**Answer**

Yes. Since more than 80 percent of the design capacity of the system is serving storage space, the refrigeration system requirements apply.

Example 10-29

**Question**

A new refrigerated warehouse is being constructed, which will include a 1,500 ft² cooler space, and a 2,500 ft² freezer space. Both the cooler and freezer are served by a common refrigeration system. Is the refrigeration system required to comply with the Standard?

**Answer**

Since the cooler and freezer each have less than 3,000 ft² of floor area, they are not required to comply with the Standard. However, they are considered walk-ins and must comply with the requirements of the Appliance Efficiency Regulations (Title 20).

Since the suction group serves a sum total 4,000 ft² of refrigerated floor area, the compressors and condenser are required to comply with subsections 4, 5, and 7 of §120.6(a), which specifically address refrigeration system requirements.

## 10.6.2 Building Envelope Mandatory Requirements

§120.6(a) subsections 1, 2, and 6 of the Energy Standards address the mandatory requirements for refrigerated space insulation, underslab heating, and infiltration barriers.

### 10.6.2.1 Envelope Insulation

*§120.6(a)1*

#### A. Wall and Roof Insulation

Manufacturers must certify that insulating materials comply with *California Quality Standards for Insulating Material* (C.C.R., Title 24, Part 12, Chapters 12-13), which ensure that insulation sold or installed in the state performs according to stated R-values and meets minimum quality, health, and safety standards. These Standards state that all thermal performance tests shall be conducted on materials which have been conditioned at 73.4° ± 3.6°F and a relative humidity of 50 ± 5 percent for 24 hours immediately preceding the tests. The average testing temperature shall be 75° ± 2°F with at least a 40°F temperature difference. Builders may not install insulating materials unless the product has been certified by the Department of Consumer Affairs, Bureau of Home Furnishing and Thermal Insulation. Builders and enforcement agencies shall use the Department of Consumer Affairs Directory of Certified Insulation Material to verify the certification of the insulating material.

Refrigerated spaces with 3,000 ft² of floor area or more shall meet the minimum R-Value requirements shown in Table 10-3.

**Table 10-3: Refrigerated Warehouse Insulation**

| SPACE | SURFACE | MINIMUM R-VALUE (°F·hr·ft²/Btu) |
|---|---|---|
| Freezers | Roof/Ceiling | R-40 |
| | Wall | R-36 |
| | Floor | R-35 |
| | Floor with all heating from productive refrigeration capacity | R-20 |
| Coolers | Roof/Ceiling | R-28 |
| | Wall | R-28 |

The R-values shown in Table 10-3 apply to all surfaces enclosing a refrigerated space, including refrigerated spaces adjoining conditioned spaces, other refrigerated spaces, unconditioned spaces and the outdoors. If a partition is used between refrigerated spaces that are designed to always operate at the same temperature, the requirements do not apply. The R-values are the nominal insulation R-values and do not include other building materials or internal or external "film" resistances.

Example 10-30

**Question**

A refrigerated warehouse designed to store produce at 45°F (7°C) is constructed from tilt-up concrete walls and concrete roof sections. What is the minimum R-value of the wall and roof insulation?

**Answer**

Since the storage temperature is greater than 28°F (-2°C), the space is defined as a cooler. The minimum R-value of the wall and roof insulation according to Table 10-3 is R-28.

Example 10-31

**Question**

A refrigerated warehouse is constructed of a wall section consisting of 4 inches of concrete, 6 inches of medium density (2 lb/ft³) foam insulation and another 4 inches of concrete. The nominal R-value of the foam insulation is R-5.8 per inch. What is the R-value of this wall section for code compliance purposes?

**Answer**

The insulating value of the concrete walls is ignored. The R-value of this wall section for code compliance purposes is based on the 6 inches of foam insulation at R-5.8 per inch, or R-34.8.

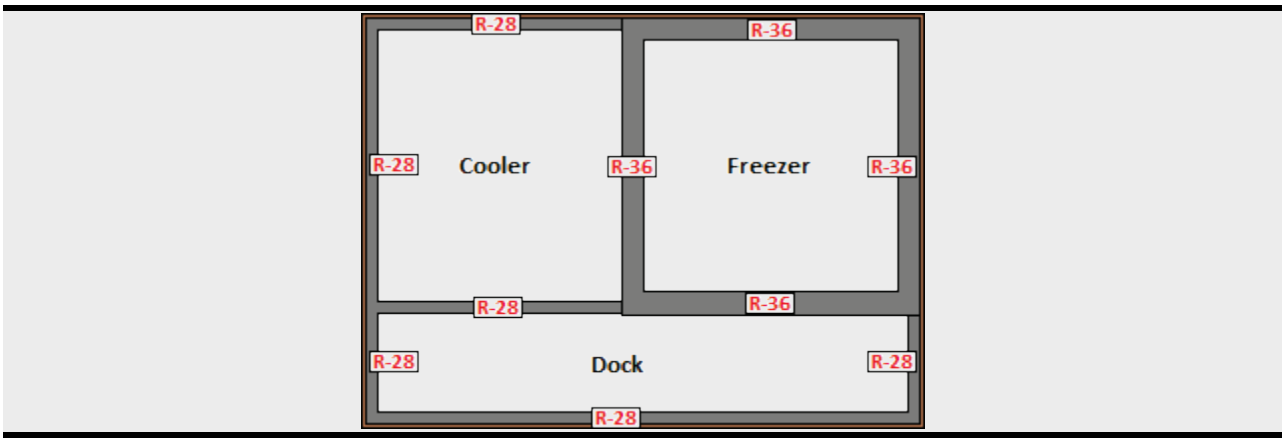
Example 10-32

**Question**

A 35°F cooler space is adjacent to a -10°F freezer space. What is the minimum required insulation R-value of the shared wall between the cooler and freezer spaces?

**Answer**

The minimum insulation R-value requirements should be interpreted to apply to all surfaces enclosing the refrigerated space at the subject temperature. Therefore, since the freezer space walls must be insulated to the minimum R-value requirements shown in Table 10-3, the R-value of the shared wall insulation must be at least R-36. The minimum insulation R-value requirement of the other three cooler walls is R-28. The figure below illustrates this example.



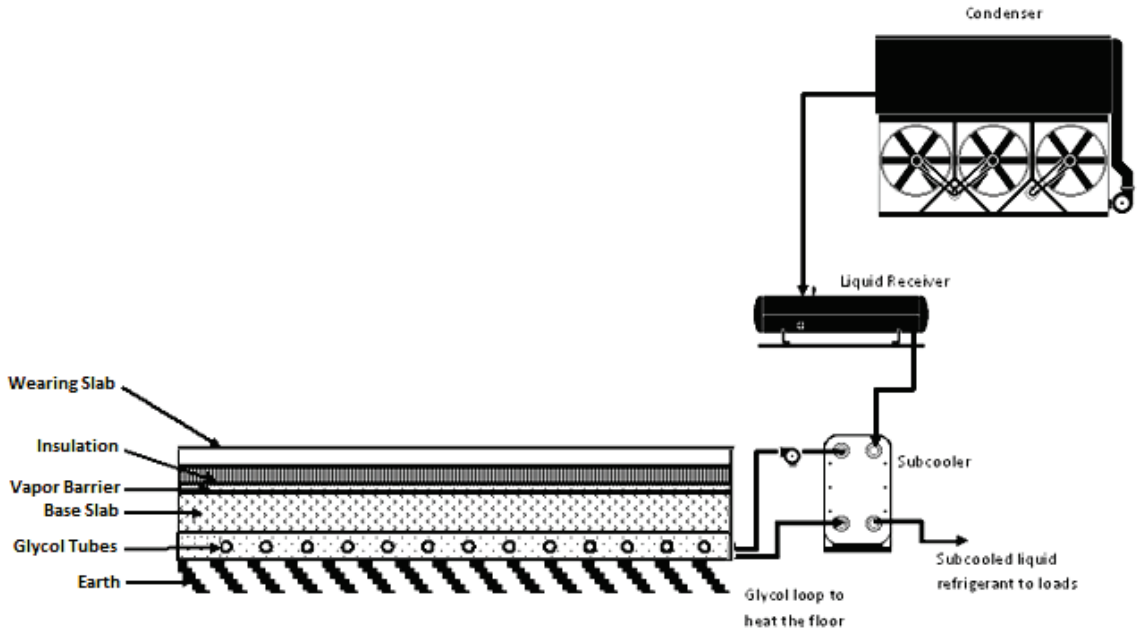

## B. Freezer Floor Insulation

Freezer spaces with 3,000 ft² of floor area or more shall meet the minimum floor insulation R-value requirements shown in Table 10-3. The requirement is a minimum R-value of R-35, with an exception if the underslab heating system increases productive refrigeration capacity, in which case the minimum R-value is R-20.

The predominant insulating material used in freezer floors is extruded polystyrene, which is commonly available in 2”-thick increments, but can optionally be purchased in 1”-thick increments as well. Extruded polystyrene has an R-value of R-5 per inch at standardized rating conditions, and extruded polystyrene panels can be stacked, so the freezer floor can be constructed with R-value multiples of 5 (R-30, R-35, R-40).

A lower floor insulation R-value of R-20 is allowed if all of the underslab heat is provided by an underslab heating system that increases productive refrigeration capacity. An example of an underslab heating system utilizing heat from a refrigerant liquid subcooler is shown in Figure 10-27.

**Figure 10-27: Underslab Heating System that Utilizes Refrigerant Subcooling**



The lower R-value requirement when this type of underslab heating system is used is justified because the increased underslab heat gain to the space due to reduced insulation is offset by the heat extracted from the refrigerant liquid—which is a direct reduction in compressor load. The minimum requirement of R-20 does not mean that R-20 is the optimum or appropriate insulation choice in all installations. Rather, R-20 is a cost-effective trade-off when underfloor heating is obtained via productive refrigeration. Higher insulation levels combined with heating from productive refrigeration would further improve efficiency.

### 10.6.2.2 Underslab Heating Controls

*§120.6(a)2*

Underslab heating systems should be used under freezer spaces to prevent soil freezing and expansion. The underslab heating element might be electric resistance, forced air, or heated fluid; however, underslab heating systems utilizing electric resistance heating elements are not permitted unless they are thermostatically-controlled and disabled during the summer on-peak period. The summer on-peak period is defined by the supplying electric utility, but generally occurs from approximately 12 PM to 6 PM weekdays during the months of May through October. The control system used to control any electric resistance underslab heating elements must automatically turn the elements off during this on-peak period. The control system used to control electric resistance underslab heating elements must be shown on the building drawings, and the control sequence demonstrating compliance with this requirement must be documented on the drawings and in the control system specifications.

### 10.6.2.3 Infiltration Barriers

*120.6(a)6*

Passageways between freezers and higher-temperature spaces, and passageways between coolers and non-refrigerated spaces, shall have an infiltration barrier such as:

- Strip curtains.
- An automatically closing door.
- Air curtain.

Examples of each are shown in the figures below.

**Figure 10-28: Strip Curtains**

**Figure 10-29: Bi-parting automatic door**

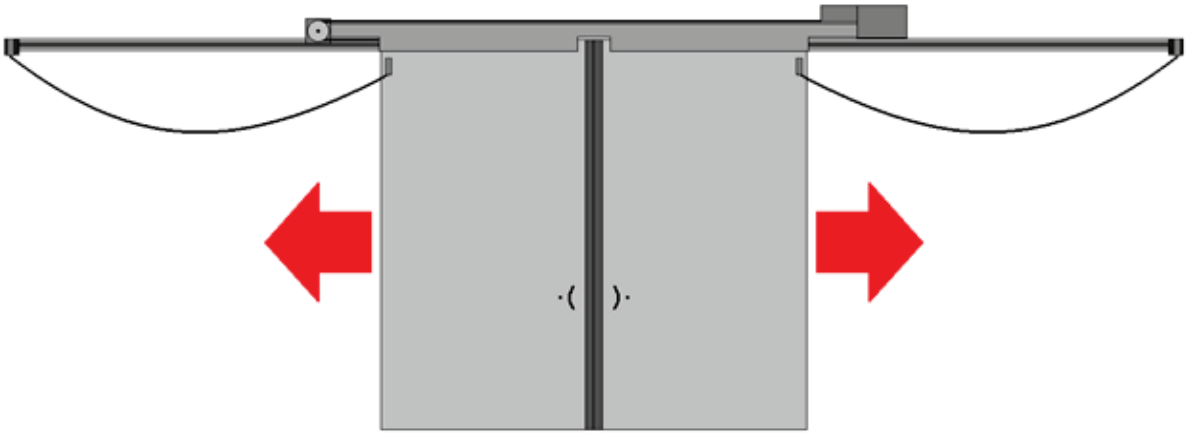


**Figure 10-30: Hinged door with spring-action door closer and door "tight" closer**

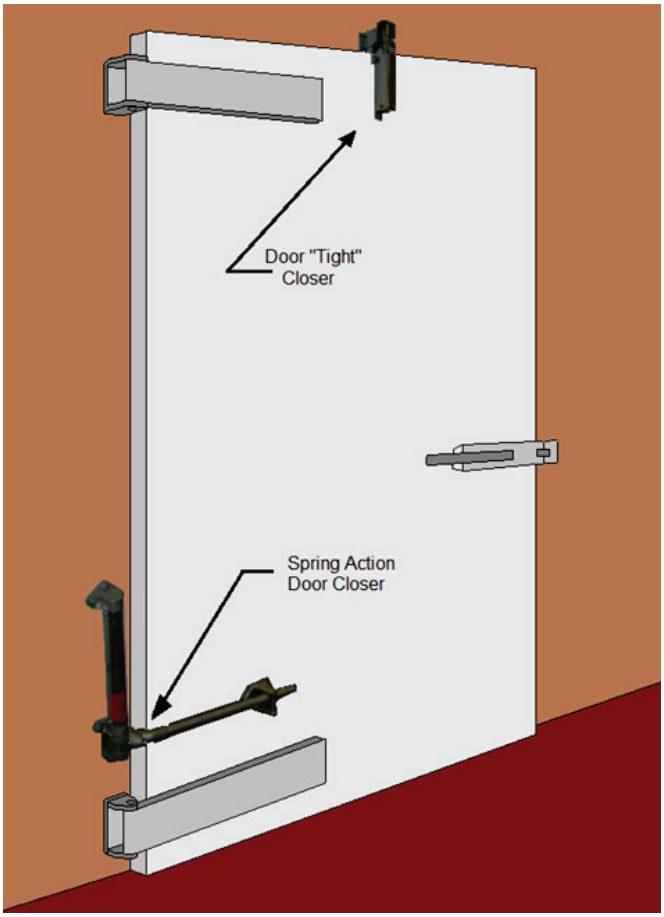
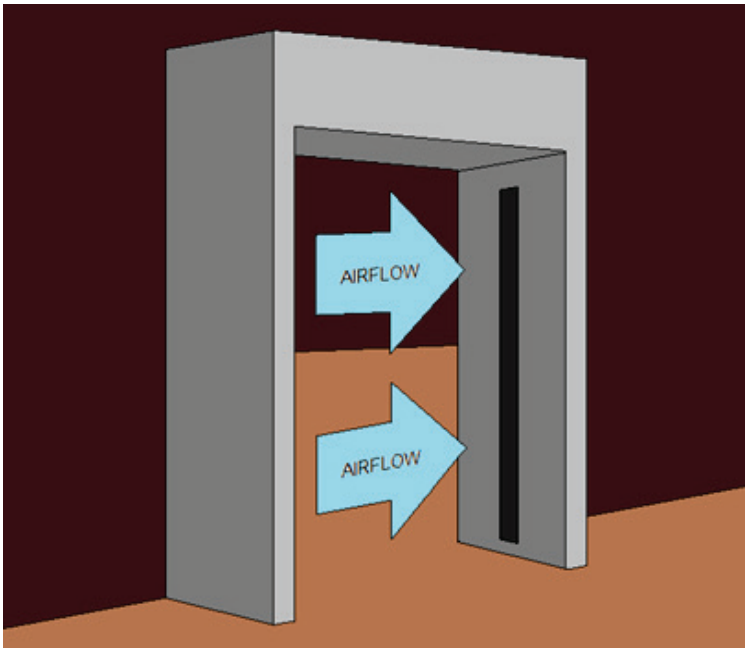


**Figure 10-31: Air Curtain**




The passageways may be for, but are not limited to, people, forklifts, pallet lifts, hand-trucks, or conveyor belts.

Strip curtains are commercial flexible plastic strips made for refrigerated openings with material type, weight and overlap design, designed for the size of the passageway opening and the temperatures of the subject spaces.

An automatically closing door is a door that fully closes under its own power. Examples include:

- Single acting or double acting hinge-mounted doors with a spring assembly or cam-type gravity hinges.
- Powered single-sliding, bi-parting or rollup doors which open based on a pull-cord, proximity or similar sensors, or by operator signal and close automatically through similar actions or after a period of time sufficient to allow passageway transit.

An air curtain is a commercial fan powered assembly intended to reduce air infiltration and designed by its manufacturer for use on refrigerated warehouse passageways, and for use on the opening size and the temperatures for which it is applied.

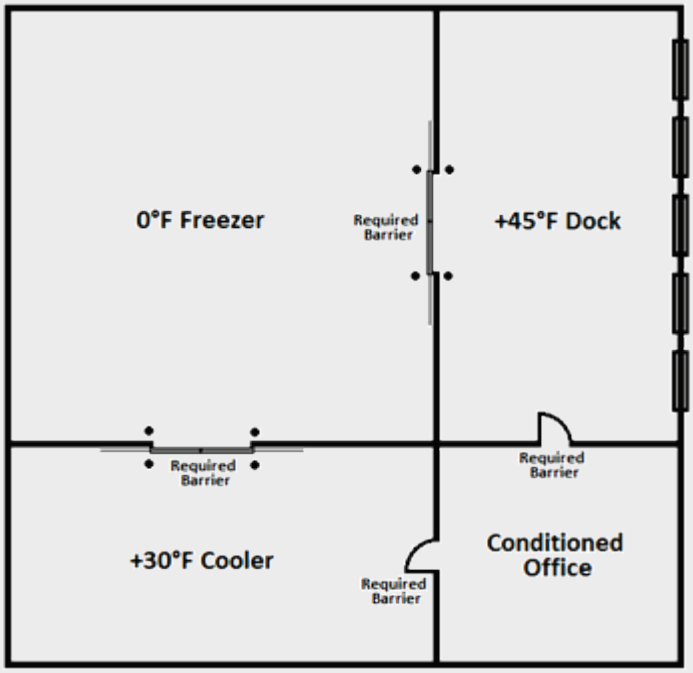
There are two exceptions to the requirements for infiltration barriers:

1. Openings with less than 16 square. feet of opening area, such as small passageways for conveyor belts
2. Loading dock doorways for trailers.

Example 10-33

**Question**

A refrigerated warehouse includes a freezer, cooler, a refrigerated dock, and a conditioned office, as shown in the following figure. Where are infiltration barriers required?




**Answer**

Infiltration barrier are required between all spaces including the hinge-mounted doors between the dock and the office. The dock doors do not require infiltration barriers.

Example 10-34

**Question**

A refrigerated warehouse is being constructed for a flower distribution company. Strip curtains cannot be used on the doors because the strips will damage the flowers when the pallet jack passes through. Is the warehouse still required to have infiltration barriers?

**Answer**

Yes, the warehouse is required to have infiltration barriers. If strip curtains cannot be used, the designer may choose another method, such as double-acting hinged doors, sliding or rollup doors with automatic door closers.

### 10.6.2.4 Acceptance Requirements

*§120.6(a)7*

The Energy Standards include acceptance test requirements for electric resistance underslab heating systems in accordance with NA7.10.1. The test requirements are described in Chapter 13 and the Reference Nonresidential Appendix NA7.10. The test requirements are described briefly in the following paragraph.

### A. Electric Resistance Underslab Heating System

*NA7.10.1*

The acceptance requirements include functional tests that are to be performed to verify that the electric resistance underslab heating system automatically turns off during a test on-peak period.

## 10.6.3 Mechanical Systems Mandatory Requirements

### 10.6.3.1 Overview

This section addresses mandatory requirements for mechanical systems serving refrigerated spaces. Mechanical system components addressed by the Energy Standards include evaporators (air units), compressors, condensers, and refrigeration system controls. The requirements for each of these components are described in the following sections. The requirements apply to all system and component types with the exception of the specific exclusions noted in §120.6(a). The following figures identify some of the common system and component configurations that fall under §120.6(a).

Figure 10-32 is a schematic of a single stage system with direct expansion (DX) evaporator coils. Figure 10-33 identifies a single stage system with flooded evaporator coils; while Figure 10-34 shows a single stage system with pump recirculated evaporators. Figure 10-35 is a schematic of a typical two-stage system with an intercooler between the compressor stages. Figure 10-36 is a single-stage system with a water-cooled condenser and fluid cooler.

**Figure 10-32: Single Stage System with DX Evaporator Coil**

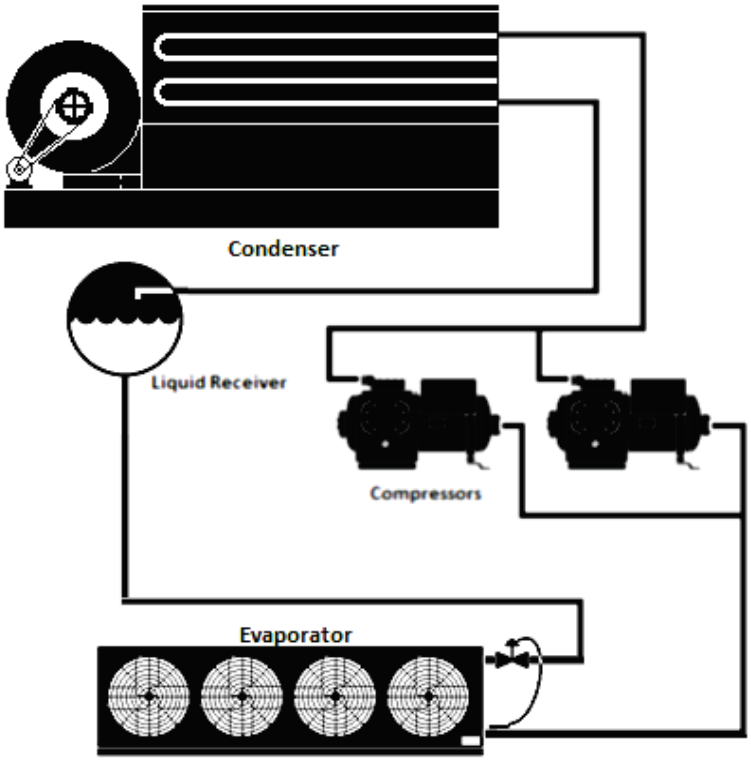


**Figure 10-33: Single Stage System with Flooded Evaporator Coil**

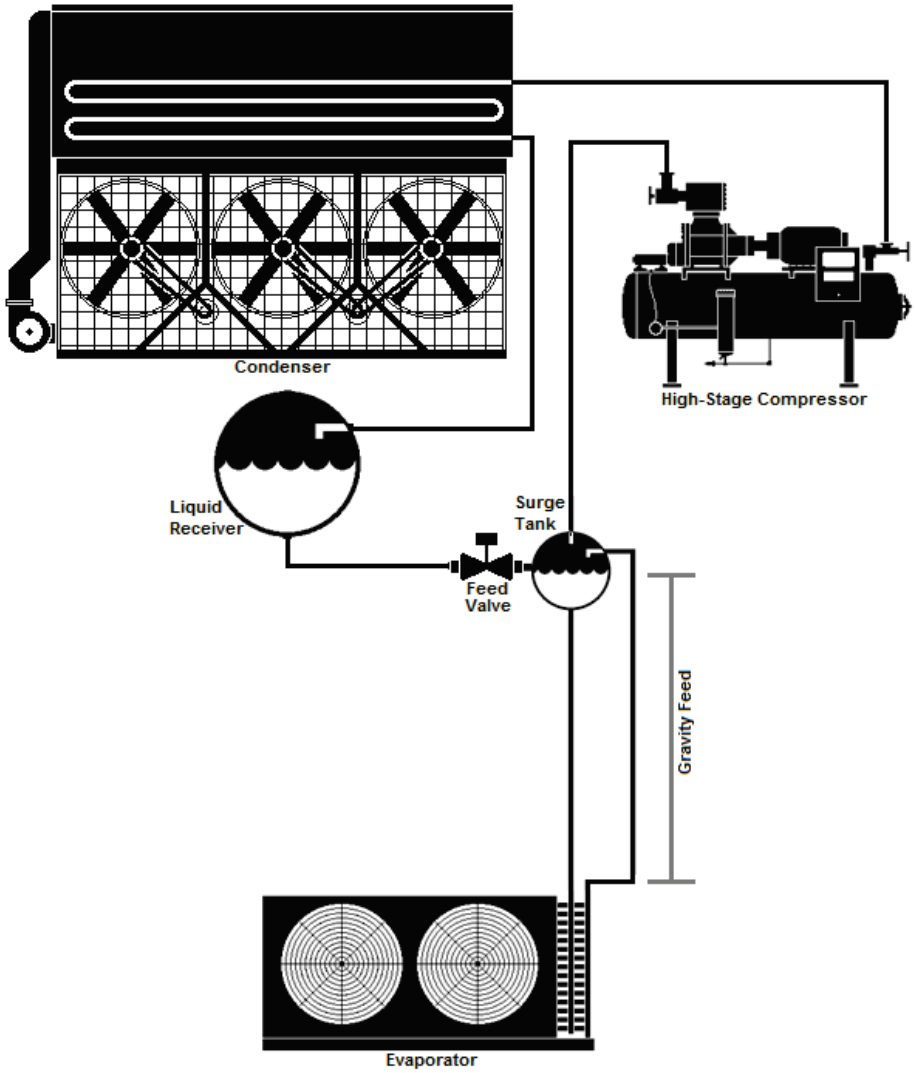


**Figure 10-34: Single Stage System with Pump Recirculated Evaporator Coils**

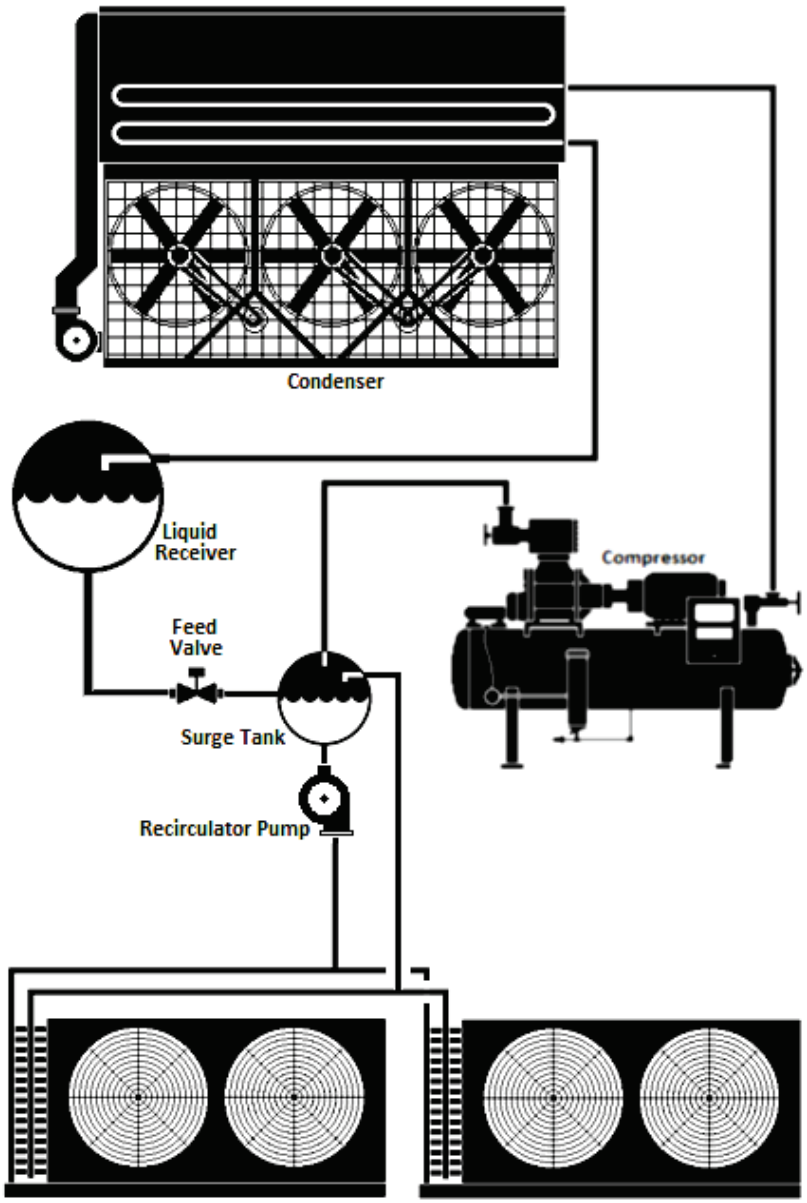


**Figure 10-35: Two-Stage System with Flooded Evaporator Coil**

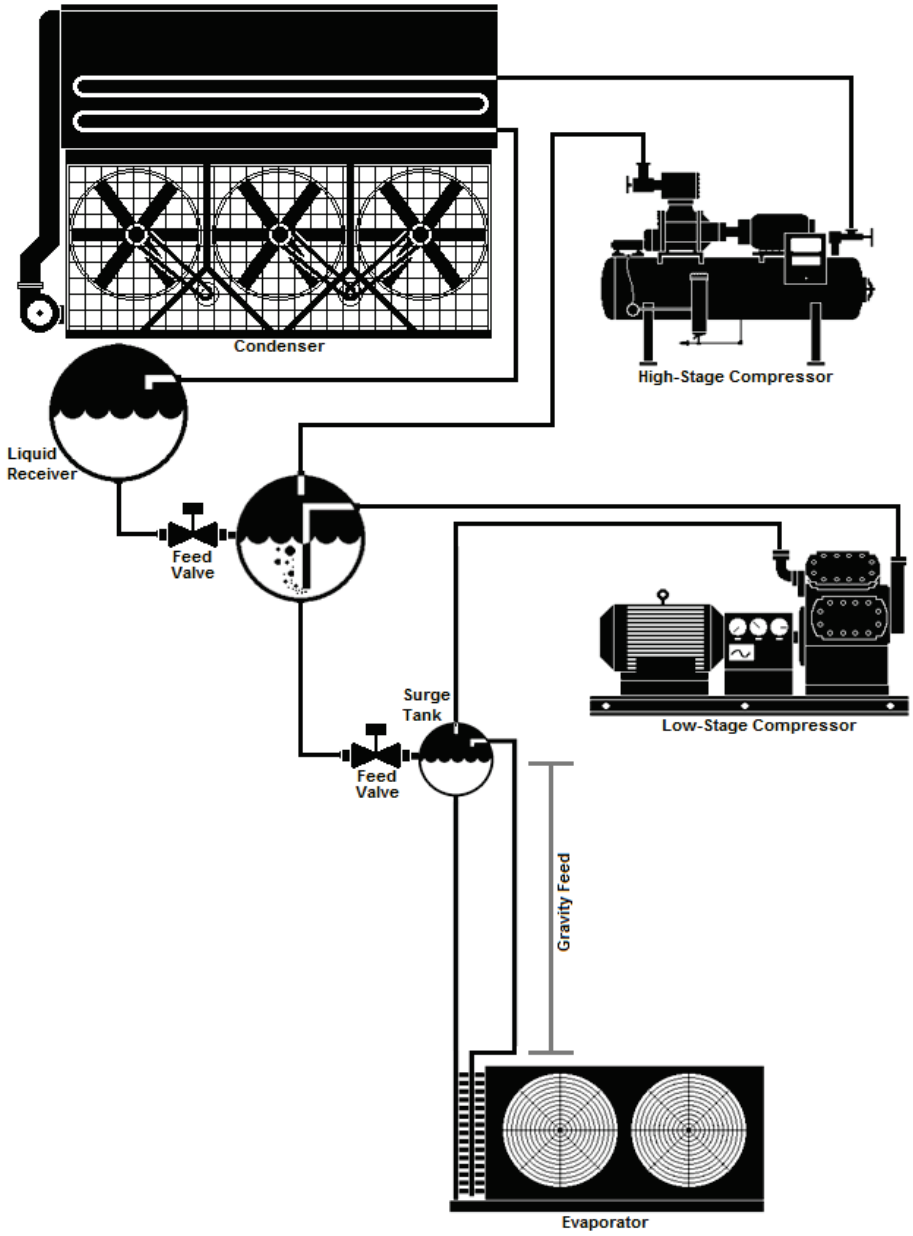
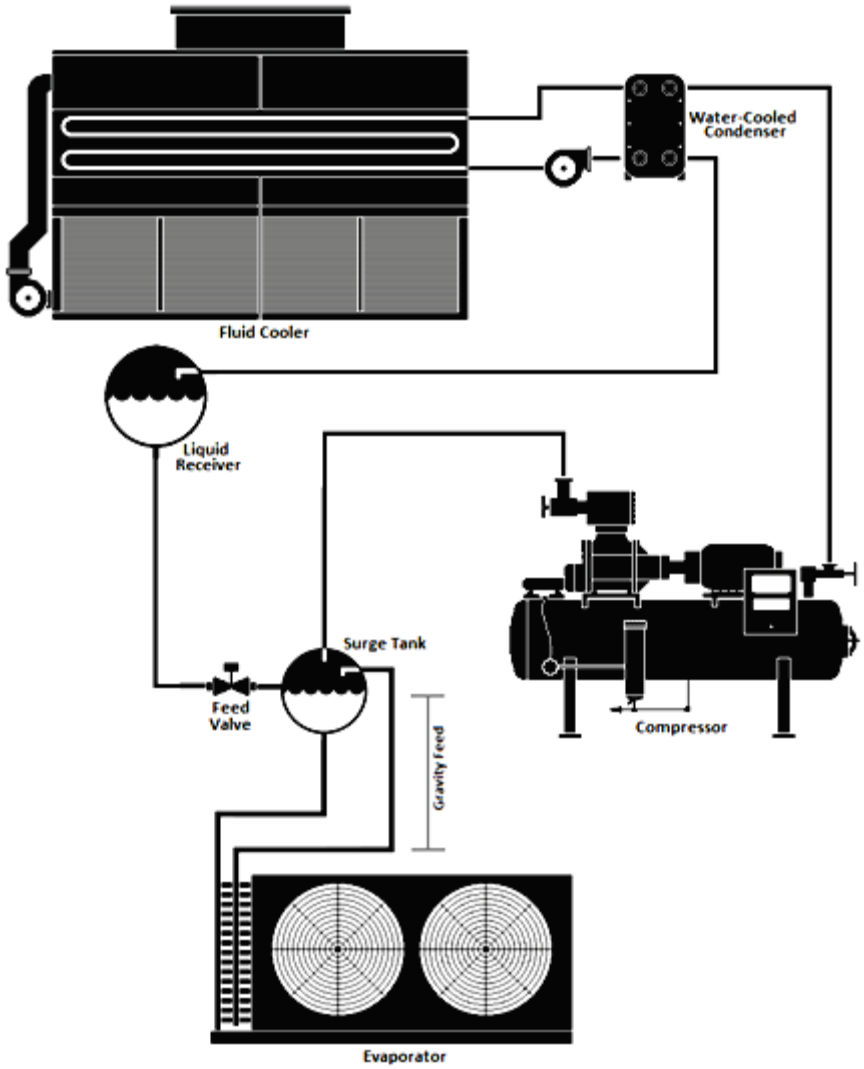


**Figure 10-36: Single System with Water-Cooled Condenser Served by Fluid Cooler**




### 10.6.3.2 Evaporators

*§120.6(a)3*

New fan-powered evaporators used in coolers and freezers must meet the fan motor type, efficiency, and fan control requirements outlined in the Energy Standards.

#### A. Allowed Fan Motor Types

Single phase fan motors less than 1 horsepower and less than 460 Volts must be either electronically-commutated (EC, also known as Brushless DC) or must have an efficiency of 70 percent or more when rated in accordance with NEMA Standard MG 1-2006 at full load rating conditions. This requirement is designed to reduce fan power in small evaporator fans.

#### B. Fan Motor Control

The speed of all evaporator fans served by either a suction group with multiple compressors or by a single compressor with variable capacity capability must be controlled in response to space temperature or humidity using a continuously variable

speed control method. Two-speed control of evaporator fans is not an acceptable control method.

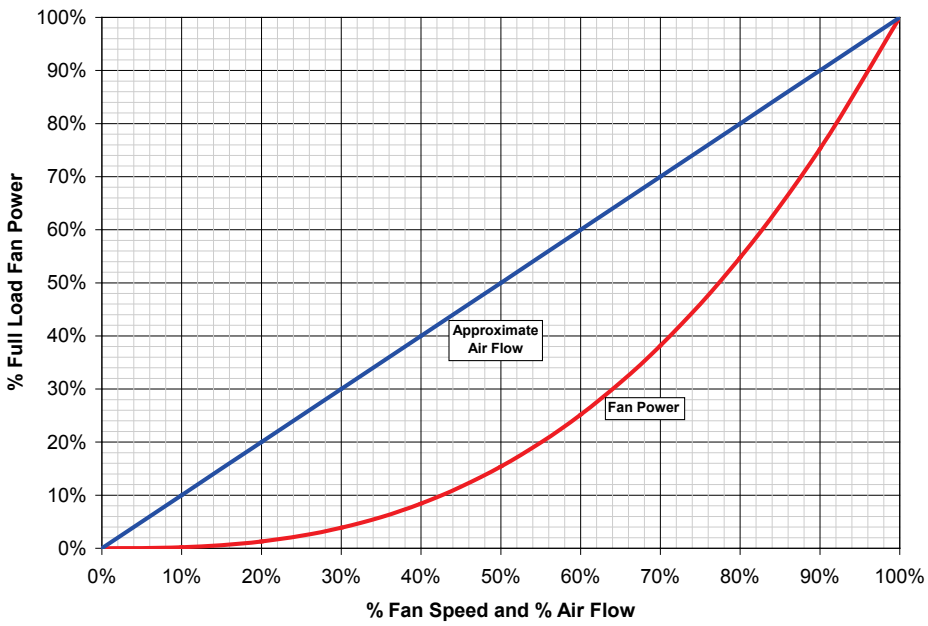
The fan speed is controlled in response to space temperature or humidity. Fan speed should increase proportionally when the space temperature is above setpoint and decrease when the space temperature is at or below setpoint, with refrigerant supply and pressure being maintained in the evaporator cooling coil. Fan speed is equivalent to air volume being circulated, resulting in direct control of cooling capacity, analogous to "variable air volume" cooling in commercial buildings. The control logic requires design and tuning to provide "variable" capacity operation.

The use of humidity as the control variable for speed control is very limited in practice but is included in the Energy Standards to accommodate special strategies for humidity-sensitive perishable product. Control logic is these applications often will employ humidity in conjunction with temperature.

The intent of this requirement is to take advantage of the "third-power" fan affinity law, which states that the percentage of required fan motor power is approximately equal to the cube of the percentage of fan speed, while the airflow is linearly proportional to the fan speed. For example, a fan running at 80 percent speed requires approximately 51 percent ($80\%^3 = 51\%$) power while providing approximately 80 percent airflow (Figure 10-37). Actual power is somewhat higher due to inefficiencies and drive losses. This shows the relationship between fan speed and both required fan power and approximate airflow.

There is no requirement in the Energy Standards for the minimum speed setting (i.e. how low the fan speed must go at minimum load). Variable speed controls of evaporator fans has commonly used minimum speeds of 80 percent or lower on direct expansion coils and 70 percent or lower on flooded or recirculated coils. The allowable minimum fan speed setting is to be determined by the refrigeration system designer. The fan speed may be adjusted or controlled to maintain adequate air circulation in order to ensure product integrity and quality.

**Figure 10-37: Relationship between Fan Speed and Required Power**



Correct fan speed control requires the associated system suction pressure to be controlled such that evaporator capacity is sufficient to meet space loads. If the evaporator suction pressure is too high relative to the desired room temperature, the evaporator fans will run at excessively high speed and energy savings will not be realized. If floating suction pressure automation is used to optimize the suction pressure setpoint, suction pressure should only be allowed to float up after fan speeds are at minimum and should be controlled to float back down prior to increasing fan speeds.

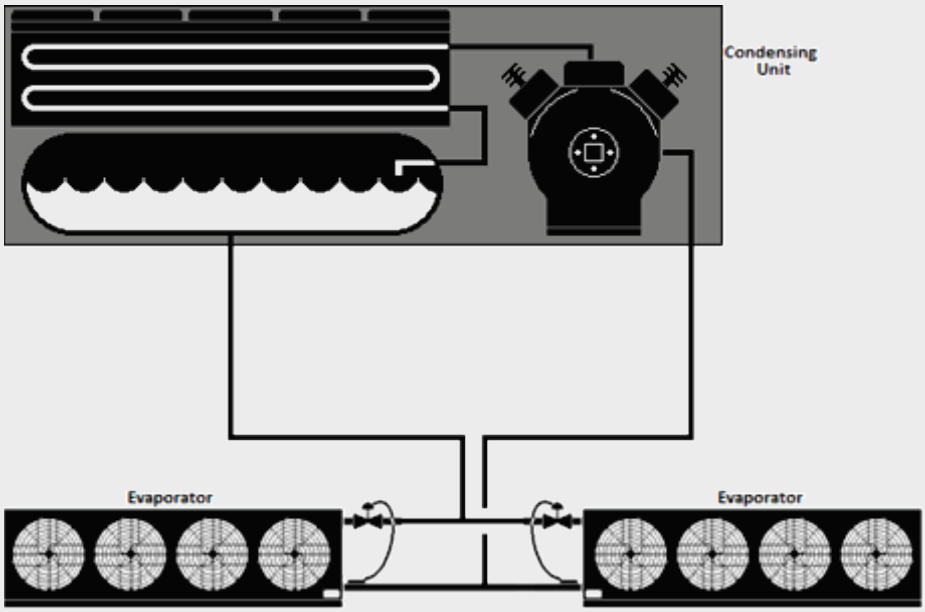
The Energy Standards have three exceptions to the evaporator variable speed requirement:

1. In case of a replacement, addition or alteration of existing evaporators with no variable speed control, the variable speed control of the evaporators is mandatory only if the replacement, addition or alteration is done for all the evaporators in an existing space. *[Exception 1 to §120.6(a)3B]*
2. A Controlled Atmosphere (CA) storage where products that require 100 percent of the design airflow at all times are stored may be exempt from the variable speed control requirement. A licensed engineer must certify that the products in the cooler require continuous airflow at 100 percent speed. Variable speed control must be implemented if the space will also be used for non-CA product or operation. *[Exception 2 to §120.6(a)3B]*
3. The variable speed control is not mandatory for spaces that are used solely for quick chilling or quick freezing of products. Such spaces have design cooling capacities that are greater than 240 Btu/hr-ft² of floor area, which is equivalent to 2 tons per 100ft² of floor area. However, variable speed control must be implemented if the spaces are used for storage for any length of time, regardless of how much refrigeration capacity is installed in the space. [Exception 3 to §120.6 (a) 3B].

Example 10-35

**Question**

A split system with a packaged air-cooled condensing unit with a single 30 HP compressor with unloaders serves two direct expansion evaporators in a 3,200 ft² cooler. Are the evaporator fans required to have variable speed control?



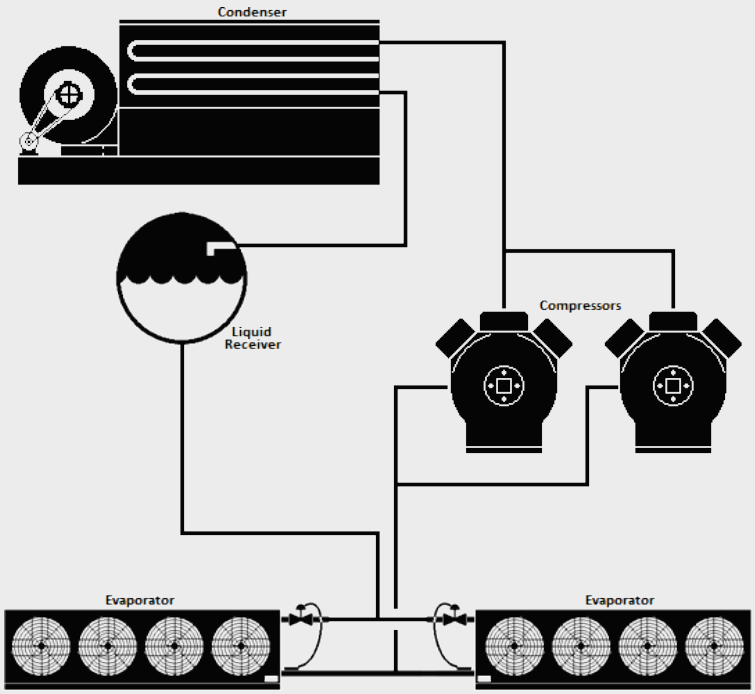
**Answer**

Yes. Since the compressor has a variable capacity capability in the form of unloaders, the evaporator fans are required to have variable speed control.

Example 10-36

**Question**

A refrigeration system utilizes two reciprocating compressors without variable capacity capability connected in parallel, and serves multiple evaporators in a 10,000 ft² cooler. Are the evaporator fans required to have variable speed control?




**Answer**

Yes. Since the evaporators are served by more than one compressor, they must have variable speed control, even though the compressors are not equipped with capacity control devices (e.g. unloaders).

In practice, the designer should consider the steps of capacity necessary to allow stable control. For small systems, the designer may consider use of control that proportionally controls both compressor capacity steps and speed steps in unison. As long as this control scheme is in response to space temperature, it would be consistent with the Energy Standards.

Example 10-37

**Question**

A -20°F (-29°C) freezer has a number of recirculated evaporator coils that were selected to meet the design load at a 10°F (5.5°C) temperature difference (TD). The evaporator fan motors utilize variable speed drives and the control system varies the fan speed in response to space temperature. What should the freezer saturated suction temperature be in order to achieve proper control and savings – by allowing fan speed control to act as the primary means of temperature control.

**Answer**

Since the coils were designed at a 10°F (5.5°C) TD and the target freezer temperature is -20°F (-29°C), the saturated evaporating temperature should be -30°F (-34°C) (-20°F minus 10°F); with the compressor controlled at a lower temperature, based on the design piping pressure drop. For example with 2°F (1°C) of piping losses, the compressor control setpoint would be -32°F (-36°C).

The purpose of this example is to show how evaporator temperature and coil capacity can be considered and maintained in order to achieve proper variable speed fan operation and savings. Settings could be further fine-tuned through observation of the required suction pressure to meet cooling loads and achieve minimum fan speeds average load periods, yet with a suction pressure no lower than necessary.

Example 10-38

**Question**

An existing refrigerated warehouse space has eight existing evaporators that do not have variable speed control. Six of the eight evaporators are being replaced with new evaporators. Do the new evaporators require variable speed control?

**Answer**

No. Since all the evaporators are not being replaced, the new evaporators do not require variable speed control.

The reason for this is that effective space temperature control would often require that the entire space utilize a consistent control scheme which could require a disproportional cost. While not required by the Energy Standards, in many instances it may still be very cost effective to add variable speed control to existing as well as new evaporators in this situation.

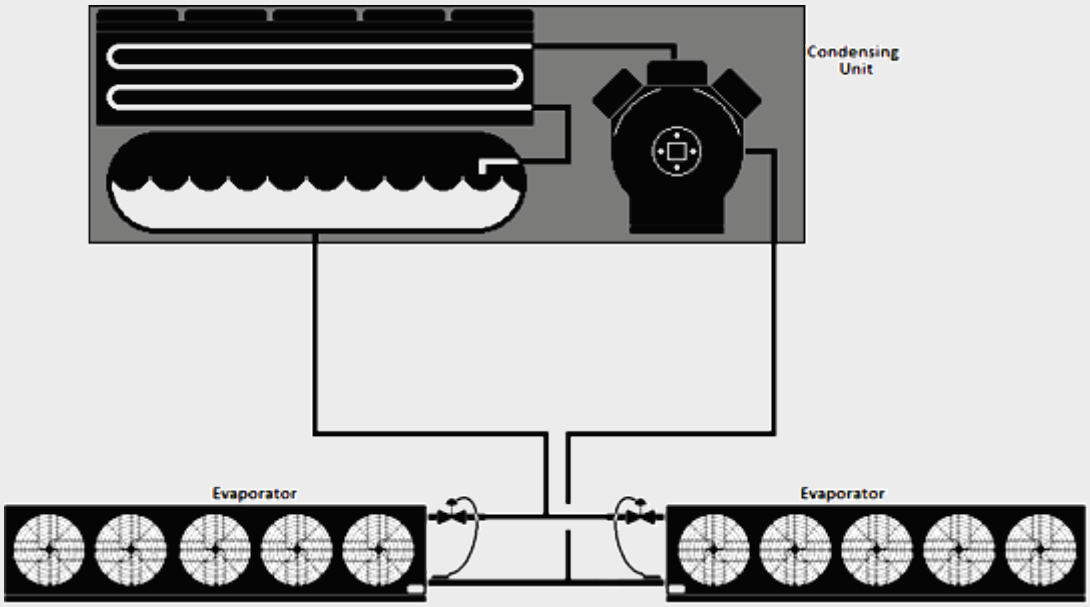
Continuously variable speed control is not mandatory for evaporators that are served by a single compressor that does not have variable capacity capability (i.e. the compressor cycles on and off in response to space temperature). For these systems, evaporator airflow must be reduced by at least 40 percent when the compressor is off. This can be accomplished in a number of ways, for example:

- Two speed evaporator fan control, with speed reduced by at least 40% when cooling is satisfied and the compressor is off.
- Turning off a portion of the fans in each evaporator to accomplish at least 40% reduction in fan power. Typically baffles are required to prevent reverse flow through fans that are turned off.
- Turning off all fans when the compressor is off. With this strategy a duty cycle can be employed to provide period forced fan operation to maintain air circulation, if the “on” period is limited to 25% of the duty cycle while the compressor is off.

Example 10-39

**Question**

A split system with a packaged air-cooled condensing unit utilizing a single cycling compressor without unloaders serves two evaporators in a cooler. Each evaporator has five fans. What options does the system designer have to meet the requirements for evaporator coils served by a single cycling compressor?




**Answer**

Multiple methods can be used to reduce airflow by at least 40% when the compressor is off, or turn all fans off with a 25% duty cycle.

Example 1: The designer may specify two-speed fans, or utilize variable frequency drives or other speed-reduction devices to reduce the fan speed to 60% or less when the compressor is off.

Example 2: The designer may utilize controls that cycle at least four of the ten fans off when the compressor is cycled off. This would most likely be accomplished by cycling two fans off on each evaporator.

### 10.6.3.3 Condensers

*§120.6(a)4*

New condensers on new refrigeration systems must follow the condenser sizing, fan control, and efficiency requirements as described in §120.6(a)4.

#### A. Condenser Sizing

§120.6(a)4A and §120.6(a)4B describe minimum sizing requirements for new condensers serving new refrigeration systems. Fan-powered evaporative condensers, as well as water-cooled condensers served by fluid coolers and cooling towers are covered in §120.6(a)4A. Fan-powered air-cooled condensers are covered by §120.6(a)4B.

Condensers must be sized to provide sufficient heat rejection capacity under design conditions while maintaining a specified maximum temperature difference between the refrigeration system saturated condensing temperature (SCT) and ambient temperature. The design condenser capacity shall be greater than the calculated combined Total Heat of Rejection (THR) of the dedicated compressors that are served by the condenser. If

multiple condensers are specified, then the combined capacity of the installed condensers shall be greater than the calculated heat of rejection. When determining the design THR for the purpose of this requirement, reserve or backup compressors may be excluded from the calculations.

There is no limitation on the type of condenser that may be used. The choice may be made by the system designer, considering the specific application, climate, water availability, etc.

The Energy Standards include an exception to §120.6(a)4A and 4B for condensers serving refrigeration systems for which more than 20% of the design cooling load comes from quick chilling or freezing space, or process (non-space) refrigeration cooling. The Energy Standard defines quick chilling or freezing space as a space with a design refrigeration evaporator capacity greater than 240 Btu/hr-ft² of floor area, which is equivalent to 2 tons per 100ft² of floor area, at system design conditions.

The sizing requirements for air-cooled condensers (§120.6(a)4B) do not apply to the condensers included in air-cooled condensing units. Condensing units include compressor(s), condenser, liquid receiver, and control electronics are packaged in a single product. However, this exception applies only if the compressor(s) in the condensing units have a nameplate size totaling less than 100HP.

Example 10-40

**Question**

A new food processing facility is being constructed that will include an 800ft² blast freezer, a holding freezer, and a loading dock. The design evaporator capacity of the blast freezer is 40 TR (tons of refrigeration). The combined evaporator capacity of the freezer and loading dock is 60 TR. Does the condenser group have to comply with the sizing requirements in §120.6(a)4A?

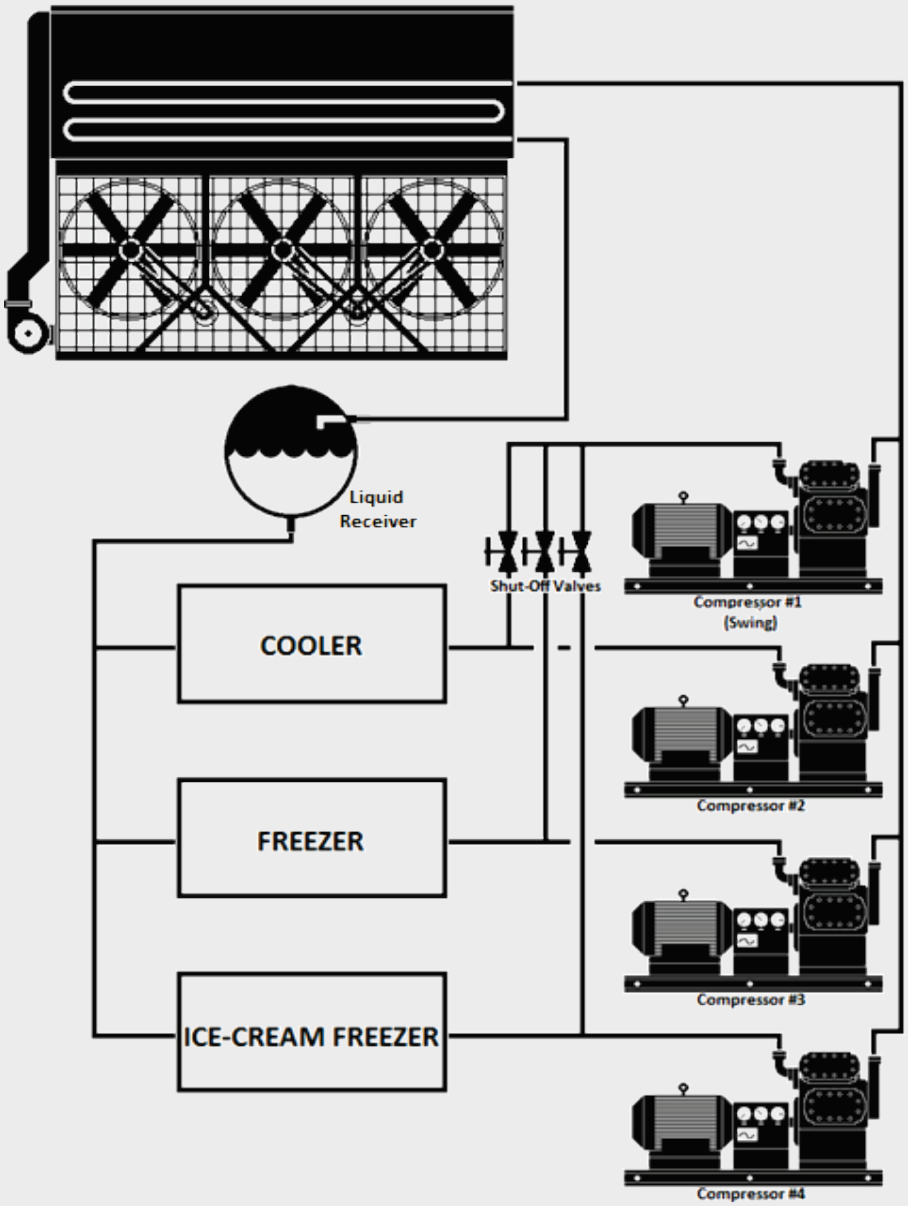
**Answer**

The blast freezer evaporator capacity divided by the floor area is 100TR/800ft², which is equal to 12.5 TR/100ft². That means this particular blast freezer is deemed quick freezing space by the Energy Standards. Therefore, the condenser group serving the refrigeration system does not have to comply with §120.6(a)4A, since 40% (i.e. greater than 20%) of the design refrigeration capacity is from quick freezing.

Example 10-41

**Question**

The refrigerated warehouse system shown below has a backup or “swing” compressor. Does the heat rejection from this compressor need to be included in the condenser sizing calculations?




**Answer**

It depends.

A swing compressor may be designed solely for back-up of multiple suction groups or it may be included in one suction group and necessary to meet the design load of that suction group, but in an emergency is also capable of providing back-up for other compressors. If the compressor is solely for use as back-up, it would be excluded from the heat rejection calculation for the purposes of the Energy Standards. In this case, the calculations would include the heat of rejection from Compressors 2, 3, and 4 and would exclude Compressor 1.

### 1. Sizing of Evaporative Condensers, Fluid Coolers, and Cooling Towers

*§120.6(a)4A*

§120.6(a)4A provides maximum design SCT values for evaporative condensers as well as systems consisting of a water-cooled condenser served by a cooling tower or fluid cooler. For the purpose of this Section, designers should use the 0.5 percent design wet-bulb temperature (WBT) from Table 10-4 – Design Day Data for California Cities in the Reference Joint Appendices JA2 to demonstrate compliance with this requirement. The maximum design SCT requirements are listed in Table 10-4 below.

**Table 10-4: Maximum Design SCT Requirements for Evaporative Condensers and Water-Cooled Condensers Served by Cooling Towers and Fluid Coolers**

| 0.5% DESIGN WET-BULB TEMPERATURE | MAXIMUM DESIGN SCT |
|---|---|
| ≤ 76°F (24°C) | Design WBT plus 20°F (11°C) |
| Between 76°F (24°C) and 78°F (26°C) | Design WBT plus 19°F (10.5°C) |
| ≥ 78°F (26°C) | Design WBT plus 18°F (10°C) |

Example 10-42

**Question**

A refrigerated warehouse is being constructed in Fresno, California. The refrigeration system will be served by an evaporative condenser. What is the sizing requirement for the condenser selected for this system?

**Answer**

The 0.5% design wetbulb temperature (WBT) from Joint Appendix JA-2 for Fresno, California, is 73°F. Therefore, the maximum design SCT for the refrigerant condenser is 73°F + 20°F = 93°F. The selected condenser for this system must be capable of rejecting the total system design THR at 93°F SCT, and 73°F WBT.

Example 10-43

**Question**

What is the minimum size for a condenser for a refrigeration system with the following parameters?

Located in Fresno, California
Design SST: 10°F
Suction group: 3 equal-sized dedicated screw compressors (none are backup units)
Evaporative condenser
200 TR (tons refrigeration) cooling load

**Answer**

From the previous example, it was determined that the design wetbulb temperature (WBT) to demonstrate compliance for Fresno, California is 73°F and the maximum design SCT for the evaporative condenser is 93°F (73°F + 20°F). We will assume the system designer determined a 2°F loss between the compressors and condenser. The designer first calculates the total heat of rejection (THR) for the suction group at the design conditions of 10°F SST and 95°F SCT. The selected compressors each have a rated capacity of 240 tons of refrigeration and will absorb 300 horsepower at the design conditions. Therefore, the calculated THR for one compressor is:

240 TR /Compressor x 3 compressor x 12,000 Btuh/TR + 300HP x 2,545 Btuh/HP = 10,930,500 Btuh

To comply with the Energy Standards, a condenser (or group of condensers) must be selected that is capable of rejecting at least 10,930,500 Btu/hr at 93°F SCT and 73°F WBT.

### 2. Sizing of Air-Cooled Condensers

*§120.6(a)4B*

§120.6(a)4B provides maximum design SCT values for air-cooled condensers. For the purpose of this Section, Designers should use the 0.5 percent design dry-bulb temperature (DBT) from Table 10-4 – Design Day Data for California Cities in the Reference Joint Appendices JA2 to demonstrate compliance with this requirement.

Standard practice is for published condenser ratings to assume the capacity of air-cooled condensers is proportional to the temperature difference (TD) between SCT and DBT, regardless of the actual ambient temperature entering the condenser. For example, the capacity of an air-cooled condenser operating at an SCT of 80°F with a DBT of 70°F is assumed to be equal to the same unit operating at 110°F SCT and 100°F DBT, since the TD across the condenser is 10°F in both examples. Thus, unlike evaporative condensers, the requirement for air-cooled condensers does not have varying sizing requirements for different design ambient temperatures.

However, the Energy Standards have different requirements for air-cooled condensers depending on the space temperatures served by the refrigeration system. The maximum design SCT requirements are listed in Table 10-5 below:

**Table 10-5: Maximum Design SCT Requirements for Air-Cooled Condensers**

| REFRIGERATED SPACE TYPE | SPACE TEMPERATURE | MAXIMUM SCT |
|---|---|---|
| Cooler | ≥ 28°F (-2°C) | Design DBT plus 15°F (8.3°C) |
| Freezer | < 28°F (-2°C) | Design DBT plus 10°F (5.6°C) |

Often, a single refrigeration system and its associated condenser will serve a mix of cooler and freezer spaces. In this instance, the maximum design SCT shall be a weighted average of the requirements for cooler and freezer spaces, based on the design evaporator capacity of the spaces served.

Example 10-44

**Question**

An air-cooled condenser is being sized for a system that has half of its installed capacity serving cooler space and the other half serving freezer space. What is the design TD to be added to the design dry bulb temperature?

**Answer**

Using air-cooled condensers for coolers have a design approach of 15°F (8.3°C) and for freezers a design approach of 10°F (5.6°C). When a system serves freezer and cooler spaces, a weighted average should be used based on the installed capacity. To calculate the weighted average, multiply the percent of the total installed capacity dedicated to coolers by 15°F (8.3°C). Next, multiply the percent of the total installed capacity dedicated to freezers by 10°F (5.6°C). The sum of the two results is the design condensing temperature approach. In this example, the installed capacity is evenly split between freezer and cooler space. As a result, the design approach for the air-cooled condenser is 12.5°F (6.9°C).

*(50% x 15°F) + (50% x 10°F) = 7.5°F + 5°F = 12.5°F*

### B. Fan Control

Condenser fans for new air-cooled or evaporative condensers, or fans on cooling towers or fluid coolers used to reject heat on new refrigeration systems, must be continuously variable speed. Variable frequency drives are commonly used to provide continuously variable speed control of condenser fans although controllers designed to vary the speed of electronically commutated motors may be used to control these types of motors. All

fans serving a common high side, or cooling water loop for cooling towers and fluid coolers, shall be controlled in unison. Thus, in normal operation, the fan speed of all fans within a single condenser or set of condensers serving a common high-side should modulate together, rather than running fans at different speeds or staging fans off. However, when fan speed is at the minimum practical level usually no higher than 10-20%, the fans may be staged off to further reduce condenser capacity. As load increases, fans should be turned back on prior to significantly increasing fan speed, recognizing a control band is necessary to avoid excessive fan cycling.

To minimize overall system energy consumption, the condensing temperature setpoint must be continuously reset in response to ambient temperatures, rather than using a fixed setpoint value. This strategy is also termed ambient-following control, ambient-reset, wetbulb following and drybulb following—all referring to control logic which changes the condensing temperature target in response to ambient conditions at the condenser. The control system calculates a target saturated condensing temperature that is higher than the ambient temperature by a predetermined temperature difference (i.e. the condenser control TD). Fan speed is then modulated according to the calculated target SCT. The target SCT for evaporative condensers or water-cooled condensers (via cooling towers or fluid coolers) must be reset according to ambient wet bulb temperature, and the target SCT for air-cooled condensers must be reset according to ambient dry bulb temperature.

The condenser control TD is not specified in the Energy Standards. The nominal control value is often less than the condenser design TD; however the value for a particular system is left up to the system designer. Since the intent is to utilize as much condenser capacity as possible without excessive fan power, common practice for refrigerated warehouse systems is to optimize the control TD over a period of time such that the fan speed is in a range of approximately 60-80% during normal operation (i.e. when not at minimum SCT). While not required, evaporative condensers and systems utilizing fluid coolers and cooling towers may also vary the condenser control TD as a function of actual WBT, to account for the properties of moist air, which reduce the effective condenser capacity at lower wetbulb temperatures.

The minimum saturated condensing temperature setpoint must be 70°F (21°C) or less. For systems utilizing halocarbon refrigerants with glide, the SCT setpoint shall correlate with a midpoint temperature (between the refrigerant bubble-point and dew point temperatures) of 70°F (21°C) or less. As a practical matter, a maximum SCT setpoint is also commonly employed to set an upper bound on the control setpoint in the event of a sensor failure and to force full condenser operation during peak ambient conditions. This value should be set high enough that it does not interfere with normal operation.

Split air-cooled condensers are sometimes used for separate refrigeration systems, with two circuits and two rows of condenser fans. Each condenser half would be controlled as a separate condenser. If a condenser has multiple circuits served by a common fan or set of fans, the control strategy may utilize the average condensing temperature or the highest condensing temperature of the individual circuits as the control variable for controlling fan speed.

Alternative control strategies are permitted to the condensing temperature reset control required in §120.6(a)4E. The alternative control strategy must be demonstrated to provide equal or better performance, as approved by the Executive Director.

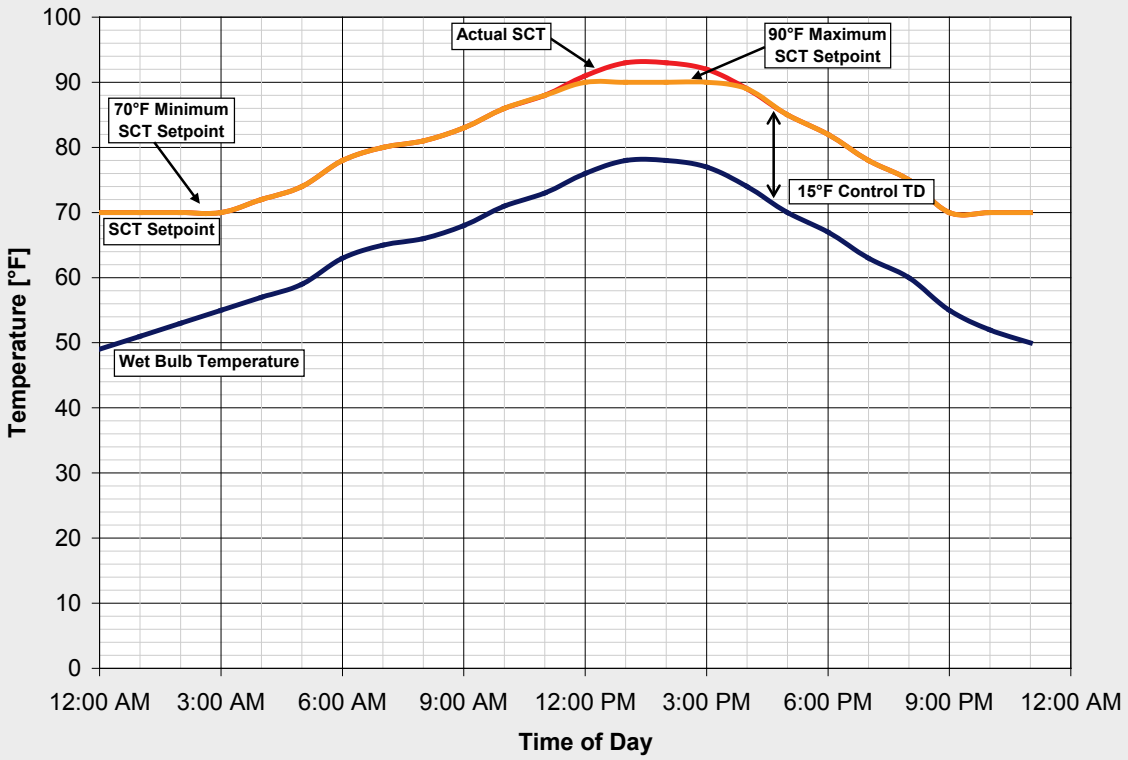
Example 10-45

**Question**

A refrigerated warehouse with evaporative condensers is being commissioned. The control system designer has utilized a wet bulb-following control strategy to reset the system saturated condensing temperature (SCT) setpoint. The refrigeration engineer has calculated that adding a TD of 15°F (8.3°C) above the ambient wet bulb temperature should provide a saturated condensing temperature setpoint that minimizes the combined compressor and condenser fan power usage throughout the year. What might the system SCT and SCT setpoint trends look like over an example day?

**Answer**

The following figure illustrates what the actual saturated condensing temperature and SCT setpoints could be over an example day using the wet bulb-following control strategy with a 15°F (8.3°C) TD and also observing the 70°F (21°C) minimum condensing temperature requirement. As the figure shows, the SCT setpoint is continuously reset to 15°F (8.3°C) above the ambient wet bulb temperature until the minimum SCT setpoint of 70°F is reached. The figure also shows a maximum SCT setpoint (in this example, 90°F (32.2°C) which may be utilized to limit the maximum control setpoint, regardless of the ambient temperature value or TD parameter.



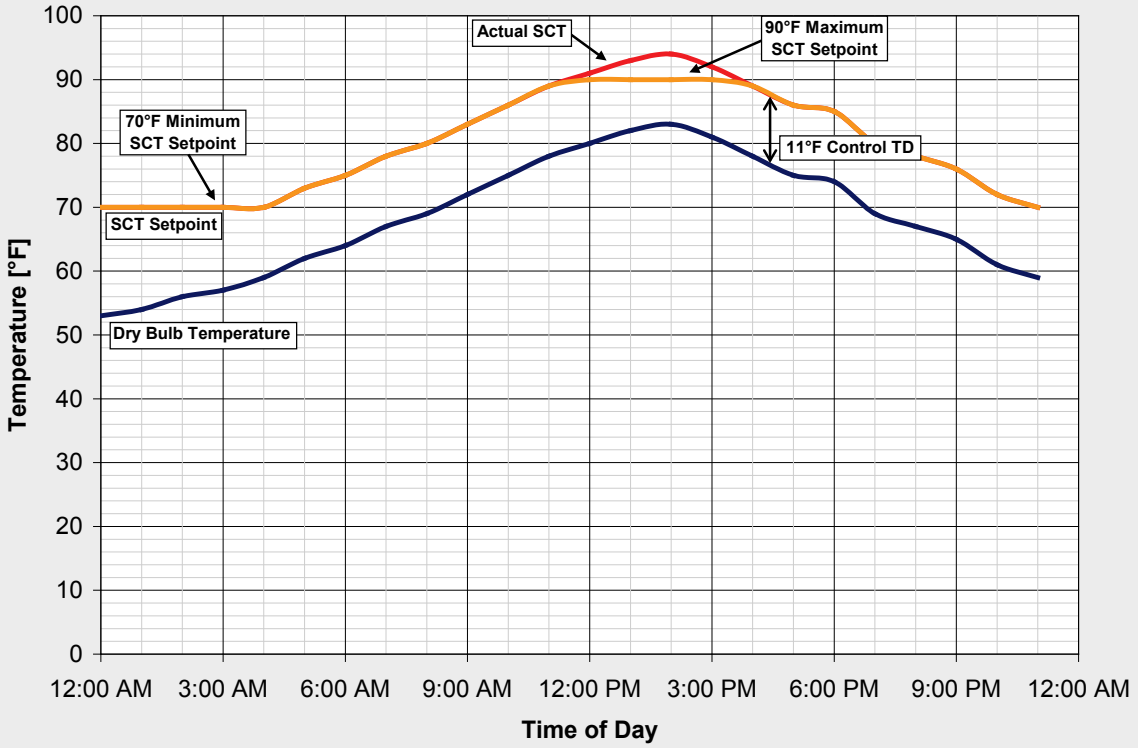

Example 10-46

**Question**

A cold storage facility with an air-cooled condenser is being commissioned. The control system designer has utilized a drybulb-following control strategy to reset the system saturated condensing temperature (SCT) setpoint. The refrigeration engineer has calculated that adding a TD of 11°F (6.1°C) above the ambient drybulb temperature should provide a saturated condensing temperature setpoint that minimizes the combined compressor and condenser fan power usage throughout the year. What might the system SCT and SCT setpoint trends look like over an example day?

**Answer**

The following figure illustrates the actual saturated condensing temperature and SCT setpoints over an example day using the drybulb-following control strategy with an 11°F (6.1°C) TD and also observing the 70°F (21°C) minimum condensing temperature requirement. As the figure shows, the SCT setpoint is continuously reset 11°F (6.1°C) above the ambient drybulb temperature, but is bounded by the minimum and maximum SCT setpoints. The figure also shows a maximum SCT setpoint (in this example, 90°F (32.2°C) which may be utilized to limit the maximum control setpoint, regardless of the ambient temperature value or TD parameter.




## C. Condenser Specific Efficiency

*§120.6(a)4F*

Requirements for Design Condensing Temperatures relative to design ambient temperatures, as described above for §120.6(a)4A&B, help assure that there is enough condenser capacity to keeping condensing temperatures compressor head pressures at reasonable levels. However the sizing requirements do not address condenser efficiency. For example, rather than providing amply sized condenser surface area, a condenser selection could consist of a small condenser area using a large motor to blow a large amount of air through the heat exchanger surface to achieve the design condenser TD. However, this would come at the expense of excessive fan motor horsepower. Also, relatively high fan power consumption can result from using condenser fans that have poor fan efficiency or low fan motor efficiency. §120.6(a)4F addresses these and other factors affecting condenser fan power by setting minimum specific efficiency requirements for condensers.

All newly installed indoor and outdoor evaporative condensers, and outdoor air-cooled condensers to be installed on new refrigeration systems shall meet the minimum specific efficiency requirements shown in Table 10-6.

**Table 10-6: Fan-powered Condensers – Minimum Specific Efficiency Requirements[1]**

| CONDENSER TYPE | REFRIGERANT TYPE | MINIMUM SPECIFIC EFFICIENCY | RATING CONDITION |
|---|---|---|---|
| Outdoor Evaporative-Cooled with THR Capacity > 8,000 MBH | All | 350 Btuh/Watt | 100°F Saturated Condensing Temperature (SCT), 70°F Outdoor Wetbulb Temperature |
| Outdoor Evaporative-Cooled with THR Capacity < 8,000 MBH and Indoor Evaporative-Cooled | All | 160 Btuh/Watt | |
| Outdoor Air-Cooled | Ammonia | 75 Btuh/Watt | 105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Drybulb Temperature |
| | Halocarbon | 65 Btuh/Watt | |
| Indoor Air-Cooled | All | Exempt | |

Condenser specific efficiency is defined as:

*Condenser Specific Efficiency = Total Heat Rejection (THR) Capacity / Input Power*

The total heat rejection capacity is at the rating conditions of 100°F Saturated Condensing Temperature (SCT) and 70°F outdoor wetbulb temperature for evaporative condensers, and 105°F SCT and 95°F outdoor drybulb temperature for air-cooled condensers. Input power is the electric input power draw of the condenser fan motors (at full speed), plus the electric input power of the spray pumps for evaporative condensers. The motor power is the manufacturer's published applied power for the subject equipment, which is not necessarily equal to the motor nameplate rating. Power input for secondary devices such as sump heaters shall not be included in the specific efficiency calculation.

As shown in Table 10-6 the Energy Standards have different minimum efficiencies depending on the type of condenser that is being used. The different classifications of condenser are:

- Outdoor, evaporative, THR greater than 8,000 MBH at specific efficiency rating conditions
- Outdoor, evaporative, THR less than 8,000 MBH at specific efficiency rating conditions
- Indoor, evaporatively cooled
- Outdoor, air-cooled, ammonia refrigerant
- Outdoor, air-cooled, halocarbon refrigerant
- Indoor, air-cooled

The data published in the condenser manufacturer's published rating for capacity and power shall be used to calculate specific efficiency. For evaporative condensers, manufacturers typically provide nominal condenser capacity, and tables of correction factors that are used to convert the nominal condenser capacity to the capacity at various applied condensing temperatures and wetbulb temperatures. Usually the manufacturer publishes two sets of correction factors: one is a set of "heat rejection" capacity factors,

[1] Table is copied directly from TABLE 120.6-B FAN-POWERED CONDENSERS – MINIMUM EFFICIENCY REQUIREMENTS

while the others are “evaporator ton” capacity factors. Only the “heat rejection” capacity factors shall be used to calculate the condenser capacity at the efficiency rating conditions for the purpose of determining compliance with this section.

For air-cooled condensers, manufacturers typically provide the capacity at a given temperature difference (TD) between SCT and drybulb temperature. Manufacturers typically assume that air-cooled condenser capacity is linearly proportional to TD; the catalog capacity at 20°F TD is typically twice as much as at 10°F TD. The condenser capacity for air-cooled condensers at a TD of 10°F shall be used to calculate efficiency. If the capacity at 10°F TD is not provided, the capacity shall be scaled linearly.

Depending on the type of condenser, the actual manufacturer’s rated motor power may vary from motor nameplate in different ways. Air cooled condensers with direct-drive OEM motors may use far greater input power than the nominal motor horsepower would indicate. On the other hand, evaporative condenser fans may have a degree of safety factor to allow for higher motor load in cold weather conditions (vs. the 100°F SCT/70°F WBT specific efficiency rating conditions). Thus, actual motor input power from the manufacturer must be used for direct-drive air-cooled condensers while for large (i.e. > 8,000 MBH) evaporative condensers and other belt drive condensers, the full load motor rating is generally conservative but manufacturer’s applied power should be used whenever possible to more accurately determine specific efficiency.

Example 10-47

**Question**

An evaporative condenser is being considered for use in an outdoor application on a new refrigerated warehouse. The refrigerant is ammonia. The condenser manufacturer’s catalog provides the following information:

| Model Number | Base Heat Rejection (MBH) |
|---|---|
| A441 | 4410 |
| B487 | 4866 |
| C500 | 4998 |
| D551 | 5513 |
| E559 | 5586 |
| F590 | 5895 |
| G591 | 5909 |
| H598 | 5983 |
| I631 | 6306 |
| J637 | 6365 |

| Condensing Temperature (°F) | Entering Wetbulb Temperature (°F) | | | | | |
|---|---|---|---|---|---|---|
| | 62 | 64 | 66 | 68 | 70 | 72 |
| 95 | 0.88 | 0.92 | 0.97 | 1.02 | 1.08 | 1.16 |
| 96.3 | 0.84 | 0.88 | 0.92 | 0.97 | 1.02 | 1.09 |
| 97 | 0.83 | 0.86 | 0.90 | 0.94 | 0.99 | 1.05 |
| 98 | 0.80 | 0.83 | 0.87 | 0.91 | 0.96 | 1.01 |
| 99 | 0.77 | 0.80 | 0.84 | 0.87 | 0.92 | 0.97 |
| 100 | 0.75 | 0.78 | 0.81 | 0.84 | 0.88 | 0.93 |

For this example, model number D551 is being considered. Elsewhere in the catalog, it states that condenser model D551 has two 7.5 HP fan motors and one 5 HP pump motor. Fan motor efficiencies and motor loading factors are not provided. Does this condenser meet the minimum efficiency requirements?

**Answer**

First, the condenser capacity must be calculated at the efficiency rating condition. From Table 10-4, we see that the rating conditions for an outdoor evaporative condenser are 100°F SCT, 70°F WBT. From the Base Heat Rejection table above, we see the nominal capacity for model D551 is 5,513 MBH. From the Heat Rejection Capacity Factors table, we see that the correction factor for 100°F SCT, 70°F WBT is 0.88. The capacity of this model at specific efficiency rating conditions is 5,513 MBH / 0.88 = 6,264 MBH. Since 6,264 MBH is less than 8,000 MBH, we can see from Table10-4 that the minimum specific efficiency requirement is 160 (Btu/hr)/Watt.

To calculate input power, we will assume 100% fan and pump motor loading and minimum motor efficiency since the manufacturer has not yet published actual motor input power at the specific efficiency rating conditions. We look up the minimum motor efficiency from Nonresidential Appendix NA-3: Fan Motor Efficiencies. For a 7.5 HP 4-pole open fan motor, the minimum efficiency is 91.0%. For a 5 HP 6-pole open pump motor, the minimum efficiency is 89.5%. The fan motor input power is calculated to be:

*2 Motors x 7.5 HP/Motor x 746 Watts/HP x 100% assumed loading/ 91% Efficiency = 12.297 Watts*

The pump motor input power is calculated to be:

*1 Motors x 5 HP/Motor x 746 Watts/HP x 100% assumed loading/ 89.5% Efficiency = 4.168 Watts*

The combined input power is therefore:

*12.297 Watts + 4.168 Watts = 16.464 Watts*

Note: Actual motor power should be used when available (see notes in text).

Finally, the efficiency of the condenser is:

*(6,264 MBH x 1000 Btuh/MBH) / 16.464 Watts = 381 Btuh/Watt*

381 Btu/hr per Watt is higher than the 160 Btu/hr per Watt requirement. This condenser meets the minimum efficiency requirements.

### D. Condenser Fin Spacing

According to §120.6(a)4G air-cooled condensers shall have a fin density no greater than 10 fins per inch. Condensers with higher fin densities have a higher risk of fouling with airborne debris. This requirement does not apply to air-cooled condensers that utilize a micro-channel heat exchange surface, since this type of surface is not as susceptible to permanent fouling in the same manner as traditional tube-and-fin condensers with dense fin spacing.

## 10.6.3.4 Compressors

*§120.6(a)5*

Compressors on new refrigeration systems must follow the design and control requirements as described in §120.6(a)5.

### A. Minimum Condensing Temperature

Floating head control is one of the largest energy savings measures applied to refrigeration systems. This control attempts to keep condensing temperatures as low as possible (while not consuming too much condenser fan energy) as this reduces compressor head pressure which directly impact compressor energy. When ambient temperatures are low the primary constraint on how low the condensing temperature can be reset is the design requirements of the compressor and associated system components.

§120.6(a)5A addresses the compatibility of the compressor design and components with the requirements for floating head control. All compressors that discharge to the

condenser(s) and all associated components (coalescing oil separators, expansion valves for liquid injection oil cooling, etc.) must be capable of operating at a condensing temperature of 70°F (21°C) or less. Note that oil separator sizing is often governed by the minimum condensing temperature as well as other factors, such as the maximum suction temperature. Suction temperatures above the design value may occur under floating suction temperature control schemes.

The system designer should also keep in mind that other design parameters such as piping run-lengths or evaporator defrost requirements must be considered to meet this requirement.

### B. Screw Compressor Control at Part-Load

New open-drive screw compressors in new refrigeration systems with a design saturated suction temperature (SST) of 28°F or lower shall vary compressor speed as the primary means of capacity control. The compressor speed shall reduce to the manufacturer-specified minimum speed before unloading via slide valve. Similarly, when the load increases, the compressor shall increase to 100% slide valve before increasing speed. This requirement applies only to compressors discharging to the condenser (i.e. single stage or the high stage of a two-stage system) and only to suction groups that consist of a single compressor.

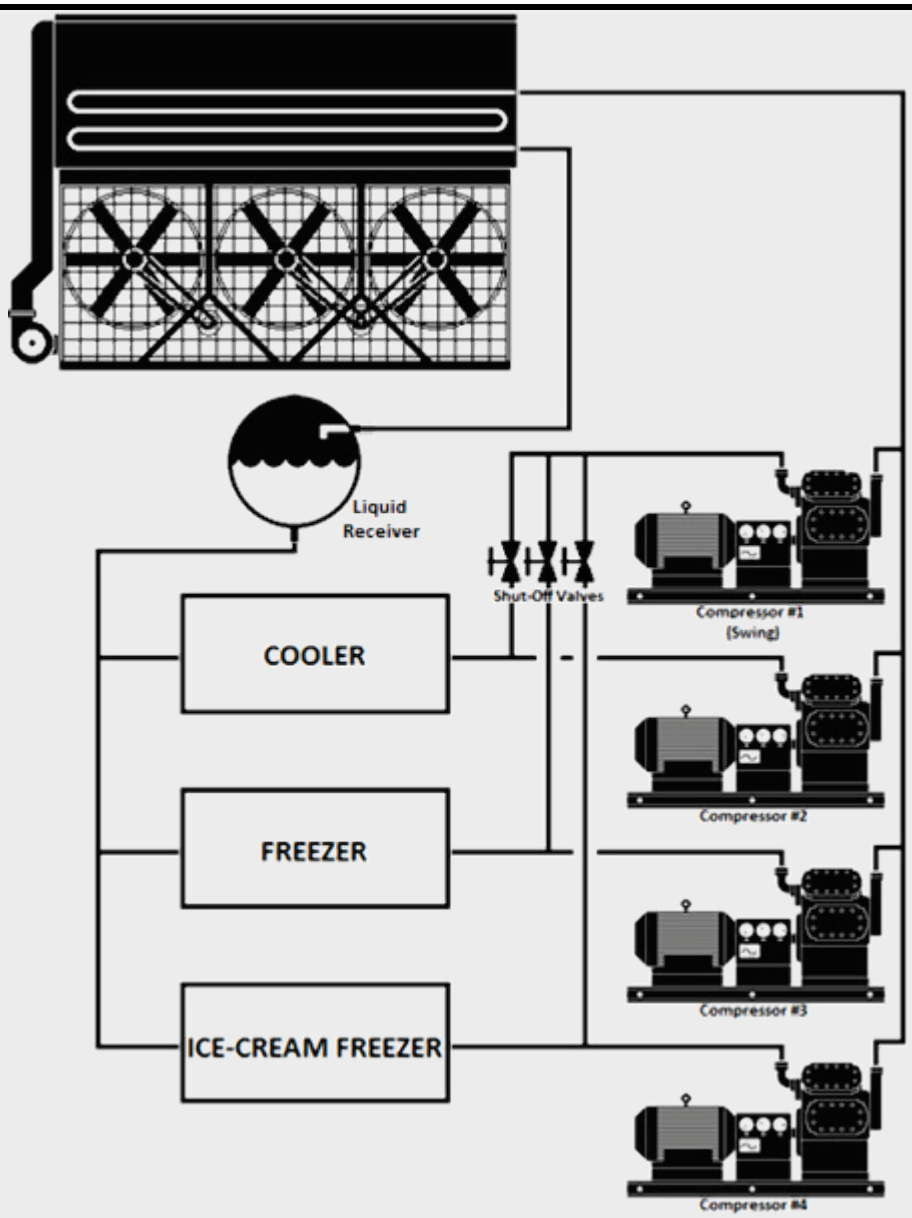
An exception to §120.6(a)5 is provided for compressors on a refrigeration system with more than 20% of the design cooling load from quick chilling or freezing space, or non-space process refrigeration cooling. The “refrigeration system” refers to the entire associated system, (i.e. the refrigerant charge) not the suction group. While variable speed compressor control may be cost effective in many instances and may be considered by the system designer, this exception exists to allow for situations such as seasonal processes with low operating hours or loads that may be precisely matched to a fully loaded compressor.

New screw compressors with a motor nameplate power greater than 150HP shall incorporate the capability to automatically vary the volume ratio (i.e. variable Vi) in order to optimize efficiency at off-design operating conditions.

Example 10-48

**Question**

The system shown below has three 200 HP open-drive screw compressors serving three different suction levels and one 200 HP backup or swing open-drive screw compressor that can be connected by valve into any of the three suction lines. Does this count as having more than one compressor per suction group and exempt the compressors from the requirements in §120.6(a)5B?




**Answer**

Probably not. The Exception 1 to §120.6(a)5B only applies when a suction group has two or more dedicated compressors. A compressor that is used solely as backup does not count as a dedicated compressor. As a result, all compressors (1, 2, 3, and 4) in the example above must comply with §120.6(a)5B and use variable speed control as the primary means of capacity control.

However, if Compressor 1 is actually required to meet the design load of one of the suction groups, it could be considered part of that suction group and variable speed control would not be required. Whether a swing compressor is really a back-up compressor or part of a suction group should be apparent from the design loads and capacities listed in the design documents.

### 10.6.3.5 Acceptance Requirements

*§120.6(a)7*

The Energy Standards have acceptance test requirements for:

- Electric underslab heating controls.
- Evaporator fan motor controls.
- Evaporative condensers.
- Air-cooled condensers.
- Variable speed compressors.

These test requirements are described in Chapter 13 and the Reference Nonresidential Appendix NA7.10. They are described briefly in the following paragraphs.

#### A. Electric Underslab Heating Controls

*NA7.10.1*

Controls for underslab electric heating controls, when used for freeze protection on freezer floors, are tested to ensure heat is automatically turned off during summer on-peak electric periods.

#### B. Evaporator Fan Motor Controls

*NA7.10.2*

Evaporator equipment and controls are checked for proper operation. The controls are tested to ensure the fan speed automatically varies in response the temperature and/or humidity of the space.

#### C. Evaporative Condensers

*NA7.10.3.1*

Evaporative condensers and variable speed fan controls are checked to ensure the required minimum SCT setpoint of 70°F or lower is implemented, and the condenser fans continuously vary in unison to maintain a target temperature difference between the SCT and the wet bulb temperature. Trends of wet bulb temperature and SCT can be used to verify the controls over time.

The condenser control TD or offset is a key parameter in fine-tuning the operation of the fans and maximizing the energy savings. In best practice, this control setting should be adjusted during average loaded so that the fan average 60-80% speed when in the control range (i.e. between the minimum and maximum SCT setpoints).

#### D. Air-cooled Condensers

*NA7.10.3.2*

Air-cooled condensers and variable speed fan controls are checked to ensure the required minimum SCT setpoint of 70°F or lower is implemented, and the condenser fans continuously vary in unison to maintain a target temperature difference between the SCT and dry bulb temperature. Trends of dry bulb temperature and SCT can be used to verify the controls over time.

The condenser control TD is a key parameter in fine-tuning the operation of the fans and maximizing the energy savings. This control setting should be adjusted during average loaded so that condenser capacity is effectively utilized but fan speed is not excessive.

#### E. Variable Speed Compressors

*NA7.10.4*

The controls and equipment for the variable speed control of screw compressors is checked and certified as part of the acceptance requirements. The compressor should unload capacity by reducing speed to the minimum speed setpoint before unloading by slide valve or other means. Control system trend screens can also be used to verify that the speed varies automatically in response to the load.

### 10.6.4 Additions and Alterations

*§140.9*

#### 10.6.4.1 Requirements

Requirements related to refrigerated warehouse additions and alterations are covered by the Energy Standards in §141.1(a). The specific requirements for additions and alterations for commercial refrigeration are included in §120.6(a). Definitions relevant to refrigerated warehouses include:

- An addition is a change to an existing refrigerated warehouse that increases refrigerated floor area and volume. Additions are treated like new construction.
- When an unconditioned or conditioned building; or unconditioned or conditioned part of a building adds refrigeration equipment so that it becomes refrigerated, this area is treated as an addition.
- An alteration is a change to an existing building that is not an addition or repair. An alteration could include installing new evaporators, a new lighting system, or a change to the building envelope, such as adding insulation.
- A repair is the reconstruction or renewal of any part of an existing building or equipment for the purpose of its maintenance. For example, a repair could include the replacement of an existing evaporator or condenser.

Any addition or altered space must meet all applicable mandatory requirements. Repairs must not increase the preexisting energy consumption of the repaired component, system, or equipment; otherwise, it is considered to be an alteration.

Example 10-49

**Question**

The new construction is an addition to an existing refrigerated warehouse. The new space is served by an existing refrigeration plant. Does the refrigeration plant need to be updated to meet the Energy Standards?

**Answer**

No. The new construction must comply with the Energy Standards; however, the existing refrigeration plant equipment is exempt from the Energy Standards.

Example 10-50

**Question**

The new construction includes an addition to refrigerated space and expansion of the existing refrigeration plant. Is the existing refrigeration equipment subject to the Energy Standards requirements?

**Answer**

No. Only the new equipment installed in the added refrigerated space and any new compressors added to the existing plant are subject to the requirements of the Energy Standards. If a new refrigeration system was installed with a new condenser for the addition then the new condenser must also comply with the Energy Standards.

Example 10-51

**Question**

An upgrade to an existing refrigerated storage space includes replacing all of the existing evaporators with new evaporators. Do the new evaporators need to comply with the Energy Standards?

**Answer**

Yes. A complete renovation of the evaporators in the space is considered to be an alteration. The alteration requirements apply when all of the evaporators in the space are changed.

Example 10-52

**Question**

An existing refrigerated storage space is adding additional evaporators to meet an increase in the refrigeration load. Do the new evaporators need to comply with the Energy Standards?

**Answer**

No. The alteration requirements apply only when all of the evaporators in the space are changed.

Example 10-53

**Question**

An existing evaporator is being replaced by a new evaporator as part of system maintenance. Does the new evaporator need to comply with the Energy Standards?

**Answer**

No. Replacement of an evaporator during system maintenance is considered a repair. However, the energy consumption of the new evaporator must not exceed that of the equipment it replaced.

## 10.7 Laboratory Exhaust

### 10.7.1 Overview

In the climates of California, laboratories have average annual energy intensities 10 to 20 times larger than offices when normalized by building area. The energy use of laboratories is driven from long hours of operation and the large quantities of outside air. Many lab buildings also have large internal loads.

Research in the climates of California showed annual cost of lab air at ~$3 to $5 per cfm/yr or $5 to $10/ft2/yr. At these costs the paybacks on retrofitting constant volume labs to

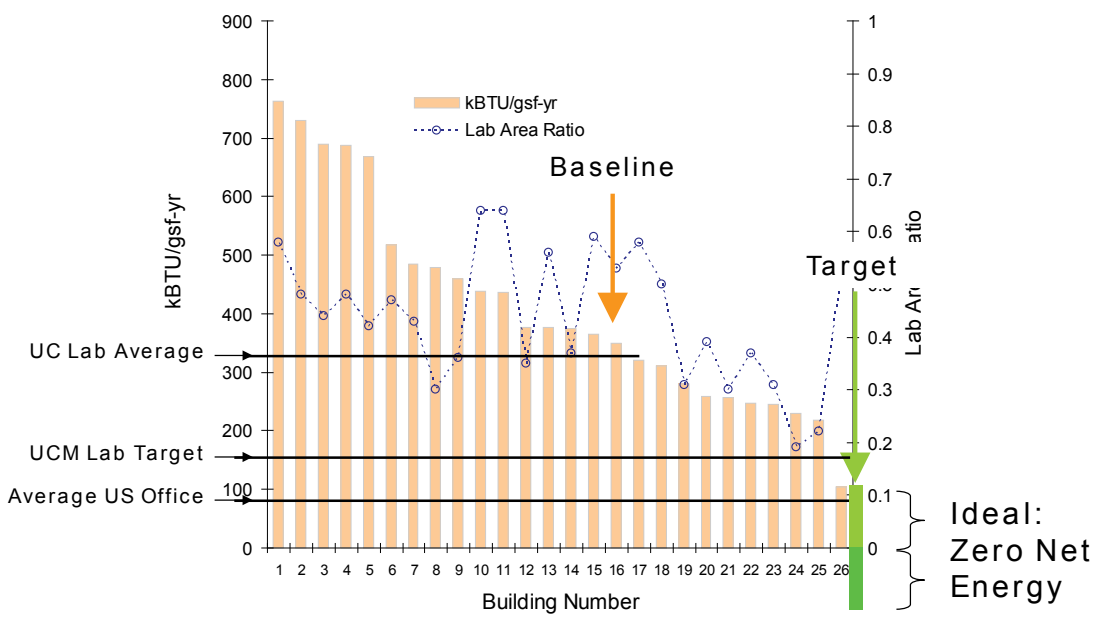
variable air volume have been less than 10 years. With new construction the paybacks are much shorter.

The energy and demand savings are strongly dependent on the facility's characteristics, including the following:

- The ratio of lab to non-lab space.
- The minimum airflow required by code or the facility EH&S department. These range from 4 air-changes per hour (ACH) to 18 ACH or higher.
- The climate.

Figure 10-38 below shows benchmarking data from Labs 21 for lab buildings in the San Francisco Bay area. The total energy use intensity in kBtu/gsf/yr is shown on the left access. The 26 labs are arranged from highest to lowest normalized energy use. The right access is the "Lab Area Ratio" the ratio of lab area to total building area. There are three reference lines on this graph: the University of California campus wide average laboratory building end use intensity; the University of California Merced campus goal for their laboratories; and the average national energy end use for office buildings.

**Figure 10-38: Laboratory Benchmarking from Labs 21 for San Francisco Bay Area**




Using the criteria for cost effectiveness in the Standard and very conservative estimates of the first costs (using costs from VAV retrofits not new construction) this measure was shown to be cost effective in all California Climate Zones up to 14 ACH of minimum ventilation

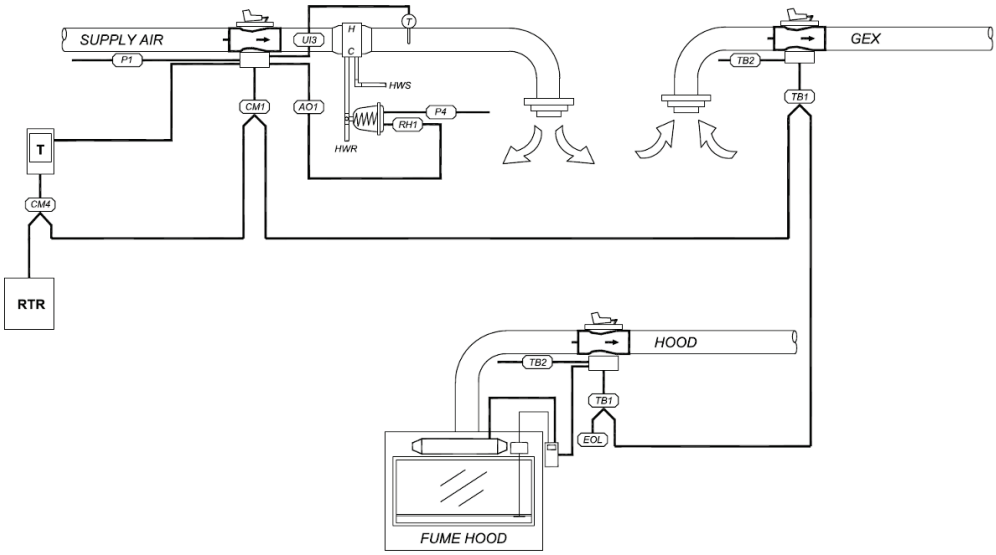
Using off the shelf variable air volume controls can greatly reduce the energy use in laboratory buildings. The Energy Standards requires VAV controls on all zones not required to be constant volume by the AHJ, facility EH&S department or other applicable health and safety code. Furthermore in ANSI/AIHA Z9.5 and NFPA 45 would allow lower minimum airflows in many hoods which would further increase the savings from variable air volume design.

Figure 10-39 below show the zone components for a variable air volume (VAV) laboratory. There are three zone valves shown in this image: one each on the supply air to the zone;

the fume hood (if one exists), and; the general exhaust valve (GEX) if one is needed. These zone valves can be venturi type valves as shown in this image or standard dampers like those used for VAV boxes in offices. The dampers or venturi valves must be designed to resist corrosion and damage from the exhaust. The hood valve when it is used is controlled to automatically maintain the design sash face velocity as the hood sash is opened or closed. The role of the supply valve is to maintain space pressurization by tracking the sum of the hood and general exhausts in the space. The supply valves are typically provided with a reheat coils to maintain space comfort for heating. The GEX is typically used to control the cooling, on call for cooling it opens, and the supply valve in turn opens to maintain space pressure. In some systems the supply modulates like a typical VAV box in response to the thermostat and the GEX modulates to maintain space pressure.

All three valves are made to control as either variable volume or constant volume depending on the application. A hood might for instance be required to be constant volume for dilution. If this is the case, a constant volume bypass hood should be employed. Even with a constant volume hood you will need a pressure independent hood valve if the exhaust it is attached to includes any variable volume zones. The same is true for constant volume supply or general exhaust: if any zone on a supply or exhaust duct is variable volume, all zone ducts on it must have pressure independent controls.

**Figure 10-39: Zone Components for A VAV Lab**




The fume exhaust is generally blown out of a stack. The design of the stack and the velocity of the discharge is selected to disperse all contaminants so that they are sufficiently dilute by the time they are near any occupants. For contaminants like radio isotopes for which there is no acceptable level of dilution, the exhaust system typically has some form of filtration that captures the particles of concern. On general lab exhaust there is typically an inlet bypass damper on the exhaust fan that modulates to keep a constant volume of exhaust moving at the stacks. Using multiple stacks in parallel you can stage off stacks and fans to save more energy.

### 10.7.2 Mandatory Measures

There are no mandatory measures specific to laboratory exhaust. The equipment efficiencies in §110.1 and §110.2 apply.

### 10.7.3 Prescriptive Measures

§140.9(c) requires that all laboratory exhaust with minimum circulation rates of 10 ACH or lower shall be designed for variable volume control on the supply, fume exhaust and general exhaust.

An exception is provided for laboratory exhaust systems where constant volume is required by code, the Authority Having Jurisdiction (AHJ), or the facility Environmental Health and Safety (EH&S) division (Exception 1 to §140.9(c)). Examples include: hoods using perchloric acid; hoods with radio isotopes; and exhaust systems conveying dust or vapors that need a minimum velocity for containment.

A second exception is provided for new zones added to an existing constant volume exhaust system (Exception 2 to §140.9(c)).

### 10.7.4 Additions and Alterations

As noted in the previous section variable volume controls are not required if you are adding zones to an existing constant volume system.

## 10.8 Compressed Air Systems

*§120.6(e)*

### 10.8.1 Overview

§120.6(e) applies to all new compressed air systems and all additions or alterations to a compressed air system with a total installed compressor capacity of ≥ 25 hp. For alternations there is an exception for systems that include one or more centrifugal compressors.

Before discussing the requirements these key terms must be understood:

A. **Online compressors:** all compressors that are available to serve peak load. Online compressors do not include back-up compressors whose only purpose is to be available when a compressor fails. Online compressors are all compressors that are physically connected to compressed air piping excluding back-up compressors.

B. **Back-up compressors:** are those compressors not used to meet peak compressed air flow loads. Back –up compressors can be physically connected to the compressed air piping system and can be automatically controlled to turn on if one of the other compressors on the system fails. Back-up compressors do not normally operate.

C. **Online capacity:** total combined capacity in actual cubic feet per minute (acfm) of compressed air at a given pressure during times of peak compressed air load.

D. **Trim compressor:** is a compressor that is designated for part-load operation, handling the short term variable trim load of end uses, in addition to the fully loaded base compressor. In general the trim compressor will be controlled by a VSD but it also can be a compressor with good part load efficiency. If the trim compressor does not have good part load efficiency broadly across its operating range, then it will take more compressors to meet the Energy Standards requirements.

E. **Base compressor:** the opposite of a trim compressor, a base compressor is expected to be mostly loaded. If the compressed air system has only one compressor, the requirements of the Energy Standards require that the single compressor be treated as a trim compressor.

F. **Specific power:** the ratio of power to compressed air flow rate at a given pressure typically given the units of kW/100 acfm. The lower the specific power the more efficient the compressor is at a given compressed air loads.

G. **Total effective trim capacity:** the combined effective trim capacity of all trim compressors where effective trim capacity for each compressor is the range of capacities in acfm which are within 15 percent of the specific power at its most efficient operating point. This is displayed in Figure 10-42.

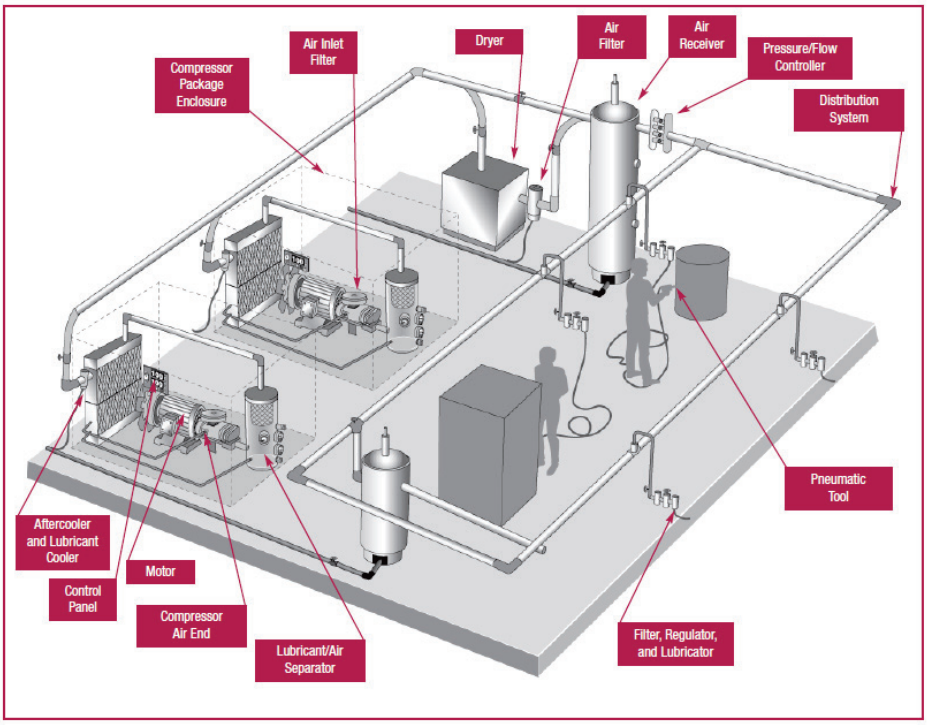
H. **Largest net capacity increment:** is the largest increase in capacity when switching between combinations of base compressors that is expected to occur under the compressed air system control scheme. See Example 10-54.

I. **Primary Storage:** are tanks or other devices that store compressed air capacity. Also known as an air receiver, they reduce peak air demand on the compressor system and reduce the rate of pressure change in a system. As primary storage these devices are near the air compressors and are differentiated from remote storage that might be out near an end use device.

As described in the following paragraphs, there are 3 main requirements in this section:

- Trim Compressor and Storage - §120.6(e)1.
- Controls - §120.6(e)2.
- Acceptance - §120.6(e)3.

**Figure 10-40: Zone Components for A VAV Lab**




*Source: Improving Compressed Air System Performance: A Sourcebook for Industry, USDOE 2003*

## 10.8.2 Mandatory Measures

*§120.6(e)*

### 10.8.2.1 Trim Compressor and Storage

*§120.6(e)1*

This requirement targets the performance of a compressed air system across its full range.

There are two alternate paths to comply with this requirement:

1. Using a variable speed drive (VSD) controlled compressor(s) as the Trim Compressor (§120.6(e)1A):
   - The VSD trim compressor(s) must have a capacity (acfm) of at least 1.25 times the largest net capacity increment (see Example 10-54).
   - Primary storage of at least one gallon per acfm (1 gal/acfm) of the largest trim compressor.
2. Using a compressor or set of compressors as the Trim Compressor (§120.6(e)1B) without requiring a VSD controlled compressor:
   - The trim compressor(s) must have a total effective trim capacity no less than the largest new capacity increment.
   - Primary storage of at least two gallons per acfm (2 gal/acfm) of the largest trim compressor.
   - Effective trim capacity is the rang of compressed air flow rates where the specific power (W/acfm) is no greater than 115% of the minimum specific power (see Figure 10-42).

Both of these paths aim to reduce the amount of cycling of fixed speed compressors by utilizing a better-suited compressor that operates well in part-load.

#### A. Compliance Option 1: VSD-controlled Trim Compressor

*§120.6(e)1A*

Many base load compressors are designed to provide peak efficiency near their rated capacity with a significant drop off in efficiency at lower flow rates (in acfm). Compressed air systems often avoid the losses in efficiency associated with part load compressed air flows by staging multiple compressors so that in most cases base compressors operate near full load. A trim compressor is designed to have close to peak efficiency over a broad range of compressed air flow rates. To make sure the compressed air system is operating efficiently over its entire range, it is important to have a trim compressor sized to handle the gaps between base compressors. The minimum size of the trim compressor(s) is determined calculating the *Largest Net Capacity Increment* - the biggest step increase between combinations of base compressors.

With equally sized compressors this is fairly intuitive: in a system with two-100 hp (434 acfm) rotary screw compressor system, the largest step increase would be the size of one of the compressors (434 acfm). For systems with uneven compressor sizes, it requires going through the following steps:

1. Determine all combinations of base compressors (including all compressors off).
2. Order these combinations in increasing capacity.
3. Calculate the difference between every adjacent combination.
4. Choose the largest difference.

This largest difference is what must be covered by the trim compressor(s) in order to avoid a control gap.

Once the *Largest Net Capacity Increment* is calculated, this value can be used to satisfy the first compliance option. Option 1 mandates that the rated capacity of the VSD compressor(s) be at least 1.25 times the largest net increment.

For compliance option 1, the system must include primary storage that has a minimum capacity of 1 gallon for every acfm of capacity of the largest trim compressor.

Example 10-54

**Question**

Given a system with three base compressors with capacities of 200 acfm (Compressor A), 400 acfm (Compressor B) and 1,000 acfm (Compressor C), what is the *Largest Net Capacity Increment?*

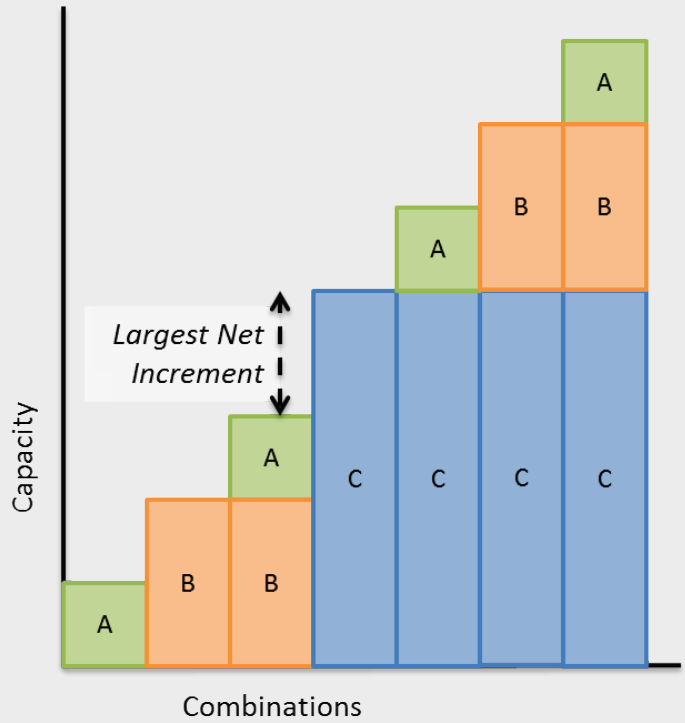
**Answer**

As shown in the image below there are 8 possible stages of capacity ranging from 0 acfm with no compressors to 1,600 acfm with all three compressors operating. The largest net increment is between stage 4 with compressors A and B operating (200+400=600 acfm) to stage 5 with compressor C operating (1,000 acfm)

**Combinations of Base Compressors**

| Base Compressors | |
|---|---|
| A | 200 |
| B | 400 |
| C | 1000 |

| Capacity | Combination |
|---|---|
| 0 | None |
| 200 | A |
| 400 | B |
| 600 | A + B |
| 1000 | C |
| 1200 | A + C |
| 1400 | B + C |
| 1600 | A + B + C |




For this system the *Largest Net Capacity Increment* is 1,000 acfm-600 acfm = 400 acfm

Example 10-55

**Question**

Using the system from the previous example, what is the minimum rated capacity of VSD compressor(s) that are needed to comply with Option 1?

**Answer**

As previously shown, the *Largest Net Capacity Increment* is 1,000 acfm-600 acfm = 400 acfm. The minimum rated capacity for VSD compressor(s) is 400 acfm X 1.25 = 500 acfm.

Example 10-56

**Question**

What is the required minimum primary storage capacity for the trim compressor from the previous example to comply with Option 1?

**Answer**

Assuming there is a VSD compressor with a rated capacity of 500 acfm, per §120.6(e)1A it must have 1 gallon of storage per acfm of rated capacity or 500*1 = 500 gallons of storage.

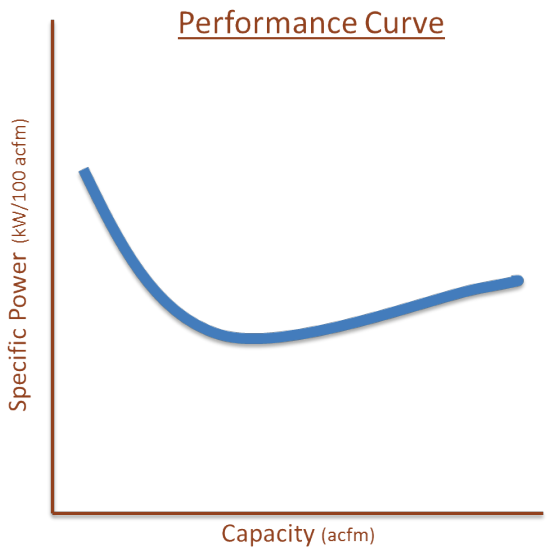
### B. Compliance Option 2: Other Compressors as Trim Compressor

*§120.6(e)1B*

The second compliance option offers more flexibility but requires looking at both the *Largest Net Capacity Increment* of the system, as well as the *Effective Trim Capacity* of the trim compressor(s).

The *Effective Trim Capacity* is the range across which a trim compressor has adequate part-load performance. Performance is measured in power input over air volume output or specific power (kw/100 acfm). Many VSD compressors come with a compressor performance graph in a CAGI data sheet that looks similar to the graph in Figure 10-41.

**Figure 10-41: Example Compressor Power vs. Capacity Curve**



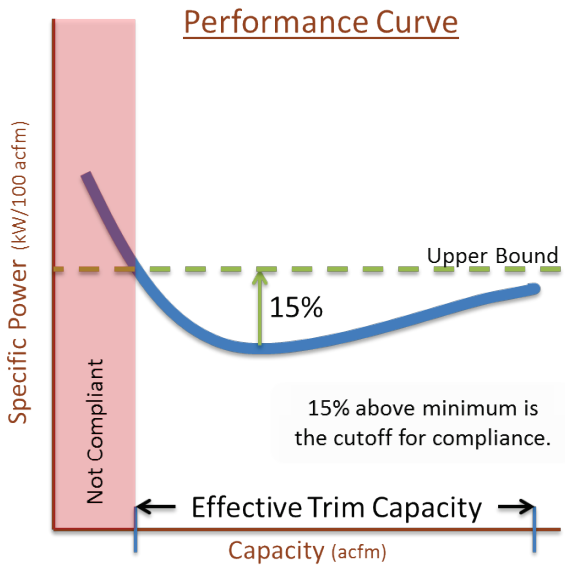

The capacity of the compressor is along the x-axis, while the power is on the y-axis. The curve in Figure 10-41 is a typical shape of a performance curve for a VSD compressor. The lower the specific power, the more energy efficient the compressor is at that condition.

The *Effective Trim Capacity* uses the minimum of the compressor power vs. capacity curve to determine the range of adequate part-load performance. This can be done in the following steps and is illustrated in the graph below.

1. Find the minimum specific power across the range.
2. Find the upper bound by calculating 1.15 times the minimum specific power.
3. Determine the endpoints of the capacity (acfm) where the specific power is less than or equal to the upper bound.

4. The capacity difference in units of acfm between these two endpoints is the effective trim capacity.

**Figure 10-42: Determination of Effective Trim Capacity from a Compressor Curve**




This definition of *Effective Trim Capacity*, along with the *Largest Net Capacity Increment* of the system, will be used to assist in sizing the trim compressor appropriately in the next section.

For compliance option 2, the system must include primary storage that has a minimum capacity of 2 gallons for every acfm of capacity of the largest trim compressor.

Example 10-57

**Question**

Continuing with the system from the previous examples, what is the required minimum *Effective Trim Capacity* of the trim compressor(s) to comply with Option 2*?*

**Answer**

As previously shown, the *Largest Net Capacity Increment* is 1,000 acfm-600 acfm = 400 acfm. Per §120.6(e)1 the minimum *Effective Trim Capacity* is equal to the *Largest Net Capacity Increment* or 400 acfm.

Example 10-58

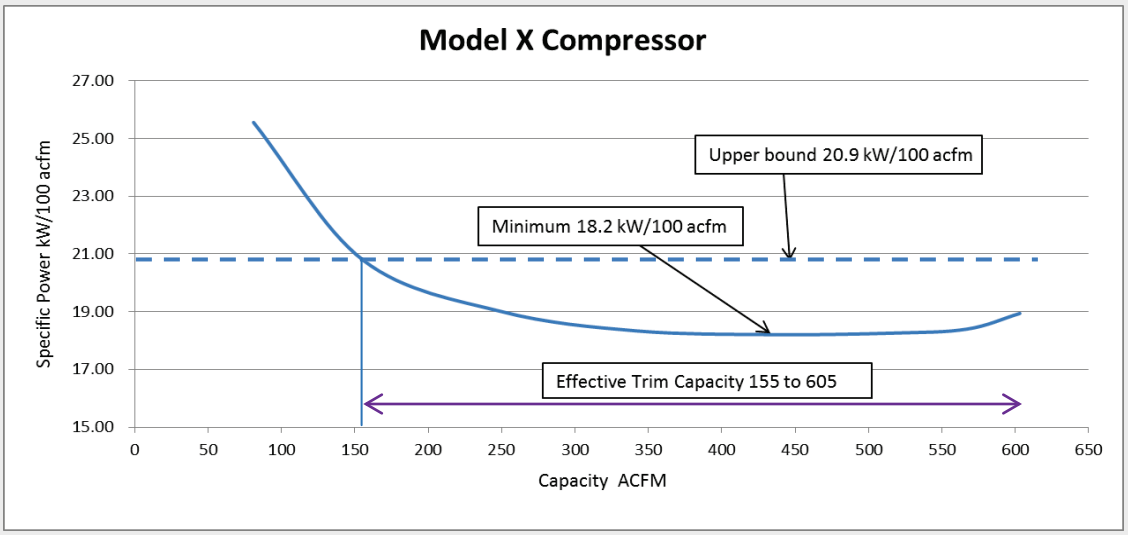
**Question**

A manufacturer provided the following data for their compressor; would this provide the minimum *Effective Trim Capacity* for this system to comply with Option 2?

| Input Power (kW) | Capacity (acfm)[a,d] | Specific Power (kW/100 acfm)[d] |
|---|---|---|
| 20.7 | 81.0 | 25.56 |
| 32.4 | 156.0 | 20.77 |
| 47.5 | 250.0 | 19.00 |
| 62.7 | 342.0 | 18.33 |
| 79.0 | 434.0 | 18.20 |
| 94.2 | 516.0 | 18.26 |
| 104.3 | 567.0 | 18.40 |
| 114.2 | 603.0 | 18.94 |

**Answer**

From the manufacturer's data the minimum specific power is 18.2 kW/100 acfm. The upper limit would be 18.2 * 1.15 = 20.9 kW/100 acfm. Interpolating from the manufacturer's data this appears to go from 155 acfm to 605 acfm for an Effective Trim Capacity of 605-155= 450 acfm. This is larger than the *Largest Net Capacity Increment* of 400 acfm so this compressor would comply as a trim compressor for this system.




Example 10-59

**Question**

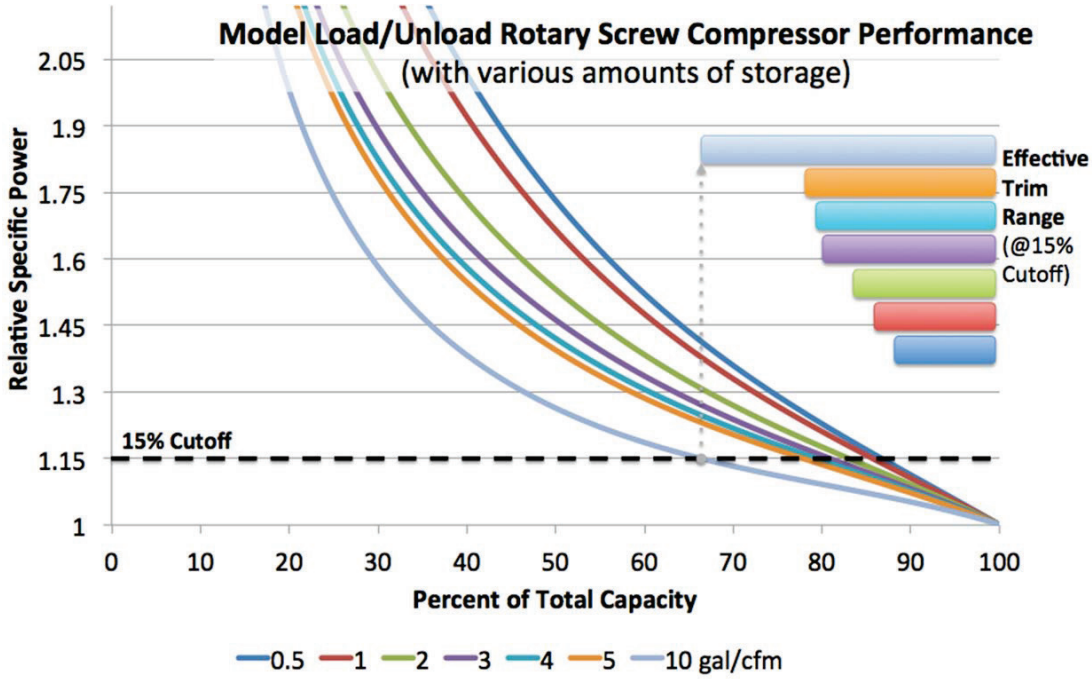
What is the required minimum primary storage capacity for the trim compressor from the previous example to comply with Option 2?

**Answer**

This compressor has a rated capacity of 603 acfm, per §120.6(e)1B it must have 2 gallons of storage per acfm of rated capacity or 603*2 = 1,206 gallons of storage.

The last example used a VSD compressor, but other technologies can be used for compliance option 2 such as a compressor with unloaders and sufficient compressed air storage to achieve relatively high part load efficiencies over a broad range of compressed air flow rates. Generally, higher levels of storage improve part-load performance. The following data in Figure 10-43 and for this example was generated from theoretical curves used in AirMaster+, a tool created by the U.S. Department of Energy.

**Figure 10-43: Normalized efficiency curves for a screw compressor with load/unload controls for various amounts of storage**



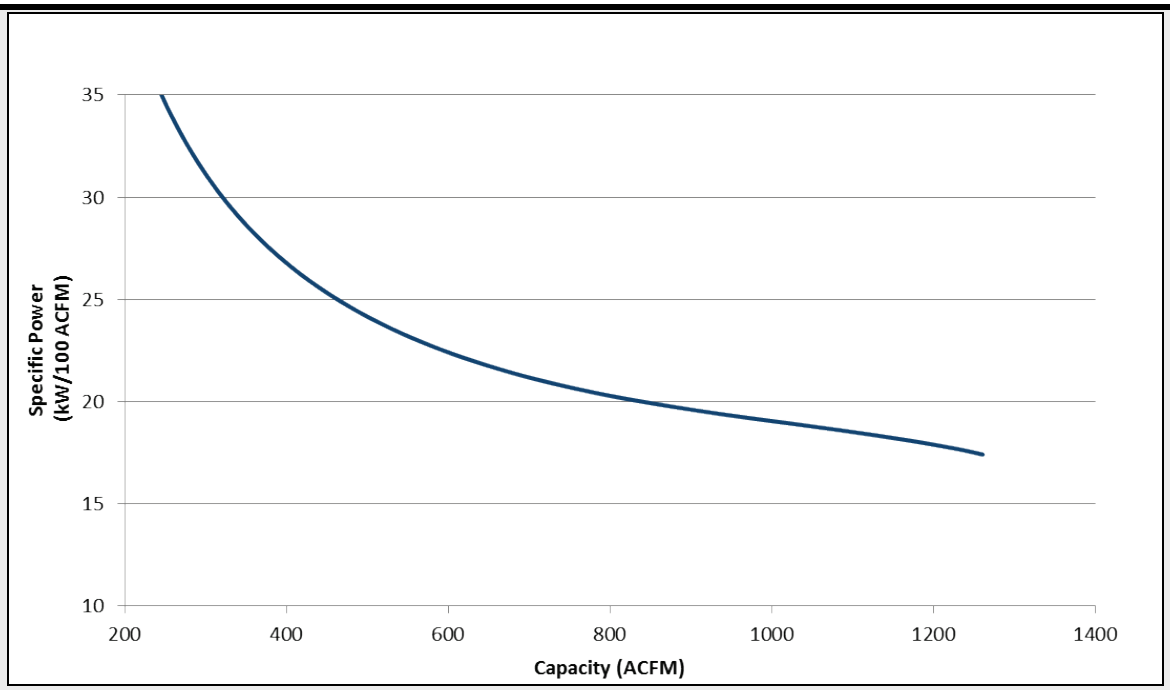

*Source: Derived from Fact Sheet 6 – Compressed Air Storage, Improving Compressed Air Storage: a Sourcebook for Industry, U.S. Department of Energy, 2003*

The next example examines a 250-hp load-unload, single stage, rotary screw compressor coupled with 10 gallons/cfm of storage. This combination of compressor and storage was chosen to meet the part-load performance mandated by code.

Example 10-60

**Question**

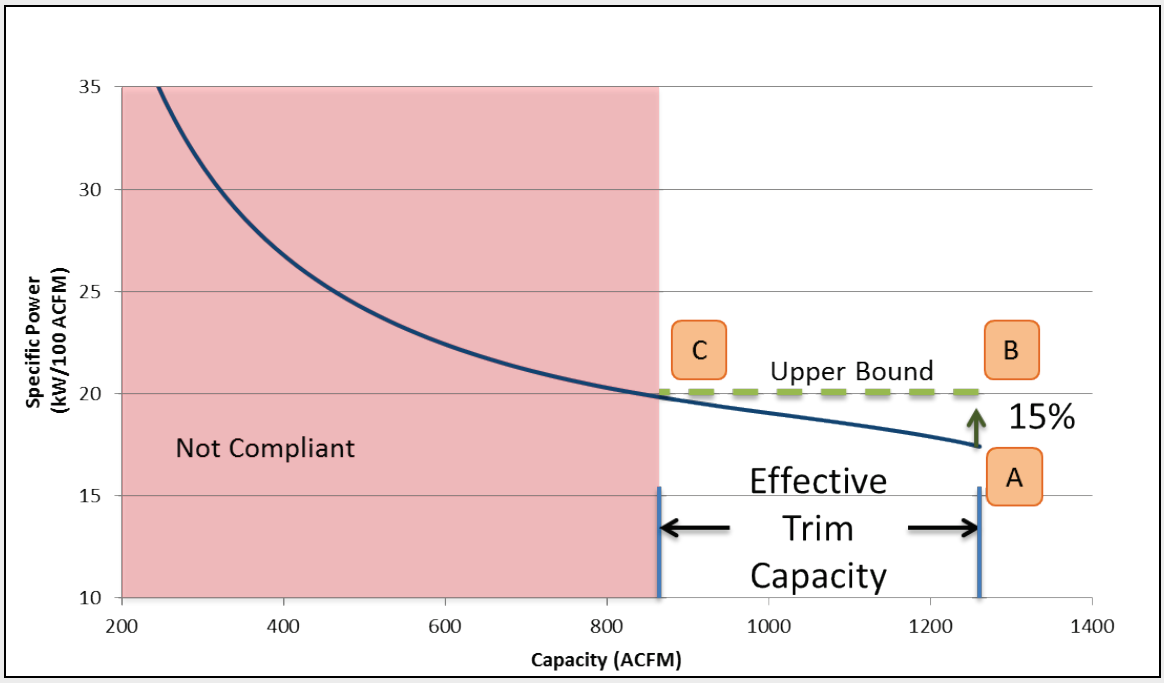
Part-load data was approximated below for a 250-hp load-unload, single stage, rotary screw compressor (with a capacity of 1,261 acfm) coupled with 10 gallons/cfm of storage; would this provide the minimum *Effective Trim Capacity* for this system to comply with Option 2?




**Answer**

Using the previous examples, a compressor with an effective trim capacity of at least 400 acfm is necessary.

Looking at the graph, the minimum specific power (labeled as A below) occurs at full load - a capacity of 1261 acfm, with a specific power of 17.4 kW/100 acfm. Using this minimum specific power, the upper bound is 17.4 * 1.15 = 20.01 kW/100acfm or 15% higher than the minimum specific power. This puts the ends of the effective trim capacity at 1261 acfm (labeled as B) and 845 acfm (labeled as C), resulting in an effective trim capacity of 1261 – 845 = 416 acfm. This is larger than the Largest Net Capacity Increment of 400 acfm so this compressor would comply as a trim compressor for this system.



Example 10-61

**Question**

What is the required minimum primary storage capacity for the trim compressor from the previous example to comply with Option 2?

**Answer**

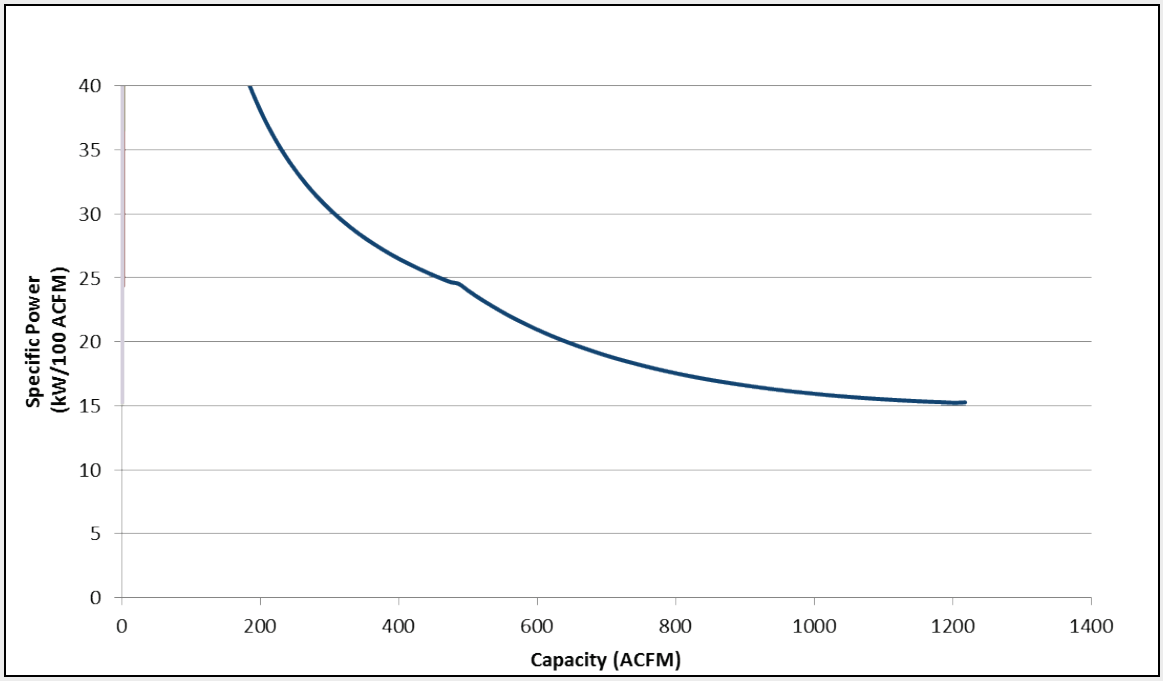
This compressor has a rated capacity of 1261 acfm, and per §120.6(e)1B it must have 2 gallons of storage per acfm of rated capacity or 1261 * 2 = 2,522 gallons of storage.

However, a minimum of 10 gallons of storage per acfm was needed for the screw compressor with a load/unload controls to have a large enough Effective Trim Capacity. The minimum required primary storage to meet both the Effective Trim Capacity and storage requirements in §120.6(e)1B, is 10 gal per acfm of rated trim compressor capacity, thus the minimum primary storage capacity required is 1261 * 10 = 12,610 gallons.

Example 10-62

**Question**

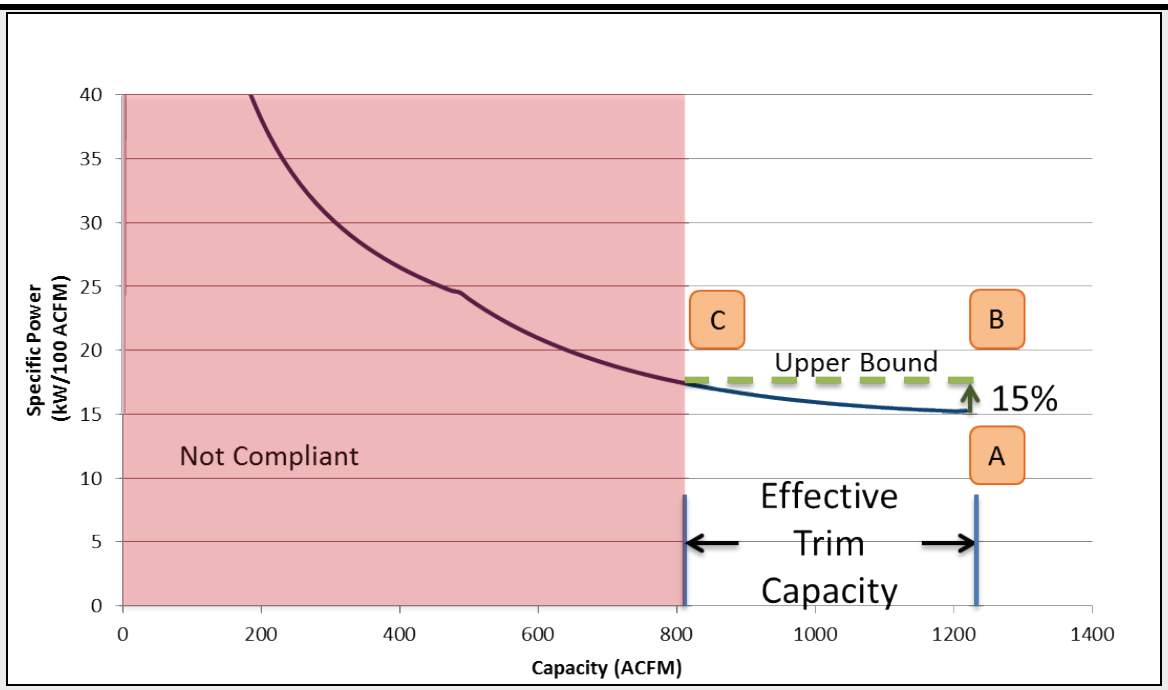
Part-load data was approximated below for a 250-hp variable capacity compressor; would this provide the minimum *Effective Trim Capacity* for this system to comply with Option 2?




**Answer**

Using the previous examples, a compressor with an effective trim capacity of at least 400 acfm is necessary.

Looking at the graph, the minimum specific power (labeled as A below) occurs at full load - a capacity of 1218 acfm, with a specific power of 15.3 kW/100 acfm. Using this minimum specific power, the upper bound is 15.3 * 1.15 = 17.56 kW/100 acfm or 15% higher than the minimum specific power. This puts the ends of the effective trim capacity at 1218 acfm (labeled as B) and 804 acfm (labeled as C), resulting in an effective trim capacity of 1218 – 804 = 414 acfm. This is larger than the Largest Net Capacity Increment of 400 acfm so this compressor would comply as a trim compressor for this system.




Example 10-63

**Question**

What is the required minimum primary storage capacity for the trim compressor from the previous example to comply with Option 2?

**Answer**

This compressor has a rated capacity of 1218 acfm, and per §120.6(e)1B it must have 2 gallons of storage per acfm of rated capacity or 1218 * 2 = 2,236 gallons of storage.

### 10.8.2.2 Controls

*§120.6(e)2*

This requirement applies to new facilities that are being altered with ≥100 hp of installed compressor capacity. The section requires an automated control system which will optimally stage the compressors to minimize energy for the given load. With new systems, this ideally means that at any given load, the only compressors running at part-load are the trim compressors. Because not all systems are required to upgrade the trim compressor, the installed controls must stage the compressors in the most efficient manner.

This requirement also mandates the measurement of air demand. The control system must be able to measure or calculate the current system demand (in terms of actual cubic feet per minute of airflow). There are two ways to accomplish this, including but not limited to the following sensors:

- A flow meter.
- A combination of pressure transducers and power meters.

### 10.8.2.3 Acceptance

*§120.6(e)3*

New systems and altered systems which are subject to the trim compressor requirements of §120.6(e)1 or the staging control requirements of §120.6(e)2 must be tested per NA7.13.

### 10.8.3 Prescriptive Measures

*§140.9*

There are no prescriptive measures for compressed air systems.

### 10.8.4 Additions and Alterations

These requirements apply to existing systems which are being altered and which have a total compressor capacity of ≥ 25 hp. These requirements will be triggered by replacing a compressor, adding a compressor, or removing a compressor.

These requirements will not apply to:

- Adding a VFD to a fixed speed compressor
- Repairing a compressor
- Replacing a compressor drive motor
- Adding compressed air controls
- Adding air dryers
- Adding oil separators
- Adding compressed air storage capacity
- Removing an air compressor without adding any air compressors

For alterations or additions to an existing compressed air system, the requirements for trim compressor size and storage only apply when the combined capacity of compressor replacements and/or additions are 50 percent or more of the existing online capacity (not including back-up compressor capacity) of the compressed air system. Since the capacity of a compressor is closely correlated to its horsepower, for simplicity of compliance and enforcement, the combined horsepower of the alterations and/or additions can be used to determine the total combined capacity.

Example 10-64

**Question**

If a 50 hp compressor was added to a compressed air system with only one existing 100 hp compressor, would the requirements of §120.6(e) apply?

**Answer**

Yes, because 50 hp is equal to or greater than 50 percent of the existing 100 hp capacity.

Example 10-65

In some cases after a compressed air assessment, it is recognized that the online compressor capacity can be downsized. As an example, a plant has a system with 100 hp online compressor and a 100 hp back-up compressor. After the assessment, it is identified that the maximum hp needed to meet the air demand is only 50 hp. The project scope calls for the installation of one 50 hp compressor and to leave both existing 100 hp compressors as back-up compressors.

**Question**

What requirements of §120.6(e) apply?

**Answer**

After the retrofit, the online compressor horsepower is 50 hp. Since the total online compressor hp is greater than 25 hp, the requirements of §120.6(e) 1, 2 and 3 must be considered. The new 50 hp compressor accounts for 50 percent of the preexisting online capacity. Thus the trim compressor requirements of §120.6(e)1 would apply.

Since this system only has one on-line compressor, the requirements for a load controller that optimizes compressor selection as described in §120.6(e)2 is not required.

Since this system was required to install a trim compressor, the acceptance test requirement of §120.6(e)3 would apply.

The requirements for a staging control only apply if after the alterations/additions, there is more than one online compressor and their total combined horsepower is greater than 100 hp.

The acceptance testing mentioned in §120.6(e)3 test for system blow-off and short cycling. These issues are impacted by sizing and specification of the trim compressor or the selection and programming of an optimal staging control. Thus the acceptance tests are required only for alterations/additions where a trim compressor and/or staging controls are required.

## 10.9 Process Boilers

### 10.9.1 Overview

A process boiler is a type of boiler with a capacity (rated maximum input) of 300,000 Btus per hour (Btu/h) or more that serves a process. A process is an activity or treatment that is not related to the space conditioning, service water heating, or ventilating of a building as it relates to human occupancy.

### 10.9.2 Mandatory Measures

*§120.6(d)*

#### 10.9.2.1 Combustion Air

*§120.6(d)1*

Combustion air positive shut-off shall be provided on all newly installed process boilers as follows:

- All process boilers with an input capacity of 2.5 MMBtu/h (2,500,000 Btu/h) and above, in which the boiler is designed to operate with a non-positive vent static pressure. This is sometimes referred to as natural draft or atmospheric boilers. Forced draft boilers, which rely on a fan to provide the appropriate amount of air into the combustion chamber, are exempt from this requirement.
- All process boilers where one stack serves two or more boilers with a total combined input capacity per stack of 2.5 MMBtu/h (2,500,000 Btu/h). This requirement applies to natural draft and forced draft boilers.

Combustion air positive shut-off is a means of restricting air flow through a boiler combustion chamber during standby periods, and is used to reduce standby heat loss. A

flue damper and a vent damper are two examples of combustion air positive shut-off devices.

Installed dampers can be interlocked with the gas valve so that the damper closes and inhibits air flow through the heat transfer surfaces when the burner has cycled off, thus reducing standby losses. Natural draft boilers receive the most benefit from draft dampers because they have less resistance to airflow than forced draft boilers. Forced draft boilers rely on the driving force of the fan to push the combustion gases through an air path that has relatively higher resistance to flow than in a natural draft boiler. Positive shut-off on a forced draft boiler is most important on systems with a tall stack height or multiple boiler systems sharing a common stack.

#### 10.9.2.2 Combustion Air Fans

*§120.6(d)2*

Combustion air fans with motors 10 horsepower or larger shall meet one of the following for newly installed boilers:

- The fan motor shall be driven by a variable speed drive, or
- The fan motor shall include controls that limit the fan motor demand to no more than 30 percent of the total design wattage at 50 percent of design air volume.

Electricity savings result from run time at part-load conditions. As the boiler firing rate decreases, the combustion air fan speed can be decreased.

#### 10.9.2.3 Excess Oxygen ≥ 5 MMBtu/h to ≤ 10 MMBtu/h

*§120.6(d)3 and 4*

Newly installed process boilers with an input capacity of 5 MMBtu/h (5,000,000 Btu/h) to 10 MMBtu/h (10,000,000 Btu/h) shall maintain excess (stack-gas) oxygen concentrations at less than or equal to 5.0 percent by volume on a dry basis over firing rates of 20 percent to 100 percent. Combustion air volume shall be controlled with respect to firing rate or measured flue gas oxygen concentration. Use of a common gas and combustion air control linkage or jack shaft is prohibited.

One way to meet this requirement is with parallel position control. Boilers mix air with fuel (usually natural gas although sometimes diesel or oil) to supply oxygen during combustion. Stoichiometric combustion is the ideal air/fuel ratio where the mixing proportion is correct, the fuel is completely burned, and the oxygen is entirely consumed. Boilers operate most efficiently when the combustion air flow rate is slightly higher than the stoichiometric air-fuel ratio. However, common practice almost always relies on excess air to insure complete combustion, avoid unburned fuel and potential explosion, and prevent soot and smoke in the exhaust. Excess air has a penalty, which is increased stack heat loss and reduced combustion efficiency.

Parallel positioning controls optimize the combustion excess air to improve the combustion efficiency of the boiler. It includes individual servo motors allowing the fuel supply valve and the combustion air damper to operate independently of each other. This system relies on preset fuel mapping (i.e., a pre-programmed combustion curve) to establish proper air damper positions (as a function of the fuel valve position) throughout the full range of burner fire rate. Developing the combustion curve is a manual process, performed in the field with a flue-gas analyzer in the exhaust stack, determining the air damper positions as a function of the firing rate/fuel valve position. Depending on type of burner, a more consistent level of excess oxygen can be achieved with parallel position compared to single-point positioning

control, since the combustion curve is developed at multiple points (firing rates), typically 10 to 25 points. Parallel positioning controls allow excess air to remain relatively low throughout a burner's firing range. Maintaining low excess air levels at all firing rates provides significant fuel and cost savings while still maintaining a safe margin of excess air to insure complete combustion.

#### 10.9.2.4 **Excess Oxygen > 10 MMBtu**

*§120.6(d)4*

Newly installed process boilers with an input capacity greater than 10 MMBtu/h (10,000,000 Btu/h) shall maintain excess (stack-gas) oxygen concentrations at less than or equal to 3.0 percent by volume on a dry basis over firing rates of 20 percent to 100 percent. Combustion air volume shall be controlled with respect to measured flue gas oxygen concentration. Use of a common gas and combustion air control linkage or jack shaft is prohibited.

One way to meet this requirement is with oxygen trim control. This control strategy relies on parallel positioning hardware and software as the basis but takes it a step further to allow operation closer to stoichiometric conditions. Oxygen trim control converts parallel positioning to a closed-loop control configuration with the addition of an exhaust gas analyzer and PID controller. This strategy continuously measures the oxygen content in the flue gas and adjusts the combustion air flow, thus continually tuning the air-fuel mixture.

Detecting and monitoring excess air is easy because oxygen not consumed during combustion is present in the exhaust gases. Detecting and monitoring carbon monoxide assures the air/fuel ratio is not too rich as the excess air is trimmed. Based on the exhaust gas analysis, a controller maintains close to stoichiometric combustion by commanding a servo motor to adjust the combustion air damper and another servo motor to adjust the fuel supply valve.

### 10.9.3 Prescriptive Measures

There are no prescriptive measures for process boilers.

## 10.10 Elevators

### 10.10.1 Overview

§120.6(f) applies to all nonresidential new construction elevators, as well existing elevators undergoing major alterations involving mechanical equipment, lighting and/or controls. The goal behind this measure is to save energy by reducing light power density of the elevator cab lighting and requiring a minimum wattage per cfm for ventilation fans in cabs without air conditioning. Both the lighting and ventilation fans are to be controlled in such a way to shut off when the elevator has been unoccupied for an extended period of time.

## 10.10.2 Mandatory Measures

*§120.6(f)*

### 10.10.2.1 Elevator Lighting Power Density

*§120.6(f)1*

The lighting power density of an elevator cab shall not exceed 0.6 watts per square foot (W/sq ft). This is determined by taking the total wattage of the elevator lighting and dividing by the area of the elevator in square feet. Interior signal lighting and interior display lighting are not included in the total wattage of the elevator lighting.

Example 10-66

**Question**

An elevator with a length of 6 ft and a width of 8 ft has 9 LED lamps at 3 Watts each. Does this comply with §120.6(f).1?

**Answer**

Yes. 9 Lamps * 3 Watts/Lamp = 27 Watts. The square footage of the cab is 6ft*8ft = 48 $ft^2$. The lighting power density is equal to 27 Watts/48 $ft^2$ = 0.56 $W/ft^2$, which is less than 0.6 $W/ft^2$.

### 10.10.2.2 Elevator Ventilation CFM Fan Performance

*§120.6(f)2*

Ventilation fans for cabs without space conditioning shall not exceed 0.33 watts per cubic feet per minute of airflow (W/cfm) at maximum speed. Elevator cabs with space conditioning are excluded from this measure.

### 10.10.2.3 Elevator Lighting and Fan Shutoff Control

*§120.6(f)3*

When the elevator cab is stopped and unoccupied with doors closed for over 15 minutes, the cab interior lighting and ventilation fans shall automatically switch off until elevator cab operation resumes. This can be accomplished with an occupancy sensor, or more elaborate built in elevator controls.

## 10.10.3 Prescriptive Measures

There are no prescriptive measures for elevators.

## 10.10.4 Additions and Alterations

- An elevator installation is considered an addition when the location of the installation did not previously contain an elevator.
- An alteration is a change to an existing elevator system that is not an addition or repair. An alteration could include installing new controls or a new lighting system.
- A repair is the reconstruction or renewal of any part of an existing elevator system for the purpose of its maintenance. For example, the replacement of lights or cosmetic features.

Any addition or altered space must meet all applicable mandatory requirements. Repairs must not increase the preexisting energy consumption of the repaired component, system, or equipment; otherwise, it is considered to be an alteration.

# 10.11 Escalators and Moving Walkways

## 10.11.1 Overview

§120.6(g) applies to nonresidential new construction escalators and moving walkways in airports, hotels, and transportation function areas, as well as existing escalators and moving walkways undergoing major alterations involving mechanical equipment or controls in the same locations. The goal behind this measure is to save energy by reducing the full speed run time of the escalator by slowing it down when unoccupied.

## 10.11.2 Mandatory Measures

*§120.6(g)*

### 10.11.2.1 Escalator and Moving Walkway Speed Control

*§120.6(g)1*

Escalators and moving walkways located in airports, hotels, and transportation function areas shall automatically slow to the minimum permitted speed in accordance with ASME A17.1/CSA B44 when not conveying passengers.

The ASME A17.1/CSA B44 2013 requirements for intermittent speed control on escalators and moving walkways are summarized below. These requirements are necessary to ensure maximum passenger safety when speeding up or slowing down escalators and moving walkways. In order to be compliant with the Energy Standards, the escalator or moving walkway must also be compliant with ASME A17.1/CSA B44 2013. Additional safety requirements may exist in Title 8.

Variation of the escalator and moving walkway speed after start-up shall be permitted provided the escalator and moving walkway installation conforms to all of the following:

1. The acceleration and deceleration rates shall not exceed 0.3 m/s² (1.0 ft/s²).
2. The rated speed is not exceeded.
3. The minimum speed shall be not less than 0.05 m/s (10 ft/min).
4. The speed shall not automatically vary during inspection operation.
5. Passenger detection means shall be provided at both landings of the escalator such that:
    a. Detection of any approaching passenger shall cause the escalator or moving walkway to accelerate to, or maintain the, full speed conforming to (1) through (4) above.
    b. Detection of any approaching passenger shall occur sufficiently in advance of boarding to cause the escalator or moving walkway to attain full operating speed before a passenger walking at normal speed [1.35 m/s (270 fpm)] reaches the combplate.
    c. Passenger detection means shall remain active at the egress landing to detect any passenger approaching against the direction of escalator or moving walkway travel and shall cause the escalator or moving walkway to accelerate to full rated speed

and sound the alarm at the approaching landing before the passenger reaches the combplate.

6. Automatic deceleration shall not occur before a period of time has elapsed, since the last passenger detection that is greater than 3 times the amount of time necessary to transfer a passenger between landings.
7. Means shall be provided to detect failure of the passenger detection means and shall cause the escalator or moving walkway to operate at full rated speed only.

**Figure 10-44: Example of Pedestrian Detection Method using Motion Sensors**

*Source: wwwtelcosensors.com/solutions/industries/elevators*

Escalator speed control is required only in airports, hotels, and transportation function areas. A transportation function area is defined in §100.1 of the Energy Standards as the ticketing area, waiting area, baggage handling areas, concourse, in an airport terminal, bus or rail terminal or station, subway or transit station, or a marine terminal. The reason behind limiting the scope of this measure was to focus on escalators and moving walkways that experience pedestrian flow rates in waves, and are more likely to operate 24 hours a day. An escalator in a busy shopping mall that only operates 12 hours a day may experience a constant pedestrian flow rate throughout the day and would rarely slow down and therefore save little energy. For these continuously busy applications during the operating hours of the escalator, the speed control would not be cost effective.

### 10.11.3 Prescriptive Measures

There are no prescriptive measures for escalators or moving walkways.

### 10.11.4 Additions and Alterations

- An escalator or moving walkway installation is considered an addition when the location of the installation did not previously contain an escalator.
- An alteration is a change to an existing escalator or moving walkway system that is not an addition or repair. An alteration could include installing new controls or motor.
- A repair is the reconstruction or renewal of any part of an existing escalator or moving walkway system for the purpose of its maintenance. For example, a repair could include the replacement of a damaged step or handrail.

Any addition or altered space must meet all applicable mandatory requirements. Repairs must not increase the preexisting energy consumption of the repaired component, system, or equipment; otherwise, it is considered to be an alteration.