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10 Covered Processes

10.1 Introduction

This chapter of the Nonresidential Compliance Manual addresses covered processes for the *2022 Building Energy Efficiency Standards* (Energy Codes) (§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
- 10.12 — Controlled Environment Horticulture
- 10.13 — Steam Traps

10.1.2 Compliance Forms

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.

Under the prescriptive compliance approach, the project designer is responsible for completing the Process Compliance Forms & Worksheets. The project designer is required to complete all applicable sections of the NRCC-PRC-E. This form 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.

10.1.3 What Is New for the 2022 California Energy Code?

Significant changes for covered process in the 2022 update to the Energy Codes include both new processing loads being covered as well as additional requirements being applied to process loads that were covered by the energy code previously.

Newly covered process loads include:

- Controlled environment horticulture (mandatory measures).
    - Electric lighting for growing plants now must have high photosynthetic photon efficacy (PPE), which is spectrum-efficient for growing plants, and must have dimming and timeclock controls.
    - Dehumidifiers must meet federal dehumidifier standards or recover at least 75 percent of the heat used for reheating.
    - Conditioned greenhouses must have at least two glazing layers.
- Steam traps (mandatory measures).
    - Steam trap monitoring system which provides status updates of steam trap fault detection sensors.
    - Steam traps must have an integral strainer and blow-off valves, or a strainer and blow-off valve must be installed within 3 feet upstream of the steam trap.
- Compressed air systems (mandatory measures).
    - Base-compressed air-system requirements on total horsepower of compressors connected to compressed air piping.
    - Energy and air demand monitoring systems capable of measuring and logging pressure, compressor power and compressor airflow of the compressed air systems.
    - Leak testing requirements for compressed air piping.
    - Compressed air system pipe sizing requirements to minimize frictional losses in the distribution system.
- Computer room (prescriptive measures).
    - Require minimum efficiencies for alternating current uninterruptible power supplies.
- Transcritical carbon dioxide (CO_2) refrigeration systems in refrigerated warehouses (mandatory)
    - Minimum condensing temperature
    - Transcritical gas coolers — sizing requirements, minimum efficiency and air-cooled gas coolers prohibited in some climate zones

- Transcritical CO_2 refrigeration systems in commercial refrigeration (mandatory)
  - Minimum condensing temperature
  - Transcritical gas coolers — sizing requirements, minimum efficiency and air-cooled gas coolers prohibited in some climate zones
- Automatic door closers in refrigerated warehouses

Revisions to covered process loads previously regulated under the Energy Code:

- Computer room (mandatory measures).
  - Prohibit reheating and simultaneous heating and cooling.
  - Humidification shall be adiabatic.
  - Unitary air conditioners and chilled water fan systems shall be designed to vary the airflow rate as a function of actual load.
- Computer room (prescriptive measures).
  - Define supply air temperatures for air and water economizers.
  - For air economizers and water economizers, revise ambient air temperatures when full economizing occurs.
  - Require air containment for computer rooms with information technology equipment (ITE) of 10 kW or more.

10.2 Enclosed Parking Garages

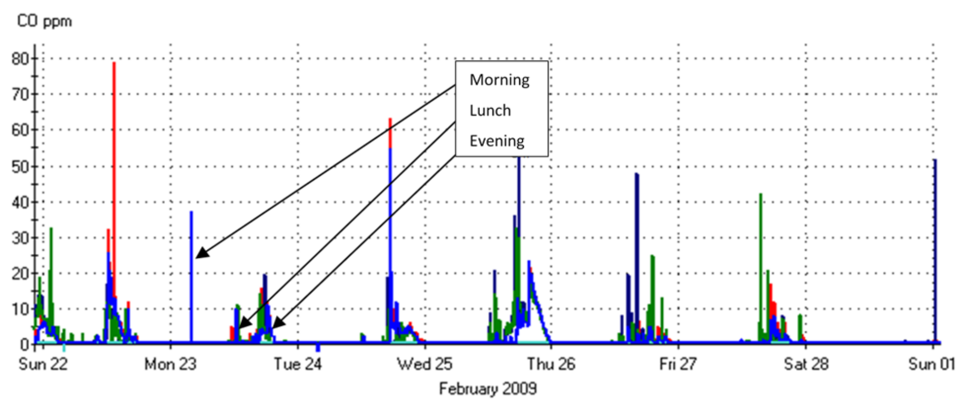
10.2.1 Overview

Garage 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
- 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



Source: California Energy Commission

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 requirement includes:

a. Minimum fan power reduction of the exhaust fan energy to ≤ 30% of design wattage at 50 percent of design flow. A two-speed or variable-speed motor can be used to meet this requirement.

b. CO concentration measured with at least one sensor per 5,000 ft^2 with each sensor located where the highest concentrations of CO are expected.

c. CO concentration of 25 ppm or less as control set point at all times.

d. A minimum ventilation of 0.15 cfm/ft^2 when the garage is "occupied."

e. The garage maintained at a neutral or negative pressure with respect to adjacent occupiable areas when the garage is scheduled to be occupied.

f. CO sensors certified to the minimum performance requirements listed under §120.6(c) of the Energy Codes.

g. 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 percent 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 sensor serves an area no more than 5,000 ft^2. Furthermore, the standard requires a minimum of two sensors per "*proximity zone*." *Proximity zones* are 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 toward 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:

a. Certified by the manufacturer to:
    1. Accuracy of +/- 5%.
    2. 5% or less drift per year.
    3. Require calibration no more than once per year.
b. Be factory-calibrated
c. 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. At a minimum, the following must be monitored:
    1. If any sensor has not been calibrated according to the manufacturer’s recommendations within the specified calibration period, the sensor has failed.
    2. During unoccupied periods, the system compares the readings of all sensors. For example, if any sensor is more than 15 ppm above or below the average of all sensors for longer than four hours, the sensor has failed.
    3. During occupied periods, the system compares the readings of sensors in the same proximity zone. For example, if the 30-minute rolling average for any sensor in a proximity zone is more than 15 ppm above or below the 30-minute rolling average for other sensor(s) in that proximity zone, the sensor has failed.

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:

1. Direct replacement of exhaust air limitations.
2. Type I exhaust hood airflow limitations.
3. Makeup and transfer air requirements.
4. 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:

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

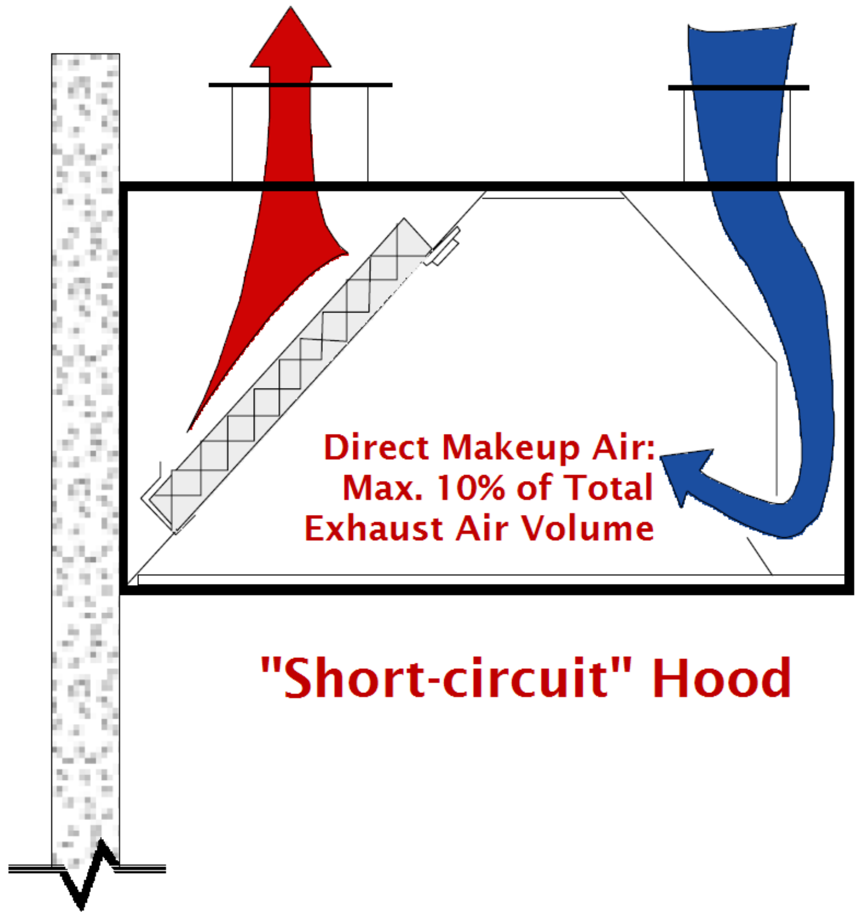
A. Limitation of Short-Circuit Hoods

§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 and Electric (PG&E), the American Gas Association (AGA), and the California Energy Commission (CEC) have shown that in short-circuit hoods, direct supply greater than 10 percent 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 reduction results in higher consumption of energy and conditioned makeup air.

Figure 10-2: Short-Circuit Hood



10.3.3.1 Maximum Exhaust Ratings for Type I Kitchen Hoods

§140.9(b)1B

The Energy Codes 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 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: 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 (light-duty through extra-heavy-duty). The values in this table are typically less than the minimum airflow rates for hoods that are not Underwriter Laboratories (UL) specification-listed products. These values are supported by ASHRAE research for use with UL-listed hoods. (For more detail see ASHRAE research project report RP-1202.) 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 Codes 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
Back shelf/Pass-over	210	210	280	Not Allowed

Energy Codes Table 140.9-A

10.3.3.2 Kitchen Ventilation

§140.9(b)2

This section covers two requirements:

1. Limitations to the amount of mechanically heated or cooled airflow for kitchen hood makeup air §140.9(b)2A
2. 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 Codes define as the following:

a. ***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, and hotel/motel buildings, cooling of a space by direct or indirect evaporation of water alone is not considered mechanical cooling.

b. ***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.

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.

10.3.3.2 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 the greater of:

1. The supply flow required to meet the space heating or cooling load.
2. 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. This amount includes 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.

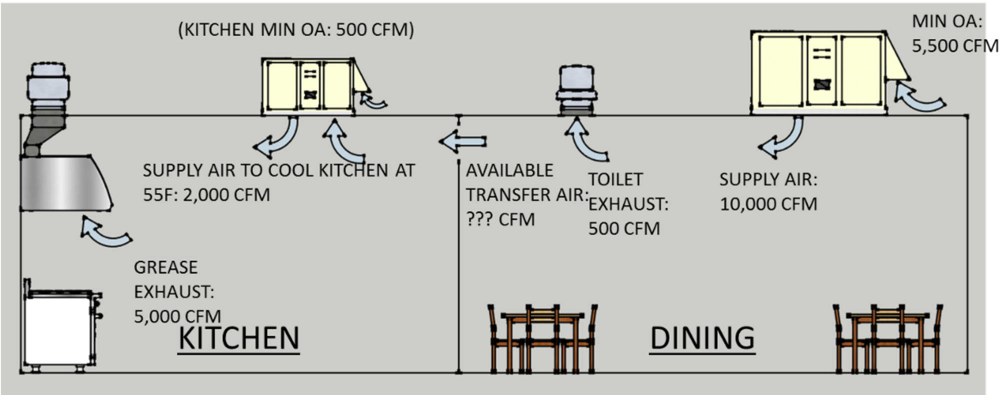
An exception is provided for existing kitchen makeup air units (MAU) that are not being replaced as part of an addition or alternation.

While the requirement to use available transfer air refers only 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 makeup 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, subtract 500 cfm for the toilet exhaust and 0 cfm for building pressurization.

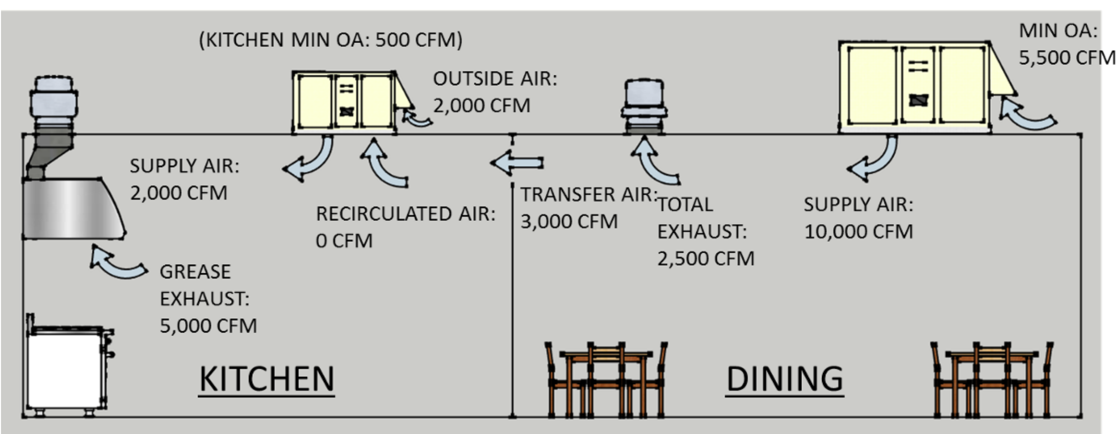
5,500 cfm – 500 cfm – 0 cfm = 5,000 cfm

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:

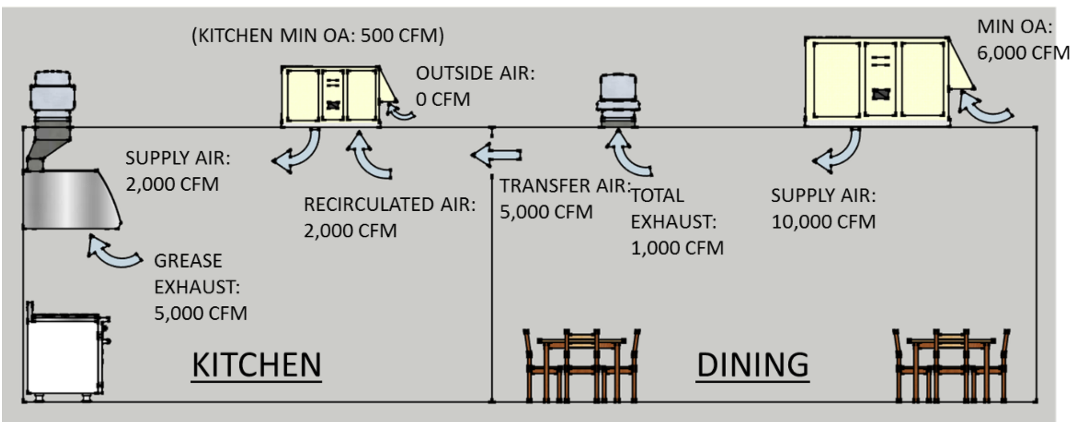
Yes. This example meets the first provision of §140.9(b)2A. The supply flow required to meet the cooling load is 2,000 cfm. Thus, up to 2,000 cfm of mechanically conditioned makeup air can be provided to the kitchen. The supply from the MAU, 2,000 cfm, is not as large as the hood exhaust, 5,000 cfm. 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 percent of the makeup air, 5,000 cfm, is provided by transfer air from the adjacent dining room. 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:

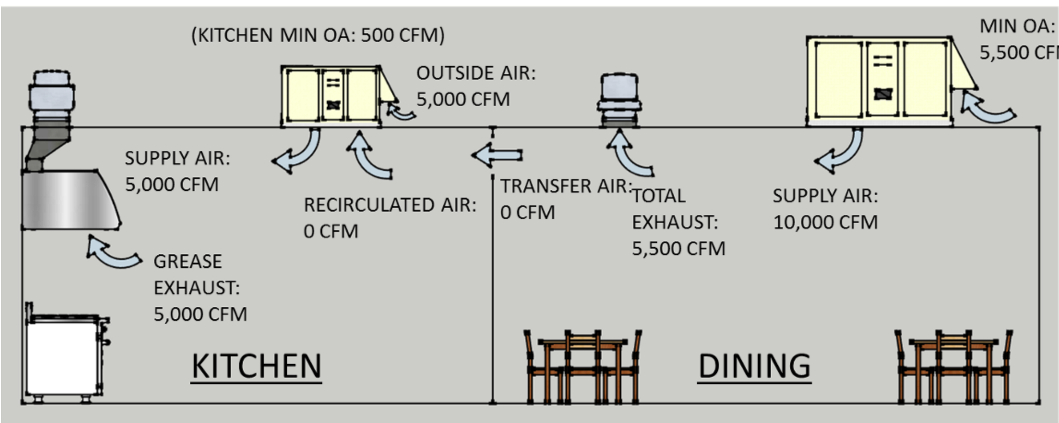
1. The total outside airflow to be conditioned has been reduced from 7,500 cfm in the previous example (2,000 cfm at the MAU and 5,500 cfm at the dining room unit) to 6,000 cfm.
2. The dining room exhaust fan has dropped from 2,500 cfm to 1,000 cfm reducing both fan energy and first cost of the fan.

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 is mechanically heated or cooled. Per §140.9(b)2A, the maximum amount of makeup air that can be mechanically heated or cooled cannot exceed the greater of:

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 (5,500 - 500 = 5000 cfm; from Example 10-2).

The 5,000 cfm of conditioned makeup air exceeds 2,000 cfm. This example assumes that the required exhaust for the dining space is 500 cfm of bathroom exhaust, and the remaining 5,000 cfm of dining outdoor air is available for transfer to the kitchen.

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 and 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. The listed energy recovery devices have a sensible heat recovery effectiveness ≥40% on ≥50% of the total exhaust flow.
4. Seventy-five percent or more of the makeup air volume is:
   a. Unheated or heated to no more than 60°F.
   b. Uncooled or cooled without the use of mechanical cooling.

Exception to 140.9(b)2B: Existing hoods not being replaced as part of an addition or alteration.

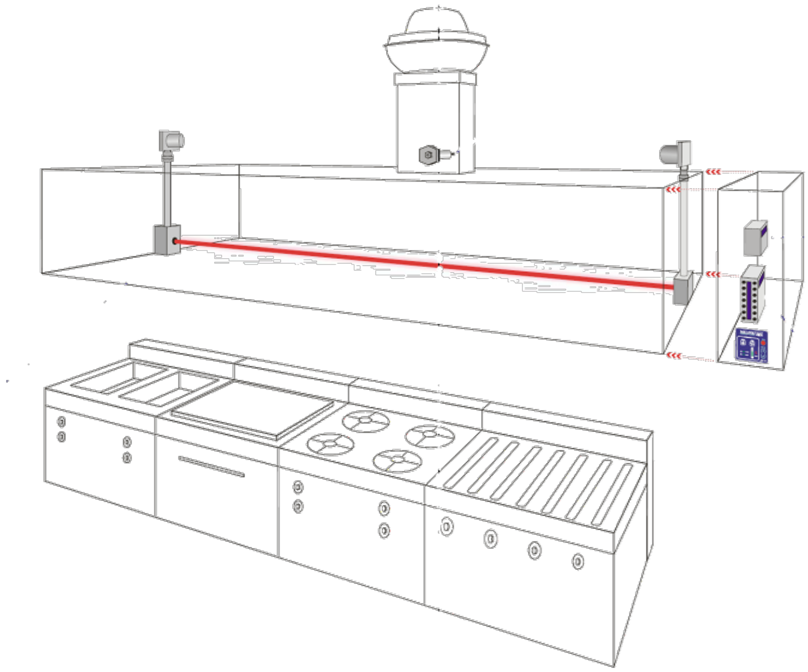
Transfer Air: The concept of transfer air was addressed in the discussion of §140.9(b)2A above.

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

a. 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.
b. Include failsafe controls that result in full flow upon cooking sensor failure.
c. Include an adjustable timed override to allow occupants the ability to temporarily override the system to full flow.
d. Be capable of reducing exhaust and replacement air system airflow rates to the larger of:
   1. 50 percent of the total design exhaust and replacement air system airflow rates.
   2. The ventilation rate required in §120.1.

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

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



Source: California Energy Commission

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) and less 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 set point to 60°F and to use evaporative (non-compressor) cooling or no cooling at all for 75 percent 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.

10.3.3.4 Healthcare Facilities

Healthcare facilities are not required to meet 140.9(b).

10.3.4 Additions and Alterations

See above sections for specific applications of these measures to additions and alterations.

10.4 Computer Rooms

10.4.1 Overview

Sections 120.6(j), 140.9(a) and 141.1(b) provides minimum requirements for *computer rooms*. A *computer room* is defined in §100.1 as "a room within a building whose primary function is to house electronic equipment and that has a design information technology equipment (ITE) equipment power density exceeding 20 watts/ft^2 (215 watts/m^2) of conditioned floor area." ITE is defined in §100.1 Definitions and "includes computers, data storage, servers, and network/communication equipment located in a computer room."

10.4.2 Mandatory Measures

There are three mandatory measures specific to computer rooms:

a. Reheat - §120.6(j)1
b. Humidification - §120.6(j)2, and
c. Fan Control - §120.6(j)3.

The equipment efficiencies in §110.1 and §110.2 also apply.

10.4.2.1 Reheat

§120.6(j)1 Section 120.6(j)1 prohibits reheating, recooling, or simultaneous heating and cooling in c*omputer rooms*. Furthermore, 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.

10.4.2.2 *§120.6(j)2* Humidification

Section 120.6(j)2 prohibits the use of nonadiabatic humidification for *computer 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 printed circuit board failure due to circuit board metallic filament formations known as conductive anodic filaments. For both of these issues, there is insufficient evidence that the risks are adequately addressed 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/ESD Standard 20.20) because it was not effective and did not supplant the need for personal grounding. The Energy Code allows for humidification but prohibits the use of nonadiabatic 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.

10.4.2.3 Fan Control

§120.6(j)3

Section 120.6(j)3 requires that fans serving *computer rooms* 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 66% 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 Prescriptive Measures

The following is a summary of measures for new construction computer rooms:

a. Economizers — §140.9(a)1

b. Power consumption of fans — §140.9(a)2

c. Air containment — §140.9(a)3

d. Alternating current-output uninterruptible power supplies — §140.9(a)4.

10.4.3.1 Economizers

§140.9(a)1

This section requires integrated air or water economizers. If an air economizer is used to meet this requirement, it must be designed to provide 100 percent of the expected system cooling load at outside temperatures of 65°F dry bulb (Tdb) and below or at outside temperatures of 50°F wet bulb (Twb) and below. This is different from the noncomputer-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 than the standard air economizer. Furthermore, air handlers with cooling capacity greater than 33,000 Btu/hr and air economizers must be equipped with fault detection and diagnostic devices meeting §120.2(i).

If a water economizer is used to meet this requirement, it must be capable of providing 100 percent of the expected system cooling load at outside temperatures of 50°F dry bulb and below or at outside temperatures of 45°F wet bulb and below.

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

There are two exceptions to this requirement:

1. **Exception 1 to §140.9(a)1**: Computer rooms with an ITE design load less than 5 tons in a building that does not have any economizers. The computer room exception applies only if none of the other cooling systems in the building includes an economizer. 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 the California climates justified this requirement.
2. **Exception 2 to §140.9(a)1**: Applies to computer rooms with an ITE design load less than 20 tons in a larger building with a central air-handling system and complying air-side economizer that can fully condition the computer rooms on weekends and evenings when the other building spaces are unoccupied. This exception allows the computer rooms to be served by fan coils or split system direct expansion (DX) units as long as the following conditions are met:
    a. The economizer system on the central air-handling unit is sized sufficiently that all the computer rooms are less than 50 percent of the total airflow capacity and the economizer system can provide full economizer cooling to the computer room at outside temperatures of 65°F dry bulb and below or at outside temperatures of 50°F wet bulb and below.
    b. The central air-handling unit is configured to serve only the computer rooms if all the other spaces are unoccupied.

Example 10-5:

Question:

A new data center is built with a total computer room load of 1,500 tons. 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 50°F dry bulb and below or 45°F wet bulb and below. The design conditions in §140.9(a) would require a different heat exchanger and cooling towers than the conditions in §140.4(e) for nonprocess spaces for a given expected load. The load on the cooling towers, while in economizer-only mode, is lower than the design load even if the computer room load is constant because the towers do not have to reject the heat from the chillers. Furthermore, 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, then 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 percent 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 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 65°F and below.
- 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.
- 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 recommended in this application as the set points were based on office occupancies. A differential dry bulb switch would provide greater energy savings.
- Moreover, because the system economizer is separate from the air handler, FDD is not required.

Example 10-7:

Question:

A new office building has a central air system with variable-air-volume (VAV) boxes with reheat with an air-side economizer that complies with §140.4(e). This building has two intermediate distribution frame (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 Code?

Answer:

Not necessarily. Both IDF rooms are required to be served by the central air system economizer of the building. 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 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 (for example, at night), the VAV boxes must be shut so that the VAV system can serve only the IDF rooms.

Example 10-8:

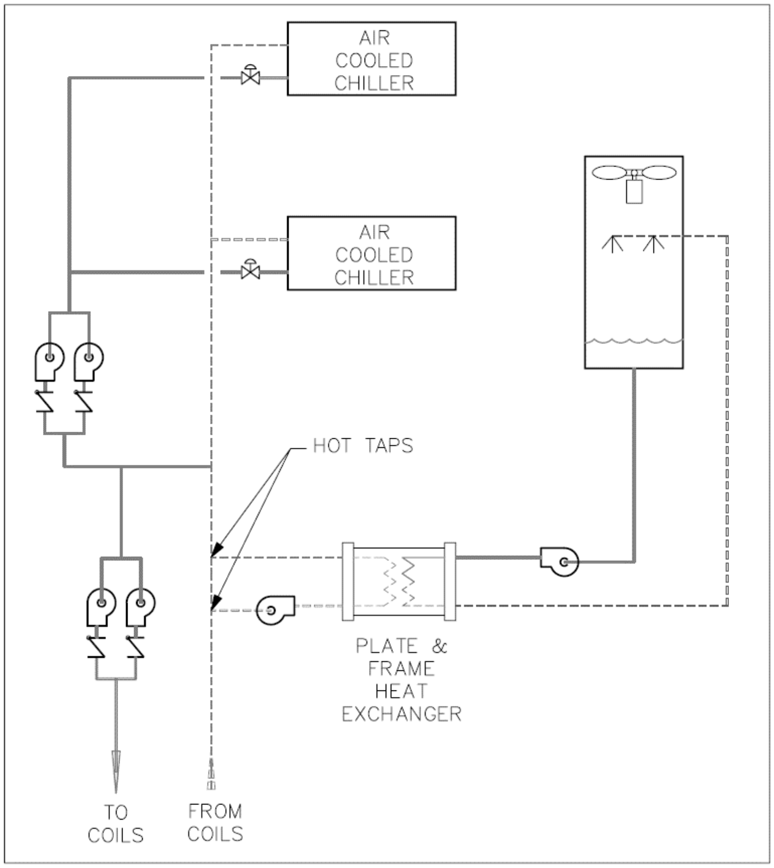
Question:

A new data center employs rear-door heat exchangers that are cooled entirely 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 *economizer* (both air and water) both have the phrase "to reduce or eliminate the need for *mechanical 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



Source: Energy Code

10.4.3.2 Power Consumption of Fans

§140.9(a)2

In §140.9(a)2, 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)

10.4.3.3 Air Containment

§140.9(a)3

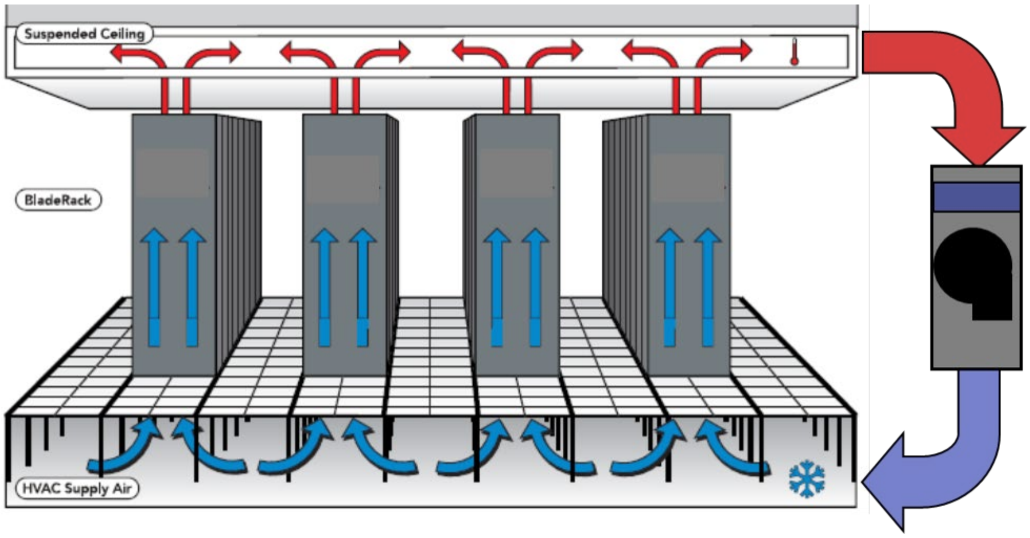
Computer rooms with a design IT equipment load exceeding 10 kW per room are required to have containment to separate the computer equipment inlets and outlets. The requirement can be met using hot-aisle containment, cold-aisle containment, or in-rack cooling. Exceptions are provided for:

a. Expansions of existing *computer rooms* that don't already have containment.

b. Computer racks with a design load of < 1 kW/rack (for example, network racks).

c. 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



Source: California Energy Commission

Figure 10-6: Example of Aisle Containment Using Hard Partitions and Doors

10.4.3.4 *§140.9(a)4* Minimum Uninterruptible Power Supply (UPS) Efficiency

Section 140.9(a)4 requires that any alternating current-output UPS installed for computer room equipment meet or exceed calculation and testing requirements identified in ENERGY STAR Program Requirements for UPSs – Eligibility Criteria Version 2.0 and that UPS meets or exceeds the minimum average efficiencies in Table 140.9-B. There are three categories of UPSs identified in Table 140.9-B:

- Voltage and frequency dependent
- Voltage independent
- Voltage and frequency independent.

10.4.4 Additions and Alterations

The following is a summary of measures for additions and alterations for computer rooms:

a. Economizers - §141.1(b)1.

The equipment efficiencies in §120.6(j), 140.9(a)2, and §140.9(a)4 also apply.

10.4.5.1 Economizers

This section requires integrated air or water economizers. If an air economizer is used to meet this requirement, it must be designed to provide 100 percentof the expected system cooling load at outside temperatures of 55°F dry bulb (Tdb) and below or at outside temperatures of 50°F wet bulb (Twb) and below. This is different from the noncomputer-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 than the standard air economizer. Furthermore, air handlers with cooling capacity greater than 33,000 Btu/hr and air economizers must be equipped with fault detection and diagnostic devices meeting §120.2(i).

If a water economizer is used to meet this requirement, it must be capable of providing 100 percent of the expected system cooling load at outside temperatures of 40°F dry bulb and below or at outside temperatures of 35°F wet bulb and below.

1. **Exception 1 to §141.1(b)**: Computer rooms with an ITE design load less than 5 tons in a building that does not have any economizers. The computer room exception applies only if none of the other cooling systems in the building includes an economizer.
2. **Exception 2 to §141.1(b)**: New cooling systems serving an existing computer room in an existing building up to a total of 50 tons of new ITE design load per building.

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 IT equipment load you would be forced to either provide economizer cooling or offset the energy loss by using the performance approach. Examples of how to meet this requirement include:

a. Provide the new capacity using a new cooling system that has a complying air or water economizer.
b. 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 §141.1(b)**: New cooling systems serving a new computer room up to a total of 20 tons of ITE load in an existing building.

   This is similar to the previous exception, but the capacity threshold is lower because you can locate a new space in a location suitable for an integrated economizer.

10.5 Commercial Refrigeration

10.5.1 Overview

This section addresses §120.6(b) of the Energy Code, 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 Code must also comply with the general provisions of the Energy Code (§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 trade-offs 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 2022 Energy Codes

In the 2022 Energy Code, adiabatic condenser efficiency and size requirements have been added. Section 120.6(b) 1D and 1E along with Table 120.6 – D have been updated with new requirements for adiabatic condenser systems using halocarbon refrigerant.

10.5.1.3 Scope and Application

§120.6(b)

Section 120.6(b) of the Energy Code applies to retail food or beverage stores that have 8,000 square feet or more of conditioned area and use either refrigerated display cases or walk-in coolers or freezers. The Energy Code has 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 Code does 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). Furthermore, the Energy Code does 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 Code.

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 lineups 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 Code for the two self-contained display cases. The refrigeration systems serving the other fixtures must comply with the Energy Code.

10.5.2 Condenser Mandatory Requirements

§120.6(b)1

This section addresses the mandatory requirements for condensers serving commercial refrigeration systems. These requirements apply only 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 the following conditions apply:

1. The total heat of rejection of the compressor system attached to the condenser or condenser system does not increase.
2. Less than 25 percent of the attached refrigeration system compressors (based on compressor capacity at design conditions) are new.
3. Less than 25 percent 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 percent "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 percent of the suction group capacity at design conditions, and the replacement display cases comprise 20 percent 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 Code?

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 percent of the suction group design capacity at design conditions.

3b. The replacement display cases comprise less than 25 percent 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, evaporative, or adiabatic condensers; fans on air- or water-cooled fluid coolers; or cooling towers used to reject heat on new refrigeration systems must be continuously variable-speed controlled. 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, usually no higher than 10–20 percent, the fans may be staged off to reduce condenser capacity. As load increases, fans should be turned back on before significantly increasing fan speed, recognizing a control band is necessary to avoid 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, the holdback valve is set below the fan control minimum saturated condensing temperature set point.

To minimize overall system energy consumption, the condensing temperature control set point must be continuously reset in response to ambient temperatures, rather than using a fixed set point value. This strategy is also termed ambient-following control, ambient-reset, wet bulb-following, and dry bulb-following — all referring to control logic that changes the condensing temperature control set point in response to ambient conditions at the condenser. The control system calculates a control set point saturated condensing temperature that is higher than the ambient temperature by a predetermined temperature difference (in other words, the condenser control temperature difference). Fan speed is then modulated so that the measured saturated condensing temperature (SCT) matches the calculated SCT control set point. The SCT control set point for evaporative condensers or water-cooled condensers (via cooling towers or fluid coolers) must be reset according to the ambient wet bulb temperature, and the SCT control set point for air-cooled condensers must be reset according to ambient dry bulb temperature. The target SCT for adiabatic condensers when operating in dry mode must be reset according to ambient dry bulb temperature. There is no requirement for SCT control during wet-mode (adiabatic) operation. Systems served by adiabatic condensers in Climate Zone 16 are exempted from this control requirement.

The condenser control TD is not specified in the Energy Code. 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 use as much condenser capacity as possible without excessive fan power, the common practice is to optimize the control TD over a period such that the fan speed is in a range of around 60–80 percent during normal operation (that is, when not at minimum SCT and not in heat recovery).

The minimum saturated condensing temperature set point must be 70°F (21°C) or less. For systems using halocarbon refrigerants with glide, the SCT set point 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 set point is also commonly employed to set an upper bound on the control set point 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 use the average condensing temperature or the highest condensing temperature of the 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.

Air-cooled condensers with separately installed evaporative precoolers added to the condenser are not considered adiabatic condensers for this standard and must meet the requirements for air-cooled equipment, including specific efficiency and ambient-following control.

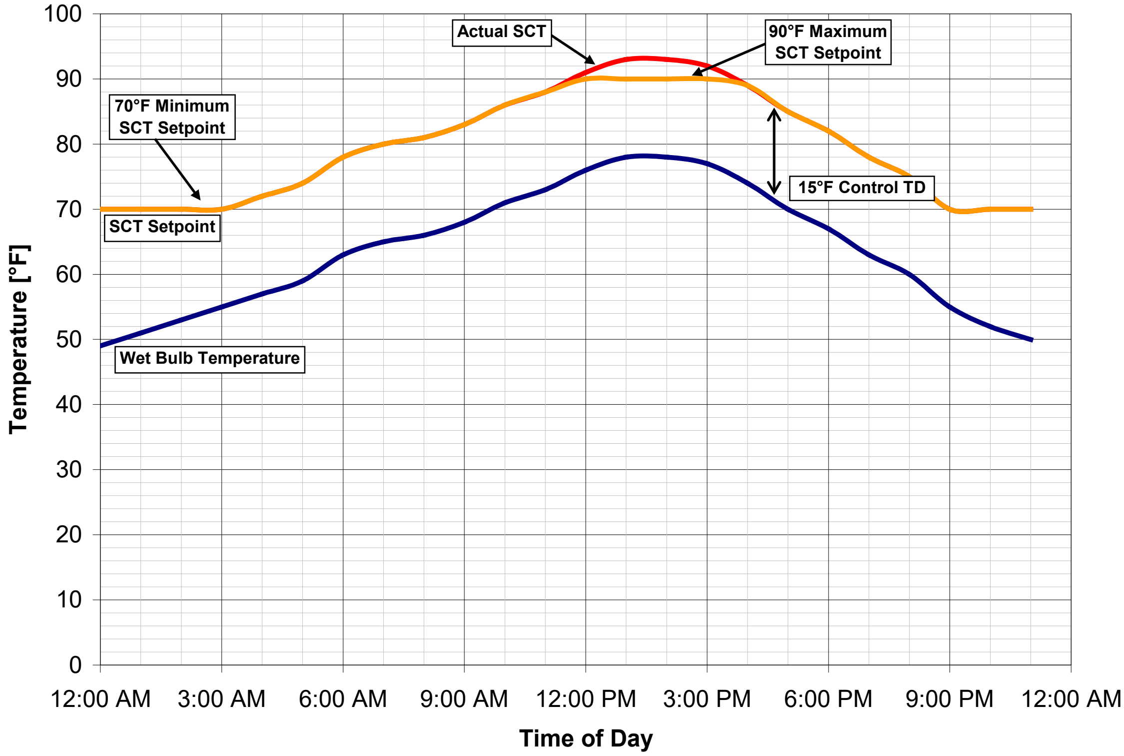
Example 10-12:

Question:

A new supermarket with an evaporative condenser is being commissioned. The control system designer has used a wet bulb-following control strategy to reset the system-saturated condensing temperature (SCT) set point. 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 set point that minimizes the combined compressor and condenser fan power usage throughout the year. What might the system SCT and SCT set point trends look like over an example day?

Answer

The following figure illustrates what the actual saturated condensing temperature and SCT set points 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 set point is continuously reset to 15°F (8.3°C) above the ambient wet bulb temperature until the minimum SCT set point of 70°F is reached. The figure also shows a maximum SCT set point (in this example, 90°F (32.2°C), which may be used to limit the maximum control set point, regardless of the ambient temperature value or TD parameter.



10.5.2.2 Condenser-Specific Efficiency

All newly installed evaporative condensers, air-cooled condensers, and adiabatic 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 Wet bulb Temperature
Air-Cooled	65 Btuh/Watt	105°F Saturated Condensing Temperature (SCT), 95°F Entering Dry bulb Temperature
Adiabatic Dry Mode	45 Btuh/Watt (Halocarbon)	105°F Saturated Condensing Temperature (SCT), 95°F Entering Dry bulb Temperature

Source: California Energy Commission

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 SCT and 70°F outdoor wet bulb temperature for evaporative condensers, and 105°F SCT and 95°F outdoor dry bulb temperature for air-cooled and adiabatic (halocarbon refrigerant only) condensers. Total heat of rejection capacity for adiabatic condensers is based on dry mode ratings (in other words, no precooling of the air). 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 wet bulb temperatures. Usually, the manufacturer publishes two sets of correction factors: one is a set of "heat rejection" capacity factors, while the other is a set of "evaporator ton" capacity factors. Only the "heat rejection" capacity factors shall be used to calculate the condenser capacity at the efficiency rating conditions for determining compliance with this section.

For air-cooled and adiabatic condensers, manufacturers typically provide the capacity at a given temperature difference (TD) between SCT and dry bulb 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 and adiabatic 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 original equipment manufacturer (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 determine specific efficiency more accurately.

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 or alteration.
3. If the condenser capacity is less than 150,000 Btuh 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 catalog 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 10½ 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 dry bulb 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 dry bulb = 10°F TD). Input power is equal to the number of motors multiplied by the input power per motor:

10 fan motors x $\frac{\textit{450 Watts}}{\textit{fan motor}}$ *= 4,500 Watts*

The specific efficiency of the condenser is therefore:

500MBH x $\frac{\textit{1,000 Btu/hr / 4,500Watts}}{\textit{4,500 Watts}}$ *= 111Btu/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 wet bulb 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 MBH/0.95 = 2,105 MBH

To calculate input power, we will assume 100 percent 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 six-pole open fan motor, the minimum efficiency is 91.7 percent. For a 2 HP six-pole open pump motor, the minimum efficiency is 87.5 percent. 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:

$$1\ \text{Motor} \times \frac{2\ \text{HP}}{\text{Motor}} \times 746\ \frac{\text{watts}}{\text{HP}} \times \frac{100\%\ \text{assumed loading}}{87.5\%\ \text{efficiency}} = 1{,}705\ \text{watts}$$

The combined input power is therefore:

8,135 watts + 1,705 watts = 9,840 watts

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

Finally, the efficiency of the condenser is:

$$\left(2{,}105\ \text{MBH} \times \frac{1{,}000\ \text{Btuh}}{\text{MBH}}\right) / 9{,}840\ \text{watts} = 214\ \text{Btuh/watt}$$

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

Example 10-15

Question:

An adiabatic condenser is being designed for a new supermarket. The refrigerant is R-407A. The condenser manufacturer's catalogue states that the subject condenser has a capacity of 550 MBH at 10°F TD between entering air dry bulb temperature and saturated condensing temperatures with R-407A refrigerant when operating in dry mode. Elsewhere in the catalog, it states that the condenser has two 5-hp fan motors that draw 4.5 kW 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 10-2, we see that the rating conditions for an air-cooled condenser are 95°F entering dry bulb temperature and 105°F SCT. The catalog capacity is rated at a 10°F temperature difference, which is deemed suitable for calculating the specific efficiency (105°F SCT - 95°F entering dry bulb = 10°F TD). Input power is equal to the number of motors multiplied by the input power per motor:

2 fan motors x 4,500 watts = 9,000 watts

The specific efficiency of the condenser is therefore:

(550MBH x 1,000 Btu/hr/MBH) / 9000 watts = 61 Btu/hr/watts

This condenser has a specific efficiency of 61 Btuh per watt, which is higher than the 45 Btuh per watt minimum requirement. This condenser meets the minimum specific efficiency requirements.

10.5.2.3 Condenser Fin Density

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 use a microchannel 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.

10.5.2.4 Adiabatic Condenser Sizing

§120.6(b)1E

New adiabatic condensers on new refrigeration systems must follow the condenser sizing, fan control, and efficiency requirements as described in §120.6(b)1E.

Condensers must be sized to provide sufficient heat rejection capacity under design conditions while maintaining a specified maximum temperature difference between the refrigeration system 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 this requirement, reserve or backup compressors may be excluded from the calculations.

Section 120.6(b)1E provides maximum design SCT values for adiabatic condensers. For 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 adiabatic condensers is proportional to the temperature difference (TD) between SCT and DBT for operation in dry mode, regardless of the actual ambient temperature entering the condenser. For example, the capacity of an adiabatic 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 during dry mode operation, since the TD across the condenser is 10°F in both examples. Thus, similar to air-cooled condensers, the requirement for adiabatic condensers does not have varying sizing requirements for different design ambient temperatures.

However, the Energy Code has different requirements for adiabatic condensers depending on the space temperatures served by the refrigeration system. The maximum design SCT requirements are listed in Table 10-5 below.

Refrigerated Load Type	Space Temperature	Maximum SCT (dry mode)
Cooler	≥ 28°F	Design DBT plus 30°F
Freezer	< 28°F	Design DBT plus 20°F

Source: California Energy Commission

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

Example 10-16:

Question:

An adiabatic condenser is being sized for a system that has half of the 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 adiabatic condensers for coolers has a design approach of 30°F and for freezers a design approach of 20°F. 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 percentage of the total installed capacity dedicated to coolers by 30°F. Next, multiply the percentage of the total installed capacity de freezers by 20°F. 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 25°F.

(50% x 20 °F) + (50% x 30°F) = 10 °F + 15 °F = 25 °F

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 set point 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

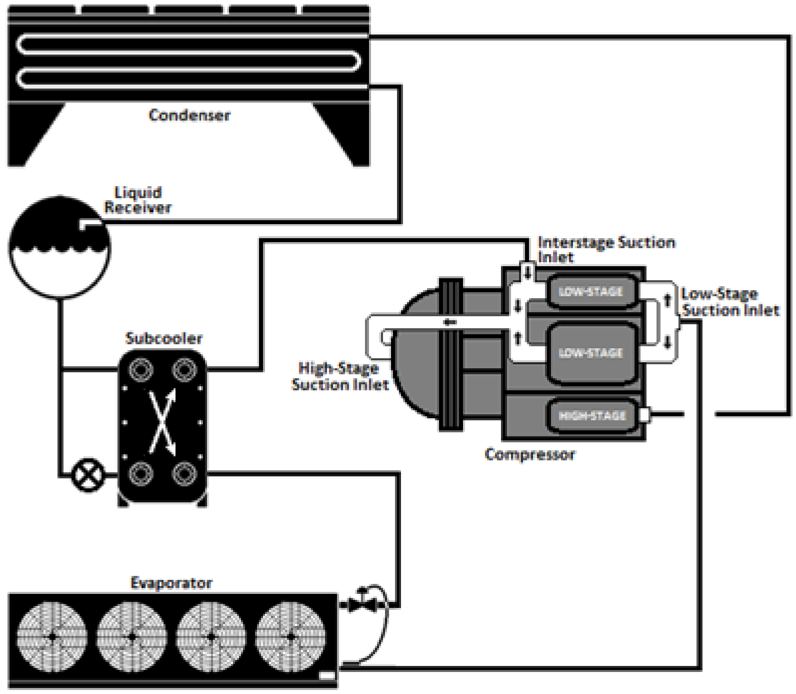


Figure 10-8: Cascade System

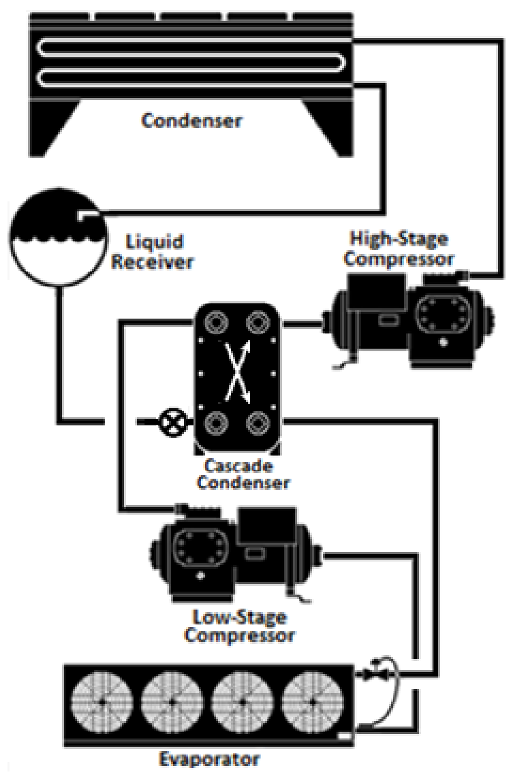
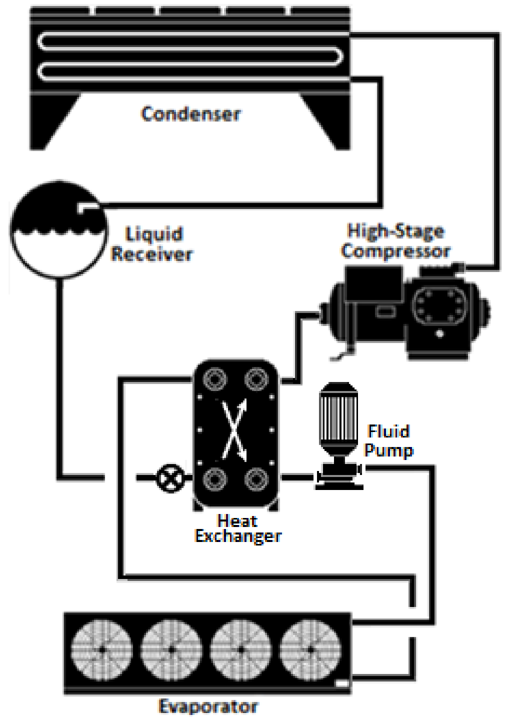


Figure 10-9: Secondary Fluid System



Example 10-17

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 that 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 10-18:

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 10-19:

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.

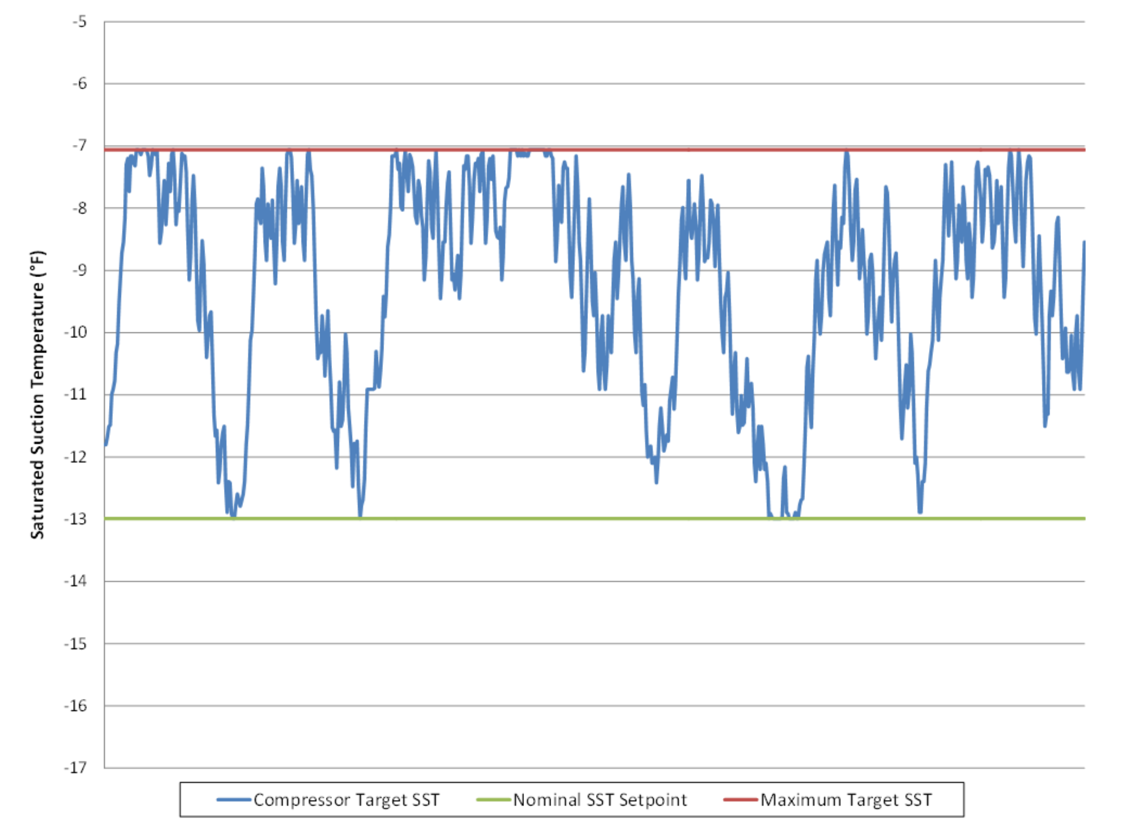
Cascade system: Only the low-temperature suction group of the system is required to have floating suction pressure control.

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 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 temperature (SST) set point is allowed to vary depending on the actual requirements of the attached loads, rather than fixing the SST set point low enough to satisfy the highest expected yearly load. The target set point is adjusted so that it is just low enough to satisfy the lowest current SET requirement of any attached refrigeration load while maintaining target fixture temperatures, but not any higher. The controls are typically bound by low and high set point 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 one week, expressed in equivalent saturation temperature. The suction pressure control set point is adjusted to meet the temperature set point 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

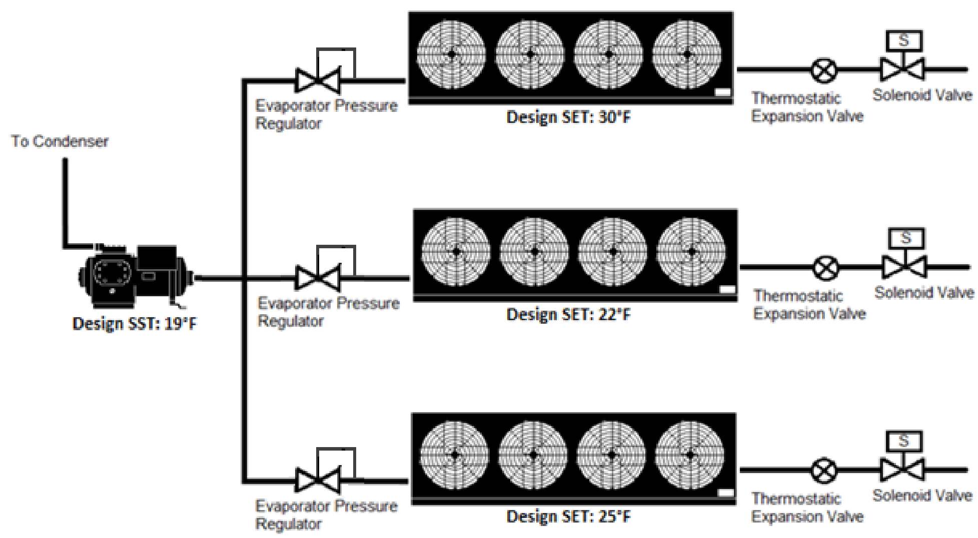
Mechanical evaporator pressure regulators (EPR valves) are often used on multiplex systems to maintain temperature by regulating the SET at each evaporator connected to the common suction group and often to function as a suction stop valve during defrost. EPR valves throttle to maintain the pressure 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 set point.

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 setting 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 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-1980s; however careful attention is still required during design, start-up, and commissioning to ensure control is effectively coordinated.

Figure 10-11: Evaporators With Evaporative Pressure Regulator Valves



B. Floating Suction Pressure Control with Electronic Suction Regulators

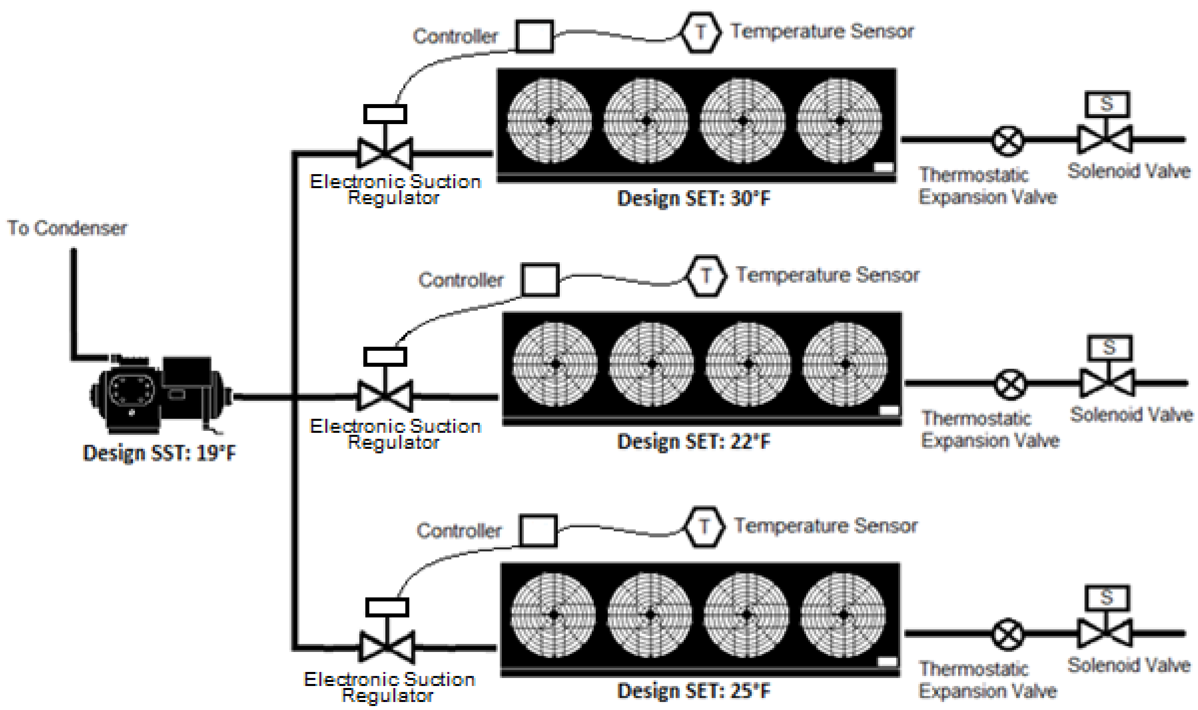
An electronic suction regulator (ESR) valve is an electronically controlled valve used in place of a mechanical evaporator pressure regulator valve. ESRs are also known in the industry as electronic evaporator pressure regulators. ESR valves are not pressure regulators; instead, they control the flow through the evaporator based on a set point air temperature at the case or walk-in. ESR valves are modulated to maintain precise temperature. This modulation provides more accurate temperature compared to an EPR that 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 set point. 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 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 the use of 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.
3. Transcritical CO_2 refrigeration systems.

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

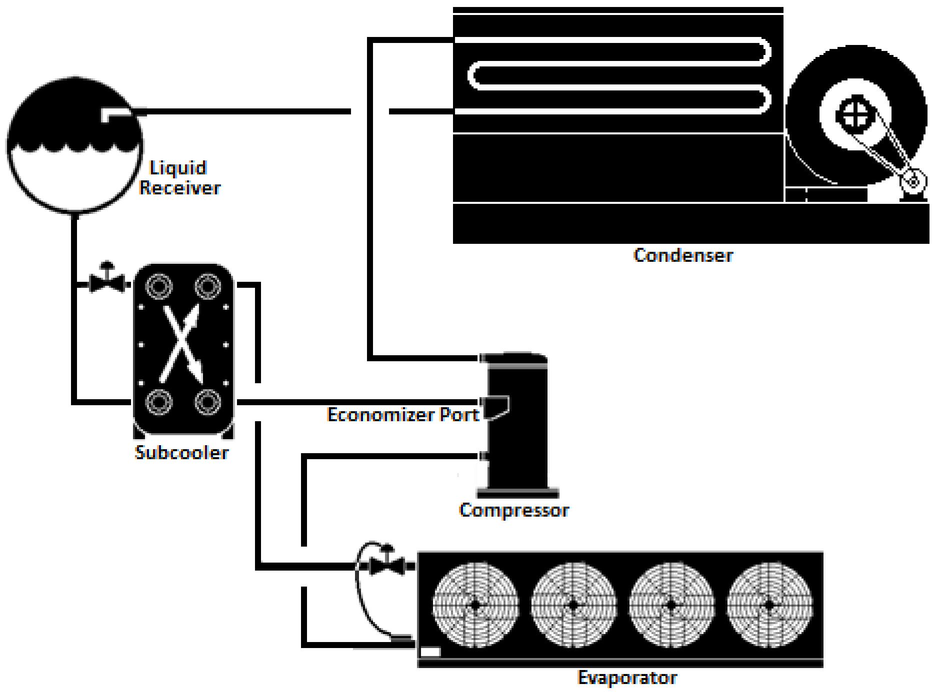
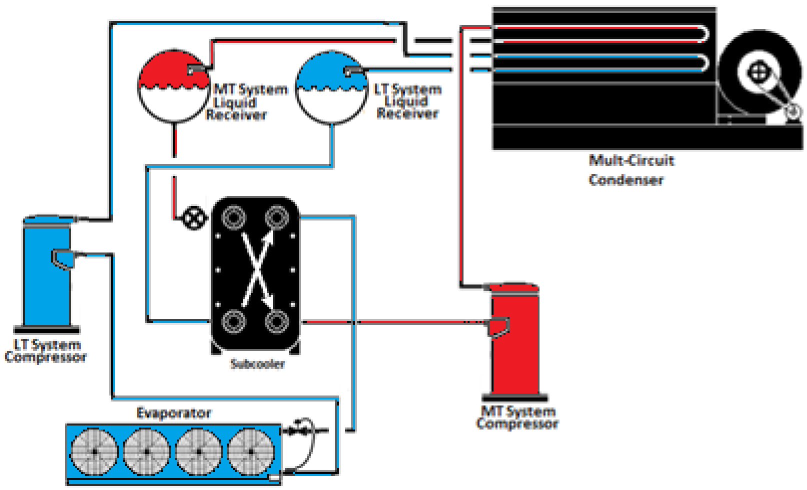


Figure 10-14: Liquid Subcooling Provided by a Separate Medium-Temperature System



10.5.3.1 Compressors for Transcritical CO_2 Refrigeration Systems

§120.6(b)2C

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 gas cooler fan energy) as this reduces compressor head pressure, which directly affects 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.

Section 120.6(b)2C addresses the compatibility of the compressor design and components with the requirements for floating head control. All compressors that discharge to the gas cooler(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 60°F (16°C) or less. 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.

The exception to the minimum SCT of 60°F for transcritical CO_2 systems requirement is:

1. Compressors with a design saturated suction temperature greater than or equal to 30°F shall be designed to operate at a minimum condensing temperature of 70°F or less.
2. Existing compressor systems that are reused for an addition or alteration.

10.5.4 Refrigerated Display Case Lighting Control Requirements

§120.6(b)3

All lighting for refrigerated display cases and glass doors of walk-in coolers and freezers shall be controlled by either automatic time switch controls or motion sensor controls or both.

A. Automatic Time Switch Control

Automatic time switch controls shall turn off the lights during nonbusiness hours.

Timed overrides for a display case lineup or walk-in case may be used to turn on the lights for stocking or nonstandard 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 percent 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 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. Several examples are presented here, but the Energy Code does not require these configurations to be used. The heat recovery design must be consistent with the other requirements in the Energy Code, 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 THR of 150,000 Btu/h or greater must be used for space heat recovery.

Exceptions to the above requirements for heat recovery are:

1. Stores 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.
3. Stores that are designed to provide less than 500,000 Btu/h in total heat rejection for all the refrigeration systems in the store combined.

The Energy Code also limits 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 10-20

Question:

A store has three new distributed refrigeration systems, A, B and C, with design 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 design THR of greater than 150,000 Btu/h, whereas Refrigeration System A has a design THR of less than 150,000 Btuh. Therefore, the store must have the minimum refrigeration heat recovery equal to 25 percent 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 10-21

Question:

How should the THR be calculated for this section?

Answer

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

Example 10-22

Question:

A 35,000 ft^2 food store is expanding 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 to replace 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 percent 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 Code. 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 multicircuit 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 Code, or there may be excessive pressure losses in the piping that could negatively affect compressor energy.

Figure 10-15: Series Direct Heat Recovery Configuration

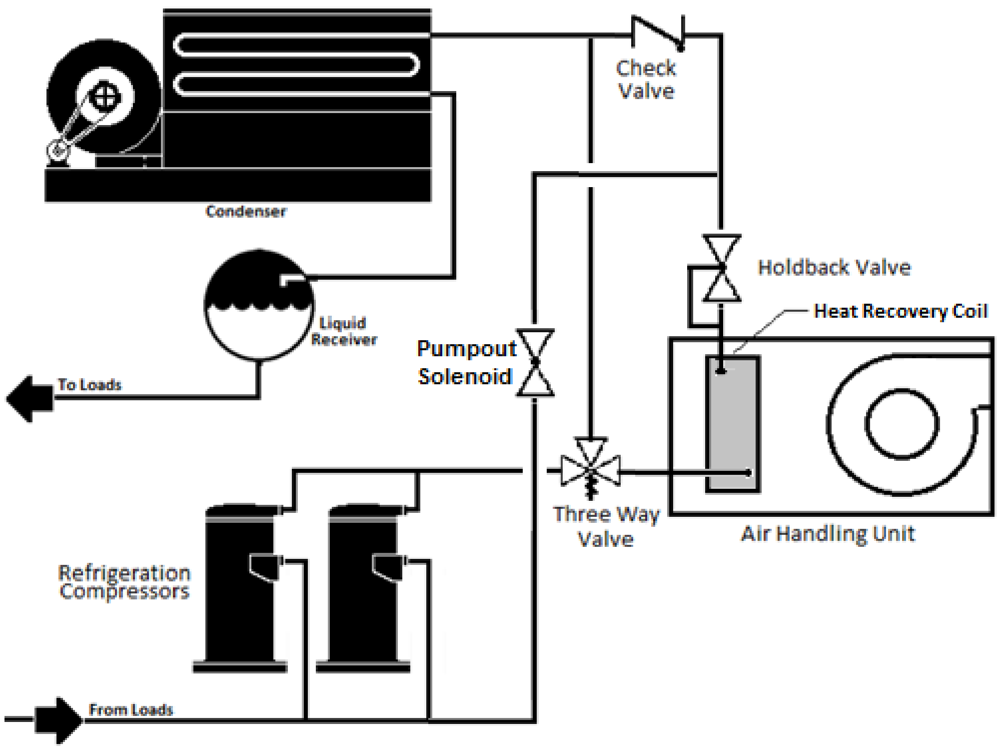
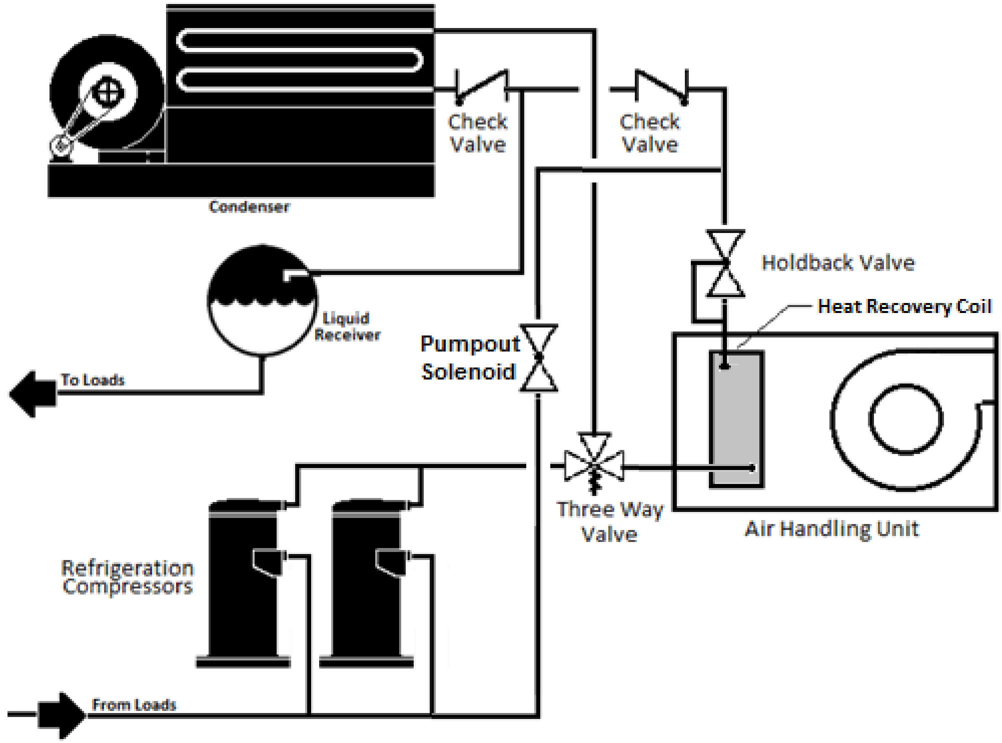


Figure 10-16 shows a parallel-connected direct-condensing configuration. In this configuration, the heat recovery coil handles the entire condensing load for the connected refrigeration system(s) when the air-handling unit is in heating mode. Reduced refrigerant charge is the primary advantage of this 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 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

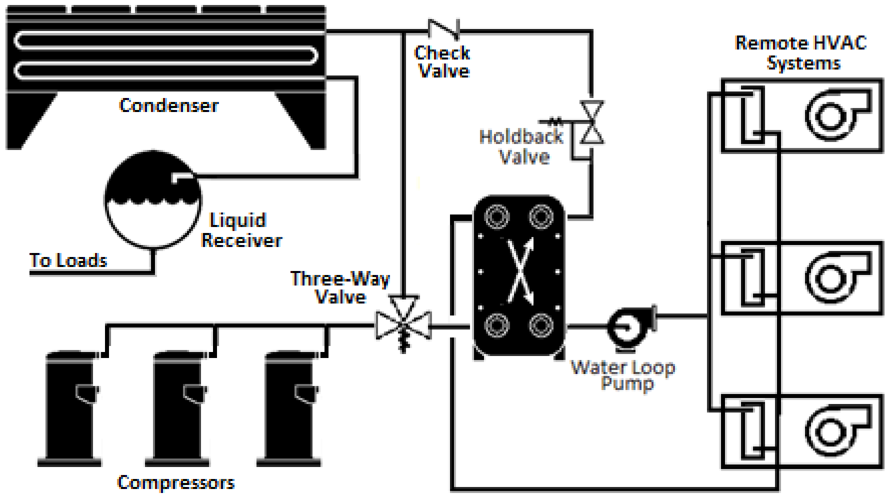


B. Indirect Heat Recovery

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.

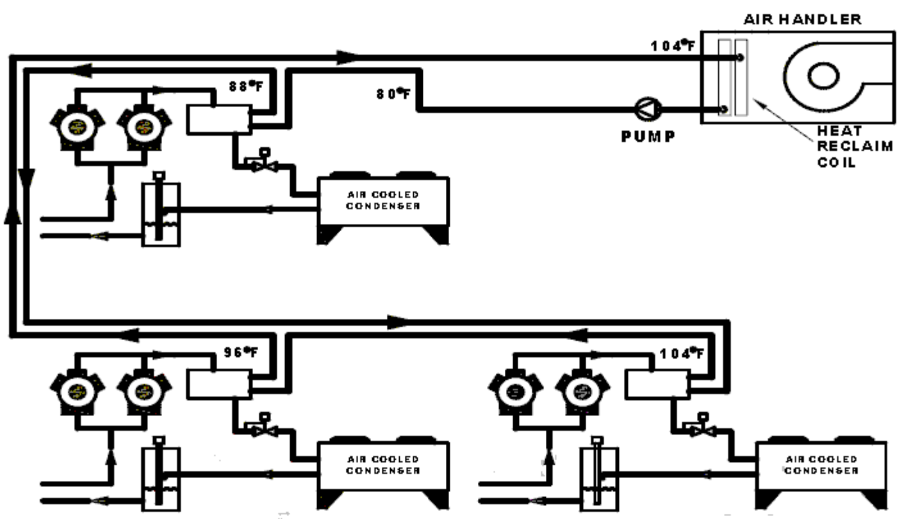
The refrigerant-to-fluid heat exchanger can be located close to the refrigeration system compressors, maximizing the available heat for recovery while keeping the overall refrigerant charge increase low. This configuration is also suitable when multiple HVAC units are employed for the refrigeration heat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



Several 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



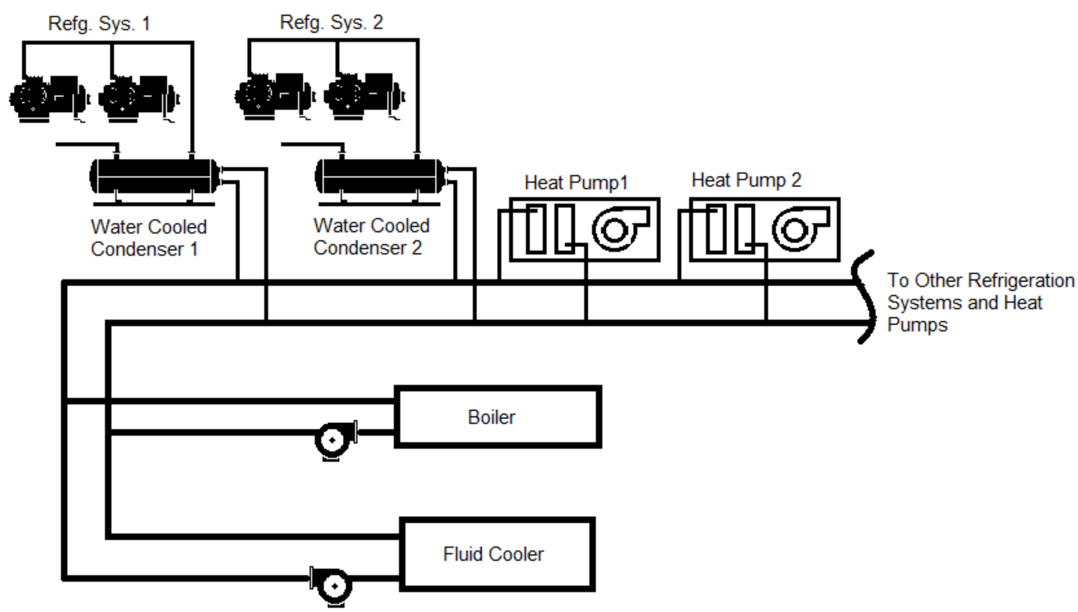
This configuration allows the refrigerant-to-water condenser temperature difference (TD) to be kept low at each refrigeration system (e.g., 8°-10°F is possible) while maintaining a sufficiently high water-side TD at the air-handling unit (e.g., 20°-25°F depending on specifics) to allow an effective selection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

10.5.5.1 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 wet bulb following control logic.

Figure 10-19: Water Loop Heat Pump Example

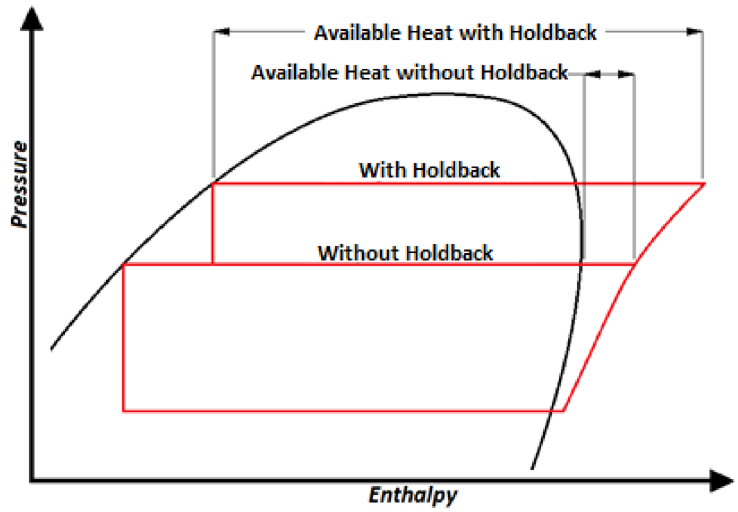


10.5.5.2 Control Considerations

A. Holdback Considerations

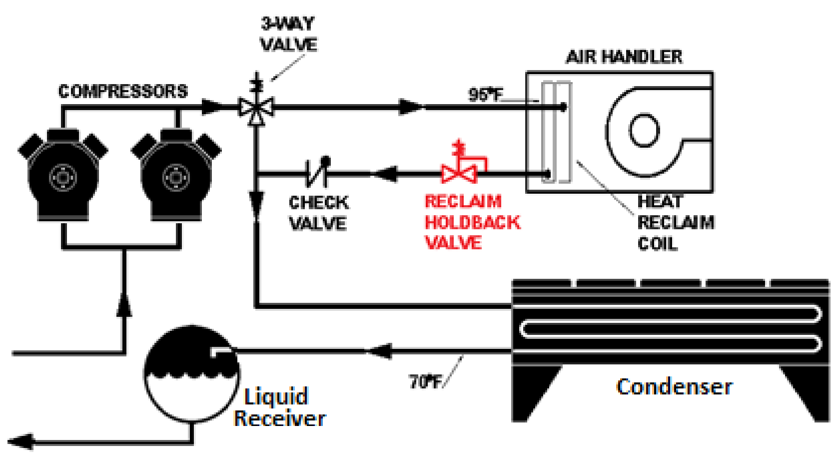
For direct and indirect systems, a holdback valve is required to control the refrigerant condensing temperature in the heat recovery coil (for direct systems) or the refrigerant-to-water condenser (for indirect systems) during heat recovery. Regulating the refrigerant pressure to achieve condensing recovers 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 the inlet and is at the exit of the recovery heat exchanger. Figure 10-21 shows a direct-condensing configuration with the proper location of the holdback valve.

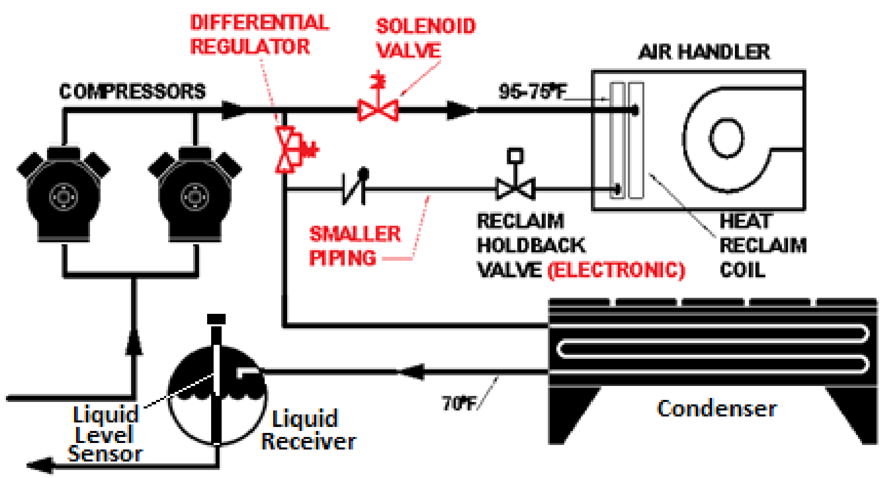
Figure 10-21: Direct-Condensing Configuration Showing Location of Holdback Valve



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 overcondensing (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

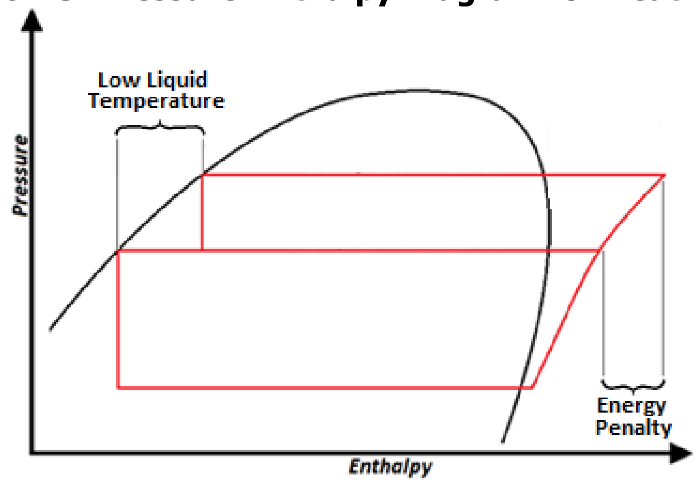


B. Heat Recovery and Floating Head Pressure

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 Codes 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 10-23

Question:

A supermarket is being constructed that will use 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 ft2

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 recover 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:

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

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

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:

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

$$F.V. = \frac{25{,}000 \text{ cfm}}{41.7 \text{ ft2}}$$

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 in 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 = 95oF - 65oF

TD = 30oF

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:

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

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

$$\text{MBH per SQ FT} = 16.72$$

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

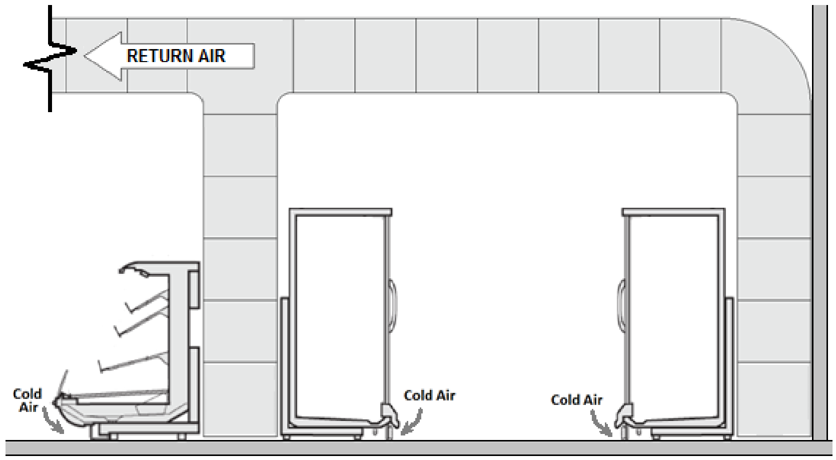
More commonly, computerized selection tools are used to select heat recovery coils, allowing vendors to provide multiple selections for comparison.

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.

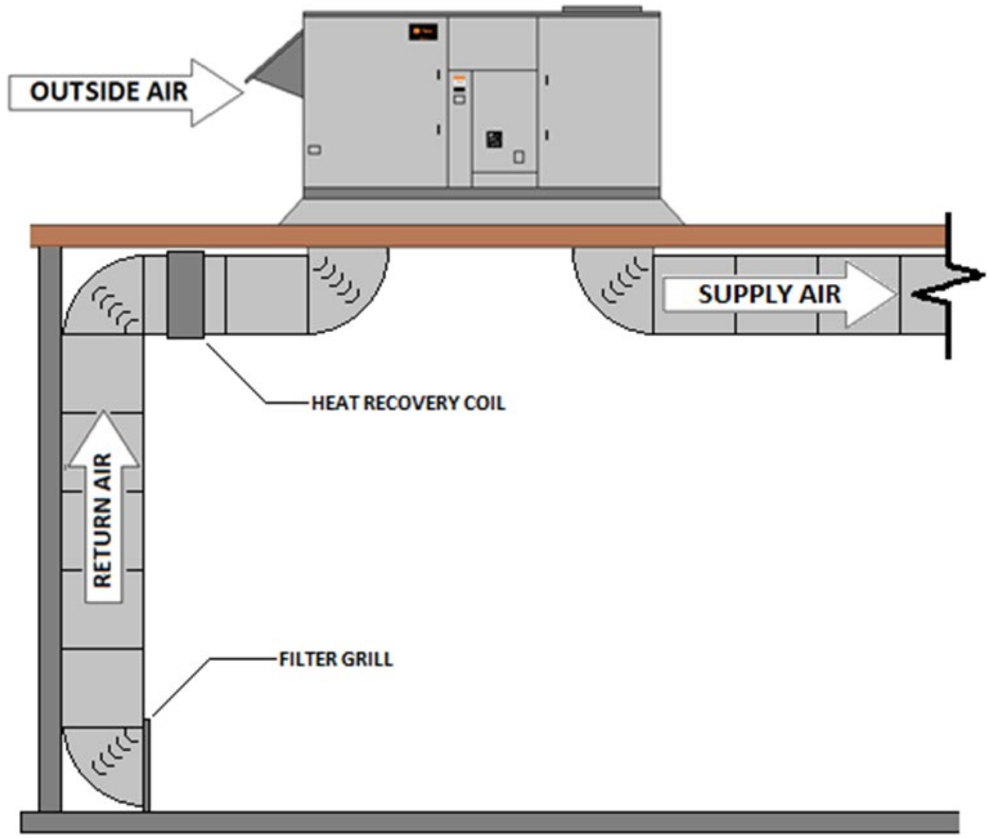
Figure 10--24: Low Return Air Example



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 in the refrigerated space, which predominantly provide heating. The fan design must allow for the additional ductwork and coil pressure drop.

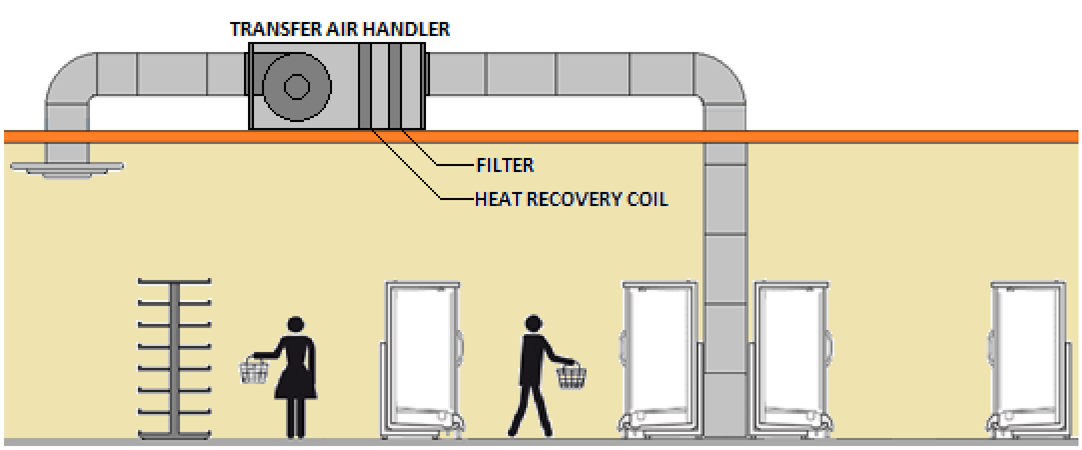
Figure 10-25: Heat Recovery Coil in Return Air Duct



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 use 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

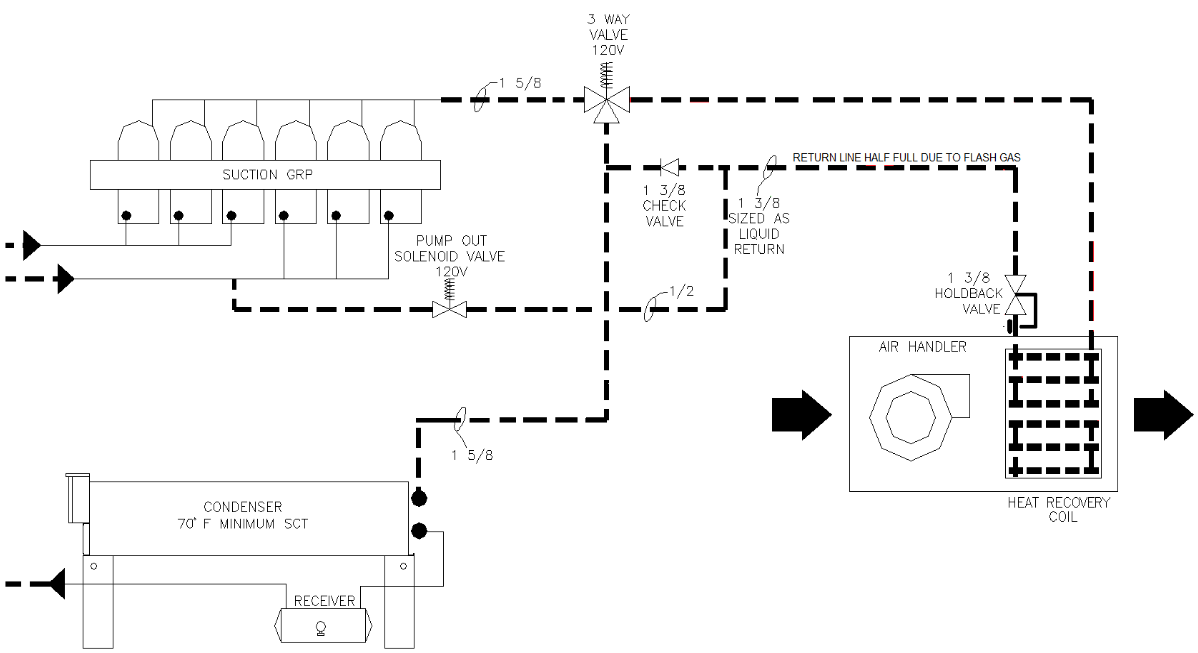
The Energy Code requires that the increase in HFC refrigerant charge from all equipment related to heat recovery for space heating shall be less than 0.35 lbs. for every 1,000 Btuh of heat recovery capacity at design conditions. Refrigerant charge may increase because of the addition of the recovery 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 10-24

Question:

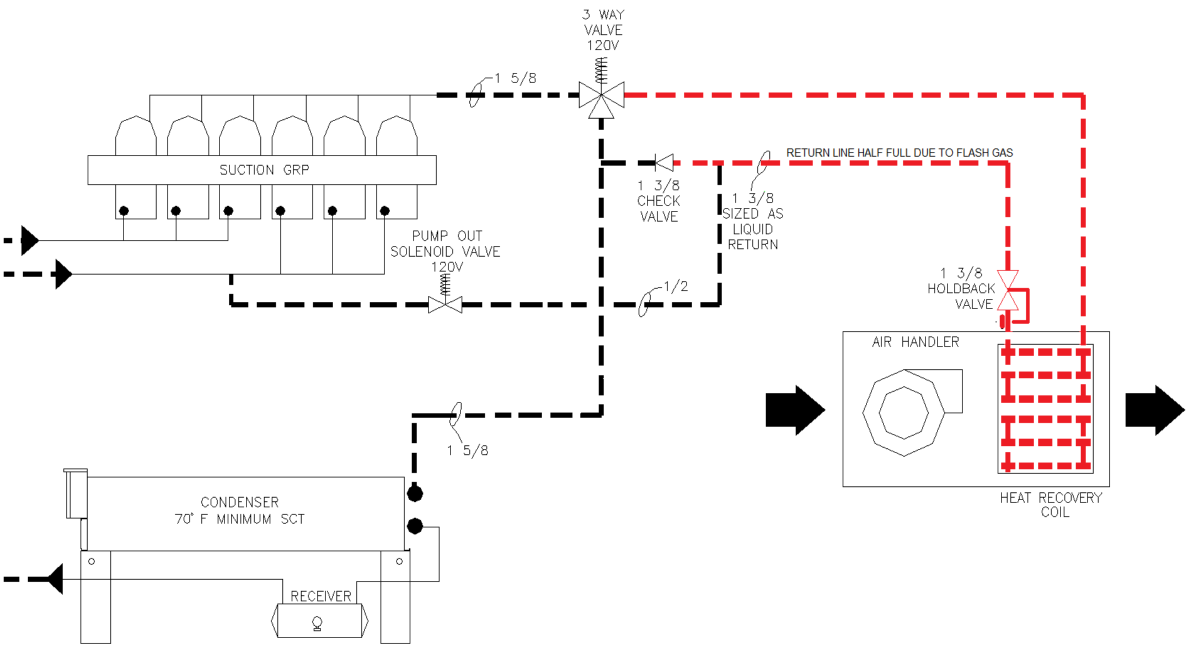
A heat recovery system is being designed for a new supermarket. The refrigerant is R-404A. The proposed design is shown below:



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 10-25

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 the manufacturer's 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 10-26

Question:

In the example above, does the recovery design comply with the requirement in the Energy Code that the recovery design shall use 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.

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

Therefore, the design complies with the Energy Code.

Example 10-27

Question:

In the example above, does the recovery design comply with the requirement in the Energy Code 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.

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

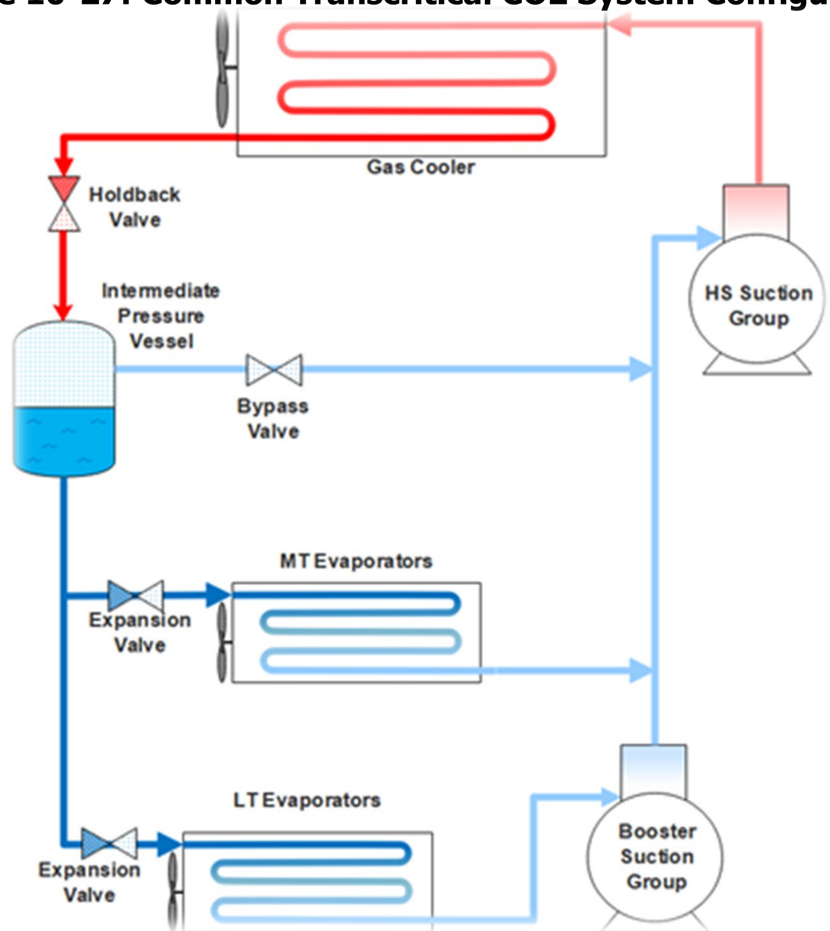
Since the refrigerant charge increases by less than 0.35 lbs/MBH, this design complies with the Energy Code.

10.5.6 Transcritical CO_2 Systems

§120.6(b)5

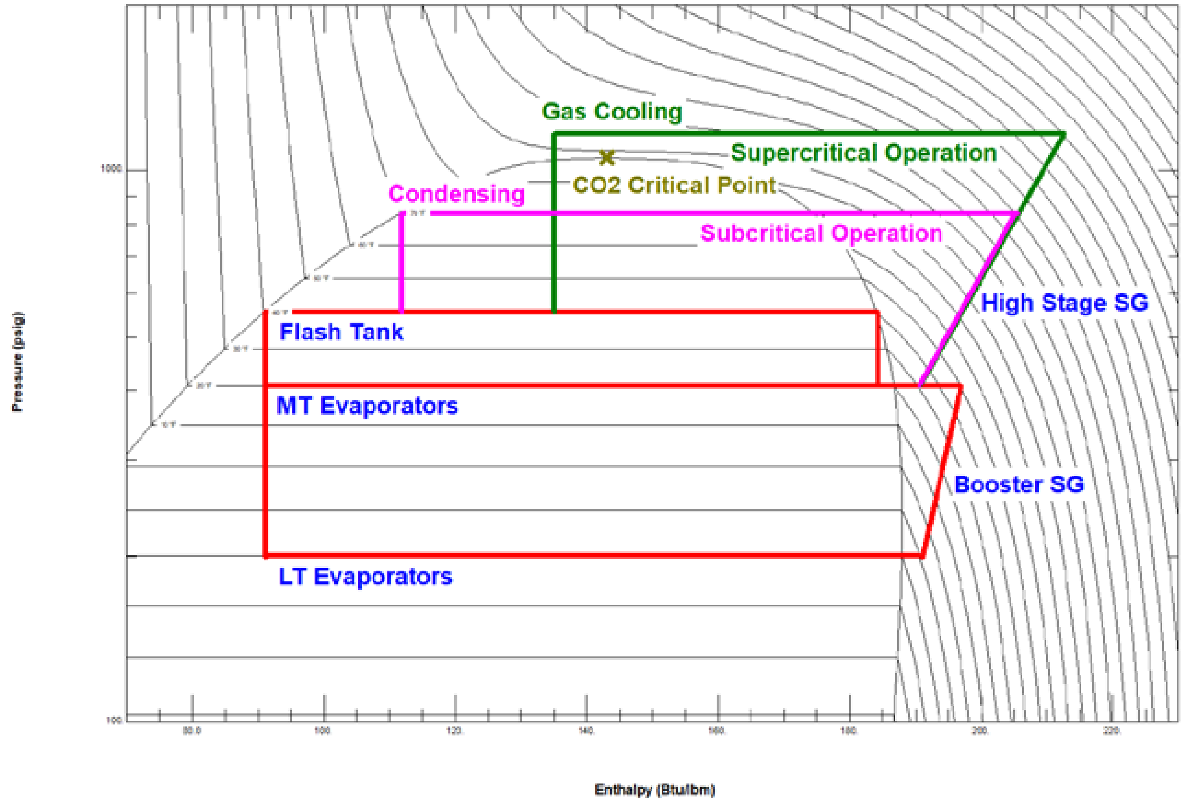
A typical transcritical CO_2 booster system is shown in Figure 10-27 below. The system consists of two suction groups: booster and high stage (HS). The compressors in the booster suction group serve low temperature (LT) loads and discharge into the suction of the high stage suction group. The compressors in the high stage suction group serve the medium temperature (MT) loads, as well as compress the gas from the booster suction group and the intermediate pressure vessel to high pressures. Heat is rejected from the high pressure gas in the gas cooler when the system is operating in supercritical mode. The discharge pressure is commonly controlled by a hold back valve in combination with the gas cooler fans. When operating in subcritical mode the gas cooler operates as a condenser, analogous to other common refrigeration systems. The gas or liquid from the gas cooler/condenser expands in the intermediate pressure vessel/flash tank. The gas from the intermediate pressure vessel is compressed by the high stage compressors, and the liquid from the flash tank is supplied to medium temperature and low temperature evaporators (loads). The evaporated gas in the evaporators is compressed by its respective suction group compressors.

Figure 10-27: Common Transcritical CO2 System Configuration



The critical point of a substance is the point above which the liquid and vapor phases become indistinguishable from each other, forming a "supercritical fluid." In the supercritical region, temperature and pressure are semi-independent variables. CO_2 has a critical point of 87.8°F, which is considered to be a low critical point compared to all commonly used refrigerants. A pressure-enthalpy (PH) diagram for CO_2 with its critical point labeled is shown in Figure 10-28. In a transcritical CO_2 refrigeration system, the high-stage suction group can operate both above and below the critical point. When ambient temperatures are high, above approximately 75°F, the HS suction group compresses CO_2 above its critical point and the system is said to be in supercritical operation. An example of a HS suction group supercritical vapor compression cycle on the PH diagram for CO_2 is represented in green in Figure 10-28. During lower ambient conditions, when CO_2 is below its critical point after compression, the system is said to be in subcritical operation. An example of a HS suction group subcritical vapor compression cycle on the PH diagram for CO_2 is represented in pink in Figure 10-28.

Figure 10-28: Pressure-Enthalpy CO2 Diagram With Transcritical Vapor Compression Cycle (Diagram Created Using REFPROP – NIST Reference Fluid Properties)



In subcritical mode, the system operates very similarly to other refrigeration systems. In supercritical mode, the overall system efficiency decreases compared with subcritical operation. This is because high ambient temperatures result in higher compressor discharge temperatures needed for heat rejection, which increases the suction-to-discharge pressure ratio to be overcome by the compressor. Additionally, when operating in supercritical mode, the gas cooler outlet stream has a higher quality (higher vapor fraction) compared to subcritical mode. Vapor in the intermediate pressure vessel does not contribute to productive refrigeration but needs to be compressed, increasing the nonproductive refrigeration load on the compressors. Available technologies that increase supercritical operation efficiency include gas ejectors and parallel compression.

10.6.1.1 Transcritical CO_2 Gas Coolers

§120.6(b)5

New fan-powered gas coolers on all new transcritical CO_2 refrigeration systems must follow the gas cooler type, sizing, fan control, and efficiency requirements as described in §120.6(b)5.

10.5.6.1 Air-Cooled Gas Coolers Restrictions

§120.6(b)5A

Section 120.6(b)5A prohibits the use of air-cooled gas coolers in Climate Zones 10 through 15, which are high ambient temperature climate zones, to reduce the number of supercritical operating hours. Alternatives to air cooled gas coolers include water cooled gas coolers connected to a cooling tower, adiabatic gas coolers, and evaporative gas coolers.

10.5.6.2 Gas Cooler Sizing

Section 120.6(b)5B and §120.6(b)5C describe minimum sizing requirements for new gas coolers serving new transcritical CO_2 refrigeration systems. Fan-powered air-cooled gas coolers are covered by §120.6(b)5B. Fan-powered adiabatic gas coolers are covered by §120.6(b)5C.

Gas coolers must be sized to provide sufficient heat rejection capacity under design conditions while maintaining a specified maximum temperature difference between the gas cooler leaving gas temperature and ambient temperature. The design gas cooler capacity shall be greater than the calculated combined total heat of rejection (THR) of the dedicated compressors that are served by the gas cooler. If multiple gas coolers are specified, then the combined capacity of the installed gas coolers shall be greater than the calculated heat of rejection. When determining the design THR for this requirement, reserve or backup compressors may be excluded from the calculations. Example 10-42 provides an example scenario of which compressors to include in the THR calculation described in this section.

10.5.6.3 Air-Cooled Gas Cooler Sizing

§120.6(b)5B

Section 120.6(b)5B provides maximum design gas cooler leaving gas temperature (LGT) values for air-cooled gas coolers. For 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 gas cooler ratings to assume the capacity of air-cooled gas coolers is proportional to the temperature difference (TD) between the LGT and DBT, regardless of the actual ambient temperature entering the gas cooler. For example, the capacity of an air-cooled gas cooler operating at a leaving gas temperature of 80°F with a DBT of 70°F is assumed to be equal to the same unit operating at 110°F LGT and 100°F DBT, since the TD across the gas cooler is 10°F in both examples. Thus, the requirement for air-cooled gas coolers is based on the temperature difference between the DBT and gas cooler leaving gas temperature. Air cooled gas coolers shall be sized so the design leaving gas temperature for air-cooled gas coolers shall be less than or equal to the design dry bulb temperature plus 6°F.

10.5.6.4 Adiabatic Gas Cooler Sizing

§120.6(b)5C

Section 120.6(b)5C provides maximum design gas cooler leaving gas temperature (LGT) values for adiabatic gas coolers. For 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 gas cooler ratings to assume the capacity of adiabatic gas coolers is proportional to the temperature difference (TD) between the LGT and DBT for operation in dry mode, regardless of the actual ambient temperature entering the gas cooler. For example, the capacity of an adiabatic gas cooler operating at a LGT of 80°F with a DBT of 70°F is assumed to be equal to the same unit operating at 110°F LGT and 100°F DBT during dry mode operation, since the TD across the gas cooler is 10°F in both examples. Thus, similar to air-cooled gas coolers, the requirement for adiabatic gas coolers is based on the temperature difference between the DBT and gas cooler leaving gas temperature. Design leaving gas temperature for adiabatic gas coolers necessary to reject the design total heat of rejection of a refrigeration system assuming dry mode performance shall be less than or equal to the design dry bulb temperature plus 15°F.

10.5.6.5 Fan Control

Section 120.6(b)5D through §120.6(b)5G describe fan control requirements for new gas coolers serving new transcritical CO_2 refrigeration systems. Fan speed control requirements are covered by §120.6(b)5D. Gas cooler pressure control requirements during subcritical and supercritical operation are described by §120.6(b)5E and §120.6(b)5F, respectively. Minimum condensing temperature set point is covered by §120.6(b)5G.

10.5.6.6 Speed Control

§120.6(b)5D

Gas cooler fans for new air-cooled, evaporatively cooled or adiabatic gas coolers, or fans on cooling towers or fluid coolers used to reject heat on new transcritical CO_2 refrigeration systems, must use continuously variable speed. Variable-frequency drives are commonly used to provide continuously variable-speed control of gas cooler 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 gas cooler or set of gas coolers 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 reduce gas cooler capacity. As load increases, fans should be turned back on before significantly increasing fan speed, recognizing a control band is necessary to avoid excessive fan cycling.

10.5.6.7 Subcritical Pressure Control

§120.6(b)5E

Section 120.6(b)5E provides pressure control requirements for gas cooler operation below the critical point. These requirements are the same as for §120.6(b)1A with the exception that the minimum condensing temperature set point must be 60°F for transcritical CO_2 systems with a design intermediate saturated suction temperature lower than 30°F. See Section 10.4.2.1 for details.

10.5.6.8 Supercritical Pressure Control

§120.6(b)5F

During supercritical mode, the gas cooler pressure set point must be continuously reset in response to ambient conditions to optimize system efficiency, rather than using a fixed set point value.

Specifying the exact relationship to be used to determine the optimal head pressure may depend on several variables beyond ambient air temperature, including the operating saturated suction temperature, system configuration, gas cooler technology type, and current load. The controls manufacturer shall consider the tradeoff between fan energy and compressor energy in developing a pressure and fan control that is responsive to environmental and system conditions.

10.6.1.1.3.4. Minimum SCT Set point

§120.6(b)5G

The minimum saturated condensing temperature set point must be 60°F (16°C) or less for air-cooled gas coolers, evaporative-cooled gas coolers, adiabatic gas coolers, air or water-cooled fluid coolers or cooling towers. As a practical matter, a maximum condensing temperature set point is also commonly employed to set an upper bound on the control set point in the event of a sensor failure and to force full gas cooler operation during peak ambient conditions. This value should be set high enough that it does not interfere with normal operation.

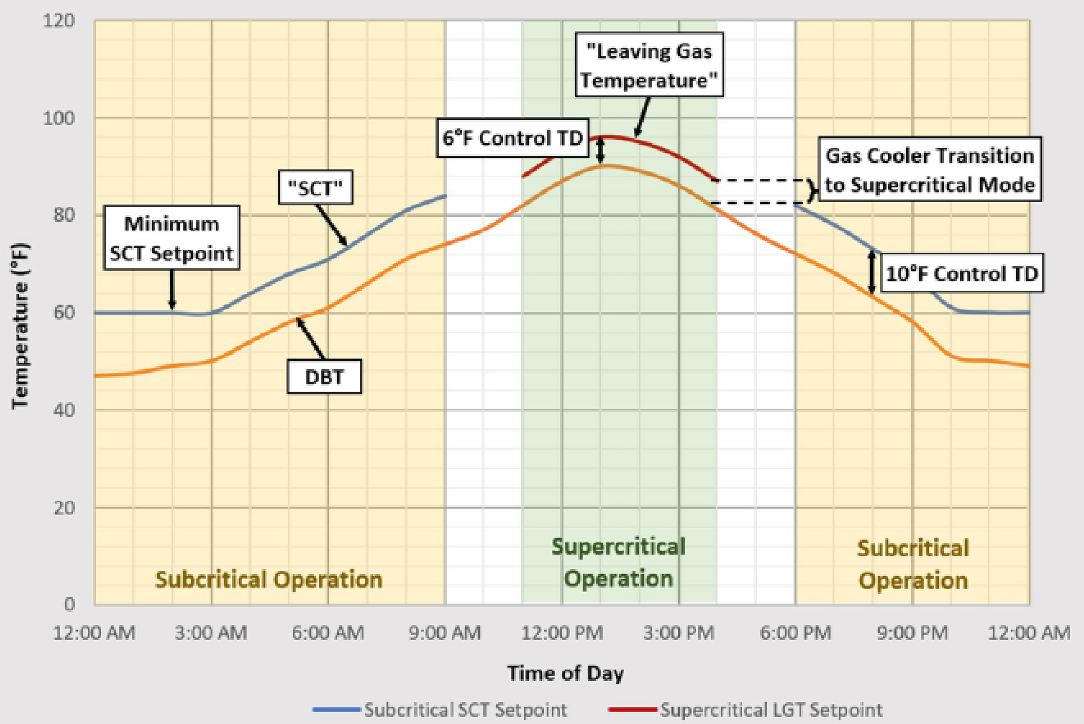
Example 10-27

Question

A commercial refrigeration facility with air-cooled gas coolers serving freezers is being commissioned. The control system designer has used a drybulb-following control strategy to reset the system saturated condensing temperature (SCT) setpoint when the system is operating subcritically, and has used a TD of 10°F (5.6°C) above the ambient drybulb temperature. The control system designer has used a proprietary gas cooler pressure control strategy to maximize system efficiency by minimizing the combined compressor and condenser fan power usage when the system is operating supercritically, with a TD of 6°F (3.3°C). What might the system DBT and SCT / gas cooler leaving gas temperature (LGT) setpoints trends look like over an example day?

Answer

The following figure illustrates what the actual SCT / LGT setpoint could be over an example day using the drybulb-following control strategy with a 10°F TD (5.6°C) when operating subcritically, a 6°F (3.3°C) TD when operating supercritically and observing the 60°F (16°C) minimum condensing temperature requirement. As the figure shows, the SCT setpoint is continuously reset to 10°F (5.6°C) above the ambient drybulb temperature until the minimum SCT setpoint of 60°F (16°C) is reached when operating subcritically and 6°F (3.3°C) above the ambient drybulb temperature when operating supercritically.



10.5.6.9 Gas Cooler-Specific Efficiency

§120.6(b)5H

Requirements for design leaving gas temperatures relative to design ambient temperatures, as described above for §120.6(b)5B and C, help assure that there is enough gas cooler capacity to keep leaving gas temperatures compressor head pressures at reasonable levels. However, the sizing requirements do not address gas cooler efficiency. For example, rather than providing amply sized gas cooler surface area, a gas cooler selection could consist of a small gas cooler area using a large motor to blow a large amount of air through the heat exchanger surface to achieve the design gas cooler TD. However, this would come at the expense of excessive fan motor horsepower. Also, relatively high fan power consumption can result from using gas cooler fans that have poor fan efficiency or low fan motor efficiency. Section 120.6(b)5H addresses these and other factors affecting gas cooler fan power by setting minimum specific efficiency requirements for gas coolers.

Table 10-6: Transcritical CO_2 Fan-Powered Gas Coolers — Minimum Specific Efficiency Requirements

Condenser Type	Refrigerant Type	Minimum Specific Efficiency	Rating Condition
Outdoor Air-Cooled	Transcritical CO_2	160 Btuh/Watt	1400 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature
Adiabatic Dry Mode	Transcritical CO_2	90 Btuh/Watt	1100 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature

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 outlet gas temperature and 90°F outdoor dry bulb temperature. Input power is the electric input power draw of the gas cooler fan motors (at full speed). 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 shall not be included in the specific efficiency calculation.

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

- Outdoor, air-cooled.
- Adiabatic (dry-mode operation).

The data published in the gas cooler manufacturer's published rating for capacity and power shall be used to calculate specific efficiency.

For air-cooled and adiabatic gas coolers, manufacturers typically provide the capacity at a given temperature difference (TDR) between SCT and dry bulb temperature. Manufacturers typically assume that gas cooler capacity is linearly proportional to TD; the catalog capacity at 20°F TD is typically twice as much as at 10°F TD. The gas cooler capacity for air-cooled gas cooler at a TD of 10°F shall be used to calculate specific efficiency. For adiabatic gas coolers, the dry mode capacity 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 gas cooler, the actual manufacturer's rated motor power may vary from motor nameplate in different ways. Air-cooled gas coolers with direct-drive OEM motors may use far greater input power than the nominal motor horsepower would indicate. Thus, actual motor input power from the manufacturer must be used for direct-drive, air-cooled gas coolers.

Example 10-48 provides an example calculation for the efficiency of a condenser, which is analogous to how the efficiency for a gas cooler would be calculated.

10.5.7 Additions and Alterations

§141.1(b)

The specific requirements for additions and alterations for commercial refrigeration are included in §120.6(b).

10.6 Refrigerated Warehouses

10.6.1 Overview

This section of the manual focuses on the Energy Code provisions unique to refrigerated warehouses. The Energy Code 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 Code must also comply with the general provisions of the Energy Code (§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)

Section 120.6(a) of the Energy Code 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 between 28°F (-2°C) and 55°F (13°C). Freezers are defined as refrigerated spaces designed to operate below 28°F (-2°C). The Energy Code does not address walk-in coolers and freezers, defined as refrigerated spaces less than 3,000 ft^2, as these are covered by the Appliance Efficiency Regulations (Title 20). However, refrigerated warehouses and spaces with a total of 3,000 ft^2 or more and served by a common refrigeration system are covered by the Energy Code and required to comply with §120.6(a).

Furthermore, areas within refrigerated warehouses designed solely for quick chilling or quick freezing of products have some exceptions for evaporators and compressors. Quick chilling and freezing spaces are defined as spaces with a design refrigeration evaporator load of greater than 240 Btu/hr-ft^2 of floor space, which is equivalent to 2 tons per 100 ft^2 of floor space. A space used for quick chilling or freezing and used for refrigerated storage must still meet the requirements of §120.6(a).

The intent of the Energy Code 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 Code to apply the compressor plant requirements to an industrial process that is not covered by the Energy Code simply because a small storage space is also attached to the suction group. Similarly, it would be outside the intent of the Energy Code to exclude a compressor plant connected to a suction group serving a large storage space covered by the Energy Code on the basis of a small process cooler or quick chill space also connected to the same suction group. For compliance, the compressor plant requirements in §120.6(a)5B 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 these compressors share a common condensing loop, it is impossible to address only the equipment serving refrigerated storage spaces. For compliance, the provisions addressing condensers, subsections 120.6(a)4A and 4B, 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 Code covers expansions and modifications to an existing facility and an existing refrigeration plant. The Energy Code does not require that all existing equipment must 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

Section 120.1(a)1 of the Energy Code, concerning ventilation requirements, does not apply to "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 greater than or equal to 3,000 ft,^{2} where mechanical refrigeration is 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 that may require ventilation for one or more reasons. Accordingly, while the Energy Code does 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 10-28

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-ft^2 at the applied conditions. Does the space have to comply with the space requirements in §120.1(a) of the Energy Codes?

Answer:

Yes. If the warehouse is 3,000 ft^2 or larger or served by a refrigeration system serving 3,000 ft^2 or more, it must meet all the requirements in subsections 1,2, 6, and 7. It also must meet the requirements of subsections 3A, 4C, 4D, 4E, 4F, 4G, 5A, and 5C. There are exceptions for 3B, 3C, 4A, 4B, and 5B.

Example 10-29

Question:

A refrigerated warehouse space is used to cool and store melons received from the field. After the product temperature is lowered, 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 all the space requirements of §120.1(a) of the Energy Code?

Answer:

Yes. While the design evaporator capacity is greater than 240 Btu/hr-ft^2 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 of §120.1(a) of the Energy Code.

Comment: This measure 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 Chapter 10.

Example 10-30

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 all the requirements of §120.1(a) of the Energy Code?

Answer

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

Example 10-31

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 this standard?

Answer:

Since the suction group serves a total 4,000 ft² of refrigerated floor area, the spaces must meet all the requirements of §120.6(a).

10.6.2 Building Envelope Mandatory Requirements

Section 120.6(a) subsections 1, 2, and 6 of the Energy Code addresses 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 that 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 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
Freezers	Wall	R-36
Freezers	Floor	R-35
Freezers	Floor with all heating from productive refrigeration capacity	R-20
Coolers	Roof/Ceiling	R-28
Coolers	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 10-32

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 10-33

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?

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.

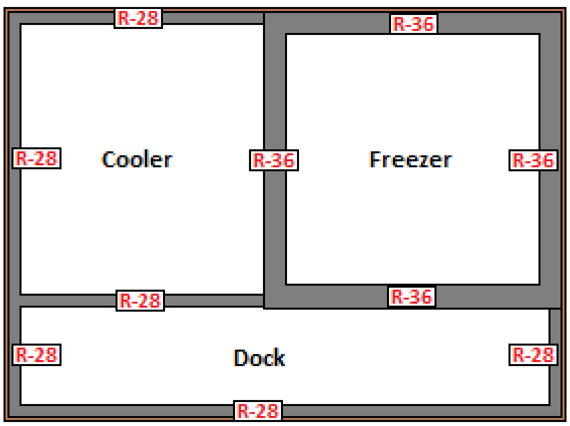
Example 10-34

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.



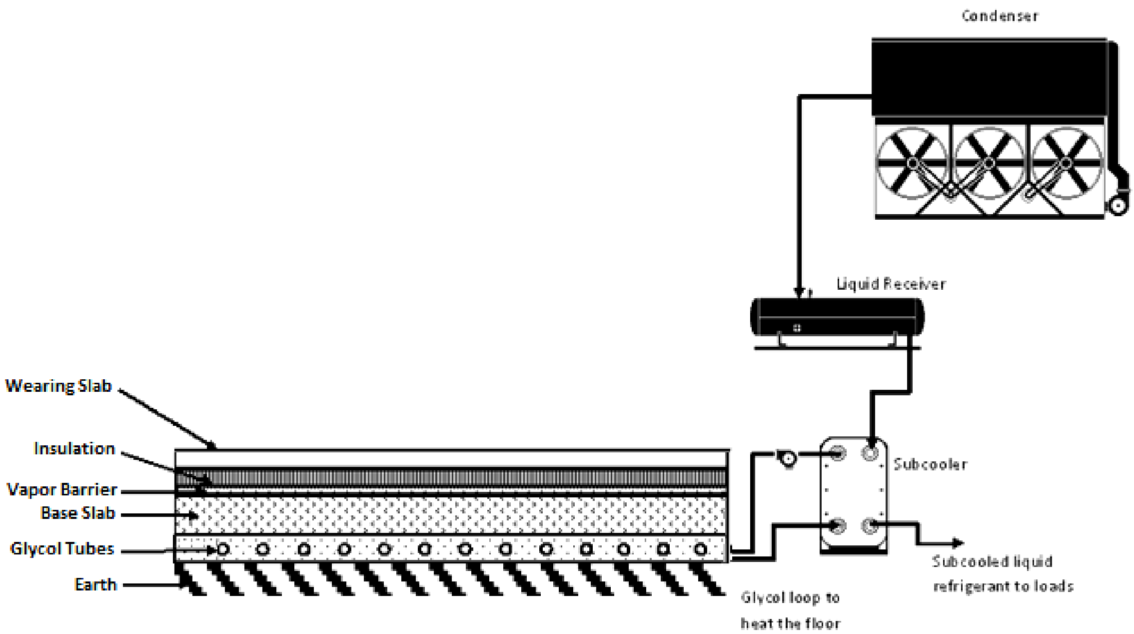
B. Freezer Floor Insulation

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 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 the underslab heat is provided by an underslab heating system that increases productive refrigeration capacity. An example of an underslab heating system using heat from a refrigerant liquid subcooler is shown in Figure 10-27.

Figure 10-27: Underslab Heating System That Uses 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 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 using 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 12 p.m. to 6 p.m. weekdays from 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 nonrefrigerated spaces, shall have an infiltration barrier such as:

h. Strip curtains.

i. An automatically closing door.

j. Air curtain.

Examples of each are shown in the figures below.

Figure 10-28: Strip Curtains

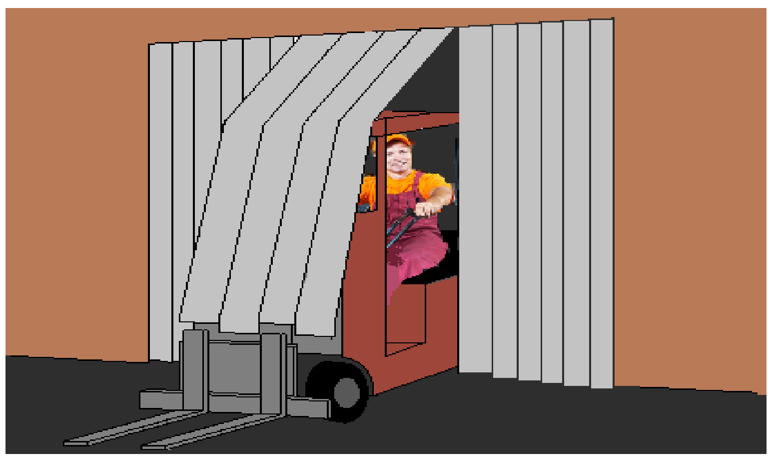


Figure 10-29: Biparting Automatic Door

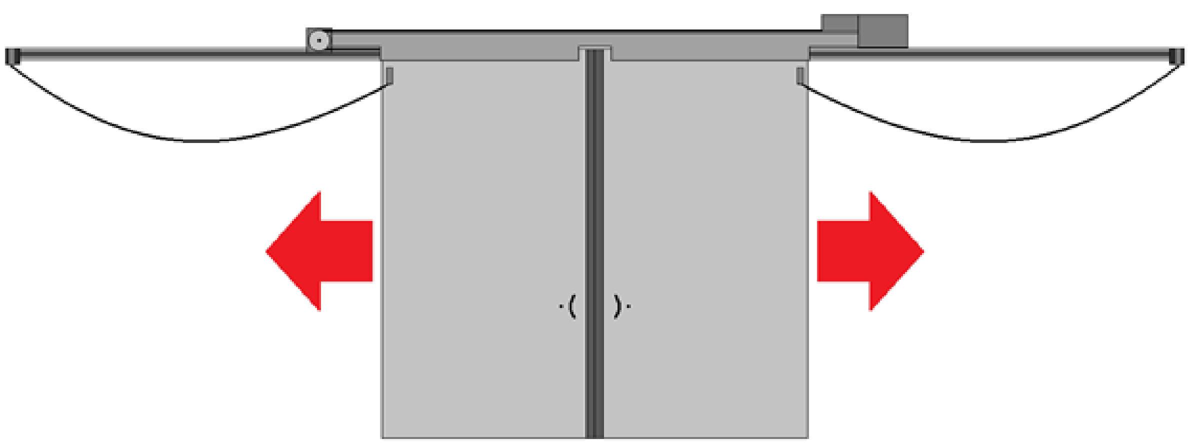


Figure 10--30: Hinged Door with Spring-Action Door Closer and Door "Tight" Closer

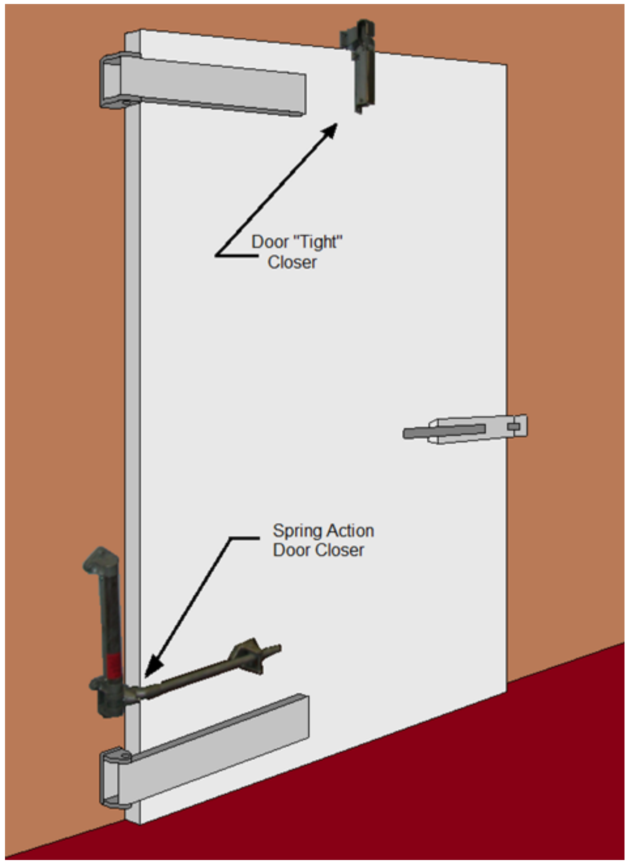
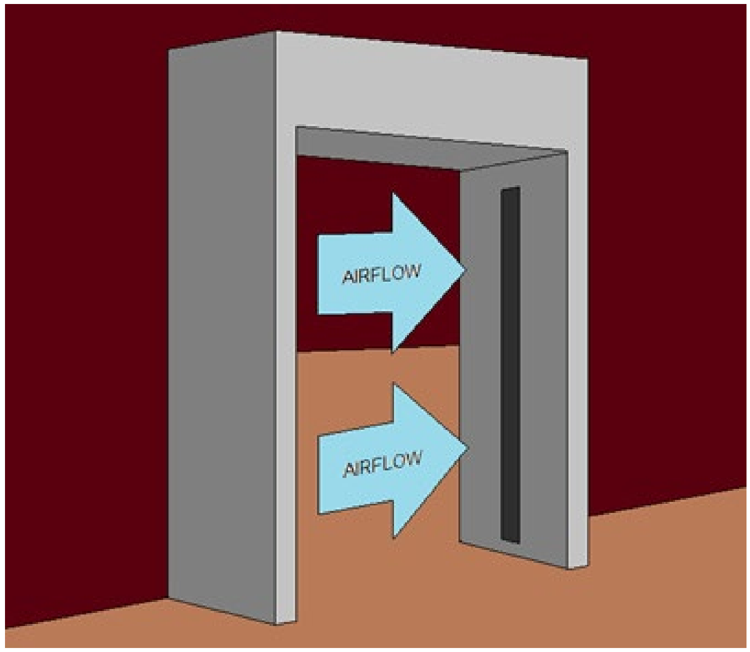


Figure 10-31: Air Curtain



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 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:

a. Single-acting or double-acting hinge-mounted doors with a spring assembly or cam-type gravity hinges.

b. Powered single sliding, biparting, or rollup doors that open based on a pull cord, proximity or similar sensors, or by operator signal and close automatically through similar actions or after a period sufficient to allow passageway transit.

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

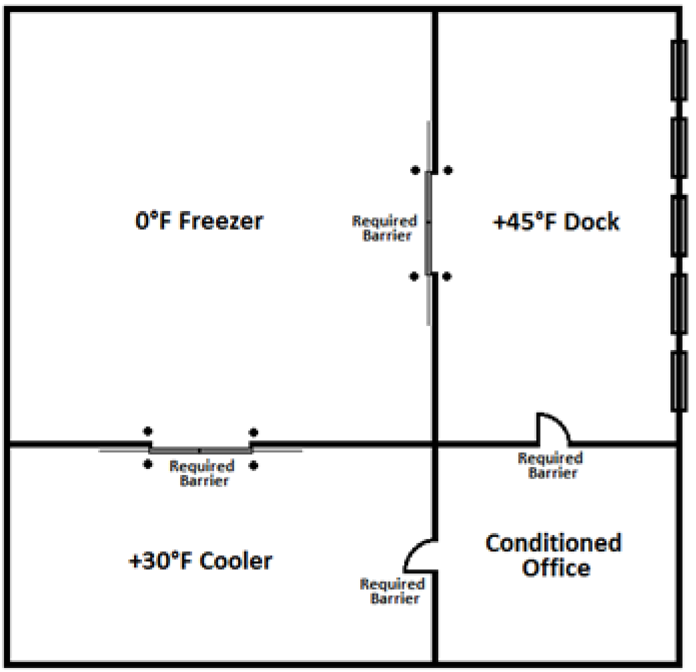
There are two exceptions to the requirements for infiltration barriers:

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

Example 10-35

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 barriers 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 10-36

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 doors, or rollup doors with automatic door closers.

10.6.2.4 Automatic Door Closers

§120.6(a)9

Infiltration through doors in refrigerated warehouses account for significant amounts of wasted energy.

The Energy Code requires that doors designed for the passage of people that are between freezers and higher-temperature spaces, or coolers and nonrefrigerated spaces, have automatic door closers.

Automatic door closers are typically hinges that closes a door from any open position and mechanisms that closes a door completely shut, if slightly ajar (from approximately 1 inch opened).

10.6.2.5 Acceptance Requirements

§120.6(a)7

The Energy Code includes 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 Code includes 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

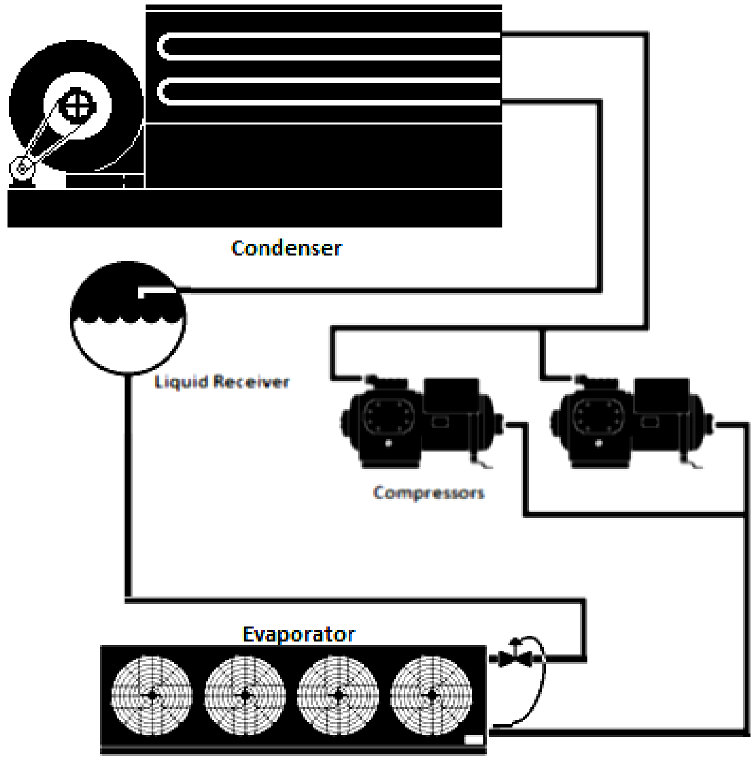


Figure 10--33: Single-Stage System with Flooded Evaporator Coil

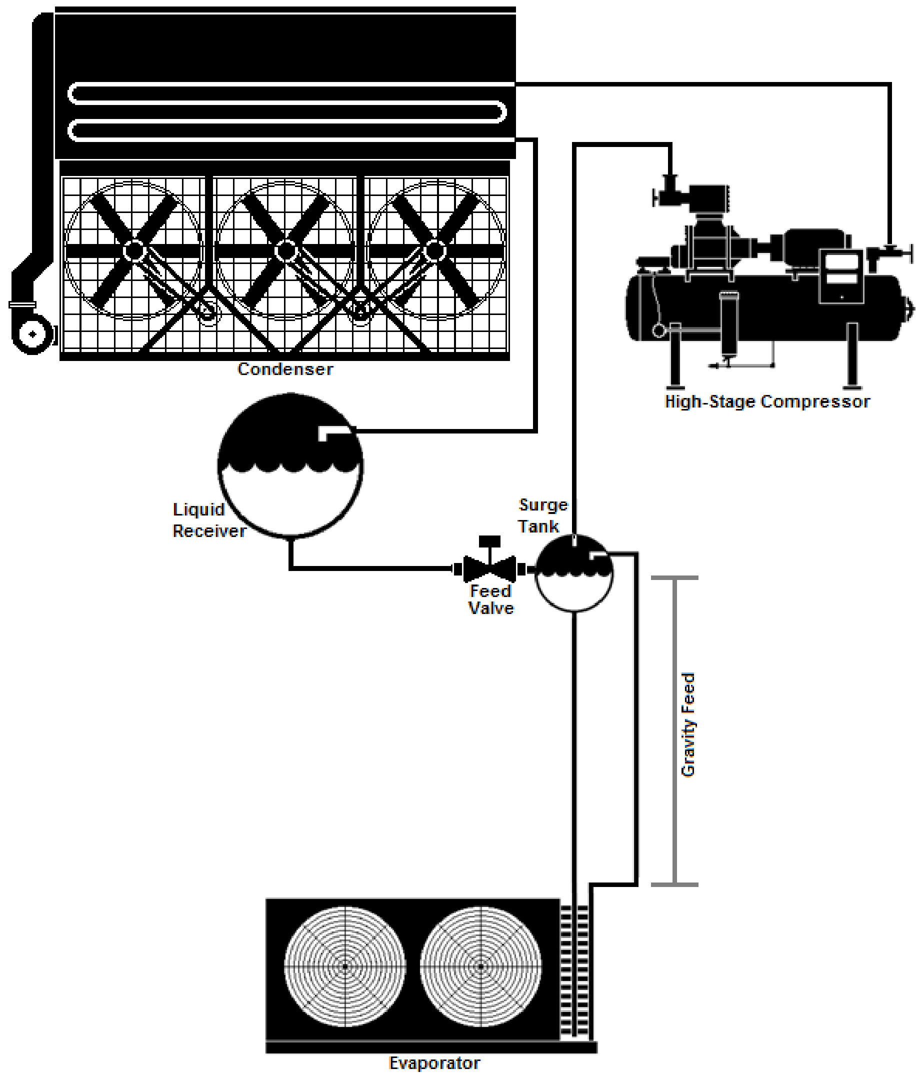


Figure 10-34: Single-Stage System with Pump Recirculated Evaporator Coils

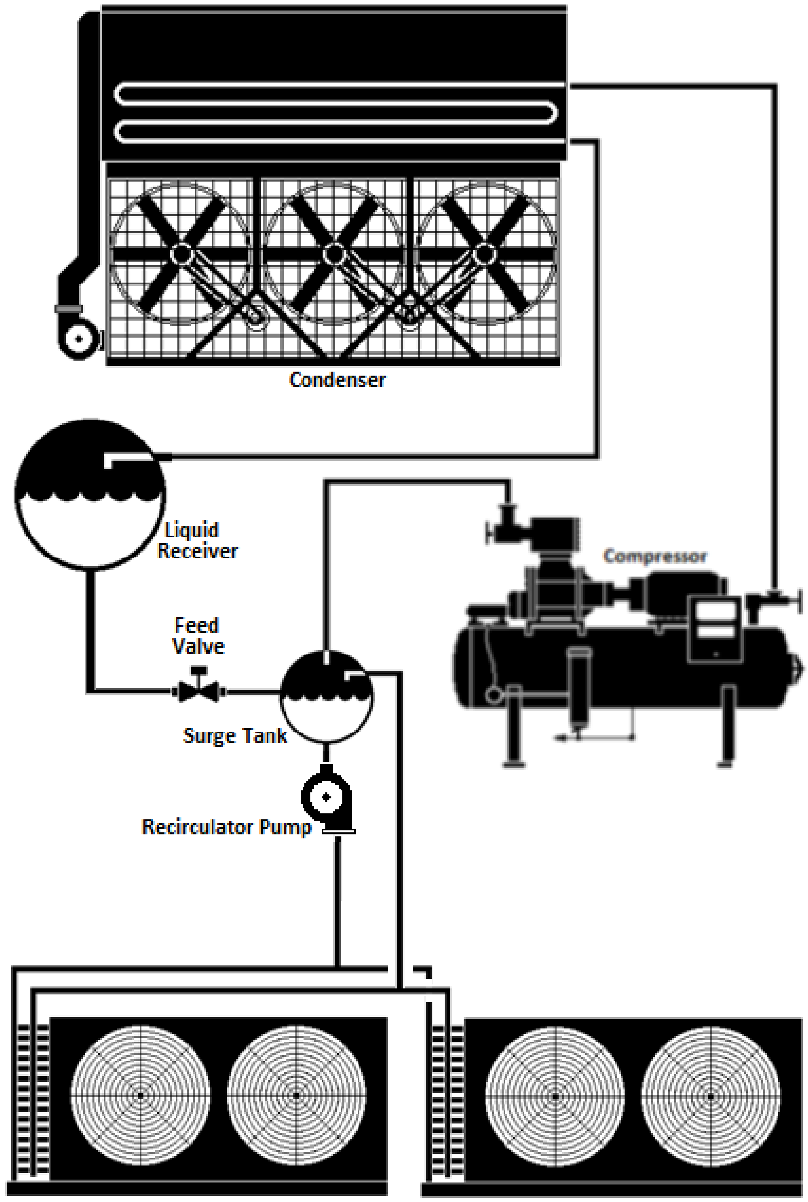


Figure 10-35: Two-Stage System with Flooded Evaporator Coil

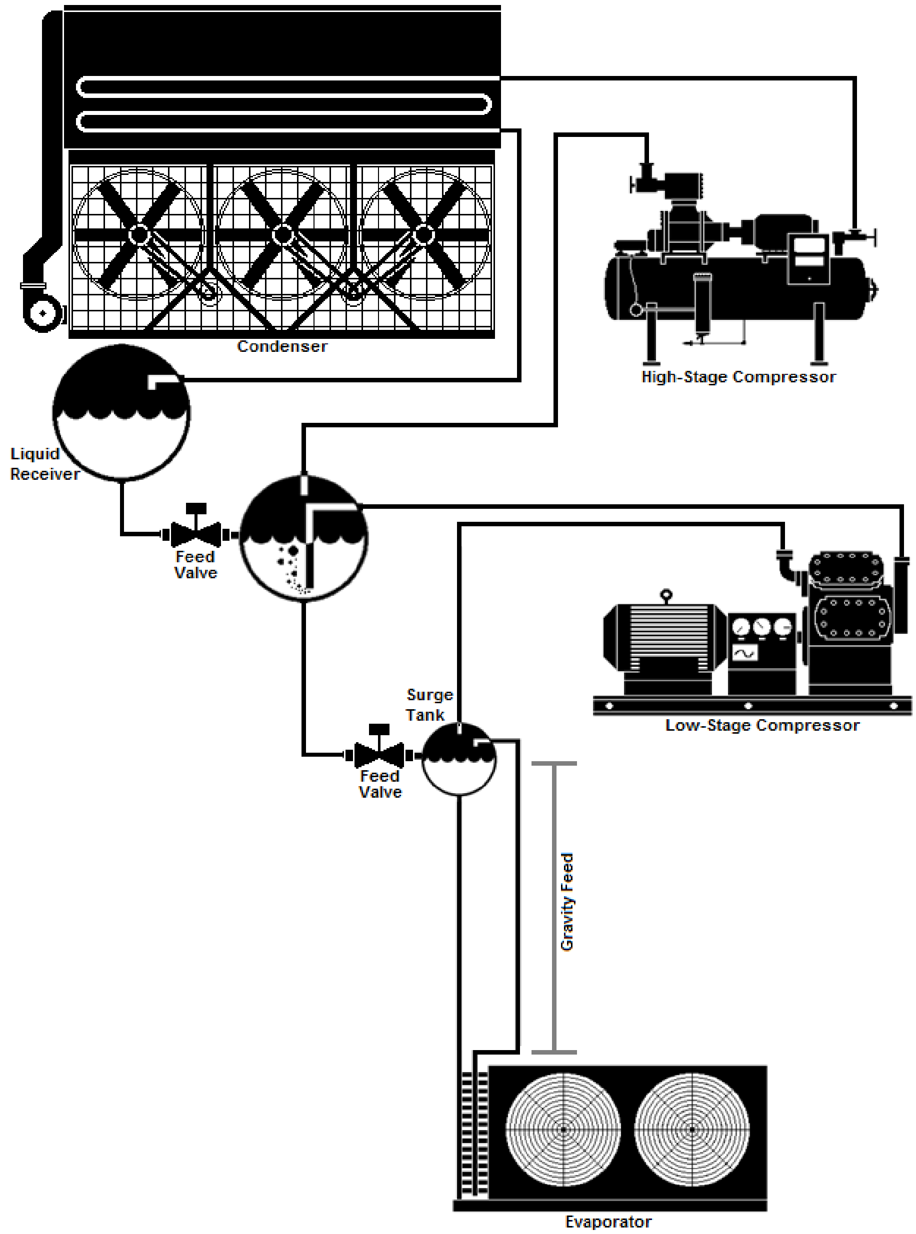
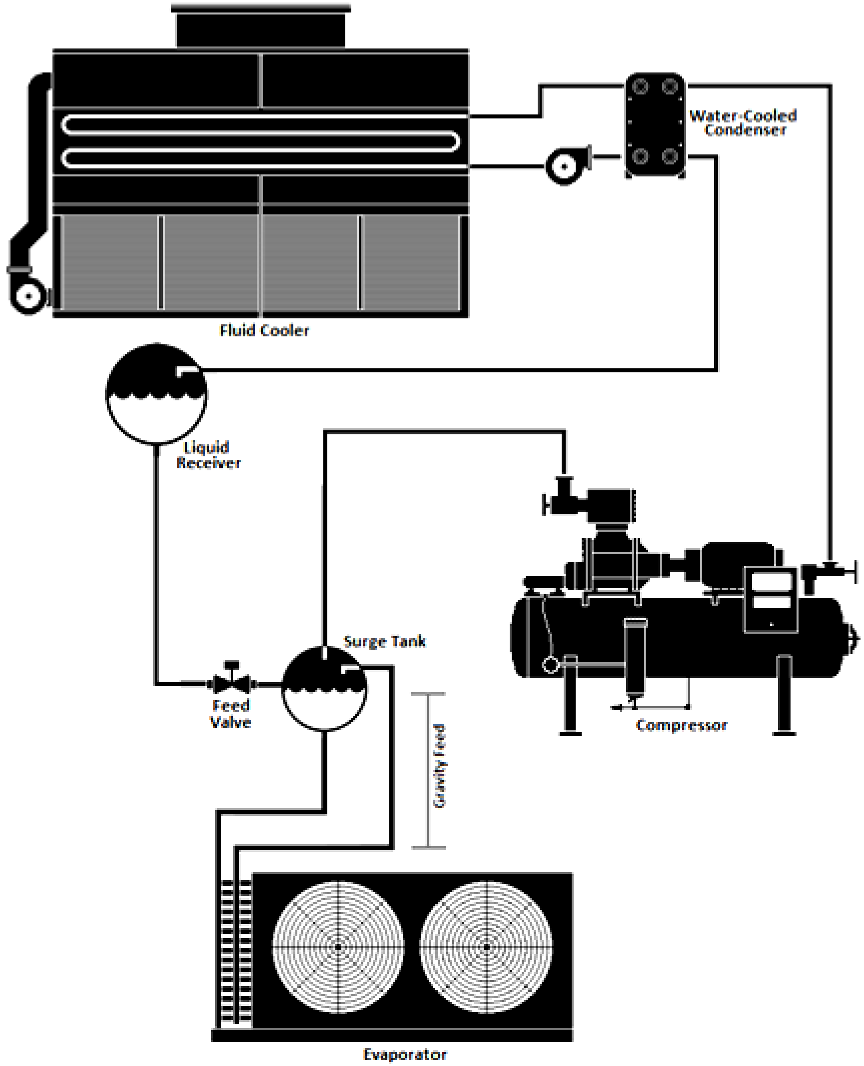


Figure 10-36: Single-Stage System with Water-Cooled Condenser Served by Fluid Cooler



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 Code.

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 Direct Current (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.

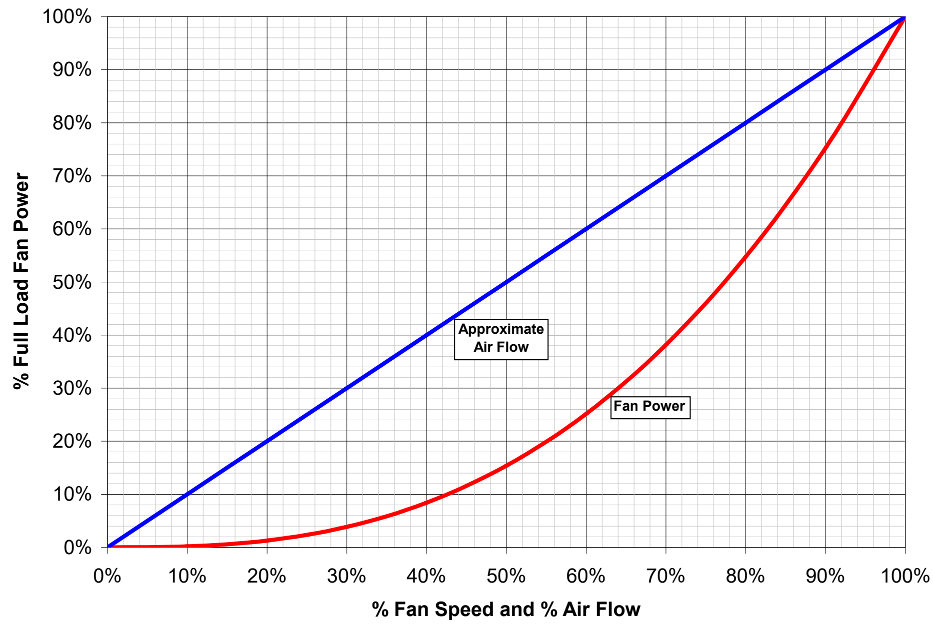
The fan speed is controlled in response to space temperature or humidity. Fan speed should increase proportionally when the space temperature is above the set point and decrease when the space temperature is at or below the set point, with refrigerant supply and pressure being maintained in the evaporator 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 Code to accommodate special strategies for humidity-sensitive perishable product. Control logic in these applications will often 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 roughly 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 about 51 percent ($80\%^3 = 51\%$) power while providing nearly 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 Code for the minimum speed setting (i.e., how low the fan speed must go at minimum load). Variable-speed controls of evaporator fans have 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 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 set point, suction pressure should be allowed only to float up after fan speeds are at minimum and should be controlled to float back down prior to increasing fan speeds.

The Energy Code has three exceptions to the evaporator variable speed requirement:

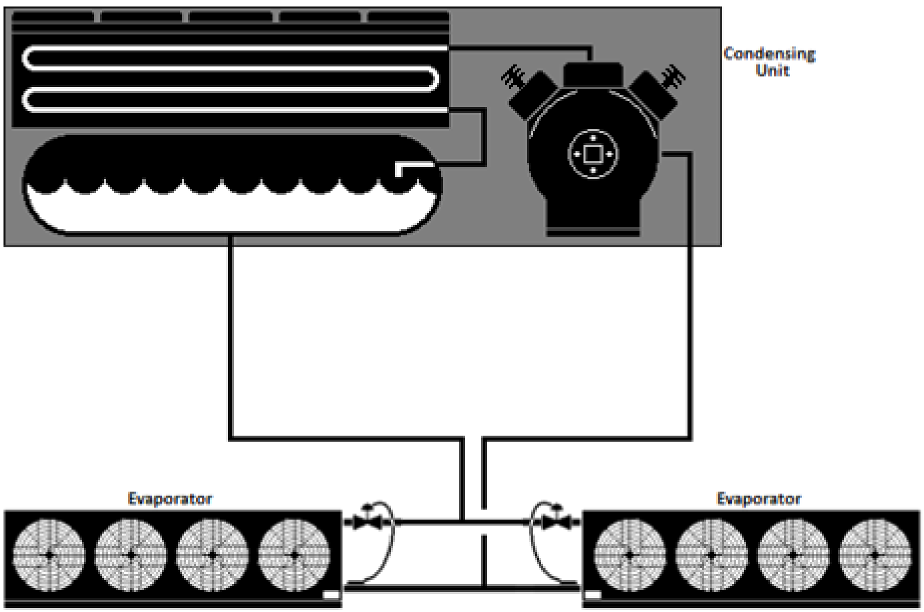
1. In case of a replacement, addition, or alteration of existing evaporators with no variable-speed control, the variable-speed control of the evaporators is mandatory only if the replacement, addition, or alteration is done for all the evaporators in an existing space. *[Exception 1 to §120.6(a)3B]*
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 10-37

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?



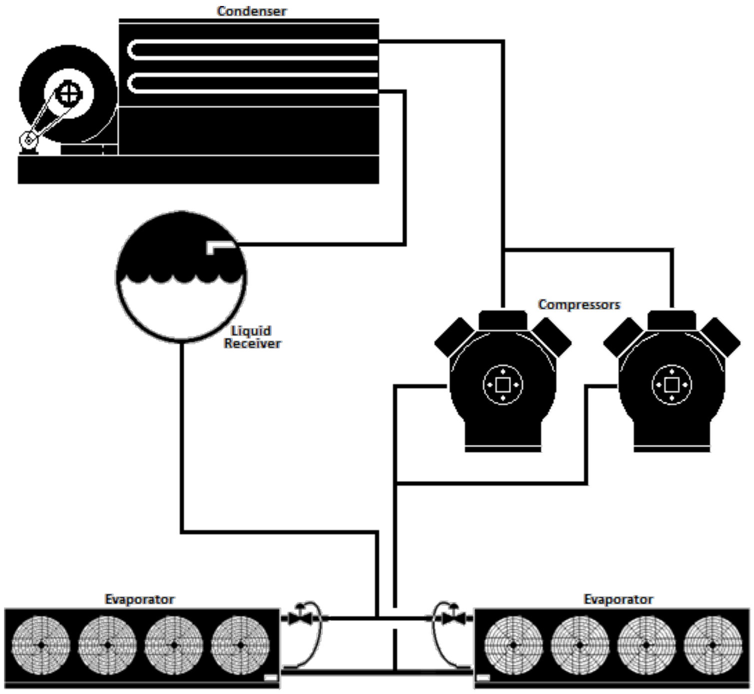
Answer:

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 10-38

Question:

A refrigeration system uses 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 proportional controls for 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 Code.

Example 10-39

Question:

A -20°F (-29°C) freezer has several 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 use variable-speed drives, and the control system varies the fan speed in response to space temperature. What should the freezer saturated suction temperature be 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 set point would be -32°F (-36°C).

This example sought to show how evaporator temperature and coil capacity can be considered and maintained to achieve proper variable-speed fan operation and savings. Settings could be 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 10-40

Question:

An existing refrigerated warehouse space has eight 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 use a consistent control scheme that could require a disproportional cost. While not required by the Energy Code, 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.

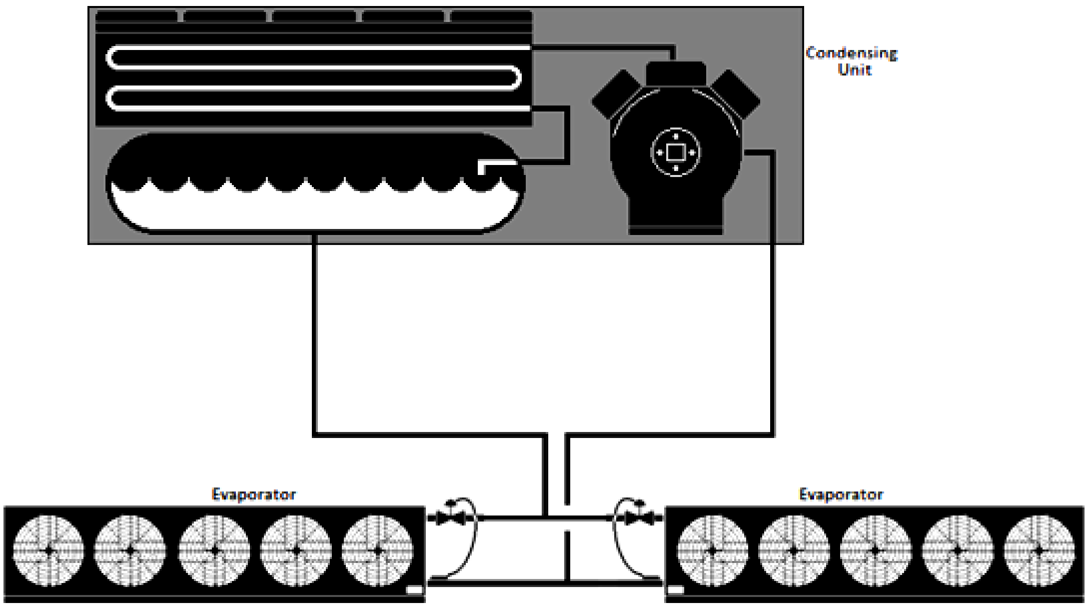
Continuously variable-speed control is not mandatory for evaporators that are served by a single compressor that does not have variable-capacity capability (i.e., the compressor cycles on and off in response to space temperature). For these systems, evaporator airflow must be reduced by at least 40 percent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

- Two-speed evaporator fan control, with speed reduced by at least 40 percent when cooling is satisfied and the compressor is off.
- Turning off a portion of the fans in each evaporator to accomplish at least 40 percent 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 percent of the duty cycle while the compressor is off.

Example 10-41

Question:

A split system with a packaged air-cooled condensing unit using 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 use controls that cycle at least 4 of the 10 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. Fan-powered adiabatic condensers are covered by §120.6(a)4C.

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

Another exception to §120.6(a)4B, for air-cooled condenser sizing, applies if a condensing unit has a total compressor power less than 100 hp. A condensing unit includes compressor(s), condenser, liquid receiver, and control electronics that are packaged in a single product.

Example 10-42

Question:

A new food processing plant is being constructed that will include an 800 ft^2 blast freezer, a holding freezer, and a loading dock. The design evaporator capacity of the blast freezer is 40 tons of refrigeration (TR). 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?

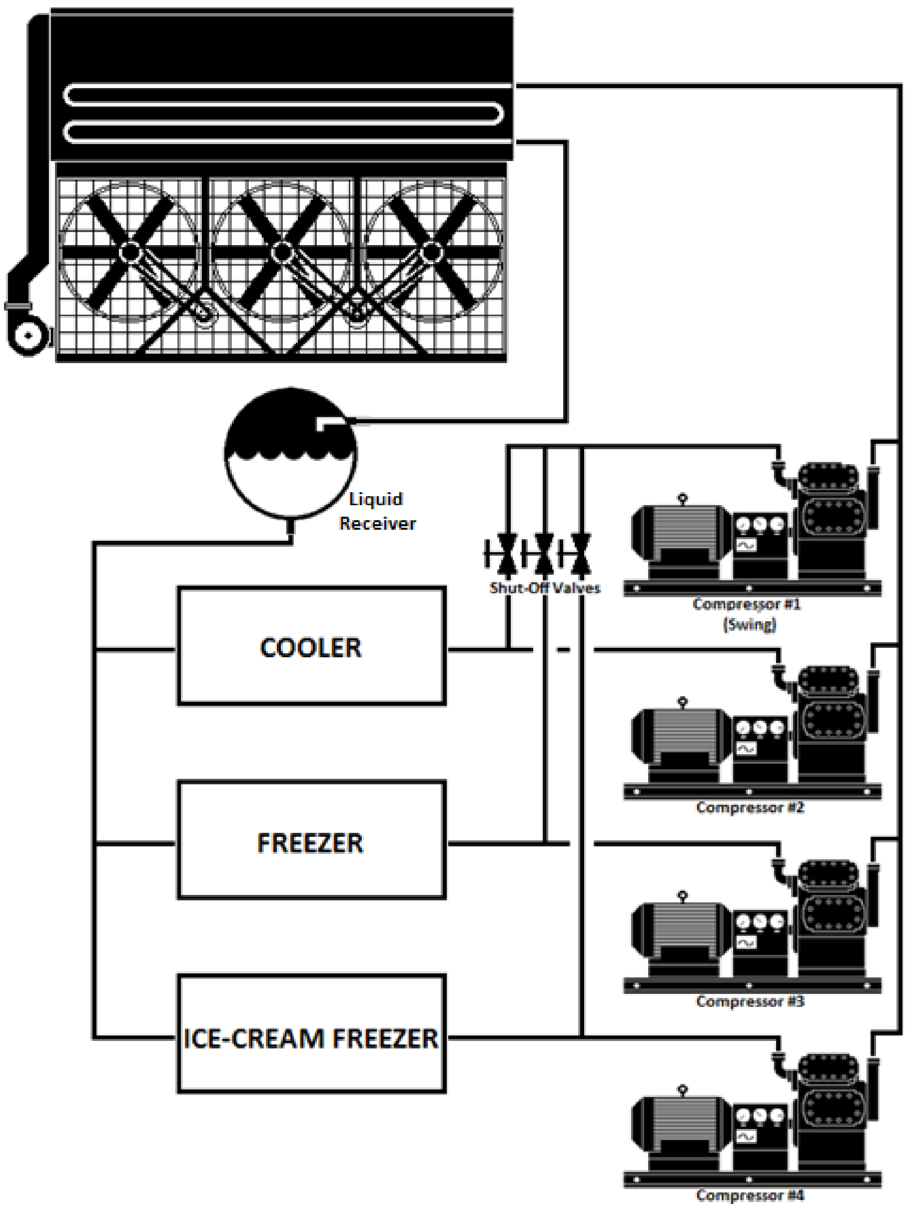
Answer:

The blast freezer evaporator capacity divided by the floor area is 40 TR/800 ft², which is equal to 5 TR/100 ft². That means this particular blast freezer is deemed quick freezing space by the Energy Code. Therefore, the condenser group serving the refrigeration system does not have to comply with §120.6(a)4A, because 40% (i.e., greater than 20%) of the design refrigeration capacity is from quick freezing.

Example 10-43

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 backup 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 backup for other compressors. If the compressor is solely for use as backup, it would be excluded from the heat rejection calculation for the purposes of the Energy Code. 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 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)
³ 78°F (26°C)	Design WBT plus 18°F (10°C)

Example 10-44

Question:

A refrigerated warehouse is being constructed in Fresno. 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 wet bulb temperature (WBT) from Joint Appendix JA-2 for Fresno 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 10-45

Question:

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

Located in Fresno

Design SST: 10°F

Suction group: Three equal-sized dedicated 100 hp screw compressors (none are backup units)

Evaporative condenser

240 TR cooling load

Answer:

From the previous example, it was determined that the design wet bulb temperature (WBT) to demonstrate compliance for Fresno 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 THR for the suction group at the design conditions of 10°F SST and 95°F SCT. Each selected compressor has a rated capacity of 240 TR 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 = 9,403,500 Btuh

To comply with the Energy Code, a condenser (or group of condensers) must be selected that is capable of rejecting at least 9,403,500 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 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 Code has 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 the 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 10-46

Question:

An air-cooled condenser is being sized for a system that has half of the associated 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:

This measure specifies a design approach of 15°F (8.3°C) for coolers and 10°F (5.6°C) for freezers. 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 percentage of the total installed capacity dedicated to coolers by 15°F (8.3°C). Next, multiply the percentage 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

3. Adiabatic Condenser Sizing

§120.6(a)4C

§120.6(a)4C provides maximum design SCT values for adiabatic condensers. These requirements are the same as for §120.6(b)1E. See section 10.5.2.3 for details.

B. Fan Control

Condenser fans for new air-cooled, evaporative, or adiabatic condensers, or fans on cooling towers or fluid coolers used to reject heat on new refrigeration systems, must use 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 reduce condenser capacity. As load increases, fans should be turned back on before significantly increasing fan speed, recognizing a control band is necessary to avoid excessive fan cycling.

To minimize overall system energy consumption, the condensing temperature set point must be continuously reset in response to ambient temperatures, rather than using a fixed set point value. This strategy is also termed ambient-following control, ambient-reset, wet bulb following and dry bulb following—all referring to the control logic that 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, the target SCT for air-cooled condensers must be reset according to ambient dry bulb temperature, and the target SCT for adiabatic condensers when operating in dry mode must be reset according to ambient dry bulb temperature. There is no requirement for SCT control during wet bulb (adiabatic) operation. This requirement for the adiabatic condenser is applicable to all systems and is independent of the type of refrigerant used

The condenser control TD is not specified in the Energy Code. 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 use as much condenser capacity as possible without excessive fan power, a common practice for refrigerated warehouse systems is to optimize the control TD over a period such that the fan speed is between approximately 60 and 80% during normal operation (i.e., when not at minimum SCT). While not required, evaporative condensers and systems using 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 wet bulb temperatures.

The minimum saturated condensing temperature set point must be 70°F (21°C) or less. For systems using halocarbon refrigerants with glide, the SCT set point 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 set point is also commonly employed to set an upper bound on the control set point 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 use 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.

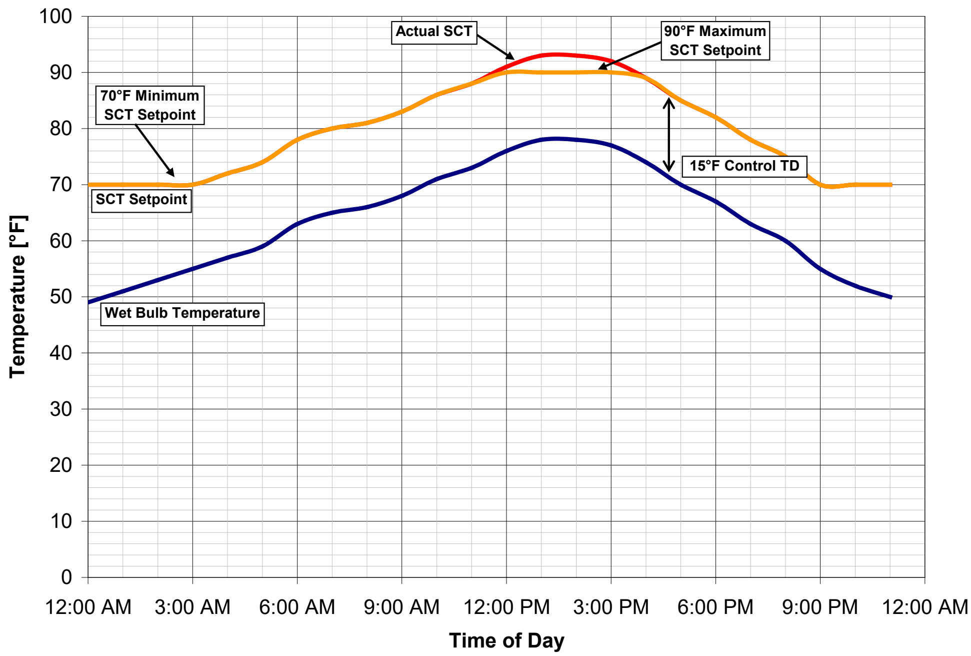
Example 10-47

Question

A refrigerated warehouse with evaporative condensers is being commissioned. The control system designer has used a wet bulb-following control strategy to reset the system saturated condensing temperature (SCT) set point. 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 set point that minimizes the combined compressor and condenser fan power usage throughout the year. What might the system SCT and SCT set point trends look like over an example day?

Answer

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



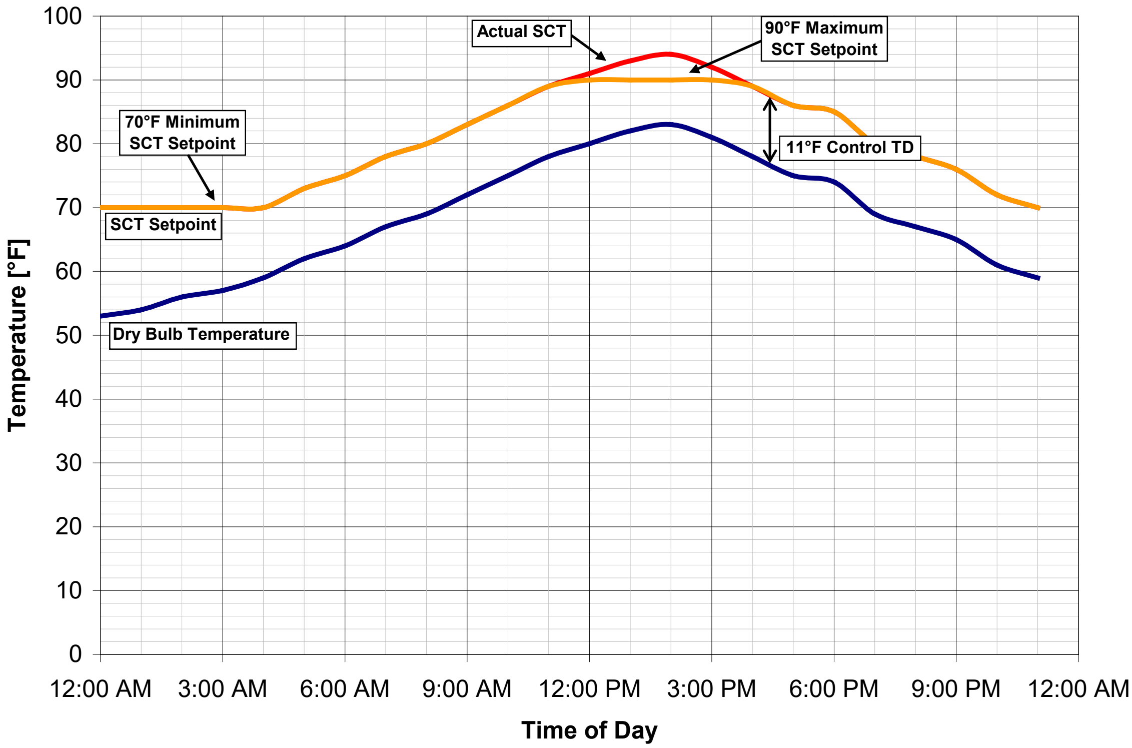
Example 10-48

Question:

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

Answer:

The following figure illustrates the actual saturated condensing temperature and SCT set points over an example day using the dry bulb-following control strategy with an 11°F (6.1°C) TD and observing the 70°F (21°C) minimum condensing temperature requirement. As the figure shows, the SCT set point is continuously reset 11°F (6.1°C) above the ambient dry bulb temperature but is bounded by the minimum and maximum SCT set points. The figure also shows a maximum SCT set point (in this example, 90°F (32.2°C) that may be used to limit the maximum control set point, 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, and C, 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 and adiabatic 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

CONDENSER TYPE	REFRIGERANT TYPE	MINIMUM SPECIFIC EFFICIENCY*	RATING CONDITION
Outdoor Evaporative Cooled with THR Capacity > 8,000 MBH	All	350 Btuh/W	100°F Saturated Condensing Temperature (SCT), 70°F Outdoor Wet bulb Temperature
Outdoor Evaporative-Cooled with THR Capacity < 8,000 MBH and Indoor Evaporative-Cooled	All	160 Btuh/W	100°F Saturated Condensing Temperature (SCT), 70°F Outdoor Wet bulb Temperature
Outdoor Air-Cooled	Ammonia	75 Btuh/W	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Dry bulb Temperature
Outdoor Air-Cooled	Halocarbon	65 Btuh/W	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Dry bulb Temperature
Adiabatic Dry Mode	Halocarbon	45 Btuh/W	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Dry bulb Temperature

Indoor Air-Cooled	All	Exempt	Exempt

Source: California Energy Commission

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 wet bulb temperature for evaporative condensers, and 105°F SCT and 95°F outdoor dry bulb 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 Code has different minimum efficiencies depending on the type of condenser that is being used. The different classifications of condenser are:

a. Outdoor, evaporative, THR greater than 8,000 MBH at specific efficiency rating conditions.
b. Outdoor, evaporative, THR less than 8,000 MBH at specific efficiency rating conditions.
c. Indoor, evaporatively cooled.
d. Outdoor, air-cooled, ammonia refrigerant.
e. Outdoor, air-cooled, halocarbon refrigerant.
f. Adiabatic (dry-mode operation), halocarbon refrigerant.
g. 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 wet bulb 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 determining compliance with this section.

For air-cooled and adiabatic condensers, manufacturers typically provide the capacity at a given temperature difference (TD) between SCT and dry bulb temperature. Manufacturers typically assume that 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. For adiabatic condensers, the dry mode capacity 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 (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 determine specific efficiency more accurately.

Example 10-49

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 four-pole open fan motor, the minimum efficiency is 91.0%. For a 5 HP six-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

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

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 use a microchannel 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.

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 affects 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 and §120.6(a)5B address the compatibility of the compressor design and components with the requirements for floating head control for compressors serving refrigeration systems other than transcritical CO_2 systems and compressors serving transcritical CO_2 systems respectively.

A. Compressors Serving Systems Other Than Transcritical CO_2

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. 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. Compressors Serving Transcritical CO_2 Systems

All compressors that discharge to the gas cooler(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 60°F (16°C) or less. 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.

An exception to §120.6(a)5B is provided for compressors that are designed to operate at a saturated suction temperature equal to or greater than 30°F. In this case, the compressors can be designed to operate at a minimum condensing temperature of 70°F or less.

C. 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 percent 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.

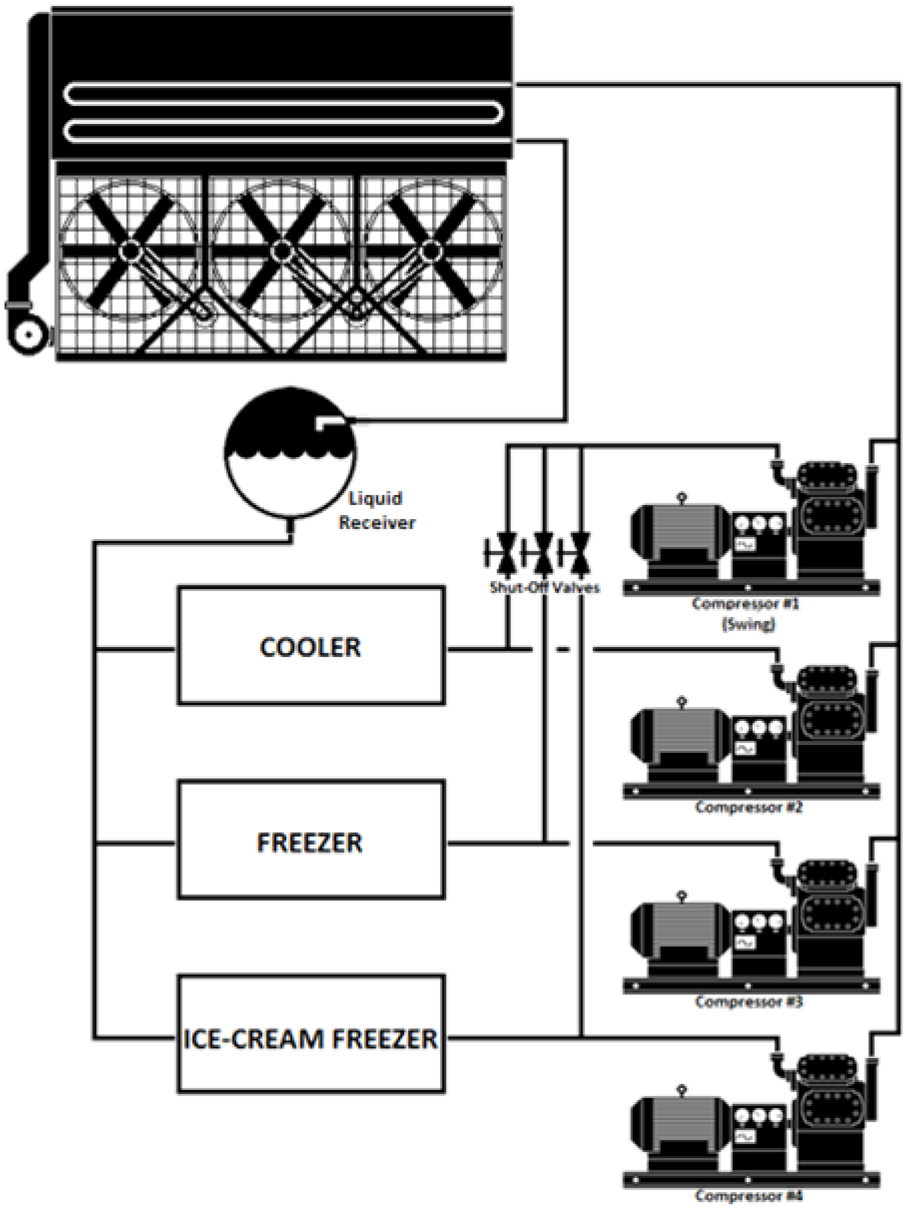
An exception to §120.6(a)5B (controlling compressor speed in response to refrigeration load) is provided for compressors on a refrigeration system with more than 20 percent of the design cooling load from quick chilling or freezing space, or nonspace process refrigeration cooling. The "refrigeration 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 150 HP shall incorporate the capability to automatically vary the volume ratio (i.e., variable Vi) to optimize efficiency at off-design operating conditions.

Example 10-50

Question

The system shown below has three 200 HP open-drive screw compressors serving three 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. Exception 1 to §120.6(a)5B applies only 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 backup 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 Codes have acceptance test requirements for:

- Electric underslab heating controls.
- Evaporator fan motor controls.
- Evaporative condensers.
- Adiabatic condensers
- Air-cooled condensers.
- Variable-speed compressors.
- Transcritical CO_2 refrigeration systems.

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 set point 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 load so that the fan average 60-80% speed when in the control range (i.e., between the minimum and maximum SCT set points).

D. Air-Cooled Condensers

NA7.10.3.2

Air-cooled condensers and variable-speed fan controls are checked to ensure the required minimum SCT set point 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 energy savings. This control setting should be adjusted during average load so that condenser capacity is effectively used but fan speed is not excessive.

E. Adiabatic Condensers

NA7.10.3.3

Adiabatic condensers and variable-speed fan controls are checked to ensure the required minimum SCT set point 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 when operating in dry mode. 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 used but fan speed is not excessive.

F. Variable-Speed Compressors

NA7.10.4

The controls and equipment for the variable-speed control of screw compressors are checked and certified as part of the acceptance requirements. The compressor should unload capacity by reducing speed to the minimum speed set point 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.

10.6.3.1 G. Transcritical CO_2 Systems

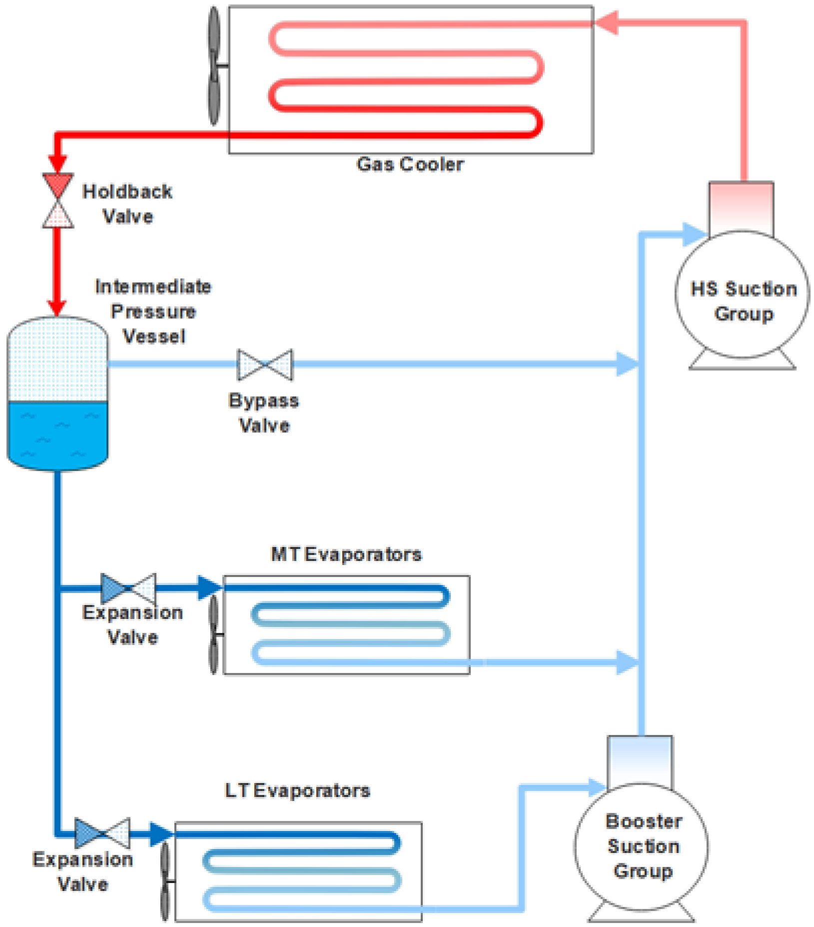
NA7.20.1

The controls and equipment of transcritical CO2 refrigeration systems are checked to verify correct operation of gas cooler controls, including testing of subcritical and supercritical operating modes.

10.6.3.2 10.6.3.2 Transcritical CO_2 Systems

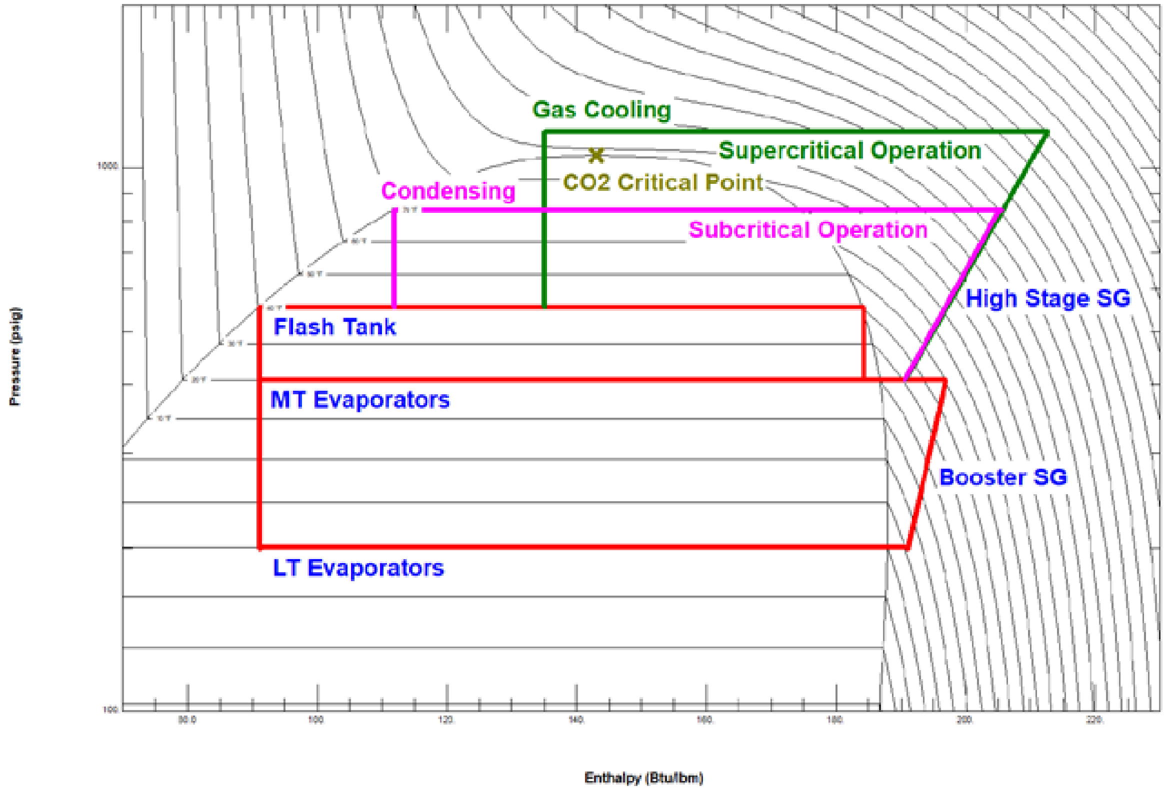
A typical transcritical CO_2 booster system is shown in Figure 10-37 below. The system consists of two suction groups: booster and high stage (HS). The compressors in the booster suction group serve low temperature (LT) loads and discharge into the suction of the high stage suction group. The compressors in the high stage suction group serve the medium temperature (MT) loads, as well as compress the gas from the booster suction group and the intermediate pressure vessel to high pressures. Heat is rejected from the high pressure gas in the gas cooler when the system is operating in supercritical mode. The discharge pressure is commonly controlled by a hold back valve in combination with the gas cooler fans. When operating in subcritical mode the gas cooler operates as a condenser, analogous to other common refrigeration systems. The gas or liquid from the gas cooler / condenser expands in the intermediate pressure vessel / flash tank. The gas from the intermediate pressure vessel is compressed by the high stage compressors, and the liquid from the flash tank is supplied to medium temperature and low temperature evaporators (loads). The evaporated gas in the evaporators is compressed by its respective suction group compressors.

Figure 10-37: Common Transcritical CO_2 System Configuration



The critical point of a substance is the point above which the liquid and vapor phases become indistinguishable from each other, forming a 'supercritical fluid'. In the supercritical region, temperature and pressure are semi-independent variables. CO_2 has a critical point of 87.8°F, which is considered to be a low critical point compared to all commonly used refrigerants. A pressure-enthalpy (PH) diagram for CO_2 with its critical point labeled is shown in Figure 10-38. In a transcritical CO_2 refrigeration system, the high stage suction group can operate both above and below the critical point. When ambient temperatures are high, above approximately 75°F, the HS suction group compresses CO_2 above its critical point and the system is said to be in supercritical operation. An example of a HS suction group supercritical vapor compression cycle on the PH diagram for CO_2 is represented in green in Figure 10-38. During lower ambient conditions, when CO_2 is below its critical point after compression, the system is said to be in subcritical operation. An example of a HS suction group subcritical vapor compression cycle on the PH diagram for CO_2 is represented in pink in Figure 10-38.

Figure 10-38: Common Transcritical CO2 System Configuration



In subcritical mode, the system operates very similarly to other refrigeration systems. In supercritical mode, the overall system efficiency decreases compared with subcritical operation. This is because high ambient temperatures result in higher compressor discharge temperatures needed for heat rejection, which increases the suction-to-discharge pressure ratio to be overcome by the compressor. Additionally, when operating in supercritical mode, the gas cooler outlet stream has a higher quality (higher vapor fraction) compared to subcritical mode. Vapor in the intermediate pressure vessel does not contribute to productive refrigeration but needs to be compressed, increasing the non-productive refrigeration load on the compressors. Available technologies that increase supercritical operation efficiency include gas ejectors and parallel compression.

10.6.3.2.1 Transcritical CO_2 Gas Coolers

§120.6(a)8

New fan-powered gas coolers on all new transcritical CO_2 refrigeration systems must follow the gas cooler type, sizing, fan control, and efficiency requirements as described in §120.6(a)8.

A. Air-cooled Gas Coolers Restrictions

§120.6(a)8A

§120.6(a)8A prohibits the use of air-cooled gas coolers in Climate Zones 9 through 15, which are high ambient temperature climate zones, to reduce the number of supercritical operating hours. Alternatives to air cooled gas coolers include water cooled gas coolers connected to a cooling tower, adiabatic gas coolers, and evaporative gas coolers.

B. Gas Cooler Sizing

§120.6(a)8B and §120.6(a)8C describe minimum sizing requirements for new gas coolers serving new transcritical CO_2 refrigeration systems. Fan-powered air-cooled gas coolers are covered by §120.6(a)8B. Fan-powered adiabatic gas coolers are covered by §120.6(a)8C.

Gas coolers must be sized to provide sufficient heat rejection capacity under design conditions while maintaining a specified maximum temperature difference between the gas cooler leaving gas temperature and ambient temperature. The design gas cooler capacity shall be greater than the calculated combined total heat of rejection (THR) of the dedicated compressors that are served by the gas cooler. If multiple gas coolers are specified, then the combined capacity of the installed gas coolers shall be greater than the calculated heat of rejection. When determining the design THR for this requirement, reserve or backup compressors may be excluded from the calculations. Example 10-50 provides an example scenario of which compressors to include in the THR calculation described in this section.

1. Air-Cooled Gas Cooler Sizing

§120.6(a)8B

§120.6(a)8B provides maximum design gas cooler leaving gas temperature (LGT) values for air-cooled gas coolers. For 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 gas cooler ratings to assume the capacity of air-cooled gas coolers is proportional to the temperature difference (TD) between the LGT and DBT, regardless of the actual ambient temperature entering the gas cooler. For example, the capacity of an air-cooled gas cooler operating at a leaving gas temperature of 80°F with a DBT of 70°F is assumed to be equal to the same unit operating at 110°F LGT and 100°F DBT, since the TD across the gas cooler is 10°F in both examples. Thus, the requirement for air-cooled gas coolers is based on the temperature difference between the DBT and gas cooler leaving gas temperature. Air cooled gas coolers shall be sized so the design leaving gas temperature for air-cooled gas coolers shall be less than or equal to the design dry bulb temperature plus 6°F.

The Energy Codes include an exception to §120.6(a)8B for air-cooled gas coolers serving refrigeration systems in Climate Zones 2, 4 and 8 where the design leaving gas temperature for shall be less than or equal to the design dry bulb temperature plus 8°F.

2. Adiabatic Gas Cooler Sizing

§120.6(a)8C

§120.6(b)8C provides maximum design gas cooler leaving gas temperature (LGT) values for adiabatic gas coolers. For 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 gas cooler ratings to assume the capacity of adiabatic gas coolers is proportional to the temperature difference (TD) between the LGT and DBT for operation in dry mode, regardless of the actual ambient temperature entering the gas cooler. For example, the capacity of an adiabatic gas cooler operating at a LGT of 80°F with a DBT of 70°F is assumed to be equal to the same unit operating at 110°F LGT and 100°F DBT during dry mode operation, since the TD across the gas cooler is 10°F in both examples. Thus, similar to air-cooled gas coolers, the requirement for adiabatic gas coolers is based on the temperature difference between the DBT and gas cooler leaving gas temperature. Design leaving gas temperature for adiabatic gas coolers necessary to reject the design total heat of rejection of a refrigeration system assuming dry mode performance shall be less than or equal to the design dry bulb temperature plus 15°F.

C. Fan Control

§120.6(a)8D through §120.6(a)8G describe fan control requirements for new gas coolers serving new transcritical CO_2 refrigeration systems. Fan speed control requirements are covered by §120.6(a)8D. Gas cooler pressure control requirements during subcritical and supercritical operation are described by §120.6(a)8E and §120.6(a)8F respectively. Minimum condensing temperature set point is covered by §120.6(a)8G.

1. Speed Control

§120.6(a)8D

Gas cooler fans for new air-cooled, evaporative-cooled or adiabatic gas coolers, or fans on cooling towers or fluid coolers used to reject heat on new transcritical CO_2 refrigeration systems, must use continuously variable-speed. Variable-frequency drives are commonly used to provide continuously variable-speed control of gas cooler 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 gas cooler or set of gas coolers 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 reduce gas cooler capacity. As load increases, fans should be turned back on before significantly increasing fan speed, recognizing a control band is necessary to avoid excessive fan cycling.

2. Subcritical Pressure Control

§120.6(a)8E

§120.6(a)8E provides pressure control requirements for gas cooler operation below the critical point. These requirements are the same as for §120.6(a)4F with the exception that the minimum condensing temperature set point must be 60°F for transcritical CO_2 systems with a design intermediate saturated suction temperature lower than 30°F. See section 10.6.3.3. B for details.

3. Supercritical Pressure Control

§120.6(a)8F

During supercritical mode, the gas cooler pressure set point must be continuously reset in response to ambient conditions to optimize system efficiency, rather than using a fixed set point value.

Specifying the exact relationship to be used to determine the optimal head pressure may be dependent on multiple variables beyond ambient air temperature, including the operating saturated suction temperature, system configuration, gas cooler technology type, and current load. The controls manufacturer shall consider the tradeoff between fan energy and compressor energy in developing a pressure and fan control that is responsive to environmental and system conditions.

4. Minimum SCT Set point

§120.6(a)8G

The minimum saturated condensing temperature set point must be 60°F (16°C) or less for air-cooled gas coolers, evaporative-cooled gas coolers, adiabatic gas coolers, air or water-cooled fluid coolers or cooling towers. As a practical matter, a maximum condensing temperature set point is also commonly employed to set an upper bound on the control set point in the event of a sensor failure and to force full gas cooler operation during peak ambient conditions. This value should be set high enough that it does not interfere with normal operation.

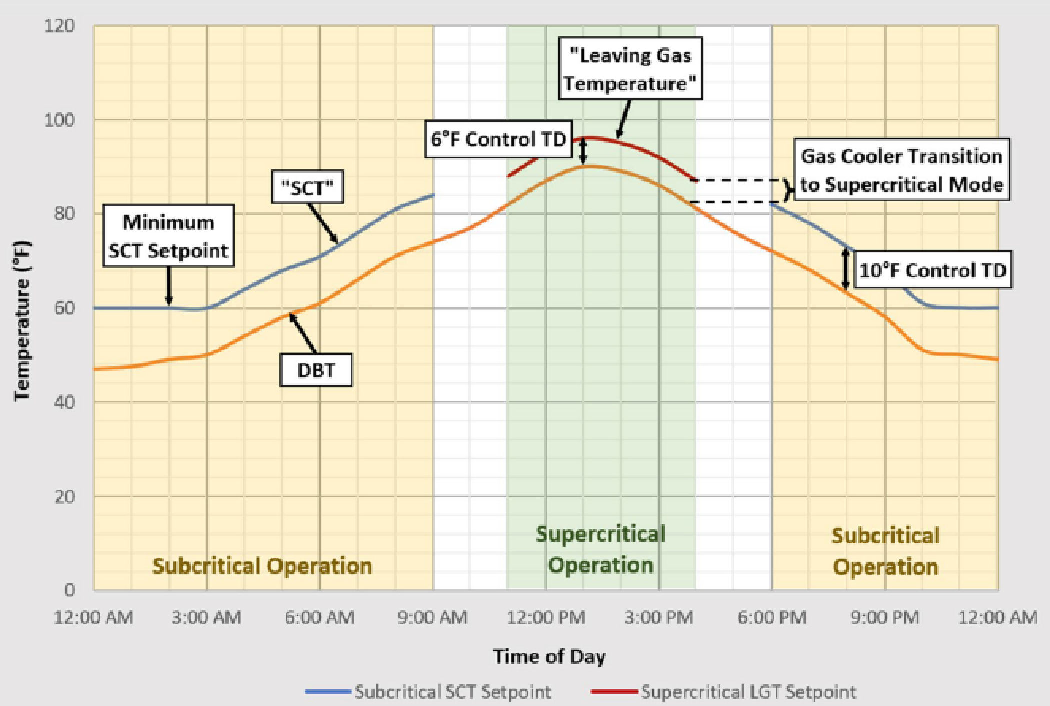
Example 10-50

Question

A refrigerated warehouse with air-cooled gas coolers serving freezers is being commissioned. The control system designer has used a drybulb-following control strategy to reset the system saturated condensing temperature (SCT) setpoint when the system is operating subcritically, with a TD of 10°F (5.6°C) above the ambient drybulb temperature. The control system designer has used a proprietary gas cooler pressure control strategy to maximize system efficiency by minimizing the combined compressor and condenser fan power usage when the system is operating supercritically, with a TD of 6°F (3.3°C). What might the system DBT and SCT / gas cooler leaving gas temperature (LGT) setpoints trends look like over an example day?

Answer

The following figure illustrates what the actual SCT / LGT setpoint could be over an example day using the drybulb-following control strategy with a 10°F TD (5.6°C) when operating subcritically, a 6°F (3.3°C) TD when operating supercritically and observing the 60°F (16°C) minimum condensing temperature requirement. As the figure shows, the SCT setpoint is continuously reset to 10°F (5.6°C) above the ambient drybulb temperature until the minimum SCT setpoint of 60°F (16°C) is reached when operating subcritically and 6°F (3.3°C) above the ambient drybulb temperature when operating supercritically.



D. Gas Cooler Specific Efficiency

§120.6(a)8H

Requirements for design leaving gas temperatures relative to design ambient temperatures, as described above for §120.6(a)8B and C, help assure that there is enough gas cooler capacity to keep leaving gas temperatures compressor head pressures at reasonable levels. However, the sizing requirements do not address gas cooler efficiency. For example, rather than providing amply sized gas cooler surface area, a gas cooler selection could consist of a small gas cooler area using a large motor to blow a large amount of air through the heat exchanger surface to achieve the design gas cooler TD. However, this would come at the expense of excessive fan motor horsepower. Also, relatively high fan power consumption can result from using gas cooler fans that have poor fan efficiency or low fan motor efficiency. §120.6(a)8H addresses these and other factors affecting gas cooler fan power by setting minimum specific efficiency requirements for gas coolers.

Table 10-7: Transcritical CO_2 Fan-Powered Gas Coolers – Minimum Specific Efficiency Requirements

Condenser Type	Refrigerant Type	Minimum Specific Efficiency	Rating Condition
Outdoor Air-Cooled	Transcritical CO_2	160 Btuh/Watt	1400 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature
Adiabatic Dry Mode	Transcritical CO_2	90 Btuh/Watt	1100 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature

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 outlet gas temperature and 90°F outdoor dry bulb temperature. Input power is the electric input power draw of the gas cooler fan motors (at full speed). 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 shall not be included in the specific efficiency calculation.

As shown in Table 10-7 the Energy Code has different minimum efficiencies depending on the type of gas cooler that is being used. The different classifications of gas coolers are:

1. Outdoor, air-cooled.
2. Adiabatic (dry-mode operation).

The data published in the gas cooler manufacturer's published rating for capacity and power shall be used to calculate specific efficiency.

For air-cooled and adiabatic gas coolers, manufacturers typically provide the capacity at a given temperature difference (TD) between SCT and dry bulb temperature. Manufacturers typically assume that gas cooler capacity is linearly proportional to TD; the catalog capacity at 20°F TD is typically twice as much as at 10°F TD. The gas cooler capacity for air-cooled gas cooler at a TD of 10°F shall be used to calculate efficiency. For adiabatic gas coolers, the dry mode capacity 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 gas cooler, the actual manufacturer's rated motor power may vary from motor nameplate in different ways. Air-cooled gas coolers with direct-drive OEM motors may use far greater input power than the nominal motor horsepower would indicate. Thus, actual motor input power from the manufacturer must be used for direct-drive air-cooled gas coolers.

Example 10-48 provides an example calculation for the efficiency of a condenser, which is analogous to how the efficiency for a gas cooler would be calculated.

10.6.4 Additions and Alterations

Requirements

Requirements related to refrigerated warehouse additions and alterations are covered by the Energy Code 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 the following:

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 an 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 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 an alteration.

Example 10-51

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 Code?

Answer

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

Example 10-52

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 Code?

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 Code. If a new refrigeration system was installed with a new condenser for the addition, then the new condenser must also comply with the Energy Code.

Example 10-53

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 Code?

Answer

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

Example 10-54

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 Code?

Answer

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

Example 10-55

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 Code?

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

§140.9(c) sets the minimum requirements for laboratory and factory exhaust systems. Laboratories have an average annual energy intensity 10-20 times larger than offices when normalized by building area. The primary drivers of laboratory building energy are long operation hours, exhaust fan energy, and makeup air conditioning in addition to typically high internal loads.

To help reduce laboratory and factory energy use, there are four categories of exhaust energy saving measures:

- Exhaust and makeup air reduction
- Reduction of conditioned makeup air
- Exhaust fan power reduction
- Fume hood automated sash closures

Laboratories in healthcare facilities are not required to meet the requirements of §140.9(c).

10.7.2 Mandatory Measures

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

10.7.3 Prescriptive Measures

Summary of measures contained in this section:

k. Airflow Reduction Requirements - §140.9(c)1
l. Exhaust System Transfer Air - §140.9(c)2
m. Fan System Power Consumption - §140.9(c)3
n. Fume Hood Automatic Sash Closure - §140.9(c)4

10.7.3.1 Airflow Reduction Requirements

§140.9(c)1 requires that all laboratory exhaust with minimum circulation rates of 10 air changes per hour (ACH) or lower shall be designed for variable-volume control on the supply, fume exhaust, and general exhaust. This requirement will enable the system to reduce zone exhaust and makeup airflow rates to the minimum allowed for ventilation or to maintain the required differential pressure for the zone.

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)1]. Examples include hoods using perchloric acid, hoods with radio isotopes, and exhaust systems conveying dust or vapors that need a minimum velocity for containment.

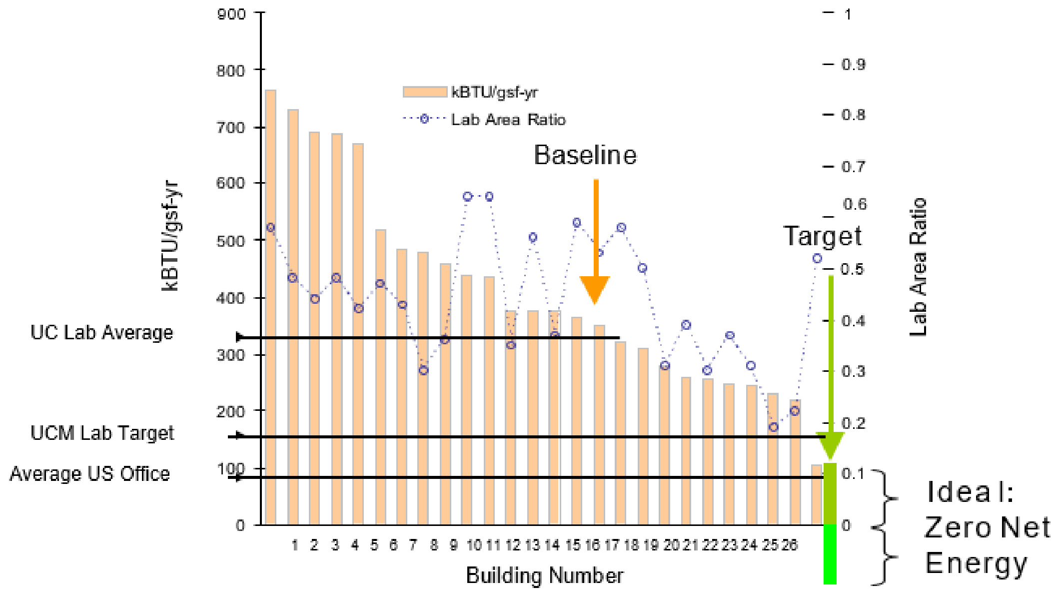
A second exception is provided for new zones added to an existing constant volume exhaust system [Exception 2 to §140.9(c)1].

The energy and demand savings depend strongly characteristics of the facility, including:

1. Ratio of lab to non-lab space.
2. Minimum airflow required by code or the facility EH&S department. These range from 4 to 18 ACH or higher.
3. Climate zone.

Figure 10-38 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 axis. The 26 labs are arranged from highest to lowest normalized energy use. The right axis 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 its laboratories; and the average national energy end use for office buildings.

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



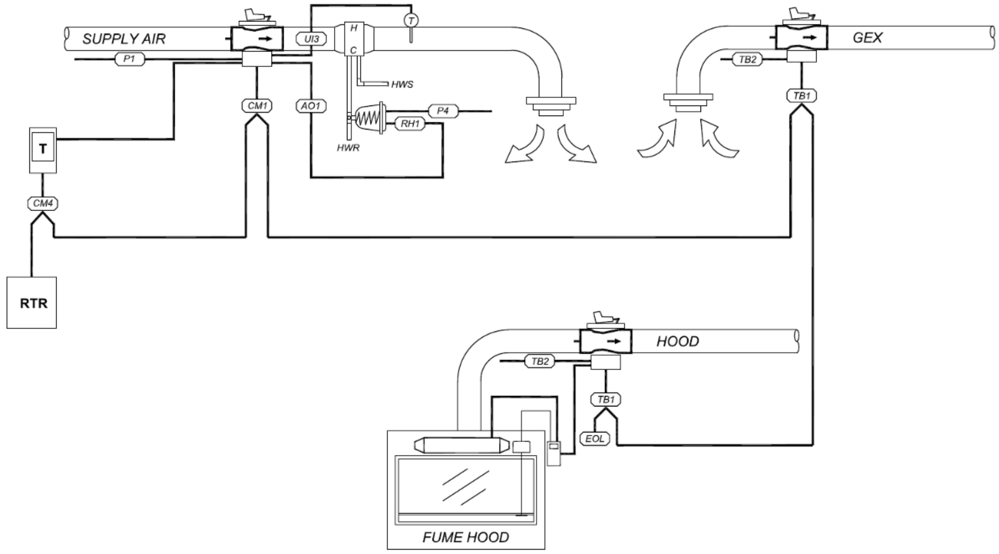
Using the criteria for cost-effectiveness in the Energy Code and very conservative estimates of the first costs (using costs from VAV retrofits not new construction), this measure was shown to be cost-effective in all California climate zones up to 14 ACH of minimum ventilation

Using off-the-shelf variable air volume (VAV) controls can greatly reduce the energy use in laboratory buildings. The Energy Code 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 codes. Furthermore, ANSI/AIHA Z9.5 and NFPA 45 allow lower minimum airflows for many hoods, which increases the savings from VAV design.

Figure 10-39 below shows the zone components for a 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. When used, the hood valve 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 reheat coils to maintain space comfort for heating. The GEX is typically used to control the cooling, on a 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 either variable volume or constant volume depending on the application. A hood might for instance be required to maintain 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 attached exhaust also serves variable-volume zones. The same rule applies 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 are 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.

10.7.3.2 Exhaust System Transfer Air

This section limits the amount of conditioned air supplied to a space with mechanical exhaust. The benefit of this requirement is to take advantage of available transfer air. By doing so, the amount of air that needs to be conditioned is limited, thus saving energy. Conditioned supply air is limited to the greater of:

1. The supply flow required to meet the space heating or cooling load.

2. The ventilation rate required by the AHJ, facility EH&S department, or by §120.1(c)3.

3. The mechanical exhaust flow minus the available transfer air.

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

Available transfer air can be from adjacent conditioned spaces or return air plenums that are on the same floor, same smoke or fire compartment, and within 15 feet. To calculate the available transfer air:

1. Calculate the minimum outside air required by adjacent spaces.
2. From 1, subtract the amount of air required by adjacent space exhaust.
3. From 2, subtract the amount of air required to maintain pressurization of adjacent spaces. This is your available transfer air.

Exceptions are provided for:

4. Laboratories classified as biosafety level 3 or higher.
5. Vivarium spaces.
6. Spaces required to maintain positive pressure differential relative to adjacent spaces.
7. Spaces that require a negative pressure relationship and the demand for transfer air may exceed the available transfer airflow rate.
8. Healthcare facilities.

10.7.3.3 Fan System Power Consumption

Newly installed laboratory and factory exhaust systems greater than 10,000 CFM have three prescriptive pathways to show compliance with this section. Regardless of the path chosen, all exhaust systems must meet the discharge requirements of ANSI Z9.5.

ANSI Z9.5-2012 Discharge Requirements:

Section 5.4 and Appendix 3 of ANSI Z9.5, Laboratory Ventilation, describe standards for laboratory exhaust system design including the discharge requirements cited by this section of the Energy Code.

10.7.3.4 Exhaust Fan System Power Consumption:

As described in greater detail for the previous VAV section, one of the major drivers of laboratory and factory energy is fan power. To reduce this demand, three prescriptive pathways have been added for the 2019 code cycle. The first and simplest pathway is an exhaust system power limit of 0.65 or 0.85 W/cfm depending on system design. The other two options do not limit exhaust system power, but instead require exhaust volume flow rate control based on either local wind conditions or exhaust chemical concentration.

Option 1: Exhaust System Efficacy:

- Systems without air treatment devices are limited to 0.65 W/cfm of exhaust air.
- Systems with air filtration, scrubbers, or other air treatment devices are limited to 0.85 W/cfm of exhaust air.
- An exception is provided for systems with code required air treatment devices that cause static pressure drop greater than 1 inch of water. For example, a local jurisdiction which has an ordinance for high exhaust filtration, which causes high filter static pressure, due to smell or other exhaust considerations.

Option 2: Wind-Based Exhaust Volume Flow Rate Control:

This compliance path saves fan energy by reducing exhaust stack airflow when local wind conditions permit. The Energy Code, ANSI Z9.5, and other best engineering practices dictate several necessary components to this type of system:

1. Anemometer Sensors:
    - **a.** Two anemometer sensors must be used to enable sensor fault detection.
    - **b.** Installation location must exhibit similar wind speed and direction to the free stream air above the exhaust stacks.
    - **c.** Sensors must be located high enough to be above the wake region created by nearby structures.
    - **d.** Sensors must be factory calibrated.
    - **e.** Sensors must be certified by the manufacturer to an accuracy of ± 40 feet per minute (fpm), ± 5.0 degrees, and to require calibration no more than every five years.
2. Dispersion Modeling:
    - **a.** Wind dispersion analysis must be used to create a look-up table for exhaust volume flow rate versus wind speed/direction.
    - **b.** Look-up table must contain at least eight wind speeds and eight wind directions to define the safe exhaust volume flow rate.
    - **c.** Exhaust volume flow rate must be based on maintaining downwind chemical concentrations below health and odor limits as defined by the 2018 American Conference of Governmental Industrial Hygienists,

Threshold Limit Values and Biological Indices, or more stringent, local, state, and federal limits if applicable.

3. Sensor Fault Management:

a. Minimum sensor failure thresholds:

- If any sensor has not been calibrated within the associated calibration period.
- Any sensor that is greater than ±30% of the four-hour average reading for all sensors.

b. Upon sensor failure, the system must revert to a safe exhaust volume flow rate based on worst-case wind conditions. Furthermore, the system must report the fault to an Energy Management Control System or other application which notifies a remote system provider.

4. Certification of requirements listed in NA7.16 for wind speed/direction control

Option 3: Contaminant Concentration-Based Exhaust Volume Flow Rate Control:

This compliance path saves fan energy by reducing exhaust airflow when the exhaust contaminant concentration is low enough to maintain safe downwind concentrations. The Energy Standards and best engineering practices dictate several necessary components to this type of system:

1. Chemical Concentration Sensors:

- Two contaminant concentration sensors must be used in each exhaust plenum to enable sensor fault detection.
- Sensors must be photo ionization detectors.
- Sensors must be factory calibrated.
- Sensors must be certified by the manufacture to an accuracy of ± 5% and require calibration no more than every six months.

2. Dispersion Modeling:

- Wind dispersion analysis must be used to determine contaminant-event thresholds (contaminant concentration levels), which control when the exhaust volume flow rate can be turned down during normally occupied hours.
- Exhaust volume flow rate must be based on maintaining downwind chemical concentrations below health and odor limits as defined by the 2018 American Conference of Governmental Industrial Hygienists, Threshold Limit Values and Biological Indices, or more stringent, local, state, and federal limits, if applicable.

3. Sensor Fault Management:

- Minimum sensor failure thresholds:
  - If any sensor has not been calibrated within the associated calibration period.
  - Any sensor that is greater than ±30% of the four-hour average reading for all sensors.
- Upon sensor failure, the system must revert to a safe exhaust volume flow rate based on worst-case wind conditions. Moreover, the system must report the fault to an energy management control system or other application that notifies a remote system provider.

- Certification of requirements listed in NA7.16 for contaminant control

Example 10-56

Question

A laboratory space has 2,500 ft^2 of conditioned floor area, a drop ceiling for plenum space, and ceiling height of 10 feet. The lab has a minimum ventilation rate of 2,500 cfm.

Is this laboratory required to have variable-volume exhaust and makeup air flow to comply with Section 140.9(c)1?

Answer

In the absence of any other code or environmental health & safety requirement for constant speed operation, Section 140.9(c)1 requires that laboratories have variable-volume exhaust and makeup airflow if the minimum ACH is less than or equal to 10. For this laboratory space, ACH is equal to the following, noting that ACH is calculated for laboratory conditioned space not including plenum volume:

ACH = 2,500 cfm x 60 min / hr / (2,500 ft^2 x 10 ft) = 6 ACH

Thus, if there is no conflicting code or safety requirement for constant volume operation, this space requires a variable-volume HVAC system.

Example 10-57

Question

A variable-volume supply fan and a variable-volume exhaust fan serving a lab system has a fan system design supply airflow and design exhaust airflow of 8,000 cfm. The system consists of one supply fan operating at an input power of 5.0 bhp and one exhaust fan operating at an input power of 8.0 bhp. The exhaust system uses a 0.6 in. pressure drop filtration device, airflow control devices, and serves fume hoods.

Does this fan system comply with the fan power requirements in Title 24?

Answer

For laboratory exhaust systems with total flow rates less than or equal to 10,000 cfm, the total fan energy of the space conditioning system and the laboratory exhaust system must comply with Section 140.4(c). First, the design fan power must be calculated in bhp, as shown below:

Design Fan Power = 5.0 bhp + 8.0 bhp = 13.0 bhp

Then, the fan power limit in section 140.4(c) is determined. From Table 140.4-A, the allowable system input power for the system is:

bhp = CFMs × 0.0013 + A

= 8,000 × 0.0013 + A = 10.4 + A

where A accounts for pressure drop adjustments.

From Table 140.4-B, the pressure drop adjustment for the exhaust flow control device (FC) is 0.5 in. of water, the pressure drop adjustment for fully ducted exhaust systems (DE) is 0.5 in. of water, and the pressure drop adjustment for the fume hoods (FH) is 0.35 in. of water. The pressure drop adjustment for fully ducted exhaust systems is included because laboratory exhaust systems are required under Title 8 to be fully ducted. An additional pressure drop adjustment is allowed to be equal to the design pressure drop of an exhaust filtration device (FD) which for this design is 0.6 in. of water column. The airflow through all these devices is 8,000 cfm, so the additional input power that is allowed is 3.8 bhp, as calculated below.

$A = [CFM_{FC} \times PD_{FC} + CFM_{DE} \times PD_{DE} + CFM_{FH} \times PD_{FH} + CFM_{FD} \times PD_{FD}] / 4,131$

A = [8,000 × 0.5 + 8,000 × 0.5 + 8,000 x 0.35 + 8,000 x 0.6] / 4131 = 3.8 bhp

The total allowed input power is 10.4 bhp plus 3.8 bhp, or 14.2 bhp. Because the design fan power of 13.0 bhp is less than 14.2 bhp, the system does comply using the procedure in section 140.4(c). If the system did not comply, one could evaluate several methods of dropping the design brake horsepower such as: lowering pressure drop though the system by increasing duct size or selecting low pressure drop valves or low pressure drop duct fittings. Alternatively, brake horsepower can be dropped by selecting a fan with higher fan efficiency at the design point.

Example 10-58

Question

A variable-volume supply fan and a variable-volume exhaust fan serving a lab system has a fan system design supply airflow and design exhaust airflow of 12,000 cfm. The system consists of one supply fan operating at an input power of 10.0 bhp served by a nominal 15 hp motor and one exhaust fan operating at an input brake horsepower of 18.0 bhp served by a nominal 25 hp motor, which at design conditions draws 14.4 kW. The exhaust system uses a 0.6 in. pressure drop filtration device and airflow control devices and serves fume hoods.

Does this fan system comply with the fan power requirements in Title 24?

Answer

For laboratory exhaust systems with total flow rates greater than 10,000 cfm, the fan energy of the space conditioning system is regulated by the requirements of Section 140.4(c) and the fan energy of the laboratory exhaust system is regulated by Section 140.9(c)3.

For laboratory exhaust systems with total flow rates greater than 10,000 cfm, the fan energy of the space conditioning system is regulated by the requirements of Section 140.4(c) and does NOT include the design exhaust fan power or the pressure drop adjustment credits for:

– Exhaust systems required by code or accreditation standards to be fully ducted.

– Exhaust airflow control devices.

– Exhaust filters, scrubbers, or other exhaust treatment.

– Exhaust systems serving fume hoods.

– Biosafety cabinets.

The fan power limit in Section 140.4(c) is determined. From Table 140.4-A, the allowable system input power for the system can be calculated for either the design motor horsepower for the fan or the brake horsepower supplied to the fan.

For the motor horsepower approach for a variable-volume system, with maximum design airflow rate, cfm_s, of 12,000 cfm, the nominal horsepower shall be no greater than:

$$hp < cfm_s \times 0.0015 = 12{,}000 \times 0.0015 = 18 \text{ hp}$$

The supply fan had a nominal horsepower of 15 hp. The space conditioning system passes using this approach.

For the fan brake horsepower approach in Section 140.4(c), the allowable system input power for the space conditioning system is:

$$bhp = CFMs \times 0.0013 + A$$

where A accounts for pressure drop adjustments.

In this case, there are no fan pressure adjustments as all the exhaust system and fume hood credits are accounted for in the allowances to Section 140.9(c)3.

$$\text{Allowable fan brake horsepower} = CFMs \times 0.0013 = 12{,}000 \times 0.0013 = 15.6 \text{ bhp.}$$

The supply fan had a design brake horsepower of 10.0 bhp, and since this design is less than 15.6 bhp, the space conditioning system passes using this approach.

The second half of this calculation is to determine whether the fan power of the laboratory exhaust systems complies with the requirements in Section 140.9(c)3. As given from the design documents, the exhaust fan draws 14.4 kW during design conditions while moving 12,000 cfm of air. The design fan watts per cfm is:

Design Exhaust Fan W/CFM = 14.4 kW x 1,000 W/kW / 12,000 CFM = **1.2 W/CFM**

As described in Section 140.9(c)3B, an exhaust system with an air filtration device will have a maximum allowable exhaust fan power of 0.85 W/CFM. Therefore, the maximum allowable exhaust fan power for this system is **0.85 W/CFM**. This is less than the fan system input power of 1.2 W/CFM. Therefore, the system does not comply with the fan power of Section 140.9(c)3B. The designer could redesign the system for lower design watts per cfm by increasing the height of the stack or alternatively design the system to vary the flow rate from the exhaust stack in response to wind speed in accordance with Section 140.9(c)3C or vary the flow rate from the exhaust stack in response to measured contaminant concentration in the exhaust plenum in accordance with Section 140.9(c)3D.

10.7.3.5 **Fume Hood Automatic Sash Closure**

10.7.3.6 Fume hood intense laboratories with VAV HVAC systems and vertical fume hood sashes are prescriptively required to install automatic sash closure systems. This measure saves energy by reducing laboratory exhaust air and makeup air conditioning. For this measure, fume hood intense means the air change rate of the space is driven by the fume hood exhaust, not minimum ventilation requirements. See Table 10-4 below, which specifies fume hood intensity by linear hood density and minimum ventilation air change rate.

10.7.3.7 The Energy Code and best engineering practices dictate several necessary components to this type of system:

1. Zone Presence Sensors:
    - Each sash closure system must have a dedicated zone presence sensor that detects people near the fume hood. Sensor should not be triggered by movement in adjacent zones.
2. Sash should automatically close within 5 minutes of sensing no presence within the fume hood zone.
3. Sash closure system safeguards:
    a. Sash automatic closing should stop when no more than 10 lbs is detected.
    b. Sash should have obstruction sensors that can detect obstructions, including transparent materials such as glassware.
4. Sash closure system must be configurable in a manual open mode.
    1. Manual open mode requires user input (push button, pedal, etc.) to open the sash and will not open automatically from presence detection.
    2. This mode is important for two reasons:
        - Safety: One example is a fume hood that has cross traffic that could cause inadvertent opening in automatic mode. This unnecessarily exposes occupants to dangerous chemicals.
        - Energy Savings: In general, a manual open configuration will save the most energy because the hood is only intentionally opened. Automatic opening mode could cause the sash to open unnecessarily, and fume hoods use more energy when fully open.

3. Automatic closing is unaffected by manual open mode
4. The Energy Code only requires the option of manual mode; sashes can still be configured in auto open mode, if preferred.

5. Certification of requirements listed in NA7.17

10.7.3.8 Fume Hood Intense Laboratories:

10.7.3.9 The intention of the fume hood intense definition is to only require automatic sash closures for spaces that have ventilation driven by fume hood exhaust, not minimum outdoor air requirements. With regard to this table, *linear feet of fume hoods* refer to the nominal hood width, not the sash opening width. The following table defines all spaces that qualify as fume hood intense:

Table 10-4: Fume Hood Intensive Laboratories

Occupied Minimum Ventilation ACH	≤ 4	> 4 and ≤ 6	> 6 and ≤ 8	> 8 and ≤ 10	> 10 and ≤ 12	> 12 and ≤ 14
Hood Density (linear feet per 10,000 ft^3 of laboratory space)	≥ 6	≥ 8	≥ 10	≥ 12	≥ 14	≥ 16

Source: California Energy Commission

Example 10-59

Question

A variable-volume laboratory space has two rooms with 10-foot ceilings, both of which have minimum ventilation rates of 6 ACH. One room has two 6-foot fume hoods and a floor area of 1,000 square feet. The second room has three 6-foot fume hoods and a floor area of 2,500 square feet.

Which fume hoods are required to have automatic sash closing controls, according to Section 140.9(c)4?

Answer

For each space, determine the fume hood density (FHD) as calculated below, noting that hoods of any sash type contribute to the nominal hood length.

FHD = 10,000 ft3 x Total nominal hood length / (lab space volume)

FHD1000 = 10,000 ft3 x 2 x 6 feet / (1,000 ft2 x 10 ft) = 12

FHD2500 = 10,000 ft3 x 3 x 6 feet / (2,500 ft2 x 10 ft) = 7.2

Using the column for minimum ventilation rate of 6 ACH in reference Table 140.9-B, fume hood densities greater than or equal to 8 are fume hood intensive. Since the 1,000 ft2 room is fume hood intensive, any hoods with vertical only sashes in that space are covered by the automatic sash closing controls prescriptive requirement. Since the 2,500 ft2 room is not fume hood intensive, that space is not required to have sash closing controls.

Example 10-60

Question

A building has two laboratory spaces with fume hoods, one with a minimum ventilation rate of 8 ACH and one with a minimum ventilation rate of 12 ACH. Both are designed to have variable-volume HVAC systems even though Section 140.9(c) only requires variable air volume when minimum ACH is 10 or less.

Which fume hoods are required to have automatic sash closing controls, according to Section 140.9(c)4?

Answer

If the spaces are deemed to be fume hood intensive according to Table 140.9-B for the corresponding minimum ACH, they are required to have sash closing controls on any vertical sash hoods. Automatic sash closing controls are required for any vertical-only hoods in fume hood intensive spaces in variable-volume laboratories.

10.7.4 Additions and Alterations

Variable Exhaust and Makeup Airflow

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

Exhaust System Transfer Air

Additions and alterations must comply with the requirements of this section. For alterations, this means that any additional exhaust and conditioned air resulting from an alteration must comply with this section.

Fan System Power Consumption

All newly installed exhaust systems greater than 10,000 cfm must meet the requirements of this section. Alterations and additions that increase an existing exhaust system’s airflow rate over the 10,000-cfm threshold do not need to meet the requirements.

Fume Hood Automatic Sash Closure

Additions and alterations must meet the requirements of this section. The addition of fume hoods to a space resulting in a density above the values of Table 140.9-B requires compliance with this section for those newly installed fume hoods.

10.8 Compressed Air Systems

10.8.1 Overview

Section 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 ≥ 25 hp. An exception is given for medical gas compressed air systems serving healthcare facilities.

Key terms and definitions:

A. **Trim compressor:** 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 the operating range, then it will take more compressors to meet the Energy Code requirements.

B. **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 Code requires that the single compressor be treated as a trim compressor.

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

D. **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 the most efficient operating point. This is displayed in Figure 10-42.

E. **Largest net capacity increment:** 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.

F. **Primary Storage:** tanks or other devices that store compressed air. 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 near an end-use device.

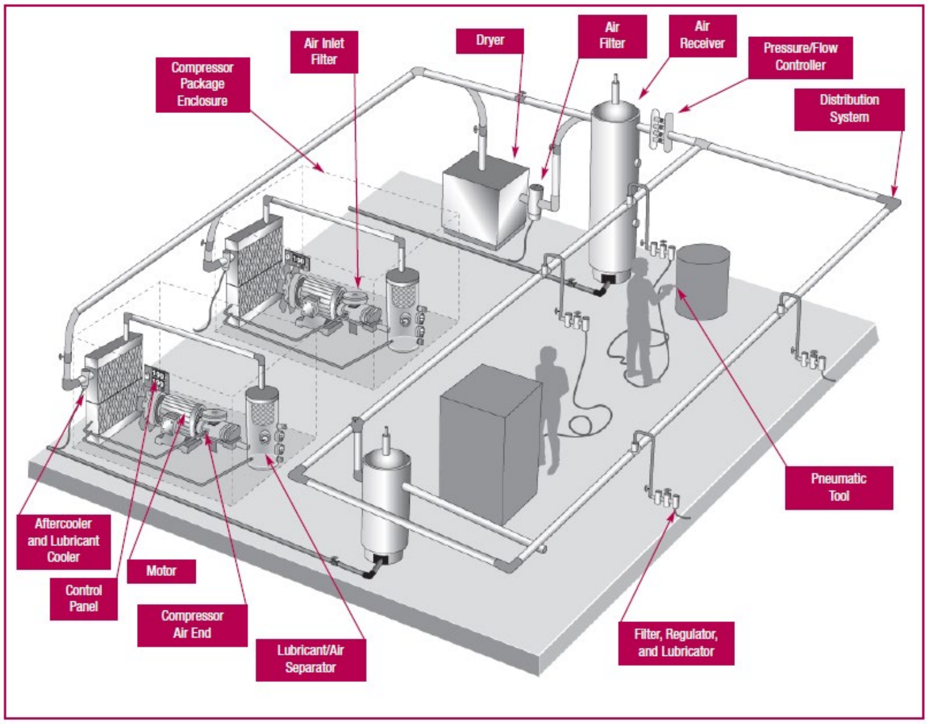
G. **Interconnection Piping:** Interconnection piping is considered to be the piping between compressor discharge outlets, conditioning equipment such as dryers and aftercoolers, and often the primary storage receiver prior to delivery to the main header. Interconnection piping often connects multiple compressors, as well.

H. **Main Header Piping:** Main header piping is the piping that delivers air from the interconnection piping to any distribution piping or sub-headers. This piping often begins at the outlet of a storage receiver or flow controller and terminates at distribution piping out to different areas of a facility. In some cases, there may not be main header piping if the distribution piping is simple enough to contain only a single diameter distribution loop or loops.

I. **Distribution Piping:** Distribution piping includes all piping after the main header and transports air to service lines.

J. **Service Line Piping:** Service lines, often called drops, are typically the smallest diameter piping that delivers air from distribution piping to individual or groups of end-uses. Any tubing such as flexible hoses or plastic tubing within end-uses is not considered service line piping and is not covered by the compressed air pipe sizing or leak testing requirements.

Figure 10-40: Typical Compressed Air System Components



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

10.8.2 Mandatory Measures

§120.6(e)

There are six main mandatory requirements in this section:

a. Trim compressor and storage - §120.6(e)1,
b. Controls - §120.6(e)2,
c. Monitoring - §120.6(e)3,
d. Leak testing of compressed air piping - §120.6(e)4,
e. Pipe sizing - §120.6(e)5, and
f. Compressed air system acceptance - §120.6(e)6.

10.8.2.1 Trim Compressor and Storage

§120.6(e)1

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

There are two 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 net capacity increment.
- Primary storage of at least two gallons per acfm (2 gal/acfm) of the largest trim compressor.
- Effective trim capacity is the range of compressed air flow rates where the specific power (W/acfm) is no greater than 115% of the minimum specific power (Figure 10-42).

Both paths aim to reduce the amount of cycling of fixed-speed compressors by using 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 the 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 the 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 a 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) 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 10-61

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?*

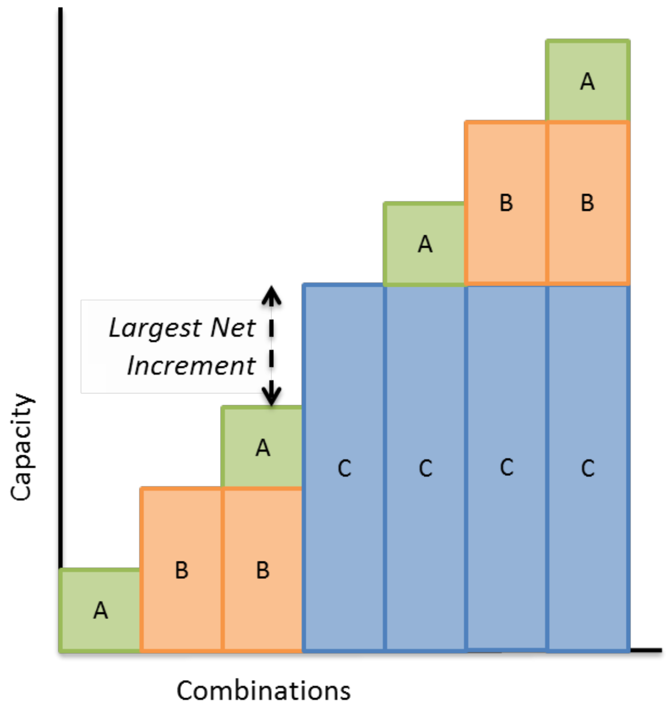
Answer

As shown in the image below, there are eight 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 10-62

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 10-63

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.

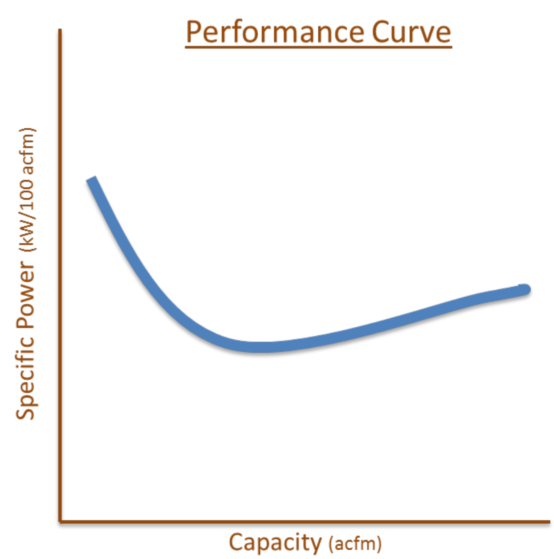
B. Compliance Option 2: Other Compressors as Trim Compressor

§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

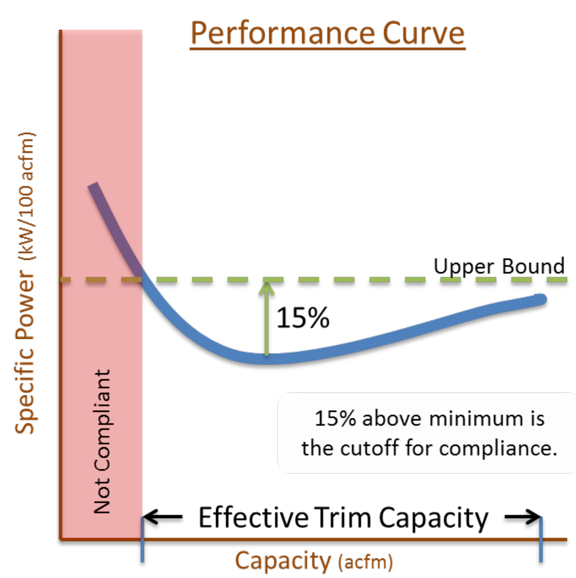


The capacity of the compressor is along the x-axis, while the power is on the y-axis. The curve in Figure 10-41 is a typical shape of a performance curve for a VSD compressor. The lower the specific power, the more energy-efficient the compressor is at that condition.

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, is used to size 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 10-64

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 10-64

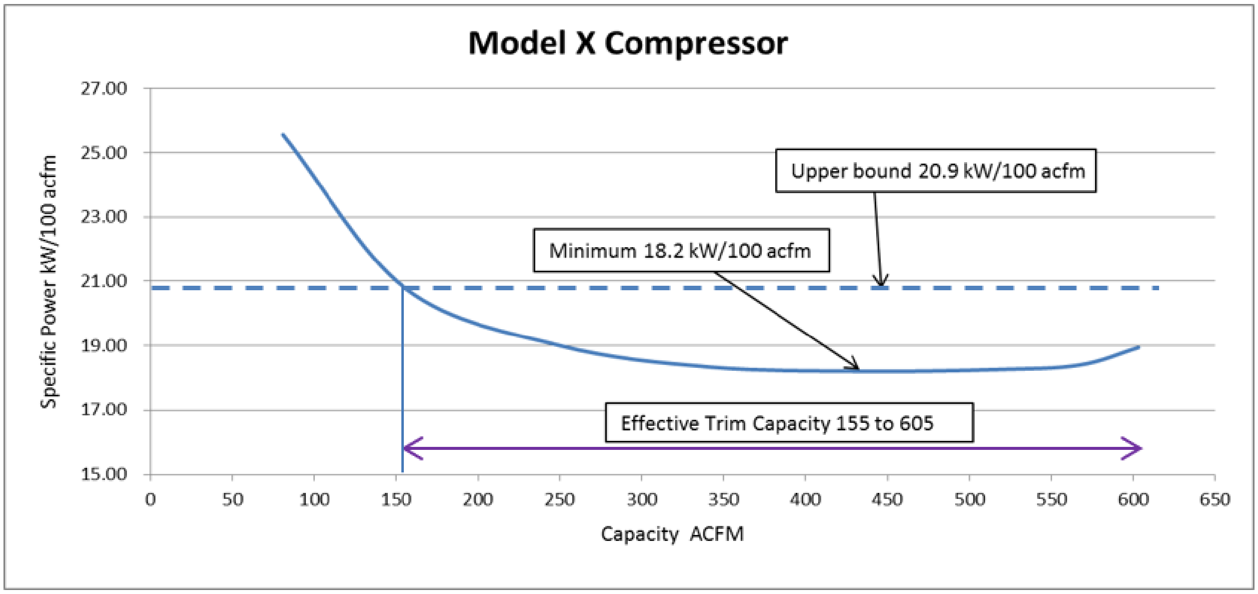
Question

A manufacturer provided the following data for its 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 10-65

Question

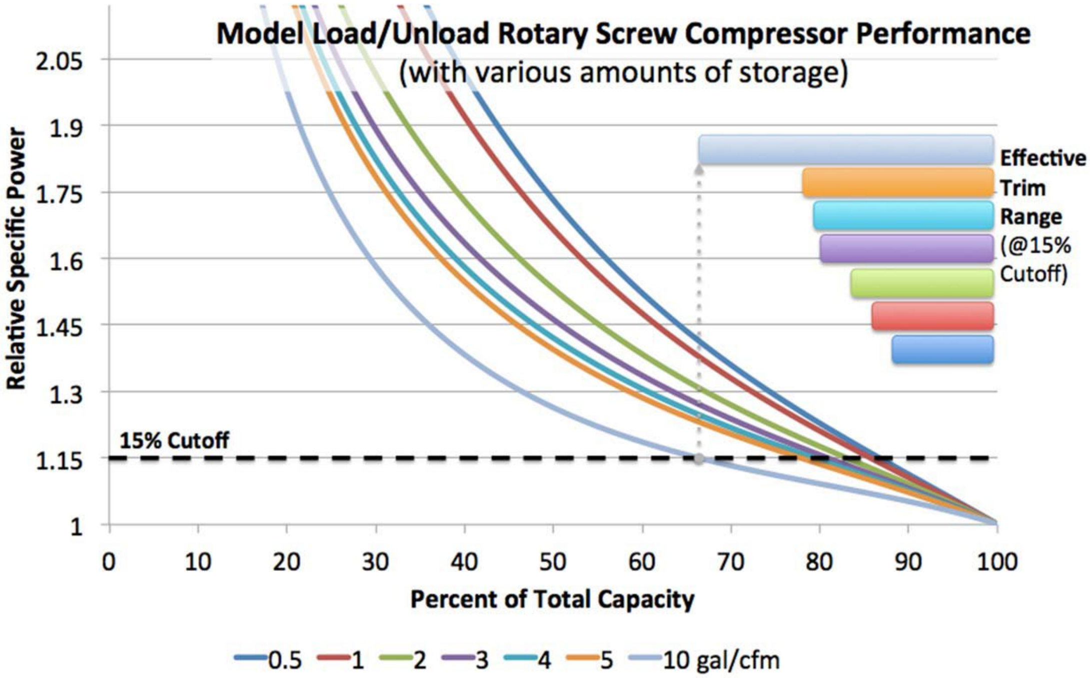
What is the required minimum primary storage capacity for the trim compressor from the previous example to comply with Option 2?

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 airflow rates. Generally, higher levels of storage improve part-load performance. The following data, in Figure 10-43 and for this example, were 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



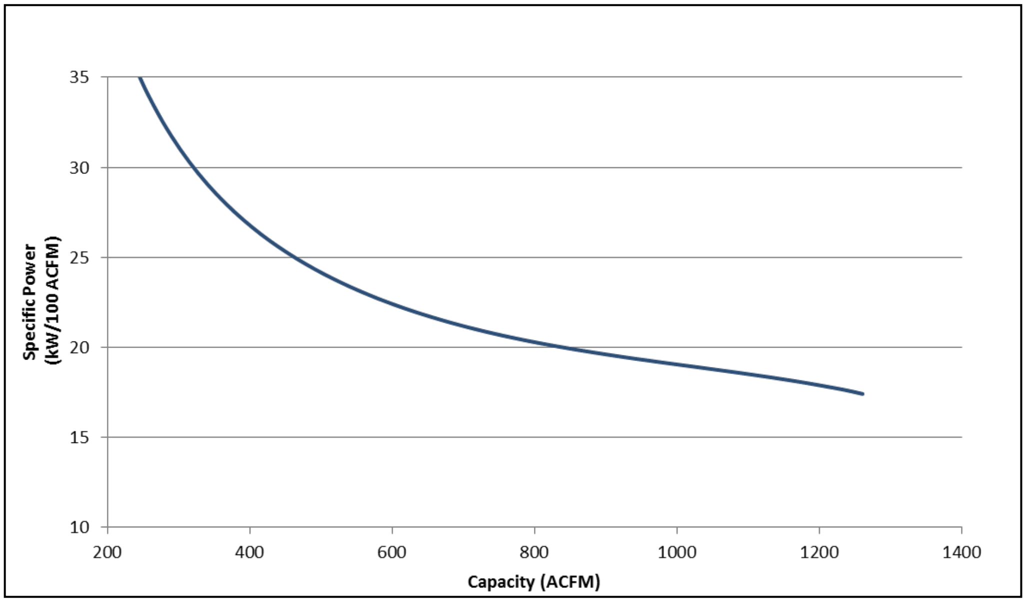
Source: Derived from Fact Sheet 6 – Compressed Air Storage, Improving Compressed Air Storage: A Sourcebook for Industry, U.S. Department of Energy, 2003

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 10-66

Question

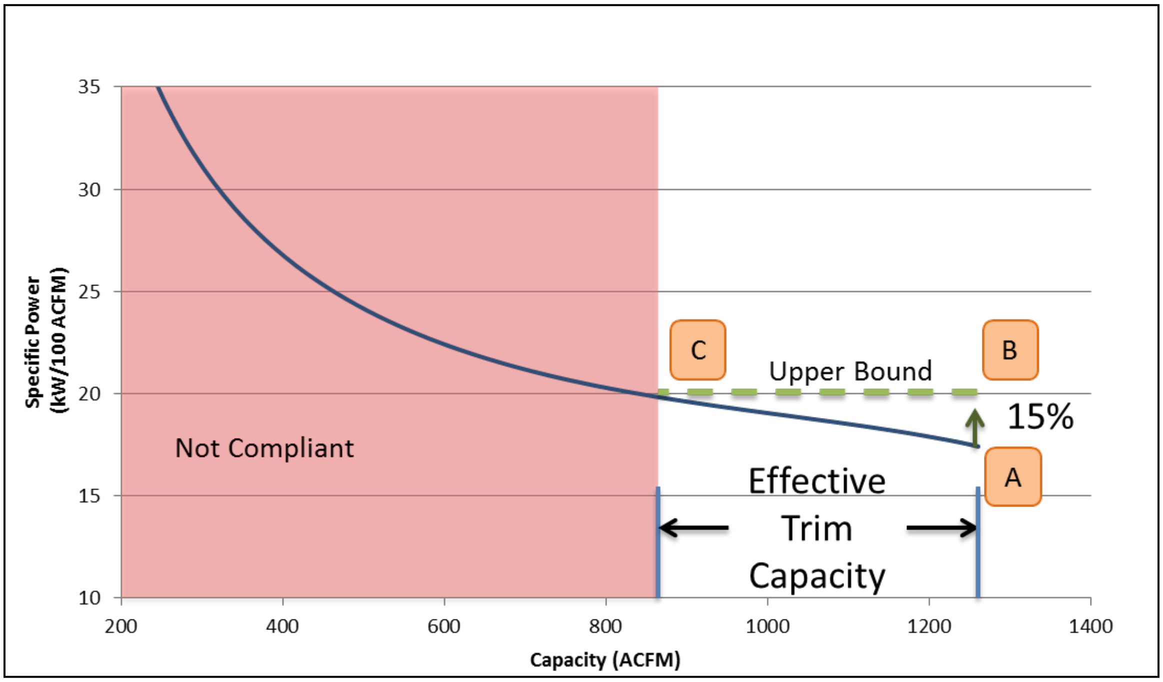
Part-load data were approximated below for a 250-hp load-unload, single-stage, rotary-screw compressor (with a capacity of 1,261 acfm) coupled with 10 gallons/cfm of storage. Would this provide the minimum effective trim capacity for this system to comply with Option 2 using the previous examples?



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 1,261 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. The shaded area labeled "not compliant" is the portion of the compressor capacity that does not contribute to the total system effective trim capacity.



Example 10-67

Question

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

Answer

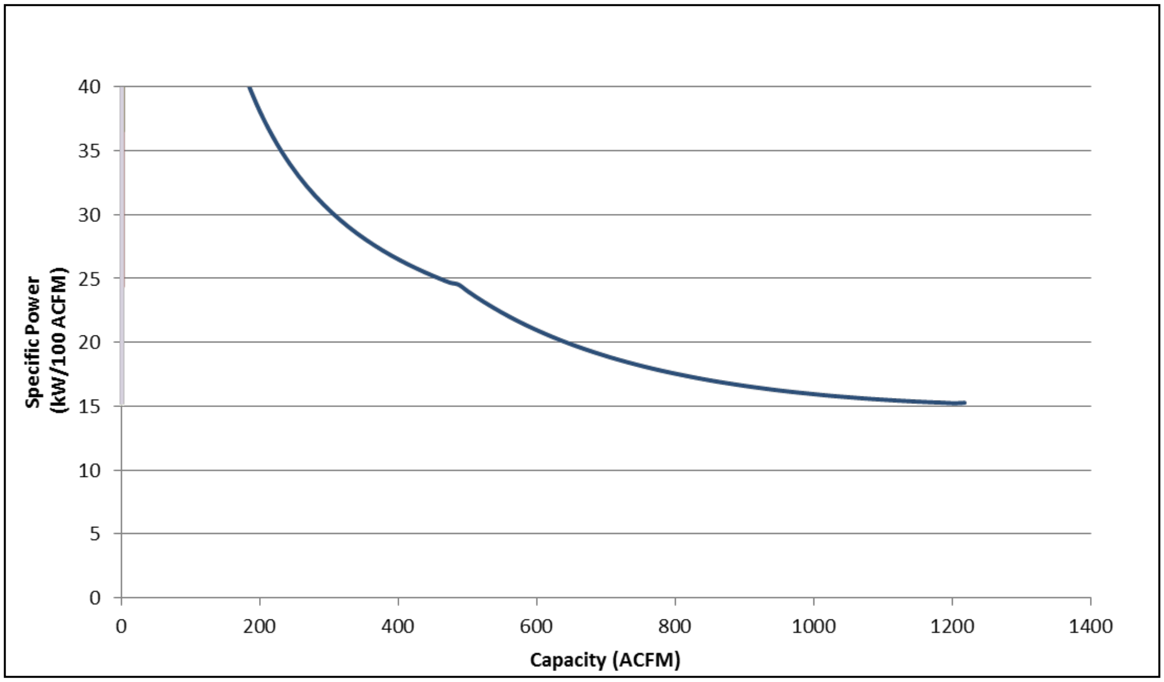
This compressor has a rated capacity of 1,261 acfm, and per §120.6(e)1B it must have 2 gallons of storage per acfm of rated capacity or 1261 * 2 = 2,522 gallons of storage.

However, a minimum of 10 gallons of storage per acfm was needed for the screw compressor with load/unload controls to have a large enough effective trim capacity. The minimum required primary storage to meet 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 1,261 * 10 = 12,610 gallons.

Example 10-68

Question

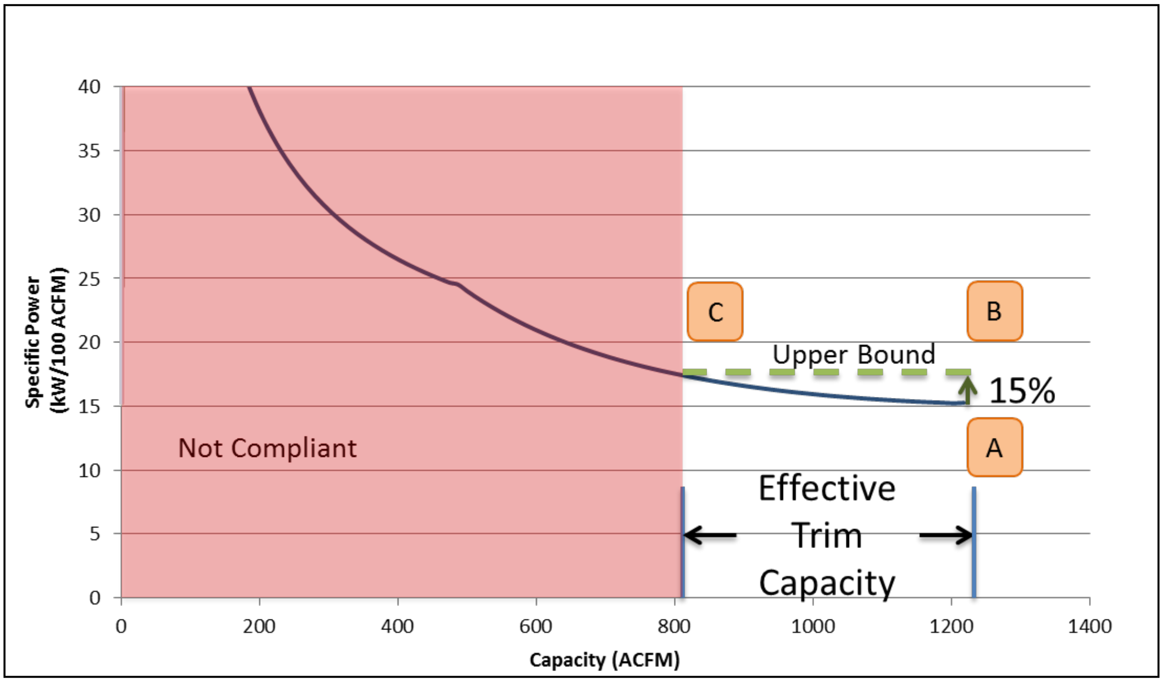
Part-load data were approximated below for a 250-hp variable-capacity compressor. Would this provide the minimum effective trim capacity for this system to comply with Option 2?



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 1,218 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 1,218 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 10-69

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 1,218 acfm, and per §120.6(e)1B it must have 2 gallons of storage per acfm of rated capacity or 1,218 * 2 = 2,236 gallons of storage.

10.8.2.2 **Controls**

§120.6(e)2

This section applies to compressed air systems with more than one on-line compressor and a combined power of ≥100 hp. This section requires an automated control system that 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:

1. A flow meter
2. A combination of pressure transducers and power meters

10.8.2.3 **Monitoring**

§120.6(e)3

This measure supports the ongoing optimization and tuning that is necessary to maintain compressed air system function and efficiency. This is done through specifying monitoring system minimum requirements for the compressed air system. Monitoring of compressed air pressure, input electrical power, and airflow enables identification of any poorly functioning plants, mechanical degradation, or increases in leak loads. Identifying these issues allows for informed maintenance and repair to systems that have unacceptable levels of waste, malfunctions, and inefficiencies. Monitoring of compressed air should be integrated into plant management protocols and facility operations as it can not only be a source for sustaining optimal energy efficiency but also a metric for factory health in an industrial setting. Monitoring of compressed air loads and efficiency is necessary for sustained system performance and energy benefits from the other requirements in §120.6(e).

Any new compressed air plant with combined nominal capacity equal to or greater than 100 hp nominal and any existing plant expanded or altered such that the combined capacity is equal to or greater than 100 hp must have a monitoring system with the following:

1. Measurement of system pressure.
2. Measurement of amps or power of each compressor.
3. Measurement or determination of total flow from the compressed air plant in cfm.
4. Data logging of pressure, power in kW, airflow in cfm, and compressed air specific efficiency in kW/100 cfm at intervals of 5 minutes or less.
5. Maintained data storage of at least the most recent 24 months.
6. Visual trending display of each recorded point, load, and specific energy.

Note that §120.6(e)3B allows for measurement of compressor input power in either amps or direct real power measurement in kW. Amps is allowable to accommodate the common practice of measuring amps and combining with voltage and power factor to estimate input power.

Note that §120.6(e)3C allows for measurement of compressed airflow output through either direct measurement of airflow or through calculation of airflow from performance data. These alternatives accommodate the two primary means of measuring airflow in the compressed air market.

Data storage and display can be accomplished at any potential user interface, from onboard packaged unit displays to integration into plant SCADA systems. Where possible, alarms for leak load limits, specific efficiency limits, and any other desired metrics should be programmed.

Example 10-70

Question

What data acquisition would be required to meet §120.6(e)3 part C, *Measurement or determination of total airflow from compressors in cfm*?

Answer

Part C specifies that two options for monitoring compressed air flow (cfm), either directly or indirectly a maximum of 5 minute intervals. Direct measurements will typically take the form of a probe or inline air flow meter. Indirect calculations would require the use of measurement of true power (kW) and estimating the airflow using compressor performance data. Performance data sheets can be provided by the air compressor manufacturer and/or national industry associations. This performance data is typically provided in the form of tables and/or performance curves. For many air compressors specific efficiency (kW/100 cfm) is a standard unit. The airflow can be indirectly measured by dividing the measured power by the specific efficiency.

Example 10-71

Question

How can the data gathered from equipment required by §120.6(e)3 be used for compressed air energy management?

Answer

Two primary energy benefits of the monitoring requirements come from maintenance of system efficiency and management of leak loads:

1. Compressed air system energy performance is typically quantified by specific power in kW/100 cfm. By monitoring specific power of a compressed air system, efficiency can be continuously managed and optimized over time. Facility staff should use regularly observe specific power monitoring to identify if and when the system begins to operate poorly. Staff should first identify what their compressor or compressed air specific power is after initial commissioning or optimization efforts. This serves as the baseline. If the specific power increases over time, facility staff should investigate or engage with their compressed air service providers to identify the source of performance degradation. This performance degradation will likely take the form of discrete, obvious spikes in specific power due to acute malfunctions or gradual increases over time. These increases in specific power affect the energy used for every cfm at all hours. Therefore, correction of any root causes or recommissioning will save energy for the entire compressed air system under all loads and hours.

2. Compressed air leak loads typically grow over time in most facilities due to unavoidable wear and tear. These leak loads are often 20-40% of total usage, resulting is large excessive, unproductive energy use and reliability issues at compressed air tools and end-uses. Monitoring of airflow in cfm can be used to identify when leak loads have grown to a level that warrants corrective action. Good practice is to target leak loads of about 10% with regular leak repair and management processes.

The monitoring system will allow facilities to observe loads in cfm for their plant. If the cfm monitoring shows that loads are growing over time without any corresponding production or operational changes, leak loads are likely growing. This leak load, energy use, and cost can be quantified by observing the cfm growth. This provides facility staff operational and financial justification for correcting leaks as well as showing the returns for doing so. Another useful approach for determining leak loads is to observe cfm trends during after-hours, breaks, or weekends with the compressors still on. This can very clearly show leak loads since most or all the output will likely be going to leaks if there are no tools or machines in use.

The monitoring system may be configured to flag or send alerts to facility staff when issues such as these are automatically identified by the monitoring system. This is optional and dependent of the monitoring system capabilities but can help streamline the process for many facilities.

10.8.2.4 **Leak Testing**

§120.6(e)4

This requirement targets the quality installation of new compressed air piping in both new construction and additions and alterations of existing piping. Piping greater than 50 adjoining feet in length shall be pressure tested to ensure minimal leakage rates after installation. The piping shall be pressurized to the system design pressure and help for at least 30 minutes without losing pressurization as indicated by test gauges measuring directly at the tested length of piping.

The pressure test should use dial pressure gauges conforming to the California Plumbing Code sections 318.3, 318.4, and 318.5. In the 2019 California Plumbing Code, these specify the following:

- Required pressure tests exceeding 10 psi (69 kPa) but less than or equal to 100 psi (689 kPa) shall be performed with gauges of 1 psi (7 kPa) incrementation or less.
- Required pressure tests exceeding 100 psi (689 kPa) shall be performed with gauges incremented for 2 percent or less of the required test pressure.
- Test gauges shall have a pressure range not exceeding twice the test pressure applied.

Piping less than or equal to 50 feet in adjoining length and connections shall be pressurized and manually inspected with a leak-detecting fluid.

In either case, if the piping fails to maintain pressure or leaks are detected, repairs or corrections must be made to the piping to eliminate the leak sources prior to re-testing.

The purpose of this requirement is to ensure quality installation of new piping. In many cases, it can be extremely difficult and disruptive to repair leaks in hard-toreach, mission-critical piping once a system is commissioned and operational. Leak testing is necessary to ensure these issues are avoided at construction and are often standard or best practices by installing contractors.

10.8.2.5 **Pipe Sizing**

§120.6(e)5

This requirement targets the performance of a compressed air systems to minimize frictional losses.

For service line piping a minimum pipe diameter of ¾ inch is required. Flexible hoses and tubing at the end-uses are not covered by this requirement.

For compressor room interconnection, main header piping, and distribution piping there are two paths for compliance:

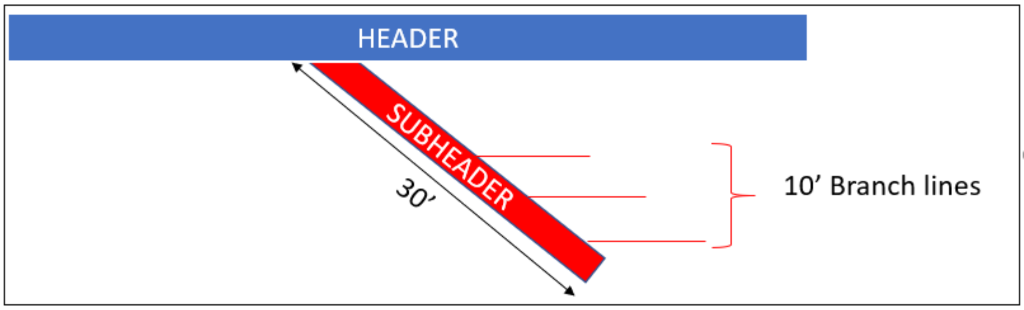
1. Piping section average velocity:
    - Interconnection and main header piping shall be sized so that at coincident peak flow conditions, the average velocity in the segment of pipe is no greater than 20 ft/sec.
    - Distribution piping shall be sized so that at coincident peak flow conditions, the average velocity in the segment of pipe is no greater than 30 ft/sec.
2. Piping total pressure drop:
    - Frictional pressure loss at coincident peak loads are less than 5 percent of operating pressure between the compressor and end-use or end-use regulator.

For the piping total pressure drop pathway, it should be noted that the pressure drop is based on piping frictional pressure losses and does not include pressure drops across components such as dryers, flow controllers, regulators, and other non-piping components.

Example 10-72

Question

An existing compressed air has installed a new 30-foot subheader with three 10-foot branch lines, what must be done to ensure this additional compressed air piping will comply?



Answer

This system is installing a total of 60 feet of adjoining compressed air system piping. To comply with §120.6(e)4, the section of piping must be isolated and undergo pressure testing for a minimum of 30 minutes with no perceptible drop in pressure. To comply with §120.6(e)5, the service line piping must have a minimum inner diameter of ¾ inch and must comply with either §120.6(e)5B OR §120.6(e)5C.

§120.6(e)5B requires a calculation of pipe size based on a targeted maximum pipe velocity. Inner diameter can be estimated through the following equation

$$d = \sqrt{\frac{576}{\pi}\frac{Q}{v}}$$

Where:

v = velocity of air in feet per **minute**

d = inside diameter of pipe in inches

Q = Maximum expected flowrate through pipe in cubic feet per minute

For the section of piping above, to minimize frictional losses by sizing the pipe for a maximum of 30 ft/second (1800 feet/minute), the resultant diameter would be 1.01 inches. Rounding to the most appropriate diameter would be 1 inch (inner diameter) pipe.

§120.6(e)5C would require the calculation of the total piping pressure drop. To satisfy this requirement the piping frictional pressure loss would need to be lower than 5 percent of the operating pressure at coincident peak load. This can be in the form of compressed air modeling software or hand calculations. Hand calculations typically reference available reference tables from handbooks and manufacturers.

Examining the section of pipe added above, the pressure loss over the section of pipe can be estimated using reference tables that will typically provide pressure loss (psi) per 100 ft of pipe. Assuming the added section is the section of pipe furthest from the compressor, we can estimate the pressure loss introduced by the added pipe.

Assuming:

Line Pressure = 100 PSIG

Air Flow = 10 CFM

Pipe Diameter = 1/2 inch

Online reference tables yield a pressure loss of 0.4 psi per 100 foot of schedule 40 steel pipe. As all branch lines are equivalent in length and further assuming all have an equivalent diameter (and area), then the length of pipe the pressure drop will need to be calculated for is 40 feet (30 feet + 10 feet). This would yield the following pressure loss:

$$p = \frac{0.4}{100} * 40$$

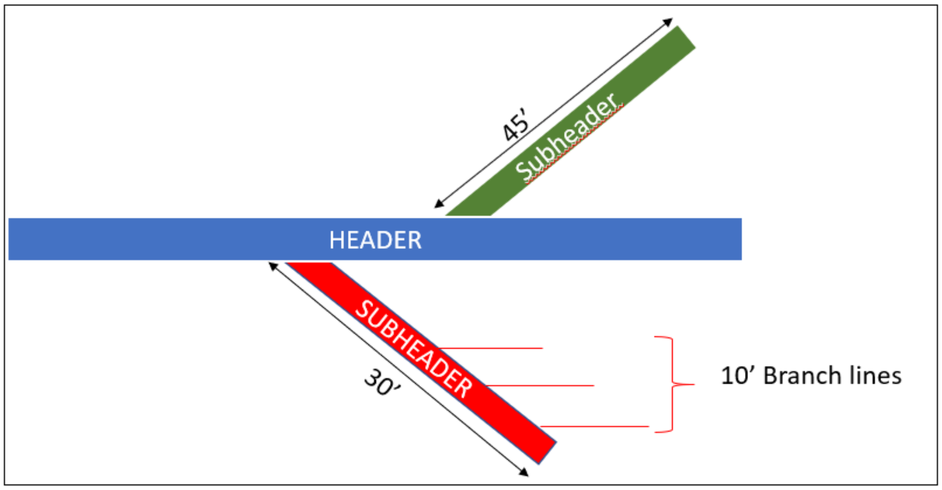
$$p = 0.16$$

The resultant pressure loss of the section of pipe added would be 0.16 psi. Again, assuming this section of the pipe is the furthest from the compressor, a similar calculation would need to be performed across the total length of existing pipe leading to the end of the 10 foot branch line. To meet the requirements of §120.6(e)5C the total pressure loss calculated in this fashion should not exceed 5 percent of the operating pressure. E.g., for a given operating pressure of 115 psig, the total frictional pressure loss calculated would need to be less than 5.75 psig.

Example 10-73

Question

Using the distribution system from the previous example, an additional 45-foot subheader in a separate location of the compressed air distribution network. What testing and pipe sizing requirements will apply?



Answer

The section of piping from the previous example must still comply with §120.6(e)4 and §120.6(e)5A and §120.6(e)5B or §120.6(e)5C.

For the situation above when installing a new 30 foot subheader and three 10 foot branch lines, the maximum velocity would be 30 feet/second. To determine the inside diameter of the pipe the only information that will need to be referenced is the pressure loss due to flow through pipes which

As the additional 45-foot subheader is a separate section of piping in the same network but adjoining to an existing header that will not be altered, only §120.6(e)4 will apply. Furthermore, pressurized leak testing is not required on this section of piping and connections may be tested with noncorrosive leak-detecting fluid or other leak detecting methods.

10.8.3 **Compressed Air** System Acceptance

§120.6(e)6

New systems and altered systems that are subject to the trim compressor requirements of §120.6(e)1, staging control requirements of §120.6(e)2, or monitoring requirements of §120.6(e)3 must be tested per NA7.13.

10.8.4 Prescriptive Measures

§140.9

There are no prescriptive measures for compressed air systems.

10.8.5 Additions and Alterations

There are six requirements in this section for compressed air systems additions and alterations:

a. Trim compressor and storage - §120.6(e)1,

b. Controls - §120.6(e)2,

c. Monitoring - §120.6(e)3,

d. Leak testing of compressed air piping - §120.6(e)4,

e. Pipe sizing - §120.6(e)5, and

f. Acceptance - §120.6(e)6.

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

These requirements do not apply to:

- 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, requirement §120.6(e)1 for trim compressor size and storage do not apply when the total combined added or replaced compressor horsepower is less than the average per-compressor horsepower of all compressors in the system, or when all added or replaced compressors are variable speed drive compressors and the compressed air system includes primary storage of at least one gallon per actual cubic feet per minute (acfm) of the largest trim compressor.

For alterations or additions to an existing compressed air system, requirement §120.6(e)2 for controls applies only when the compressed air system has three or more compressors and a combined horsepower rating of more than 100 horsepower.

For alterations or additions to an existing compressed air system, requirement §120.6(e)3 for monitoring applies only when the compressed air system has a combined horsepower rating of more than 100 horsepower. Monitoring equipment can be used to identify changes in system performance for the purpose of identifying inefficiencies such as air leaks.

For alterations or additions to an existing compressed air system, requirement §120.6(e)4 for leak testing of compressed air piping applies to added or altered piping with leak testing requirements based on the length of adjoining pipe added or altered.

For alterations or additions to an existing compressed air system, requirement §120.6(e)5 for pipe sizing only applies when the added or altered piping is greater than 50 adjoining feet in length. For service line piping a minimum pipe diameter of ¾ inch is required. Flexible hoses and tubing at the end-uses are not covered by this requirement. For compressor room interconnection, main header piping, and distribution piping there are two paths for compliance as described in 10.8.2.5.

For alterations or additions to an existing compressed air system, requirement §120.6(e)6 for compressed air system acceptance testing will apply based on which other compressed air system requirements are required. For example, if an addition or alteration to a compressed air system triggers the controls requirement in 120.6(e)3, then the acceptance requirements associated with controls are also required.

Example 10-74

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

Each subsection of §120.6(e) would need to be evaluated to determine which requirements are applicable and which requirements are not.

Requirements for trim and compressor and storage are not applicable because 50 hp is Less than the average per-compressor horsepower of all compressors in the system. Requirements for controls in §120.6(e)2 are not applicable because the compressed air system has less than three compressors. Requirements for leak testing of compressed air piping in §120.6(e)4 and pipe sizing in §120.6(e)5 are not applicable because compressed air piping is not being added.

Requirements for monitoring in §120.6(e)3 are required because the system has a combined horsepower rating greater than 100 hp. Additionally, compressed air system acceptance requirements in §120.6(e)6 pertaining to the monitoring system will also be applicable.

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 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 shutoff 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 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 shutoff 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 shutoff devices.

Installed dampers can be interlocked with the gas valve so that the damper closes and inhibits airflow 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 shutoff 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:

1. The fan motor shall be driven by a variable-speed drive, or
2. 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**

§120.6(d)3

Newly installed process boilers with an input capacity of 5 MMBtu/h (5,000,000 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 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 airflow rate is slightly higher than the stoichiometric air-fuel ratio. However, common practice almost always relies on excess air to ensure 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 the firing range of a burner. Maintaining low excess air levels at all firing rates provides significant fuel and cost savings while maintaining a safe margin of excess air to insure complete combustion.

10.9.3 Prescriptive Measures

There are no prescriptive measures for process boilers.

10.10 Elevators

10.10.1 Overview

Section 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. Elevators in healthcare facilities have an exception to the requirements of this section.

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/ft^2). This power density is determined by taking the total wattage of the elevator lighting and dividing by the floor 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 10-75

Question

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

Answer

Yes. (9 lamps) x (3 watts/lamp) = 27 watts. The square footage of the cab is (6 ft) x (8 ft) = 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 more than 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 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 an alteration.

10.11 Escalators and Moving Walkways

10.11.1 Overview

Section 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 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. To comply with the Energy Codes, the escalator or moving walkway must also comply 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/s^2 (1.0 ft/s^2).
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 comb plate.
    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 comb plate.
6. Automatic deceleration shall not occur before a specific period of time has elapsed since the last passenger detection that is greater than three 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

From 6.1.4.1 of ASME A17.1-2013/CSA B44-13, the maximum speed of escalators cannot be more than 0.5 m/s (100 ft/min), measured along the centerline of the steps in the direction of travel.

From 6.2.4.1 of ASME A17.1-2013/CSA B44-13 the maximum speed of a moving walkway depends on the maximum slope at any point on the treadway as listed below:

1. Max slope of 0-8 degrees: 0.9 m/s (180 ft/min)
2. Max slope above 8 & less than 12 degrees: 0.7 m/s (140 ft/min)

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 Code 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 operates only 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 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.

10.12 Controlled Environment Horticulture

10.12.1 Overview

Section 120.6(h) sets efficiency standards for controlled environment horticulture (CEH) spaces. These standards are divided into indoor facilities which do not use sunlight and greenhouse CEH facilities. For indoor facilities, requirements exist for lighting technology and dehumidification. For greenhouses, there are requirements for lighting and building envelope materials. These requirements impact all newly constructed CEH spaces. These requirements will help reduce the energy usage of horticulture facilities that are becoming increasingly popular business operations across the state. These requirements not dependent on the type of crop is being grown in the facility.

10.12.2 Mandatory Measures

§120.6(h)

There are two main mandatory requirements in this section:

a. Indoor horticultural growing facilities - *§120.6(h)1 - 3*, and
b. Greenhouse facilities - *§120.6(h)4.*

10.12.2.1 Indoor growing dehumidification

§120.6(h)1

Section 120.6(h)1 sets efficiency standards for dehumidification equipment for indoor facilities. Compliance can be reached with one of the four following pathways.

- Dehumidifiers subject to federal appliance standards that comply with the federal performance and testing requirements; or
- Integrated HVAC system with on-site heat recovery designed to fulfill at least 75 percent of the annual energy for dehumidification reheat; or
- Chilled water system with on-site heat recovery designed to fulfill at least 75 percent of the annual energy for dehumidification reheat; or
- Solid or liquid desiccant dehumidification system for system designs that require dewpoint of 50°F or less.

Example 10-76

Question

How do I find a dehumidifier that meets the federal regulations?

Answer

The Department of Energy's Compliance Certification Management System maintains a database of products that have been certified to meet the federal requirements.

10.12.2.2 Indoor growing horticultural lighting

§120.6(h)2

Section 120.6(h)2 requires indoor CEH growing spaces with more than 40 kilowatts (kW) of horticultural lighting load to have electric lighting systems used for plant growth and maintenance that meet all of the below requirements:

1. Luminaires with removable lamps shall contain lamps with a rated photosynthetic photon efficacy (PPE) of at least 1.9 micromoles per joule (µm/J). All other luminaires shall have a rated PPE of at least 1.9 µm/J for the wavelengths between 400 and 700 nm. Photosynthetic photon efficacy is photosynthetic photon flux for the wavelengths between 400 nm and 700 nm divided by input electric power in units of micromoles per second per watt, or micromoles per joule as defined by ANSI/ASABE S640.

   Note that some manufactures will also publish the total photon flux which includes the wavelengths outside of the 400 to 700 nm range. This higher photon flux value should not be used to calculate PPE as it includes wavelengths that are not photosynthetically active as defined by ANSI/ASABE S640.

2. Time-switch lighting controls shall be installed and comply with Section 110.9(b)1, the lighting control acceptance requirements of Section 130.4(a)4, and applicable sections of NA7.6.2.
3. Multilevel lighting controls shall be installed and comply with Section 130.1(b).

10.12.2.3 Indoor growing electrical power distribution

§120.6(h)3

Electrical power distribution systems servicing indoor CEH spaces must be designed so that a measurement device is capable of monitoring the electric energy usage of aggregate horticultural lighting load.

A 40 kW threshold equates to roughly 800-1,000 square feet of canopy for an indoor grower using 1,000 watt high pressure sodium lights every 20-25 square feet. For compliance with existing California Department of Food & Agriculture CalCannabis regulations, growers must submit canopy size calculations and a lighting diagram for indoor and mixed-light licensing types. The lighting diagram includes the maximum wattage for each light so through this diagram one can determine total horticulture lighting load.

Luminaires with a PPE of 1.9 micromoles per joule will largely be LED with a tuned spectrum for growing plants or double-ended high-pressure sodium (HPS) lamps . LEDs are becoming more popular in the market as growers become familiar with their impacts on plant quality and yield. A photosynthetic photon efficacy (PPE) of 1.9 micromoles per joule for lamps can be met by high efficiency double-ended high-pressure sodium (HPS) lamps. HPS technology has been in the horticulture market for considerable time and use of highly efficient options will allow growers to use familiar technology while still achieving energy savings.

Example 10-77

Question

How do I find the photosynthetic photon efficacy of a particular lighting fixture or lamp?

Answer

A variety of options exist to determine the PPE of a given product. The DesignLights Consortium (DLC) Qualified Products List notes the PPE level of over 415 products from popular lighting manufacturers. If your product is not found in this listing, your manufacturer’s product specification may list PPE. Additionally, there are industry test procedures that can assist in the determination of a PPE level. IES LM-46-04: IESNA Approved Method for Photometric Testing of Indoor Luminaires Using High Intensity Discharge or Incandescent Filament Lamps is the most appropriate test standard for lamps and can be used to report PPE when certain data gaps are filled. The information needed to conduct the test procedure for PPE is found in existing IES standards, LM-51 and LM-79.

10.12.2.4 Conditioned greenhouse building envelope

§120.6(h)4

Section 120.6(h)4 provides separate building envelope requirements for conditioned greenhouses. Roof/ceiling and wall insulation must meet assembly requirements of Section 120.7. Buildings have different U-factor requirements for roof/ceiling and wall insulation depending on whether they are constructed with metal or wood products and their climate zones.

Non-opaque wall and non-opaque roof assemblies shall have greenhouse glazing with two or more glazing layers separated by air or gas fill.

Examples of non-opaque glazing products that meet these requirements include double pane glass, double and triple wall polycarbonate, and double film polyethylene.

10.12.2.5 Conditioned greenhouse space-conditioning systems

§120.6(h)5

Section 120.6(h)5 requires that space-conditioning systems used in conditioned greenhouses for plant production must meet requirements applicable to the systems

10.12.2.6 Greenhouse horticultural lighting

§120.6(h)6

Pertaining to Section 120.6(h)6, greenhouse growing spaces with more than 40 kW of horticultural lighting load shall have electric lighting systems used for plant growth and maintenance that meet all of the below requirements:

1. Luminaires with removable lamps shall contain lamps with a lamp photosynthetic photon efficacy (PPE) of at least 1.7 micromoles per joule (μm/J); all other luminaires shall have a luminaire photosynthetic photon efficacy of at least 1.7 μm/J.

2. Time-switch lighting controls shall be installed and comply with Section 110.9(b)1, Section 130.4(a)4, and applicable sections of NA 7.6.2.

3. Multilevel lighting controls shall be installed and comply with Section 130.1(b).

A lamp and luminaire PPE of 1.7 μm/J can be met by high efficiency double-ended high-pressure sodium (HPS) and ceramic metal halide (CMH) lamps. HPS and CMH technology have been in the horticulture market for considerable time and use of highly efficient options will allow growers to use familiar technology while still achieving energy savings. LED technology is also an option for attaining a luminaire PPE of 1.7 μm/J.

Example 10-78

Question

How do these Title 24, Part 6 requirements interact with any CalCannabis requirements?

Answer

Title 24, Part 6 requirements for CEH greenhouses and indoor growing spaces apply to all building spaces that "are dedicated to plant production by manipulating indoor environmental conditions, such as through electric lighting, mechanical heating, mechanical cooling, or dehumidification." The Title 24, part 6 requirements apply regardless of the type of crop grown in these spaces. Title 24, part 6 defines a CEH indoor growing space as having a skylight area to roof area ratio less than 50 percent. A CEH greenhouse has a skylight area to roof area ratio of 50 percent or greater. Thus, the Title 24 definitions of greenhouse and indoor growing space are solely determined by the fraction of roof area that has glazing and is not affected by how much electric lighting is used in the space.

CalCannabis (operated by the Department of Cannabis Control) grants licenses for cannabis growing. Different types of licenses are based on factors such as lighting density and light deprivation. As of 2021, CalCannabis defines Indoor Cultivation as "Cultivation of cannabis within a permanent structure using artificial light exclusively, or within any type of structure using artificial light at a rate above 25 watts per square foot." Therefore, a CalCannabis indoor cultivation license applies to indoor spaces with no skylights and any space, even greenhouses, with more than 25 Watts of electric lighting per square foot of growing area. A CalCannabis mixed light cultivation license makes use of some amount of sunlight and must have no more than 25 Watts of electric lighting power installed per square foot of growing area.

The CalCannabis definitions of "indoor growing" and "mixed light" for licensure are not equivalent to the "indoor growing" and "conditioned greenhouse" definitions used for obtaining building permits under Title 24, Part 6 and are not relevant to Title 24 compliance.

10.12.3 Prescriptive Measures

There are no prescriptive measures for controlled environmental horticulture.

10.12.4 Additions and Alterations

Section 120.6(h) applies to major retrofits of all CEH spaces. Alterations to indoor or greenhouse horticulture lighting systems that increase lighting wattage or that include adding, replacing, or altering 10 percent of more of the horticulture luminaires servicing an enclosed space must meet the applicable requirements of Section 120.6(h).

Greenhouses being converted into conditioned greenhouses or additions to conditioned greenhouses meet requirements of 120.6(h)4. A conditioned greenhouse is an enclosed space that is provided with wood heating, mechanical heating that has a capacity exceeding 10 Btu/hr-ft2 or mechanical cooling with capacity exceeding 5 Btu/hr-ft2. In conditioned greenhouses, space-conditioning systems used for plant production shall comply with all applicable Title 24, Part 6 requirements.

Example 10-79

Question

Alterations that change the occupancy group of a building do trigger the CEH requirements. Occupancy groups are defined in Chapter 3 of Title 24, Part 2. One common change of building type that would trigger the requirements is converting an indoor warehouse into a CEH grow facility of over 10 percent of the luminaires in my greenhouse or indoor CEH facility. Do I need to meet the lighting efficiency standards in 120.6(h)2 or 6?

Answer

Lamp replacements do not trigger the horticulture lighting efficacy alteration requirements. Only replacement of 10 percent or more of the horticulture luminaires serving an enclosed space trigger the lighting efficacy requirement. When replacing 10 percent or more of the luminaires in an enclosed space, only the replacement luminaires need to meet the applicable requirements.

Example 10-80

Question

If I replace the lamps of over 10 percent of the luminaires in my greenhouse or indoor CEH facility, do I need to meet the lighting efficiency standards in 120.6(h)2 or 6?

Answer

Lamp replacements do not trigger the horticulture lighting efficacy alteration requirements. Only replacement of 10 percent or more of the horticulture luminaires serving an enclosed space trigger the lighting efficacy requirement. When replacing 10 percent or more of the luminaires in an enclosed space, only the replacement luminaires need to meet the applicable requirements.

10.13 Steam Traps

10.13.1 Overview

§120.6(i) applies to new industrial facilities where the steam trap operating pressure is greater than 15 psig and the total combined connected boiler input rating is greater than 5 million Btu/hr. §120.6(i) also applies to new steam traps added to support new, non-replacement, process equipment in existing industrial facilities where the steam trap operating press is greater than 15 psig and the total combined connected boiler input rating is greater than 5 million Btu/hr. Existing steam traps that are being replaced due to failure are not required to meet §120.6(i). An exception is given to steam traps where steam is diverted to a steam system of lower pressure for use when the steam trap fails open, as might be found in a cascading process where steam is flashed to a lower pressure process. These traps would be exempted from the requirements of this section. However, the traps on the lowest pressure section which are discharging to the drain or condensate system are still covered.

10.13.2 Mandatory Measures

§120.6(i)

There are four main mandatory requirements in this section:

a. Central steam trap fault detection and diagnostic monitoring - §120.6(i)1,

b. Steam trap fault detection - §120.6(i)2,

c. Steam trap strainer installation - §120.6(i)3, and

d. Steam trap system acceptance - §120.6(i)4.

10.13.3 Central Steam Trap Fault Detection and Diagnostic Monitoring

§120.6(i)1

There are two requirements for steam trap monitoring systems:

1. Provide status updates of all fault detection sensors to the central monitoring station at least every 8 hours, and
2. Automatically display an alarm that identifies which steam trap has a fault.

10.13.4 Steam Trap Fault Detection

§120.6(i)2

This requirement indicates that steam traps must have automatic fault detection sensors that can communicate with the central steam trap monitoring system described in §120.6(i)1.

10.13.5 Steam Trap Strainer Installation

§120.6(i)3

This requirement identifies two options for meeting steam trap strainers:

- The strainer and blow-off valve must be integrated into the steam trap, or
- The steam trap must be installed within 3 feet downstream of the strainer and blow-off valve.

10.13.6 Steam Trap System Acceptance

§120.6(i)3

Before an occupancy permit is granted for compliance with §120.6(i) the equipment and systems should be certified as meeting the Acceptance Requirement for Code Compliance, as specified by NA7.19. A Certificate of Acceptance shall be submitted to the enforcement agency.

10.13.7 Prescriptive Measures

There are no prescriptive measures for steam traps.

10.13.8 Additions and Alterations

Section 120.6(i) applies to new steam traps added to support new, non-replacement, process equipment in existing industrial facilities where the steam trap operating press is greater than 15 psi and the total combined connected boiler input rating is greater than 5 million Btu/hr. If new steam lines including steam traps are added as part of an addition or alteration to an industrial facility, these steam traps are required to meet §120.6(i) as described above.

Example 10-81

A manufacturing facility has a 100 psig steam system served by three boilers, each rated at 3 million Btu/hr input rating and 2.5 million Btu/hr output rating. Two boilers are active, and one boiler is a back-up.

Question 1: What is the combined total rating of the system?

Answer: The combined input rating is 9 million Btu/hr; it includes all boilers physically connected to the steam piping system including back-ups. Since the input rating of this steam system is greater than 5 million Btu/hr, it is large enough for the steam trap FDD requirements to apply.

Question 2: For the previously described facility, if an existing 100 psi steam jacketed kettle is replaced with a new steam jacketed kettle and steam trap, is the steam trap on this system required to be accompanied by a strainer with blow-off valve and steam trap FDD device?

Answer: No. This steam trap supports replacement of existing equipment.

Question 3: For the previously described facility, if a new 100 psi steam jacketed kettle is added to this facility, is the steam trap on this system required to be accompanied by a strainer with blow-off valve and steam trap FDD device?

Answer: Yes. This steam trap is part of new process equipment installation. A strainer with blow-off valve would be required to protect the steam trap, along with a steam trap FDD device and the central console for displaying the state of all steam traps required to have FDD monitoring.

Question 4: For the previously described facility, a new hot water generator is installed. The main steam pressure is reduced to 5 psig upstream of the inlet of the hot water generator. Would a strainer with blow-off valve and steam trap FDD device be required for this system?

Answer: No. the steam trap in question is served by 5 psig steam, this is less than the 15 psig operating pressure threshold for each steam trap at which this requirements applies.