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Covered Process

Overview

This chapter of the Nonresidential Compliance Manual addresses covered processes for the Energy Code (Section 110.2, Section 120.3, Section 120.6, Section 140.1, Section 140.9, and Section 141.1).

Organization and Content

This chapter is organized as follows:

- Overview
- Enclosed Parking Garages
- Commercial Kitchens
- Computer Rooms
- Commercial Refrigeration
- Refrigerated Warehouses
- Laboratory Systems
- Compressed Air Systems
- Process Boilers
- Elevators
- Escalators and Moving Walkways
- Controlled Environment Horticulture
- Steam Traps
- Process Pipe Insulation

Compliance Forms

Please refer to Chapter 10.1.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

What's New for 2025

Significant changes for covered process in the 2025 update to the Energy Code 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:

- There are new mandatory pipe insulation requirements for process piping supported in Section 120.3. These new requirements are covered in Process Pipe Insulation.

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

- Commercial kitchens (mandatory measures)
    - Electrical panels for kitchens must be rated to serve a minimum of 800 connected amps.
    - Each cookline appliance must have a 50A branch circuit available to it.
- Refrigerated Warehouses

- Minimum evaporator efficiency and maximum evaporator static pressure drop requirements were added to Section 120.6(a)3.
- Controlled environment horticulture (mandatory measures)
    - Electric lighting for growing plants have updated requirements for the photosynthetic photon efficacy (PPE) rating.
- Laboratory systems (prescriptive measures)
    - Added unoccupied setback guidelines.
    - Added an option for simplified exhaust controls.
    - Reduced exceptions to reheat limitations.
    - Added Occupancy L to laboratory requirements.

Enclosed Parking Garages

Overview

Please refer to Chapter 10.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Mandatory Measures

Please refer to Chapter 10.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Minimum Fan Power Reduction

Please refer to Chapter 10.2.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

CO Sensor Number and Location

Please refer to Chapter 10.2.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

CO Sensor Minimum Requirements

Please refer to Chapter 10.2.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.2.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Commercial Kitchens

Overview

There are mandatory requirements for commercial kitchens in Section 120.6(k) to ensure that kitchen infrastructure makes it easier to support electrification efforts. There are exceptions for healthcare facilities and kitchens that are already all-electric. Three types of commercial kitchen are defined in Section 100.1:

- A *full-service commercial kitchen* is defined as "a kitchen dedicated to an establishment that offers table service by waitstaff."

- An *institutional commercial kitchen* is defined as "a kitchen dedicated to a foodservice establishment that provides meals at institutions including schools, colleges and universities, hospitals, correctional facilities, private cafeterias, nursing homes, and other buildings or structures in which care or supervision is provided to occupants."
- A *quick-service commercial kitchen* is defined as "a kitchen dedicated to an establishment primarily engaged in providing fast food, fast casual, or limited services. Food and drink may be consumed on premises, taken out, or delivered to the customer's location."

There are four prescriptive energy-saving measures associated with commercial kitchen ventilation in Section 140.9(b). These four measures address:

- Direct replacement of exhaust air limitations.
- Type I exhaust hood airflow limitations.
- Makeup and transfer air requirements.
- Commercial kitchen system efficiency options.

Mandatory Measures

Reference: Section 120.6(k)

Installed appliances and equipment must meet the mandatory requirements of Section 110.1 and Section 110.2, respectively. Commercial kitchens must meet the mandatory requirements of Section 120.6(k).

Electric Commercial Kitchen Requirements

Section 120.6(k) sets mandatory requirements for commercial kitchens to have the proper electrical infrastructure to allow for an easier conversion to electrified cooklines in the future. The impacted kitchen types are quick-service commercial kitchens and institutional commercial kitchens. Kitchens in healthcare facilities and kitchens that are already planned to be all-electric are exceptions to 120.6(k).

Commercial kitchens shall meet the following mandatory requirements:

- Quick-service commercial kitchens and institutional commercial kitchens shall be serviced by a panel that has a minimum capacity of 800 connected amps.
- Quick-service commercial kitchens and institutional commercial kitchens shall have at least one dedicated branch circuit capable of supporting a 50A outlet in the kitchen.
- For commercial kitchens, the electrical service panel shall be sized to accommodate an additional 280 volt or 240 volt 50-amp breaker.

This outlet would be accessible to cookline appliances such as stoves, fryers, griddles ovens, etc. Thus, the outlet should be located so that it would be reasonably close to one of the cookline appliances. The location of the 50 Amp receptacle will typically be located directly behind the location of one of the current cookline appliances. This branch circuit wiring should be connected to overcurrent protection devices (circuit breakers) that are two-pole and sized to protect the branch circuit wiring.

The requirements for branch circuit conductors and electrical service panels are the expected requirements for an all-electric commercial kitchen. Insufficient electrical infrastructure is much more difficult to upgrade and can be costly. These requirements also cover the main electrical service panel to ensure that there will be enough power provided to the building, since this issue has been identified for all-electric retrofits.

Example 10-1

Question:

Can these facilities still install gas cooking equipment?

Answer:

Yes, quick-service commercial kitchens and institutional commercial kitchens are allowed to install gas cooking equipment as long as they meet the mandatory requirements to establish electric-ready infrastructure.

Prescriptive Measures

Kitchen Exhaust Systems

Reference: Section 140.9(b)1

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

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

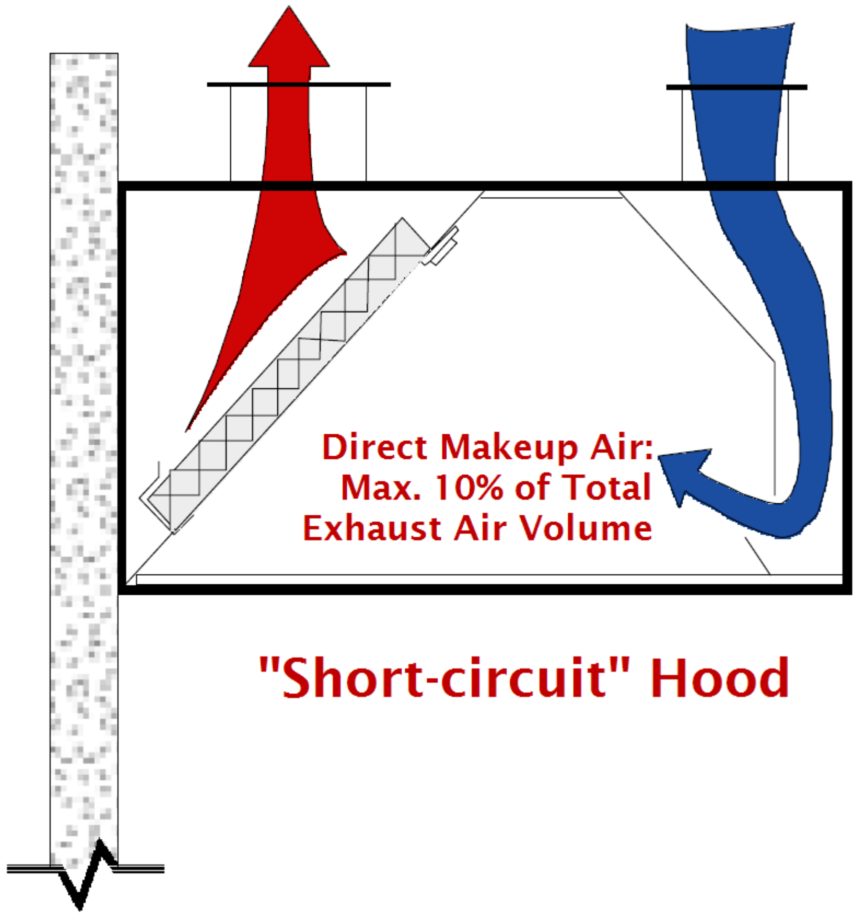
Limitation of Short-Circuit Hoods

Reference: Section 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 10-1: Short-Circuit Hood



Source: California Energy Commission

Maximum Exhaust Ratings for Type I Kitchen Hoods
Reference: Section 140.9(b)1B

The Energy Code also limits the amount of exhaust for Type I kitchen hoods based on Table 140.9-C, 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 increase energy consumption and increase energy use for conditioning of the makeup air.

There are two exceptions for this requirement:

- Exception 1 to Section 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.
- Exception 2 to Section 140.9(b)1B: Existing hoods that are not being replaced as part of an addition or alteration.

The values in Table 140.9-C 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 Code to exempt small restaurants.

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

Kitchen Ventilation

Reference: Section 140.9(b)2

This section covers two requirements:

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

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

- *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.
- *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.

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

Reference: Section 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:

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

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

To calculate the available transfer air:

- Calculate the minimum outside air (OA) needed for the spaces that are adjacent to the kitchen.
- From the amount calculated in the bullet point above, 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.
- From the amount calculated in the bullet point above, 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.

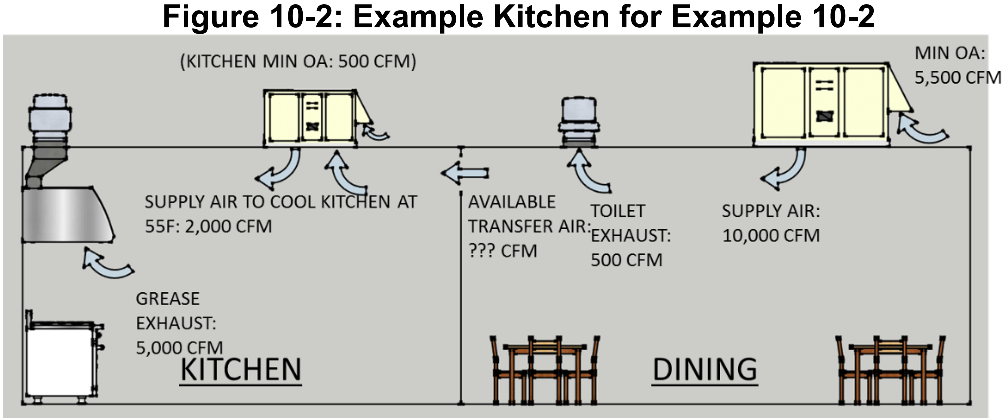
While the requirement to use available transfer air refers only to "adjacent spaces," available transfer air can come from any space in the same building as the kitchen. A kitchen on the ground floor of a large office building, for example, can draw transfer air from the return plenum and the return 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 10-2

Question:

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

Figure 10-2: Example Kitchen for Example 10-2



Source: California Energy Commission

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

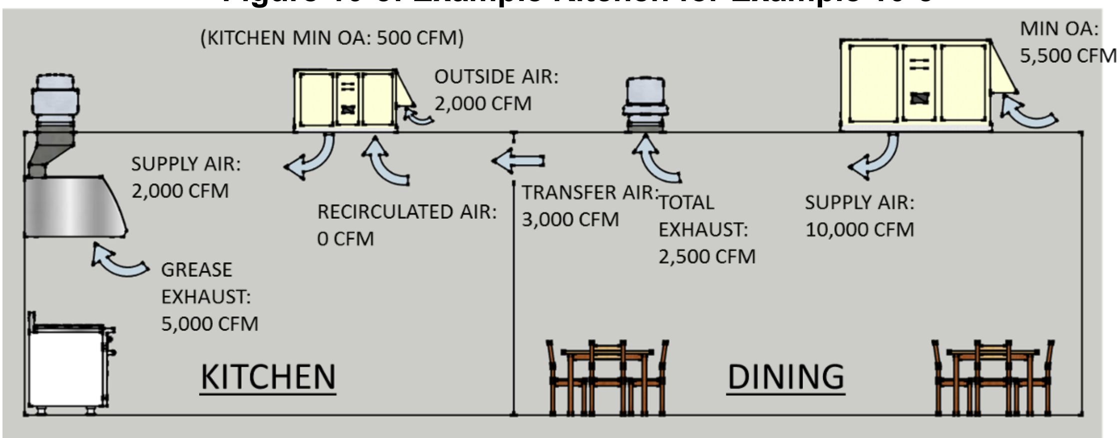
The remaining 5,000 cfm of air is available transfer air.

Example 10-3

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 Section 140.9(b)2A?

Figure 10-3: Example Kitchen for Example 10-3



Source: California Energy Commission

Answer:

Yes. This example meets the first provision of Section 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.

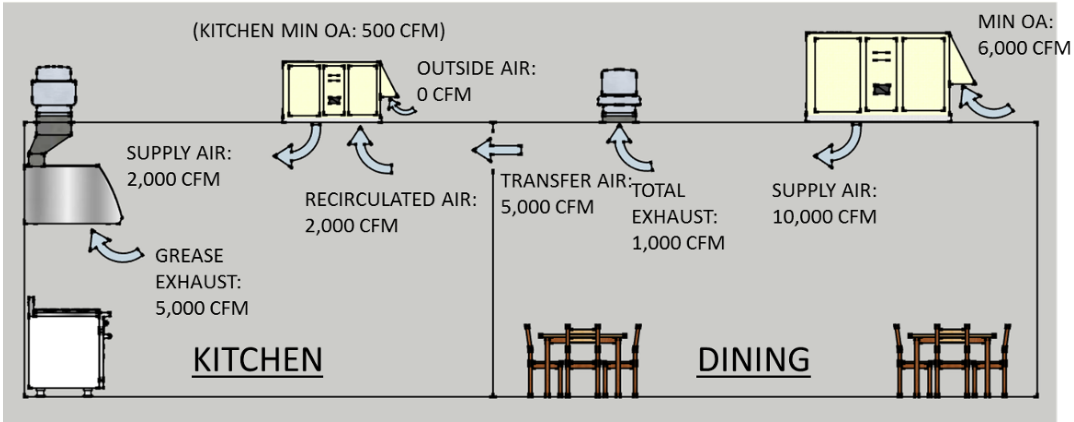
Although this is allowed under Section 140.9(b)2Ai, this is not the most efficient way to condition this kitchen, as demonstrated in the next example.

Example 10-4

Question:

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

Figure 10-4: Example Kitchen for Example 10-4



Source: California Energy Commission

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:

- 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.
- 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.

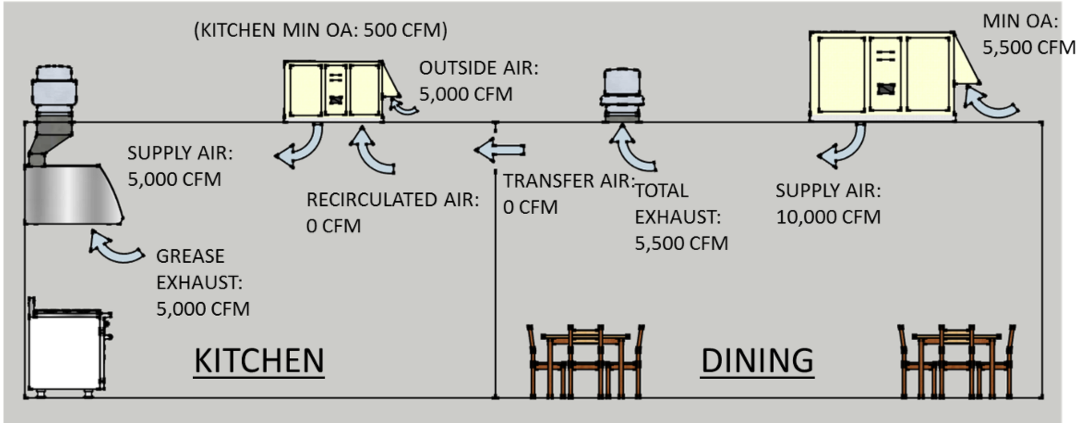
An even more efficient design would be if the kitchen MAU had a modulating OA damper that allowed it to provide up to 5,000 cfm of outside air directly to the kitchen when OA temperature < kitchen space temperature. When OA temperature > kitchen space temperature, then the OA damper on the MAU is shut, 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 10-5

Question:

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

Figure 10-5: Example Kitchen for Example 10-5



Source: California Energy Commission

Answer:

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

- Per Section 140.9(b)2Ai: 2,000 cfm, the supply needed to cool the kitchen (from above example).
- Per Section 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 above example).

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.

Additional Efficiency Measures for Large Kitchens

Reference: Section 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:

- At least 50% of all replacement air is transfer air that would have been exhausted.
- Demand ventilation control on at least 75% of the exhaust air.
- The listed energy recovery devices have a sensible heat recovery effectiveness ≥40% on ≥50% of the total exhaust flow.
- Seventy-five percent or more of the makeup air volume is:
    - Unheated or heated to no more than 60°F.
    - 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.

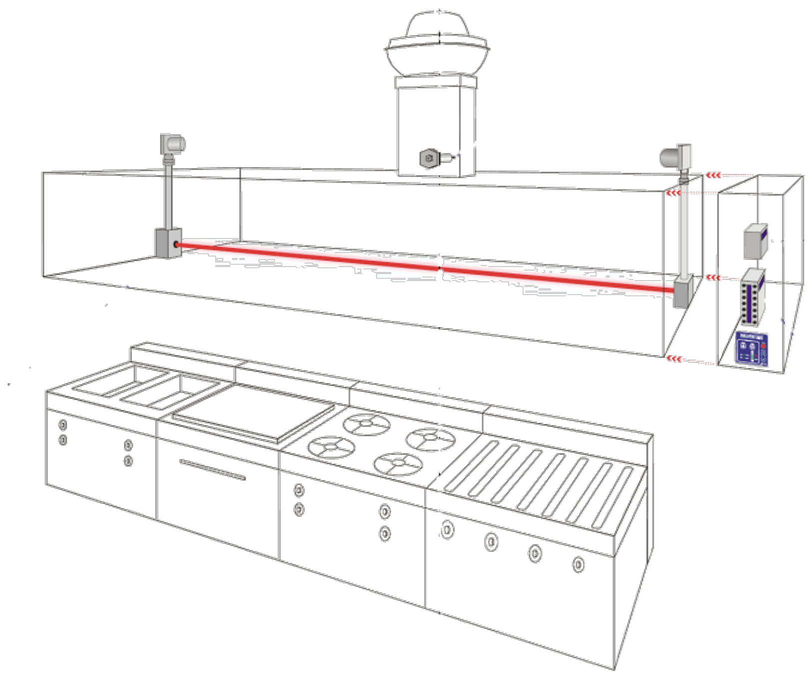
Transfer Air: The concept of transfer air was addressed in the discussion of Section 140.9(b)2A above.

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

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

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

Figure 10-6: 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.

Kitchen Exhaust Acceptance

Please refer to Chapter 10.3.3.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Healthcare Facilities

Please refer to Chapter 10.3.3.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Prescriptive requirements for commercial kitchens covered in Section 140.9 shall apply to additions and alterations to existing kitchens.

Computer Rooms

Overview

Please refer to Chapter 10.4.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Mandatory Measures

Please refer to Chapter 10.4.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Reheat

Please refer to Chapter 10.4.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Humidification

Please refer to Chapter 10.4.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Fan Control

Please refer to Chapter 10.4.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.4.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Economizers

Please refer to Chapter 10.4.3.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Power Consumption of Fans

Please refer to Chapter 10.4.3.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Air Containment

Please refer to Chapter 10.4.3.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Minimum Uninterruptible Power Supply (UPS) Efficiency

Please refer to Chapter 10.4.3.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.4.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Economizers

Please refer to Chapter 10.4.5.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Commercial Refrigeration

Overview

This section addresses Section 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 (Section 100.0 – Section 100.2, Section 110.0 – Section 110.10, Section 120.0 – Section 120.9, Section 130.0 – Section 130.5) and additions and alterations requirements (Section 141.1).

All process piping operating at a temperature below 60°F such as refrigerant piping or chilled water piping shall comply with Section 120.3. New requirements for process pipe insulation are listed in Process Pipe Insulation.

Mandatory Measures and Compliance Approaches

Please refer to Chapter 10.5.1.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Scope and Application

Reference: Section 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). *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 10-6

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 10-7:

Question:

A new retail store with 25,000 square feet of 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.

Condenser Mandatory Requirements

Please refer to Chapter 10.5.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Condenser Fan Control

Please refer to Chapter 10.5.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Condenser-Specific Efficiency

Please refer to Chapter 10.5.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Condenser Fin Density

Please refer to Chapter 10.5.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Adiabatic Condenser Sizing

Please refer to Chapter 10.5.2.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Compressor System Mandatory Requirements

Please refer to Chapter 10.5.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Floating Suction Pressure Controls

Please refer to Chapter 10.5.3.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Liquid Subcooling

Please refer to Chapter 10.5.3.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Compressors for Transcritical CO_2 Refrigeration Systems

Reference: Section 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:

- 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.
- Existing compressor systems that are reused for an addition or alteration.

Refrigerated Display Case Lighting Control Requirements

Please refer to Chapter 10.5.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Refrigeration Heat Recovery

Please refer to Chapter 10.5.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Refrigeration Heat Recovery Design Configurations

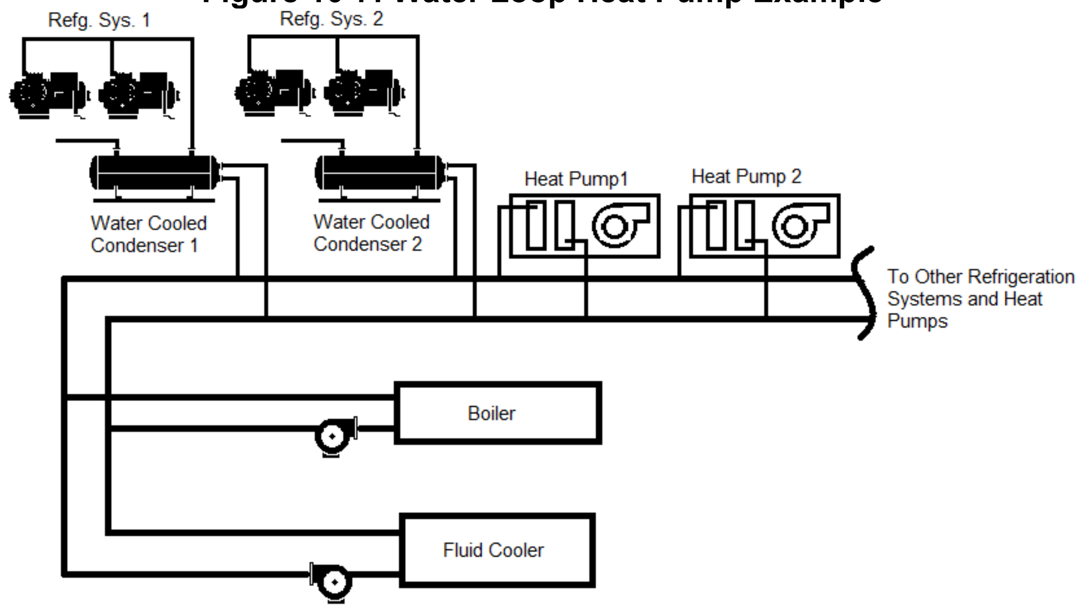
Please refer to Chapter 10.5.5.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Water Loop Heat Pump Heat Recovery

Water-source heat pumps (WLHP) can be used in conjunction with water cooled refrigeration systems, connected to a common water loop as shown in Figure 10-7: Water Loop Heat Pump Example. 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 10-7: Water Loop Heat Pump Example



Source: California Energy Commission

Control Considerations

Please refer to Chapter 10.5.5.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Recovery Coil Design Considerations

Please refer to Chapter 10.5.5.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Transcritical CO_2 Systems

Please refer to Chapter 10.5.6 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Transcritical CO_2 Gas Coolers

Reference: Section 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 Section 120.6(b)5.

Air-Cooled Gas Coolers Restrictions

Please refer to Chapter 10.5.6.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Gas Cooler Sizing

Please refer to Chapter 10.5.6.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Air-Cooled Gas Cooler Sizing

Please refer to Chapter 10.5.6.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Adiabatic Gas Cooler Sizing

Please refer to Chapter 10.5.6.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Fan Control

Please refer to Chapter 10.5.6.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Speed Control

Please refer to Chapter 10.5.6.6 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Subcritical Pressure Control

Please refer to Chapter 10.5.6.7 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Supercritical Pressure Control

Please refer to Chapter 10.5.6.8 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Minimum SCT Set Point

Please refer to Chapter 10.6.1.1.3.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Gas Cooler-Specific Efficiency

Please refer to Chapter 10.5.6.9 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.5.7 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Refrigerated Warehouses

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 and efficiency requirements, 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 (Section 100.0–Section 100.2, Section 110.0–Section 110.10, Section 120.0–Section 120.9, Section 130.0–Section 130.5) and additions and alterations requirements (Section 141.1). These topics are generally addressed in Chapter 3 of this manual.

All process piping operating at a temperature below 60°F such as refrigerant piping or chilled water piping shall comply with Section 120.3. New requirements for process pipe insulation are listed in Process Pipe Insulation.

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. Because 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.

Scope and Application

Reference: Section 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). 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 Section 120.6(a).

Areas within refrigerated warehouses designed solely for quick chilling or quick freezing of products have some exceptions for evaporators and compressor requirements. 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 Section 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 is not the intent of the Energy Code to apply 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 is not 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. As a result, the compressor plant requirements in Section 120.6(a)5C 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. The provisions addressing condensers, subsections 120.6(a)4A, 4B, and 4C, 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 Refrigerated Warehouses.

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 setpoint 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 10-8

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 Section 120.1(a) of the Energy Code?

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, 4D, 4E, 4F, 4G, 4H, 5A, 5B, and 5D. There are exceptions for 3B, 3C, 3D, 3E, 4A, 4B, 4C and 5C.

Example 10-9

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 Section 120.1(a) of the Energy Code?

Answer:

Yes. While the design evaporator capacity is greater than 240 Btu/hr-ft² and the space is used for product pull down for part of the time, the space is also used for holding product after it has been cooled. Accordingly, the space has to comply with the space requirements of Section 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 the mechanical system requirements of Chapter 10.

Example 10-10

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 Section 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 10-11

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:

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

Building Envelope Mandatory Requirements

Please refer to Chapter 10.6.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Envelope Insulation

Please refer to Chapter 10.6.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Underslab Heating Controls

Please refer to Chapter 10.6.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Infiltration Barriers

Please refer to Chapter 10.6.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Automatic Door Closers

Please refer to Chapter 10.6.2.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Acceptance Requirements

Please refer to Chapter 10.6.2.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Mechanical Systems Mandatory Requirements

Overview

Please refer to Chapter 10.6.3.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Evaporators

Please refer to Chapter 10.6.3.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Efficiency

Reference: Section 120.6(a)3D

Evaporators provide air cooling to spaces within refrigerated warehouses using refrigerants such as Ammonia (R-717), CO2 (R-744), and halocarbon refrigerants. Designers and engineers selecting evaporator equipment for a project typically have multiple models available from each manufacturer that meet all requirements. Evaporator specific efficiency requirements provide a consistent parameter for comparing evaporator models so the energy of the unit can be used in the decision process. The evaporator specific efficiency evaluates the capacity and power of the unit at a specific set of rating conditions for cooler and docks, as well as with freezers. Those rating conditions may vary from application design conditions, so multiple sets of ratings may be required for a project where specific efficiency is not directly published from the manufacturer.

Evaporator specific efficiency is the gross refrigeration capacity divided by the fan watts at full speed at the design conditions listed in Table 120.6-A-2 Fan-Powered Evaporators – Minimum Specific Efficiency Requirements and tested in accordance with AHRI 420-2023, Standard for Performance Rating of Forced Circulation Free-Delivery Unit Coolers. The rating conditions for all of the evaporators are at 0 inches static pressure, i.e., in free air and not ducted. The capacity rating for the specific efficiency calculation utilizes gross total refrigeration capacity in Btu/hr, including the capacity required to offset heat from operating the fan. The requirements are limited to evaporators which contain refrigerants which undergo a phase change (evaporation) and thus do not include evaporators where the heat transfer of the refrigerant is sensible only such as in glycol cooling coils.

Evaporator specific efficiency is not mandatory for spaces that are used solely for quick chilling or quick freezing of products. This includes, but is not limited to, spaces with design cooling capacities of greater than 240 Btu/hr-ft2 (2 tons per 100 square feet). Many units in quick applications require high airflow which as a result typically involves more fan power to achieve the cooling time requirements for the products.

Example 10-12

Question:

Evaporators for a new refrigerated warehouse space are being specified with the intent that the fan speed will be limited to a maximum 92% (55Hz) using a control system. Can the evaporator specific efficiency be rated at 95% fan speed since this is expected to give a closer representation of specific efficiency at the actual operating conditions?

Answer:

No, specific efficiency must be rated at 100% (60Hz) fan speed to reflect the worst case the fan may operate at, such as if the control system is modified at a later point in time.

Static Pressure Drop

Reference: Section 120.6(a)3E

Maximum requirements for evaporator static pressure drop provide a limit in the static pressure to prevent the need for larger horsepower evaporator motors when ducting is included in designs. The design pressure drop is limited to no more than 0.5 inches of water column. If pressure drop is too high, a redesign is warranted with consideration of duct cross-sectional areas, transitions, and fitting selection. There is no maximum static pressure drop for evaporators that are used solely for quick chilling or quick freezing of products.

Condensers

Reference: Section 120.6(a)4

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

Condenser Sizing

Section 120.6(a)4A and Section 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 Section 120.6(a)4A. Fan-powered air-cooled condensers are covered by Section 120.6(a)4B. Fan-powered adiabatic condensers are covered by Section 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 Section 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-ft² of floor area, which is equivalent to 2 tons per 100 ft² of floor area, at system design conditions.

Another exception to Section 120.6(a)4A, 4B, 4C, 4G, and 4H for condenser sizing, efficiency, and fin density applies for equipment that must meet the Appliance Efficiency Requirements for walk-in cooler or walk-in freezers, which includes equipment serving spaces less than 3,000 square feet. Refrigeration equipment with condensers serving spaces of that size are commonly classified as a condensing unit, which is a unitary packaged equipment that includes compressor(s), condenser, liquid receiver, and control electronics in a single product.

Example 10-13

Question:

A new food processing plant is being constructed that will include an 800 ft² 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 Section 120.6(a)4A?

Answer:

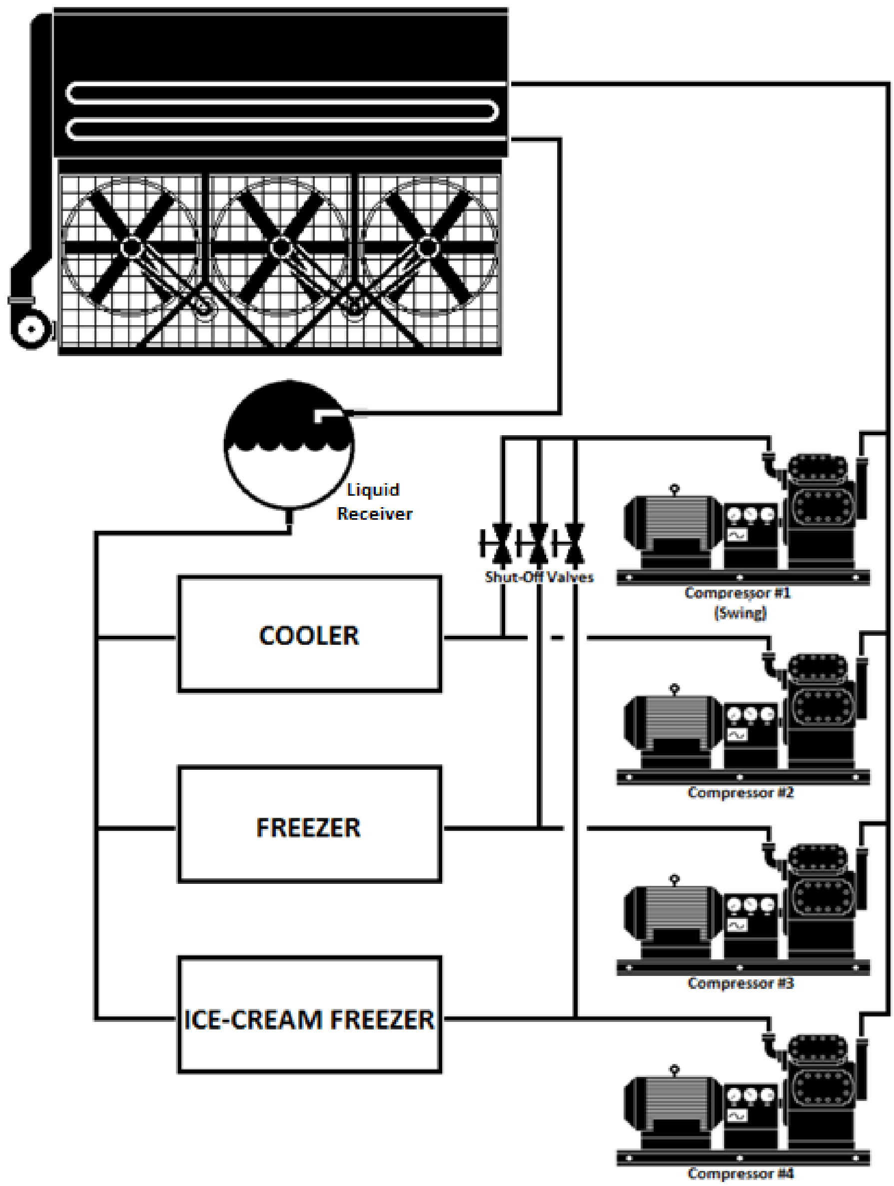
The blast freezer evaporator capacity divided by the floor area is 40 TR/800 ft², which is equal to 5 TR/100 ft². That means this particular blast freezer is deemed quick freezing space by the Energy Code. Therefore, the condenser group serving the refrigeration system does not have to comply with Section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

Example 10-14

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?

Figure 10- 8: Example Refrigerated Warehouse System for Example 10-14



Source: California Energy Commission

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.

Sizing of Evaporative Condensers, Fluid Coolers, and Cooling Towers

Reference: Section 120.6(a)4A

Section 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 2-3 – 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 10-1: Maximum Design SCT Requirements for Evaporative Condensers and Water-Cooled Condensers Served by Cooling Towers and Fluid Coolers below.

Table 10-1: Maximum Design SCT Requirements for Evaporative Condensers and Water-Cooled Condensers Served by Cooling Towers and Fluid Coolers

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

Source: California Energy Commission

Example 10-15

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 10-16

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.

Sizing of Air-Cooled Condensers

Reference: Section 120.6(a)4B

Section 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 2-3 – 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 10-2: Maximum Design SCT Requirements for Air-Cooled Condensers:

Table 10-2: 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)

Source: California Energy Commission

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 10-17

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

Adiabatic Condenser Sizing
Reference: Section 120.6(a)4C

Section 120.6(a)4C provides maximum design SCT values for adiabatic condensers. These requirements are the same as for Section 120.6(b)1E. See Condenser Mandatory Requirements for details.

Fan Control, Condensing Temperature Setpoint, and Condensing Temperature Reset
Reference: Section 120.6(a)4D, Section 120.6(a)4E, Section 120.6(a)4F

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 Section 120.6(a)4F. The alternative control strategy must be demonstrated to provide equal or better performance, as approved by the Executive Director.

Example 10-18

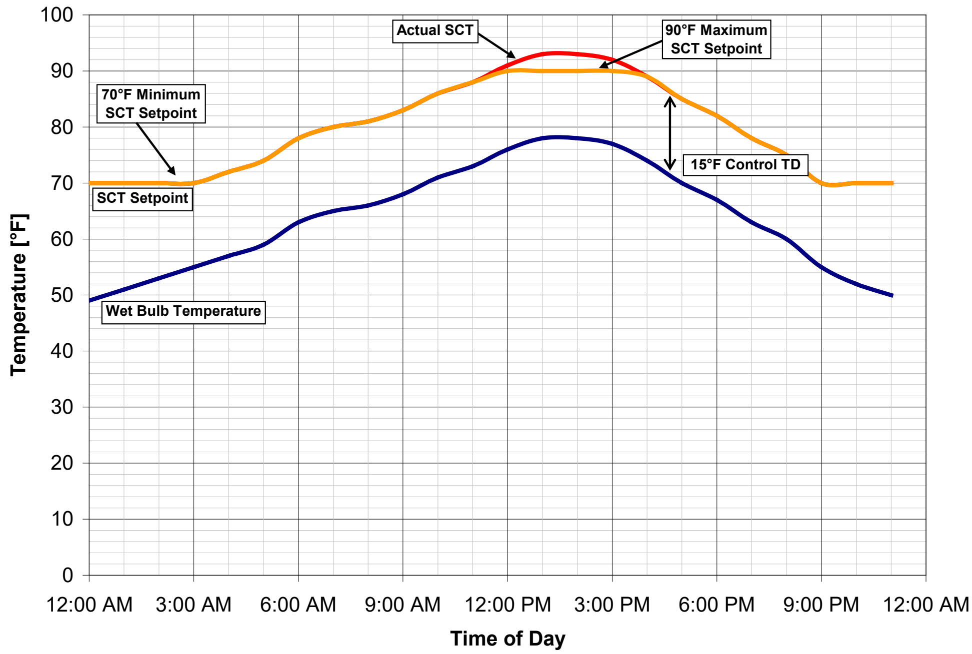
Question:

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.

Figure 10-9: SCT Setpoint and Wet-Bulb Temperature for Example 10-18



Source: California Energy Commission

Example 10-19

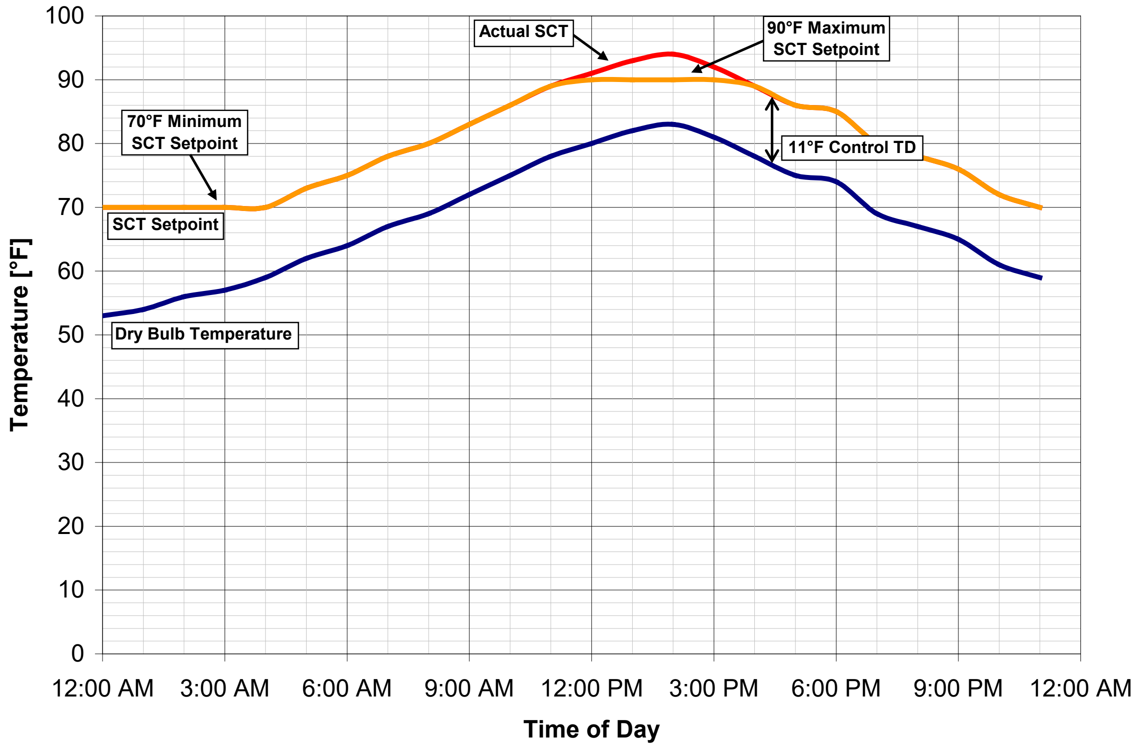
Question:

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.

Figure 10-10: SCT Setpoint and Wet-Bulb Temperature for Example 10-19



Source: California Energy Commission

Condenser Efficiency

Reference: Section 120.6(a)4G

Requirements for design condensing temperatures relative to design ambient temperatures, as described above for Section 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. Section 120.6(a)4G 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 120.6-B.

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 120.6-B, the Energy Code has different minimum efficiencies depending on the type of condenser that is being used. The different classifications of condenser are:

- Outdoor, evaporative, THR greater than 8,000 MBH at specific efficiency rating conditions.
- Outdoor, evaporative, THR less than 8,000 MBH at specific efficiency rating conditions.
- Indoor, evaporatively cooled.
- Outdoor, air-cooled, ammonia refrigerant.
- Outdoor, air-cooled, halocarbon refrigerant.
- Adiabatic (dry-mode operation), halocarbon refrigerant.
- Indoor, air-cooled.

The data published in the condenser manufacturer's published rating for capacity and power shall be used to calculate specific efficiency. For evaporative condensers, manufacturers typically provide nominal condenser capacity and tables of correction factors that are used to convert the nominal condenser capacity to the capacity at various applied condensing temperatures and 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 10-20

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:

Figure 10-11: Sample Condenser Properties for Example 10-20

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

Source: California Energy Commission

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 120.6-B, 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 Table 120.6-B 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.

Condenser Fin Spacing

According to Section 120.6(a)4H, 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.

Compressors

Please refer to Chapter 10.6.3.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Acceptance Requirements

Please refer to Chapter 10.6.3.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Transcritical CO_2 Systems

Please refer to Chapter 10.6.3.1 G and 10.6.3.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Transcritical CO_2 Gas Coolers

Please refer to Chapter 10.6.3.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Requirements related to refrigerated warehouse additions and alterations are covered by the Energy Code in Section 141.1(a). The specific requirements for additions and alterations for commercial refrigeration are included in Section 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 10-21

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 10-22

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 10-23

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 10-24

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:

Yes, the new evaporators must meet the single phase motor requirements in 120.6(a)3A, fan control for systems served by a single non-modulating compressor requirements in 120.6(a)3C, evaporator efficiency requirements in 120.6(a)3D, and maximum static pressure drop requirements in 120.6(a)3E.

Example 10-25

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. The energy consumption of the new evaporator must not exceed that of the equipment it replaced.

Laboratory Systems

Overview

Section 140.9(c) sets the minimum requirements for laboratory and factory exhaust and space conditioning 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 six categories of energy saving measures:

- Exhaust and makeup air reduction
- Reduction of conditioned makeup air
- Exhaust fan power reduction
- Fume hood automated sash closures
- Exhaust air heat recovery
- Reheat limitations (4-Pipe VAV)

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

Note that as of 2025, laboratories in Group L Occupancies are required to meet the requirements of Section 140.9(c), as are laboratories in Occupancy Groups A, B, E, F, H, I, M, R, S, and U. Prior to 2025, Group L was exempt from the Energy Code.

Mandatory Measures

The mandatory equipment efficiencies in Section 110.1 and Section 110.2 apply to laboratories.

The lab system acceptance tests are also mandatory where applicable, including:

- NA7.16.1 Construction Inspection for VAV Lab Exhaust System with Occupancy Control, per 140.9(c)1 and 140.9(c)3
- NA7.16.2 Functional Testing for VAV Lab Exhaust System with Occupancy Control, per 140.9(c)1
- NA7.16.3 Construction Inspection for Simple Turndown Control, per 140.9(c)3D.v.a
- NA7.16.4 Functional Testing for Simple Turndown Control, per 140.9(c)3D.v.a
- NA7.16.5 Construction Inspection for Wind Speed/Direction Responsive Control, per 140.9(c)3D.v.b
- NA7.16.6 Functional Testing for Wind Speed/Direction Responsive Control, per 140.9(c)3D.v.b
- NA7.16.7 Construction Inspection for Monitored Contaminant Control, per 140.9(c)3D.v.c
- NA7.16.8 Functional Testing For Monitored Contaminant Control, per 140.9(c)3D.v.c

Prescriptive Measures

Summary of measures contained in this section:

- Airflow Reduction Requirements - Section 140.9(c)1
- Exhaust System Transfer Air - Section 140.9(c)2
- Fan System Power Consumption - Section 140.9(c)3
- Fume Hood Automatic Sash Closure - Section 140.9(c)4
- Reheat Limitation - Section 140.9(c)5
- Exhaust Air Heat Recovery - Section 140.9(c)6

Airflow Reduction Requirements

Reference: Section 140.9(c)1

Section 140.9(c)1 requires that all laboratory systems be designed for variable-volume control on the supply, fume exhaust, and general exhaust. The system must be capable of reducing the total zone exhaust and makeup airflow rates down to 1.0 cfm/ft2 or lower when occupancy sensor(s) indicate the space is occupied and down to 0.67 cfm/ft2 or lower when occupancy sensor(s) indicate the space is unoccupied.

The airflow rate can be higher if the cooling load or hood sash positions demand a higher flow rate but if the cooling load is low and the hood sash positions do not require higher flow then the system must reduce the total zone flows to these minimum values.

Higher minimum flow rates are allowed if a higher minimum is required to comply with code, accreditation, or facility environmental health and safety department requirements. The minimum cannot exceed that required to comply with code, accreditation, or facility environmental health and safety department requirements. Documentation is required if this clause is used.

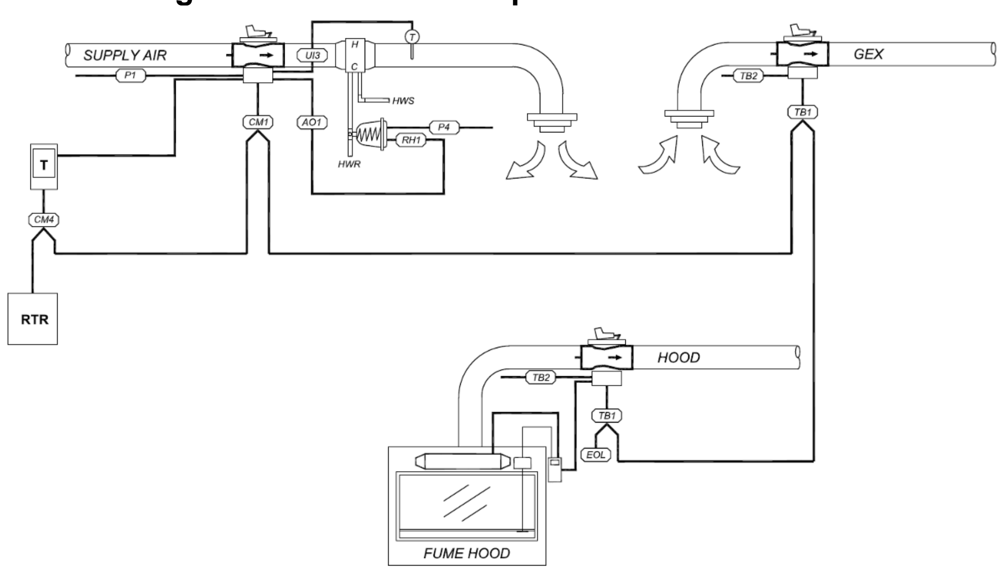
Higher minimum flow rates are also allowed if a higher minimum is required to maintain space pressurization. Typically, this would only be the case if there were many VAV fume hoods in a space and the total hood minimum flow with all the sashes closed still exceeded 0.67 or 1.0 cfm/ft2. A typical hood minimum flow rate with the sash closed is 50 cfm/ft of hood width or 25 cfm/ft2 of work surface.

The 1.0 cfm/ft2 occupied and 0.67 cfm/ft2 unoccupied are maximum minimums. Lower minimums are allowed. 1.0 cfm/ft2 is equivalent to 6 air changes per hour for a 10-foot-high ceiling. 0.67 cfm/ft2 is equivalent to 4 air changes per hour for a 10-foot-high ceiling. Many lab owners are comfortable with lower minimum air change rates, particularly when spaces are unoccupied.

Figure 10-12: Zone Components for a VAV Lab 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. When used, the hood valve is controlled to automatically maintain the design sash face velocity (e.g., 100 ft/min) 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 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 can be designed for either variable volume or constant volume depending on the application. If the hood is a constant volume bypass hood then the hood valve must be constant volume. 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 10-12: Zone Components for a VAV Lab

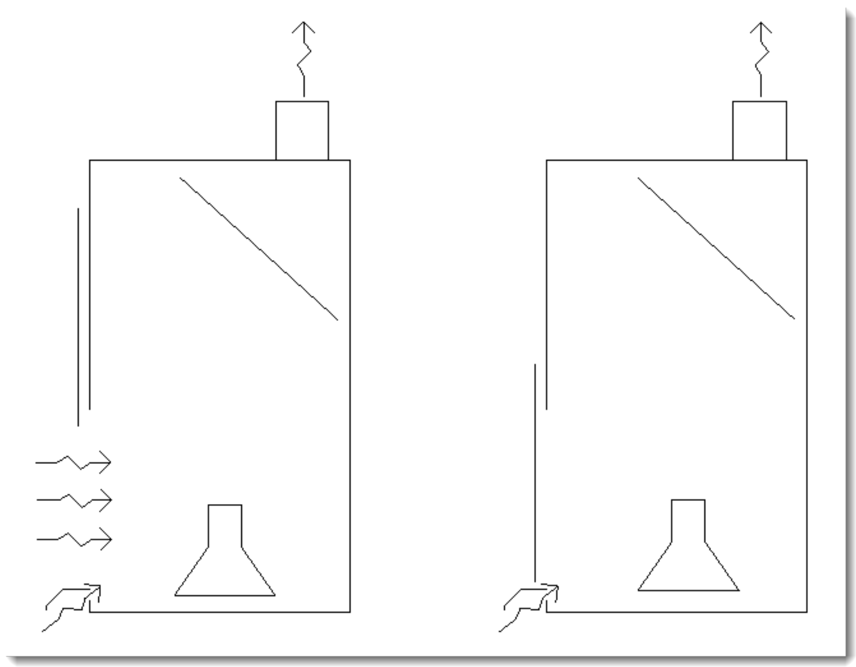


Source: California Energy Commission

Figure 10-13: Variable Air Volume Hood below illustrates a VAV hood. As the sash is moved the exhaust rate is modulated to maintain a fixed velocity (typically 100 ft/min) through the sash open area, with a minimum velocity when the sash is in the closed position (typically 50 cfm/ft of hood width or 25 cfm/ft2 of work surface).

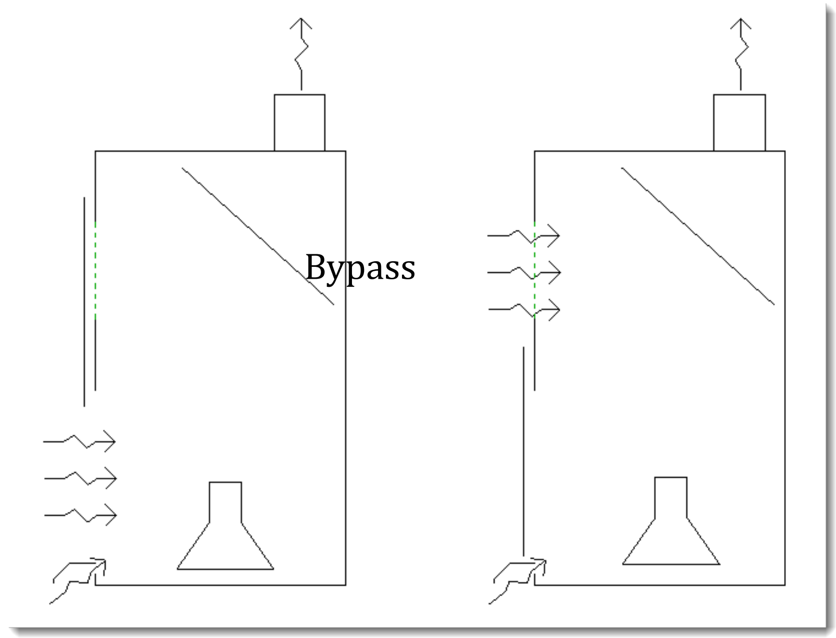
Figure 10-14: Contant Volume Hood illustrates a CAV hood. As the sash closes the bypass opens such that the sum of the flow rates through the sash and the bypass is constant.

Figure 10-13: Variable Air Volume Hood



Source: California Energy Commission

Figure 10-14: Contant Volume Hood



Source: California Energy Commission

The only exception to this requirement is for new zones on an existing constant volume exhaust system.

Example 10-26

Question:

Does Section 140.9(c)1 require VAV hoods?

Answer:

Not necessarily. Constant volume hoods are allowed if the total design exhaust rate of the hoods does not exceed 0.67 cfm/ft2 (or the regulated minimum unoccupied circulation rate documented to comply with code, accreditation, or facility environmental health and safety department requirements). However, if the total design exhaust rate of the hoods exceeds 0.67 cfm/ft2 then some or all hoods must be VAV. Only some hoods must be VAV as long as the total hood flow with the sashes closed does not exceed 0.67 cfm/ft2. If the total hood flow with the sashes closed exceeds 0.67 cfm/ft2 then all hoods must be VAV. The hood valve must follow the functionality of the hood, i.e., VAV hoods require VAV hood valves; CAV hoods require CAV hood valves.

Example 10-27

Question:

Does Section 140.9(c)1 require VAV supply valves?

Answer:

Typically, yes. A supply valve can only be CAV if its design flow rate does not exceed 0.67 cfm/ft2 (or the regulated minimum unoccupied circulation rate documented to comply with code, accreditation, or facility environmental health and safety department requirements).

Example 10-28

Question:

Does Section 140.9(c)1 require VAV GEX valves?

Answer:

Typically, yes. If the supply valve is VAV or the hood valve(s) are VAV then a VAV GEX is typically required.

Example 10-29

Question:

Our campus standard for lab minimums is 4 ACH occupied and 2 ACH unoccupied. The code language says, "shall be the greater of…user defined airflow not to exceed...6 ACH". So is 4 ACH / 2 ACH still ok?

Answer:

Yes. The user defined airflow can be any flow from 0 ACH up to 6 ACH occupied and 4 ACH unoccupied. So 4 ACH / 2 ACH is still allowed.

Example 10-30

Question:

Our EH&S Department requires at least 10 ACH for certain labs. Is this still allowed?

Answer:

Yes. The minimum may be set above 6 ACH if needed to meet EH&S minimum requirements. The AHJ may require documentation of the EH&S requirements.

Exhaust System Transfer Air

Reference: Section 140.9(c)2

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:

- The supply flow required to meet the space heating or cooling load.
- The ventilation rate required by the AHJ, facility EH&S department, or by Section 120.1(c)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:

- Calculate the minimum outside air required by adjacent spaces.
- From the first bullet point, subtract the amount of air required by adjacent space exhaust.

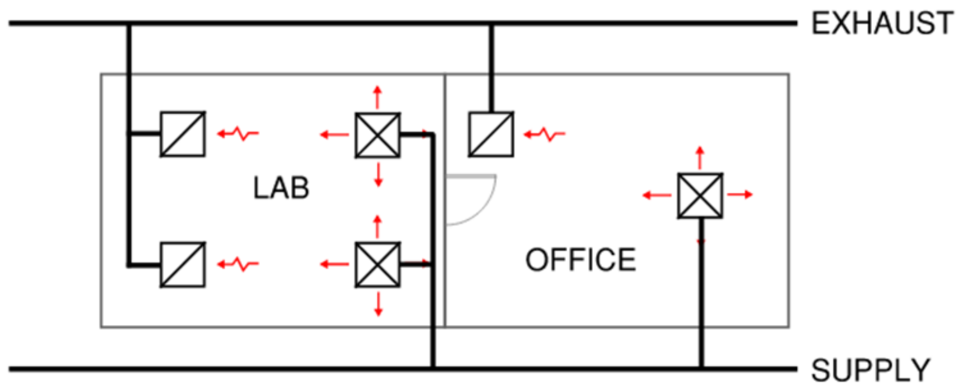
- From the second bullet point, subtract the amount of air required to maintain pressurization of adjacent spaces. This is your available transfer air.

Exceptions to Section 140.9(c)2 are provided for:

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

Figure 10-15: Does Not Comply with Transfer Air Requirement is a simple example of a system that does not comply with Section 140.9(c)2. It shows 100% of the lab space makeup air being conditioned outside air and an adjacent office that is also 100% outside air and 100% exhausted. The office space has available transfer air that can be transferred to the lab, rather than exhausted.

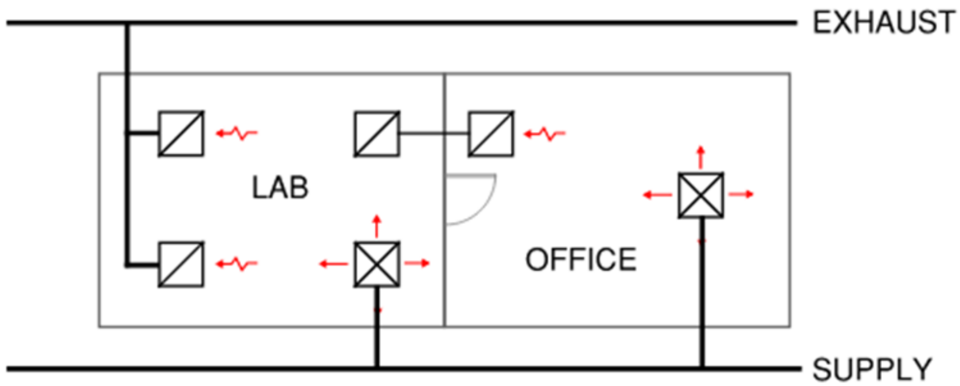
Figure 10-15: Does Not Comply with Transfer Air Requirement



Source: California Energy Commission

Figure 10-16: Complies with Transfer Air Requirement shows how the system in Figure 10-15 can be modified to comply with Section 140.9(c)2. Rather than exhausting the office space, its return air is transferred to the adjacent lab space (the pressure differential between the office and lab spaces is often sufficient to transfer the air without a fan, but a transfer fan can also be used). This reduces the amount of conditioned outside air / makeup air that must be supplied to the lab space.

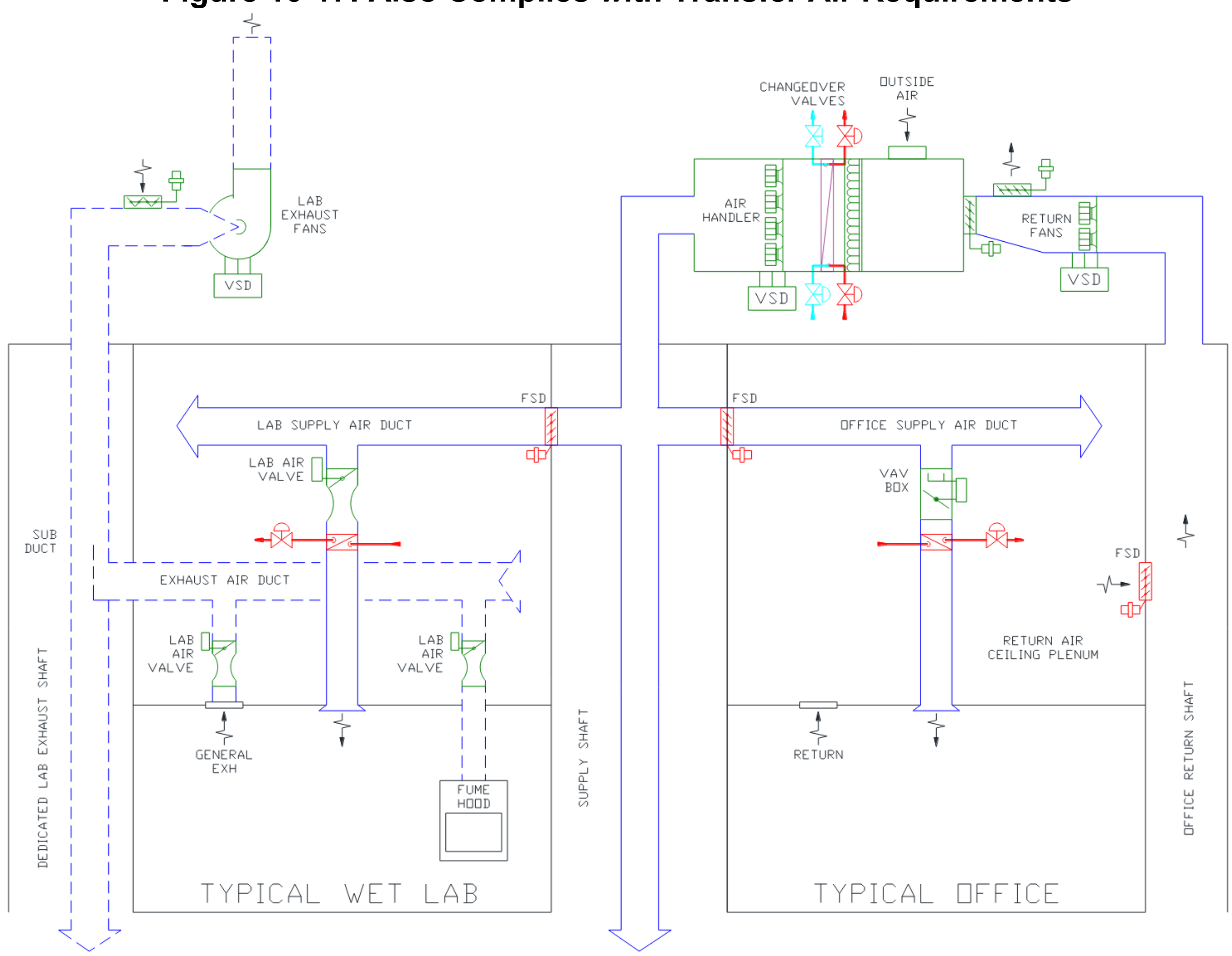
Figure 10-16: Complies with Transfer Air Requirement



Source: California Energy Commission

Figure 10-17: Also Complies with Transfer Air Requirements shows another option for complying with Section 140.9(c)2. Rather than directly transferring from office to lab spaces, the office spaces can be returned to the air handler, which can then transfer the office air to the lab spaces when the airside economizer is disabled.

Figure 10-17: Also Complies with Transfer Air Requirements



Source: California Energy Commission

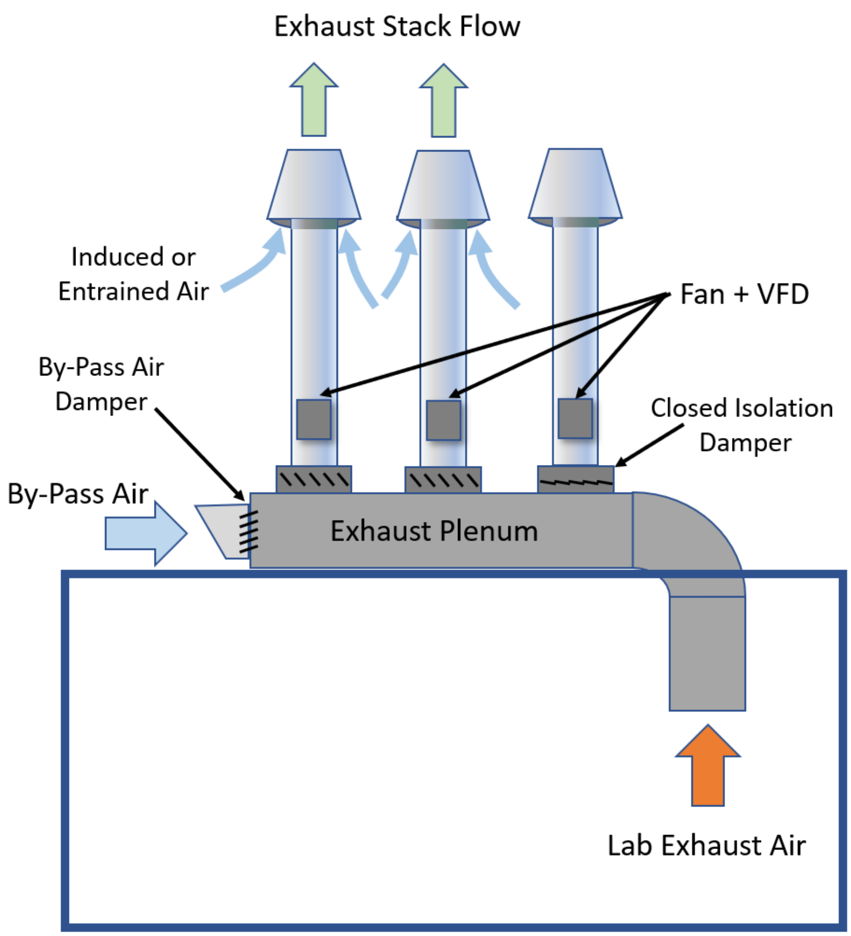
Exhaust Fan System Power Consumption

Reference: Section 140.9(c)3

Exhaust fan power consumption is a function of flowrate and pressure rise through the fan system. Because laboratory exhaust streams frequently have hazardous contaminants, there needs to be sufficient exhaust air to protect the occupants of labs by sufficiently diluting the laboratory space air. There also needs to be sufficient airflow to keep fumes generated in fume hoods from leaving through the face of the hood and instead pulled out through the exhaust ductwork to the outside. Additionally, the laboratory exhaust system must protect the general public from harmful concentrations of contaminants in the exhaust stream. This is accomplished by releasing the exhaust stream high in the air so that there is sufficient mixing and dilution before the air is at levels where people are present. This dilution and mixing are accomplished through a combination of exhaust stack height, mixing of airstreams, and exhaust velocity at the exit of the exhaust stack.

Variable air volume (VAV) exhaust systems reduce the amount of energy required to condition outside air. To keep velocities high enough to protect the general public, fans are staged so that the velocity per stack is maintained or a by-pass damper is opened so that unconditioned air can be mixed with the exhaust airstream. Some laboratory exhaust systems make use of induction fans which have openings downstream of the fan to induce or entrain air. Figure 10-18: Components of Exhaust Air Flow, illustrates lab exhaust air, bypass air, induced air, and total exhaust airflow.

Figure 10-18: Components of Exhaust Air Flow



Source: California Statewide CASE Team

The fan system air flow is the total exhaust system airflow minus induced or entrained airflow. The induction fan curves have fan inlet flow rates and higher fan exit flow rates for the same static pressure rise across the fan. This accounts for the difference between fan airflow and exiting stack airflow including induced flow. As a result, use the fan inlet flow rates for calculating exhaust fan system airflow.

Newly installed laboratory and factory exhaust systems greater than 10,000 CFM have five prescriptive options to show compliance with this section. Regardless of the path chosen, all exhaust systems must meet the discharge requirements of ANSI Z9.5-2022 Section 6.4.

Options 1 and 2 place relatively stringent fan power limitations but without the fan control requirements. Often times to meet these design fan power requirements, a relatively tall exhaust stack is needed.

Zoning or other constraints may limit the height of the exhaust stacks, and more design fan power is required to provide a higher exiting stack velocity. Options 3, 4, and 5 allow higher design fan power requirements but have fan control requirements.

The exhaust air flowrate from indoors varies due to the requirements in Section 140.9(c)1 that include different circulation air changes depending upon occupied versus unoccupied operation. Additionally, exhaust air flow rates must respond to exhaust airflow requirements from devices such as fume hoods, chemical storage cabinets, snorkels (point exhaust fixture), etc. and Section 140.9(c)4 which automates the fume hood sash opening. Reducing indoor exhaust flowrate with respect to device load and by occupancy status reduces the amount of outside air that is conditioned, brought into the space, and exhausted.

Options 1 and 2 are allowed to maintain a constant exhaust fan system airflow rate by adding unconditioned by-pass air when exhaust air flowrate from indoors drops. Thus, fan power is relatively constant while conditioned air requirements vary.

Options 3, 4, and 5 with higher design power allowances are based upon the controls descriptions in ANSI/ASSP Z9.5-2022 Laboratory Ventilation for simple turndown systems, wind responsive systems, and monitored systems. Under these control systems, the by-pass air is used less frequently, the fan speed is dropped, and fan energy is reduced most hours of the year.

Option 1: Fan Power Budget

Reference: Section 140.9(c)3B

Meet the fan power budget in Section 140.4(c)1. This option allows one to comply making use of the fan power limitations in 140.4(c)1. When using this path, one cannot make use of the process load exception to 140.4(c)1.

Option 2: 0.85/0.65 Watts/CF M

Reference: Section 140.9(c)3C

The exhaust system fan power does not exceed 0.85 w/cfm for systems with air treatment devices (e.g., scrubbers) or 0.65 w/cfm for systems without air treatment devices.

Option 3: Simple Turndown Control

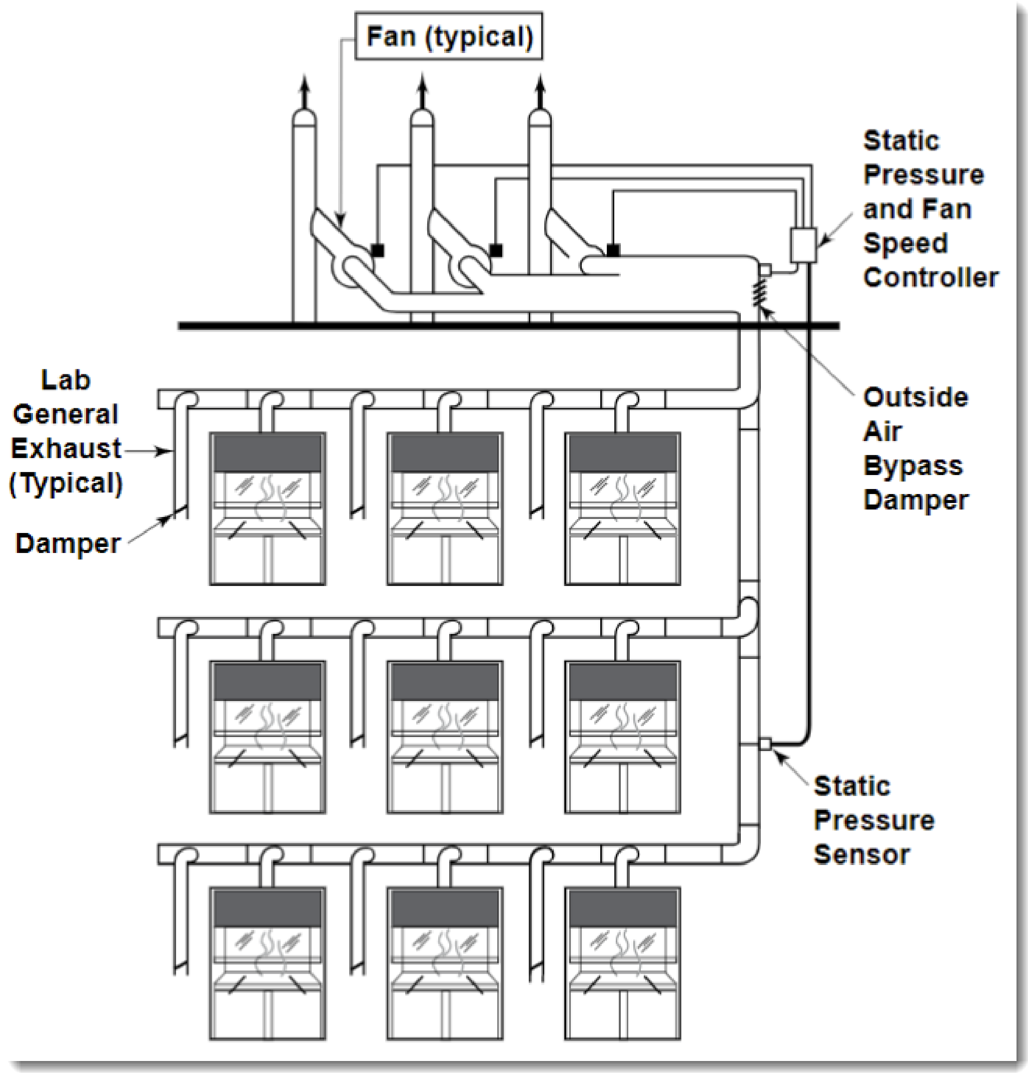
Reference: Section 140.9(c)3Dva

This option is only available if the minimum supply flow to the spaces served is less than 60% of the exhaust fan system design flow, i.e., the ability to turn down the exhaust flow to less than 60% at low load conditions. To meet this option, the exhaust system fan power cannot exceed 1.3 w/cfm, the exhaust fan must have a variable speed control, and the exhaust stack flow must track the supply flow to the spaces from 100% flow down to 60%, i.e., no bypass damper flow or strobic induction at the exhaust fan until the exhaust flow is less than 60% of design.

This option may require a very tall exhaust stack on the roof to meet the health and safety requirements in ANSI Z9.5 when the exhaust flow is less than 60% of the maximum exhaust flow.

Lab exhaust systems are often designed to maintain a stack discharge velocity at or above 3,000 feet/min. With VAV zone exhaust flows (as required by Section 140.9(c)1), there are a few options for maintaining near constant stack velocity. One option is staging exhaust fans on/off based on the load. Another option is an outside air bypass damper that supplements the exhaust from the spaces with outside air to maintain the stack velocity (see Figure 10-19: Typical Lab Exhaust Fan Control Schematic). Another option is to allow stack velocity below 3,000 fpm by analyzing hazards and providing equivalent hazard mitigation (e.g., a very tall stack). This is typically required in order to comply with Option 3. One might think that simply raising the design stack velocity to 5,000 fpm would be a simple option to allow 60% turndown while staying above 3,000 fpm. However, raising the design stack velocity to 5,000 fpm could result in exhaust system design fan power > 1.3 w/cfm or unacceptable acoustics.

Figure 10-19: Typical Lab Exhaust Fan Control Schematic



Source: California Statewide CASE Team

Option 4: Wind Responsive Control
Reference: Section 140.9(c)3Dvb

Similar to option 3, this option is only available if the minimum supply flow to the spaces served is less than 60% of the exhaust fan systems design flow. To meet this option the exhaust system fan power cannot exceed 1.3 w/cfm and the exhaust fan must have a variable speed control. The exhaust stack flow must track the supply flow to the spaces from 100% flow down to 60% for at least 70% of the hours of year. This prediction is based on dispersion model with a worst case emissions described in ANSI Z9.5. It is assumed that fan system airflow can be reduced to 60% of the design airflow for the predicted wind speeds and directions in a TMY (typical meteorological year) file 70% of the hours in a year. For the other 30% of the year with typically higher wind speeds, the bypass damper may be opened to keep exhaust fan system flowrates above 60%.

A wind responsive control system typically includes the following features:

- Anemometer Sensors:
  - Two anemometer sensors must be used to enable sensor fault detection.

  - Installation location must exhibit similar wind speed and direction to the free stream air above the exhaust stacks.
  - Sensors must be located high enough to be above the wake region created by nearby structures.
  - Sensors must be factory calibrated.
  - 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.
- Dispersion Modeling:
  - Wind dispersion analysis must be used to create a look-up table for exhaust volume flow rate versus wind speed/direction.
  - Look-up table must contain at least eight wind speeds and eight wind directions to define the safe exhaust volume flow rate.
  - 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.
- 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. Furthermore, the system must report the fault to an Energy Management Control System or other application which notifies a remote system provider.

Option 5: Contaminant Monitored Control

Reference: Section 140.9(c)3Dvc

Similar to Options 3 and 4, this option is only available if the minimum supply flow to the spaces served is less than 60% of the exhaust fan systems design flow. To meet this option, the exhaust system fan power cannot exceed 1.3 w/cfm and the exhaust fan must have a variable speed control. The exhaust stack flow must track the supply flow to the spaces from 100% flow down to 60%, unless the current measured contaminant concentration in the exhaust plenum is above a defined threshold level, in which case higher exhaust flows are acceptable (e.g., the bypass damper may be opened at exhaust flows above 60%).

This option requires a real-time contaminant monitoring system capable of measuring the concentrations of all hazardous chemicals used in the building. This may place limitations on what chemicals can be used in the building.

A contaminant monitored control system typically includes the following features:

- 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 manufacturer to an accuracy of ± 5% and require calibration no more than every six months.
- 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.
- 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.

Example 10-31

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 design airflow 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, Section 140.9(c)1 requires that laboratories have variable-volume exhaust and makeup airflow if the design cfm/ft2 > 0.67. The requirement is based on cfm/ft2, not air changes, so the ceiling height does not matter. For this laboratory space:

Design cfm/ft2 = 2,500 cfm / 2,500 ft^2 = 1.0 cfm/ft2

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

Example 10-32

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 = $CFM_s \times 0.0013 + A$

$= 8{,}000 \times 0.0013 + A = 10.4 + A$

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.

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

$A = [8{,}000 \times 0.5 + 8{,}000 \times 0.5 + 8{,}000 \text{ x } 0.35 + 8{,}000 \text{ x } 0.6] / 4131 = 3.8$ 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 10-33

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:

$hp < cfm_S \times 0.0015 = 12{,}000 \times 0.0015 = 18$ 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.

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

Fume Hood Automatic Sash Closure

Please refer to Chapter 10.7.3.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Reheat Limitation

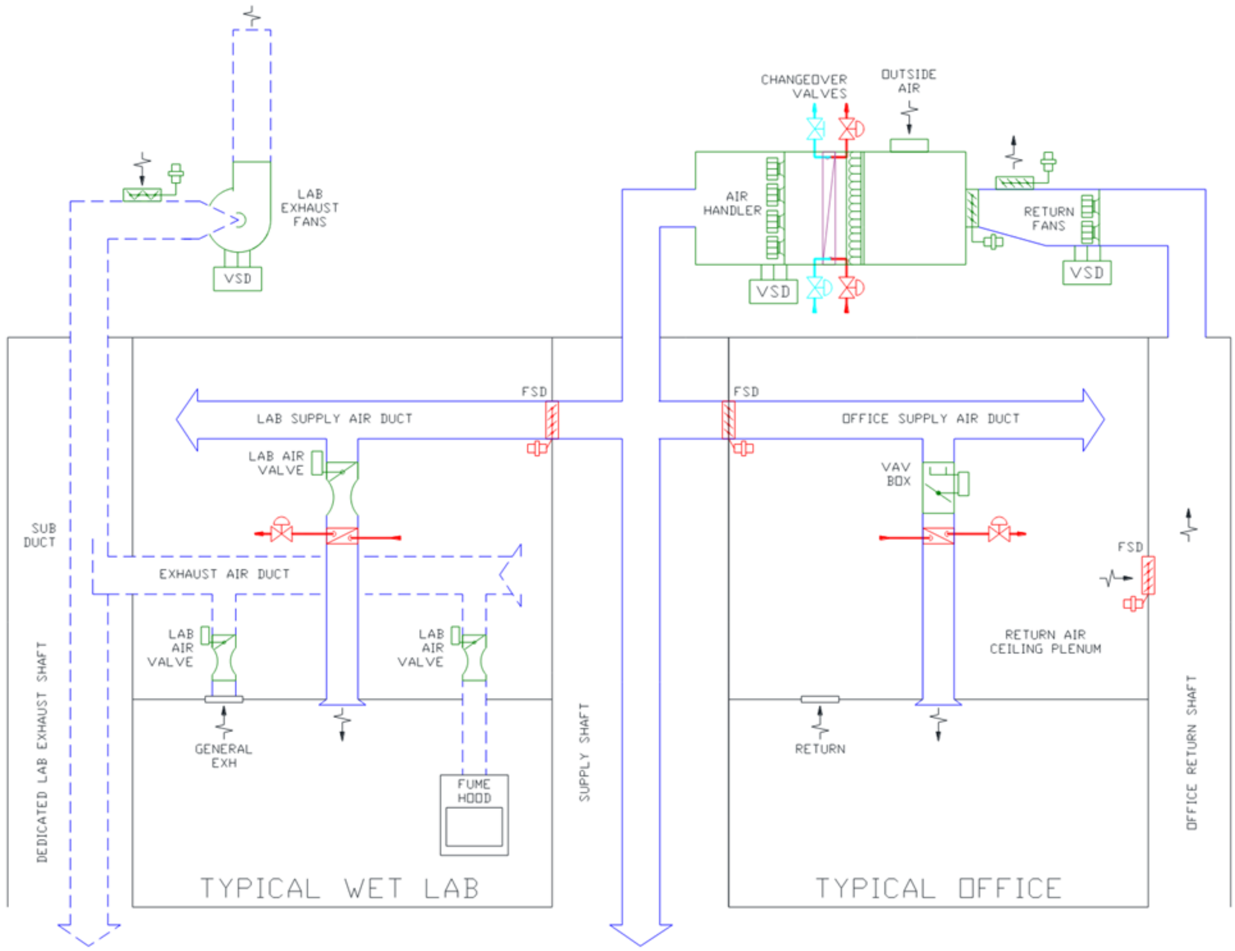
Reference: Section 140.9(c)5

Air handlers in buildings with greater than 20,000 cfm of laboratory exhaust that serve multiple space conditioning zones in laboratory spaces shall not mechanically cool air handler supply air below 80°F and shall not heat air handler supply air above 50°F, and each zone shall include heating and cooling capacity, to prevent cooling at the air handler and reheating at the zones. Most large labs use multizone AHUs.

The most common way for multizone air handlers to meet this requirement is with 4-pipe VAV, which is a VAV system with terminal units (VAV boxes) that have both hot water and chilled water coils or a single switchover coil that can be fed with hot water or chilled water. A 2-pipe VAV system (Figure 10-20: 2-Pipe VAV Does Not Comply with Section 140.9(c)5) only has hot water coils at the zones and can result in significant reheat because the air handler mechanically cools the supply air to satisfy zones with high cooling loads, and zones with low cooling loads must reheat the supply air to prevent overcooling. In a 4-pipe VAV system, reheat can be eliminated because cooling for zones with high cooling loads can be provided by the zone cooling coils.

4-pipe VAV is not the only system type that would meet this new requirement. Other systems include chilled beams, VRF, and separate cooling and heating coils at each zone, i.e., two 2-pipe coils, rather than one 4-pipe coil.

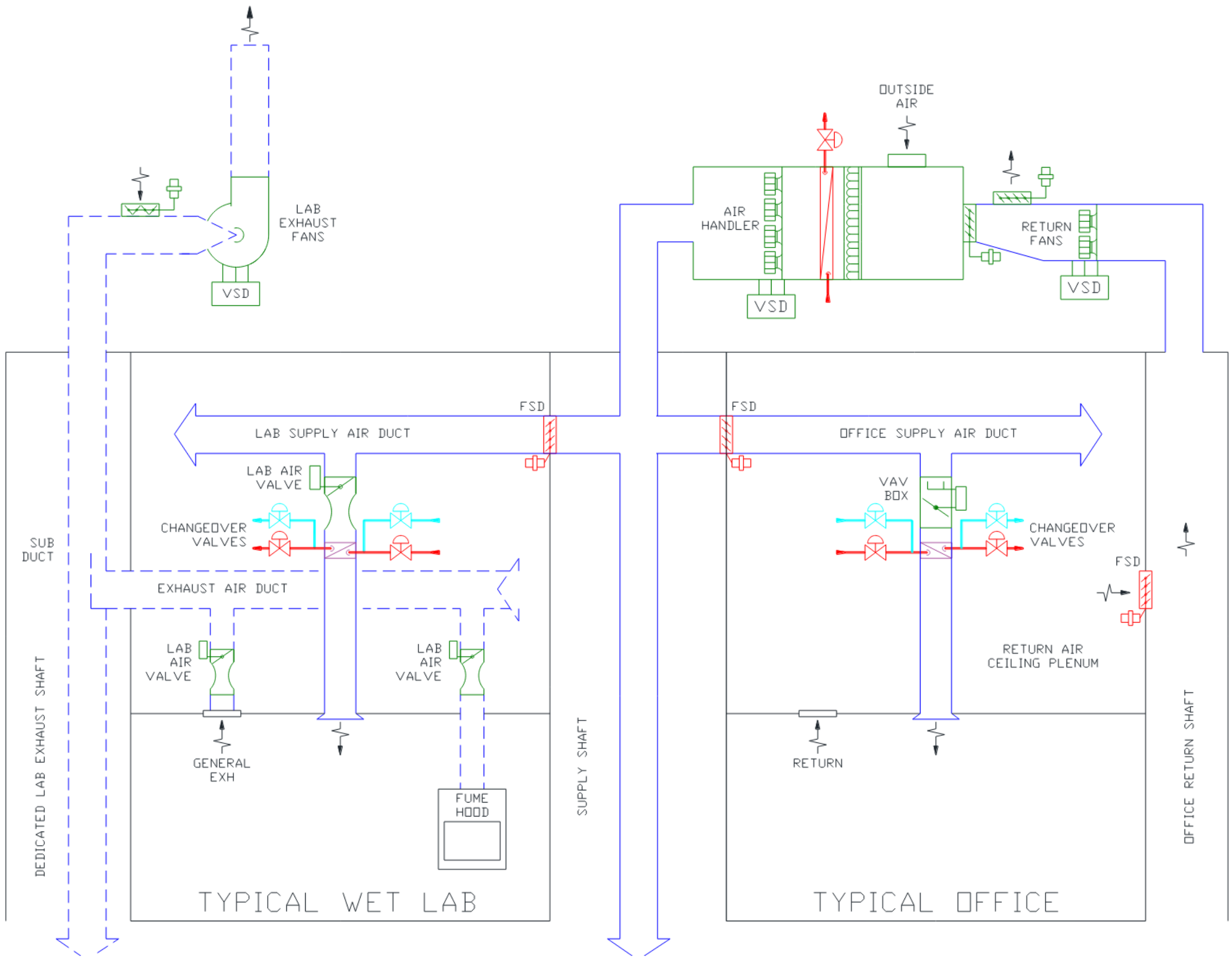
Figure 10-20: 2-Pipe VAV Does Not Comply with Section 140.9(c)5



Source: California Statewide CASE Team

Figure 10-21: 4-Pipe VAV Complies with Section 140.9(c)5 shows a 4-Pipe VAV system that meets the requirement. It shows a single changeover coil at each zone that has 4 pipes to it: chilled water supply/return and hot water supply/return. This figure shows only a hot water coil at the AHU. Cooling is not needed at the AHU if the zone cooling coil is designed to meet the design cooling load with the design outdoor air temperature (e.g., 100ºF) entering the coil. Typically, this requires a deep coil (e.g., 8 rows deep). A cooling coil (or 4-pipe changeover coil) is permitted at the AHU but the AHU cannot cool the supply air below 80ºF. Having cooling capacity at the AHU could reduce the overall project cost by allowing smaller zone coils (e.g., 4 rows deep).

Figure 10-21: 4-Pipe VAV Complies with Section 140.9(c)5



Source: California Statewide CASE Team

One consideration with a single zone coil used for both heating and cooling is that it is not possible to dehumidify the space by over-cooling with a cooling coil and then reheating with a downstream heating coil. Dehumidification is provided by the zone cooling coil in a 4-pipe VAV system (hence the need for condensate removal), but dehumidification will only occur if there is a space cooling load. This cannot be relied upon to maintain humidity if a cooling load is not present.

Fortunately, dehumidification is not required in most lab spaces in California. Per Addendum ASHRAE Standard 62.1-2022, active dehumidification is not required for climates where outdoor dewpoint is below 68°F at the ASHRAE 2 percent annual dehumidification design conditions. The proposed requirement includes an exception for these locations.

For the few lab spaces that require dehumidification for specific process requirements, active dehumidification can be provided by using separate cooling and heating coils at the zone level. This meets the proposed requirement. In fact, before 6-way control valves and changeover coils were common, it was common to minimize reheat in labs by installing separate cooling and heating coils at each lab zone. A lab can also meet the proposed requirement with a combination of changeover coils at most zones and dual coils at the few zones with strict humidity requirements.

There are three full exceptions to Section 140.9(c)5 and one partial exception.

Full Exceptions:

- Exception 1 to Section 140.9(c)5: Additions or alterations to existing air handling systems serving existing zones without heating and cooling capacity.
- Exception 2 to Section 140.9(c)5: Systems in climate zones 7 or 15 (these are the two most humid climate zones in California).
- Exception 3 to Section 140.9(c)5: Systems dedicated to vivarium spaces or to spaces classified as biosafety level 3 or higher.

Partial Exception:

- Exception 4 is only available in a handful of cities where the outdoor dew point temperature is greater than or equal to 64°F at the ASHRAE 2 percent annual dehumidification design condition. This ASHRAE website can be used to determine if this criteria is met, https://ashrae-meteo.info/v2.0/. Even in these locations, heating and cooling are required at the zone (e.g., 4-pipe VAV). The only difference is that the AHU can cool the supply air below 80°F but only when the measured outside air dewpoint is above 60°F. This allows the AHU to dehumidify in humid weather and can avoid the need to have separate cooling and heating coils at the zone level for dehumidification purposes.

Exhaust Air Heat Recovery

Reference: Section 140.9(c)6

Buildings with greater than 10,000 cfm of laboratory exhaust shall include an exhaust air heat recovery system. The heat recovery system must meet the following criteria:

- A sensible energy recovery ratio of at least 45 percent at heating design conditions and 25 percent at cooling design conditions.
    - The sensible energy recovery ratio is the ratio of the change in the outdoor air supply's dry-bulb temperature to the difference in dry-bulb temperature between the outdoor air and the entering exhaust airflow. For example, suppose the design winter outdoor temperature is 30°F and the winter return/exhaust air temperature is 70°F. The heat recovery system must be capable of raising the supply air temperature from 30°F to at least 48°F (= 0.45*(RAT-OAT)+OAT)
- Heat is recovered from at least 75 percent of all lab exhaust air volume.
    - Not every exhaust fan has to have a heat recovery coil as long as heat is recovered from 75% of the total exhaust volume.
- The system includes a run-around coil pump or other means to disable heat recovery.
    - Heat recovery is generally not beneficial when the outside air temperature is between the return air temperature and the desired AHU supply air temperature, i.e., airside economizer free cooling. For example, if RAT = 75°F and OAT = 65°F and desired SAT = 60°F, then the heat recovery should be disabled to prevent inadvertently heating the air and creating a false cooling load. The term “desired AHU SAT” is used here instead of SAT setpoint because Section 140.9(c)5 generally prohibits controlling the AHU cooling coil below 80°F but Section 140.9(c)5 does not prohibit economizer

free cooling to a desired supply air temperature that would reduce the total mechanical heating/cooling load.

- The system includes a bypass damper or other means so that the exhaust air pressure drop through the heat exchanger does not exceed 0.4 inch w.g. when heat recovery is disabled.
    - In California's mild climate there are many hours when heat recovery is not beneficial. Bypassing the heat exchanger saves fan energy.

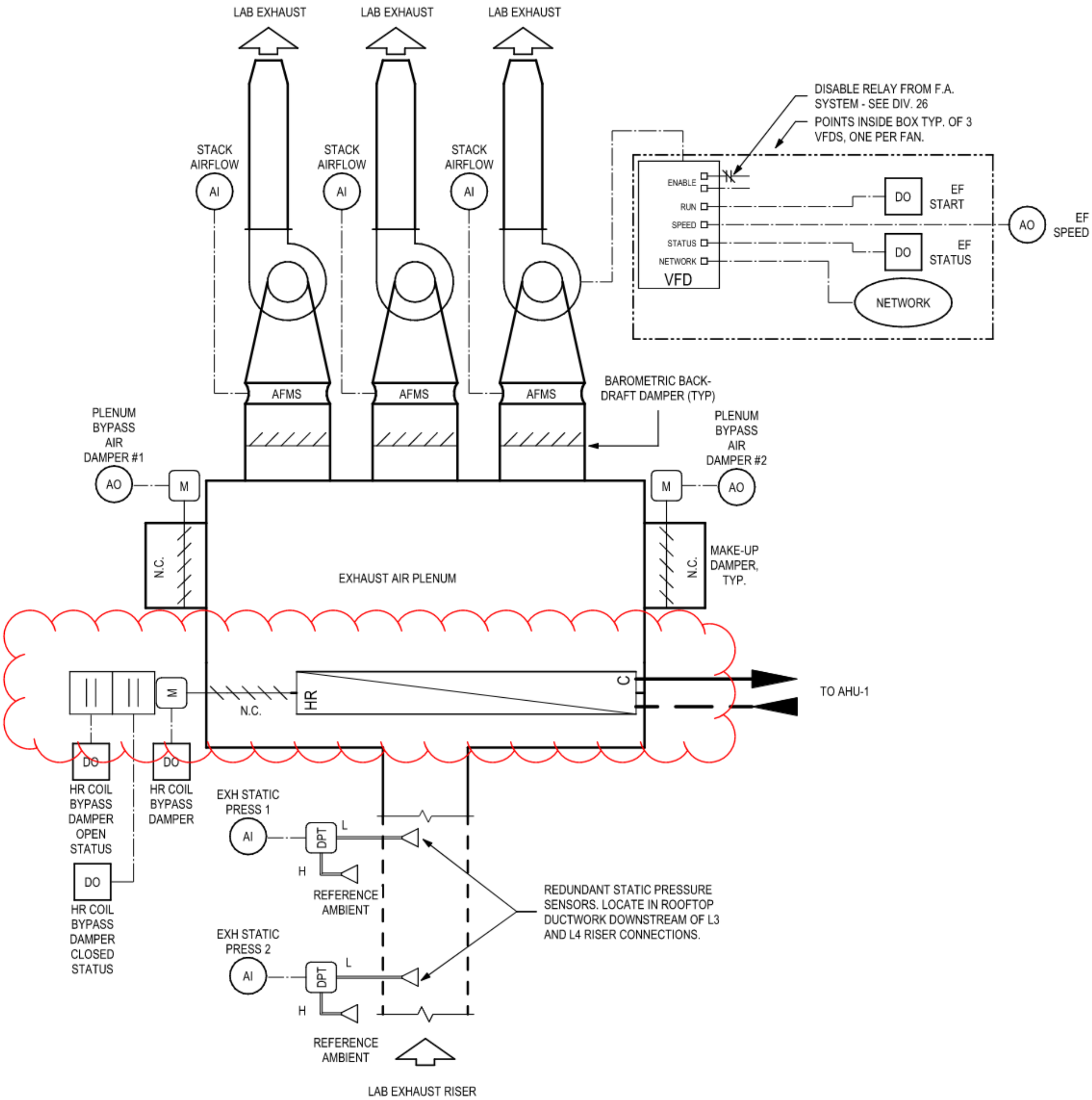
Lab exhaust heat recovery is typically achieved with a coil run-around system, as opposed to a plate-type or wheel-type air-to-air heat exchanger. This is to mitigate the risk of cross-contamination from the exhaust air stream to outside air stream.

With a run-around system, a fluid coil (water or glycol) is added into the exhaust airstream (see Figure 10-22: Typical Control Schematic of Lab Exhaust Fan System with Heat Recovery Coil). New pump(s) and piping are added to transfer heat from the exhaust coil to a coil in the supply air handler(s) (see Figure 10-23: Typical Schematic of Lab Exhaust Fan System). If the supply air handler has an existing heating coil, then that coil can also be used as the heat recovery coil. If the air handler does not have a heating coil, then a recovery coil must be added (see Figure 10-24: Typical Schematic of Lab Air Handler with Heat Recovery).

The control sequences for this type of lab exhaust heat recovery system are typically quite simple. For example:

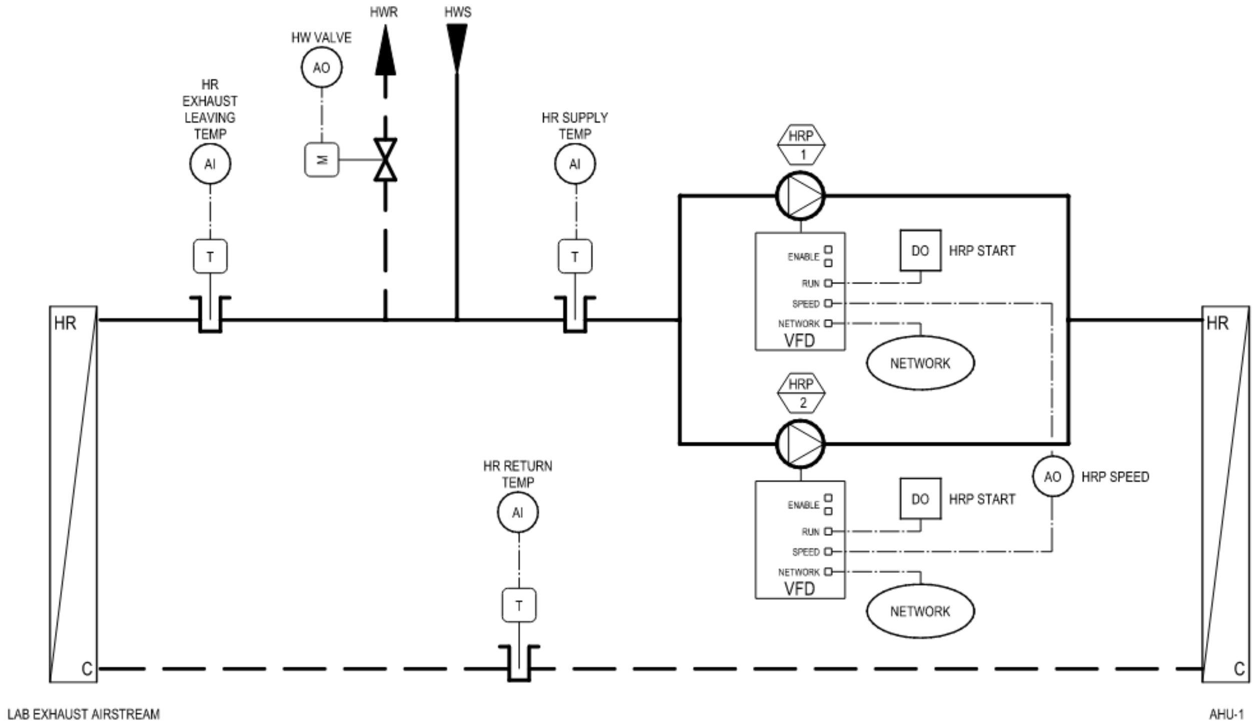
- Cooling: When the outside air temperature is above 83°F, command the lab exhaust heat recovery bypass damper zero percent open and enable the heat recovery pumps at design speed.
- Heating: A PID loop shall maintain the supply air temperature at the minimum SAT setpoint by first modulating the bypass damper from 100 percent to zero percent and then modulating the HW valve from 0 percent to 100 percent. Run the heat recovery pumps at design speed when the bypass damper is less than 100 percent.

Figure 10-22: Typical Control Schematic of Lab Exhaust Fan System with Heat Recovery Coil



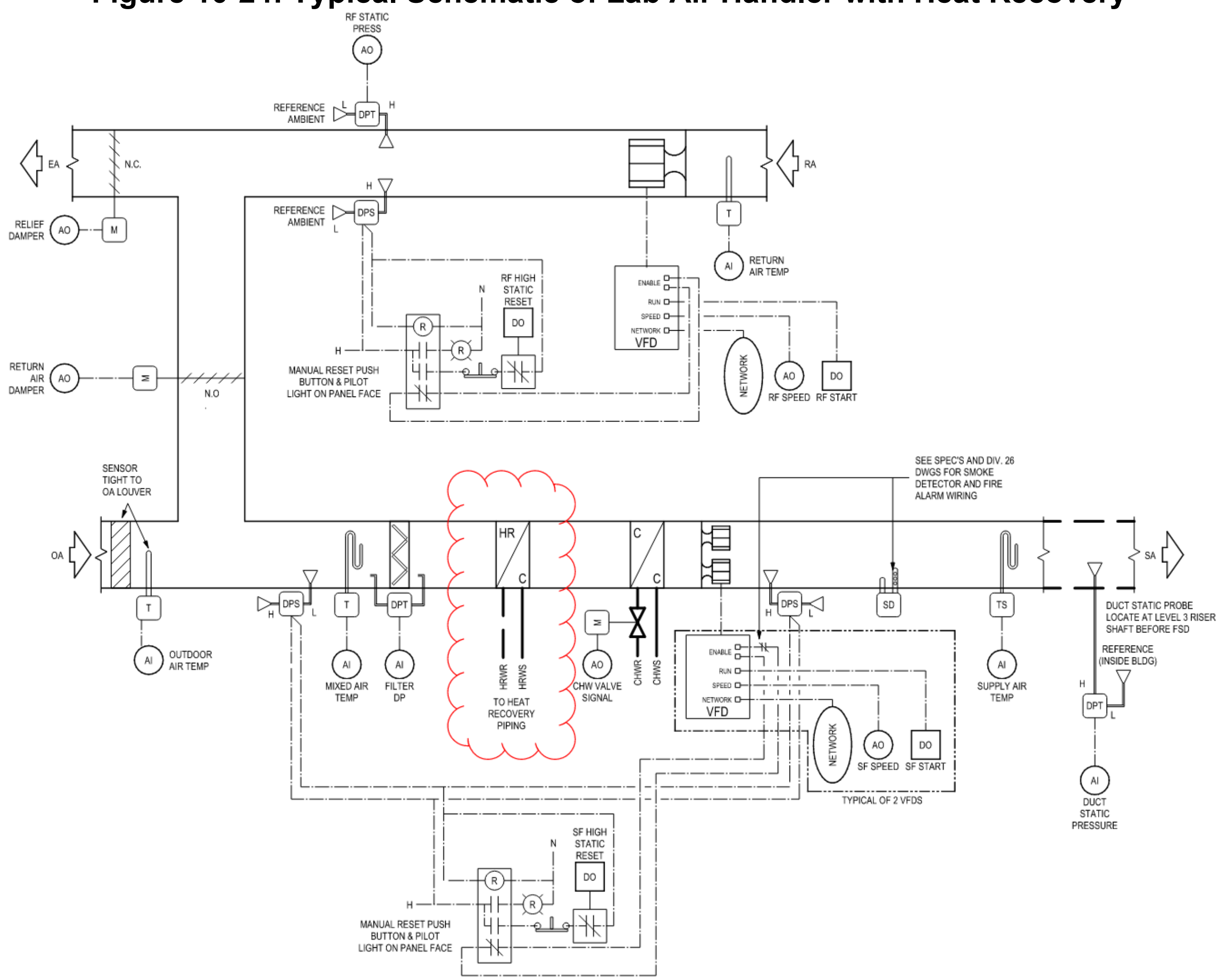
Source: California Statewide CASE Team

Figure 10-23: Typical Schematic of Lab Exhaust Fan System



Source: California Statewide CASE Team

Figure 10-24: Typical Schematic of Lab Air Handler with Heat Recovery



Source: California Statewide CASE Team

There are 4 full exceptions and one exception that requires an alternate compliance path.

Full Exceptions:

Exception 1 to Section 140.9(c)6: Additions or alterations to existing laboratory exhaust systems that do not include exhaust air heat recovery.

Exception 2 to Section 140.9(c)6: Buildings where the total laboratory exhaust rate exceeds 20 cfm/ft2 of roof area.

Heat recovery coils are typically designed for air velocities around 500 fpm. Without heat recovery coils, the exhaust plenum could be designed for velocities of 1000-2000 fpm. Adding an EAHR system can increase the size of the exhaust plenum. These plenums are typically located on the roof, near the exhaust fans. If the building is several stories tall and is packed with high load labs, then there may not be enough roof space to accommodate the larger exhaust plenums. Note that in many cases EAHR actually reduces the roof space requirements of the mechanical equipment. This is because EAHR reduces the peak heating and cooling loads. If heating is provided by air-source heat pumps (ASHPs), then fewer/small ASHPs are required on the roof. This reduction in ASHP footprint more than compensates for the increase in exhaust plenum footprint.

Exception 3 to Section 140.9(c)6: Locations that meet both of the following:

- In Climate Zone 6 or 7; and
- In a jurisdiction where gas heating is allowed. (Note: some jurisdictions do not allow gas heating)

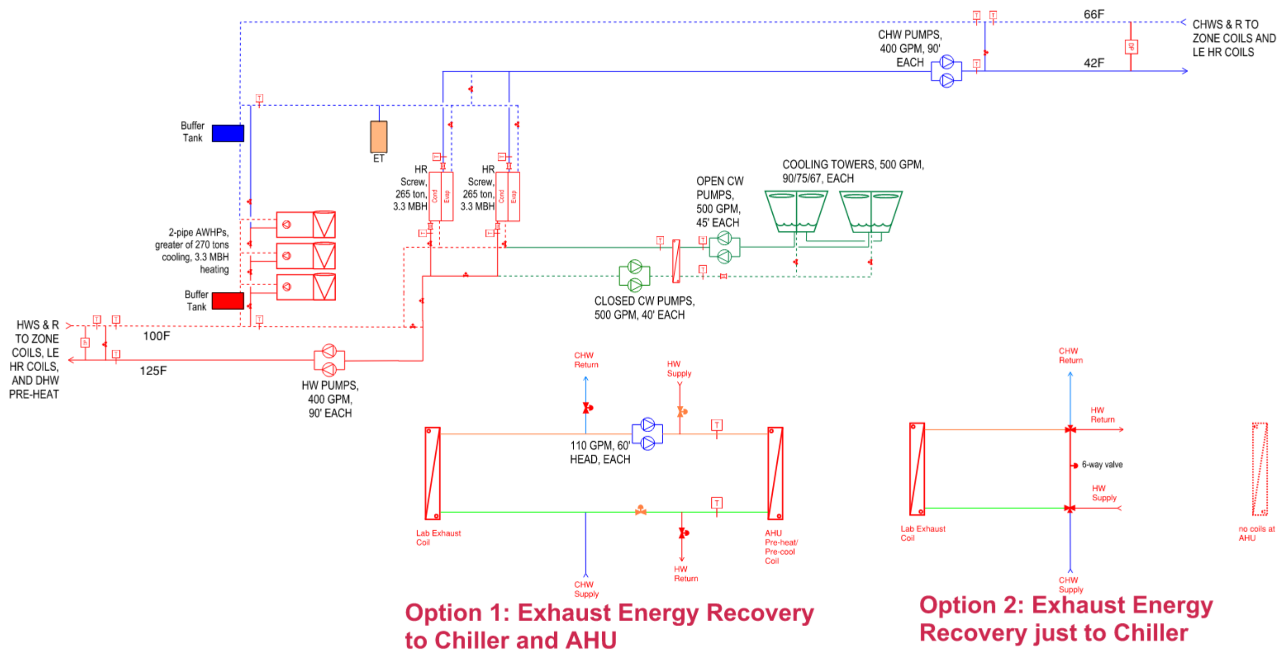
Exception 5 to Section 140.9(c)6: Exhaust systems requiring wash down systems such as exhaust systems dedicated to perchloric acid fume hoods.

Exception Requiring Alternate Compliance Path:

Exception 4 to Section 140.9(c)6: Buildings with an exhaust air heat recovery system and heat recovery chillers designed to provide at least 40% of the peak heating load from exhaust heat recovery.

Using the exhaust airstream for heat absorption/rejection by a heat recovery chiller system has similar energy savings to the proposed heat recovery requirement. Figure 10-25: Schematic of Lab Exhaust to Heat Recovery Chiller is a schematic of a lab system with options to recover heat to both the chiller and AHU or just to the chiller. When heat is recovered just to the chiller, a 6-way control valve is used at the heat recovery coil (like the changeover zone coils) so that the heat recovery coil can be used as a heat source (when there is a net heating load) or a heat sink (when there is a net cooling load).

Figure 10-25: Schematic of Lab Exhaust to Heat Recovery Chiller



Source: California Statewide CASE Team

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.

Reheat Limitations

Additions and alterations are exempt if the additions or alterations are to an existing air handling system serving existing-to-remain zones without both heating/cooling. If the addition or alteration is to an air handling system that does not already serve existing-to-remain zones without both heating/cooling, then the requirement applies.

Exhaust Air Heat Recovery

Additions and alterations are exempt if the addition or alteration is to an existing lab exhaust system that does not already have exhaust air heat recovery. If the existing exhaust system already includes exhaust air heat recovery or if the addition or alteration is a new lab exhaust system, then the requirement applies.

Compressed Air Systems

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.

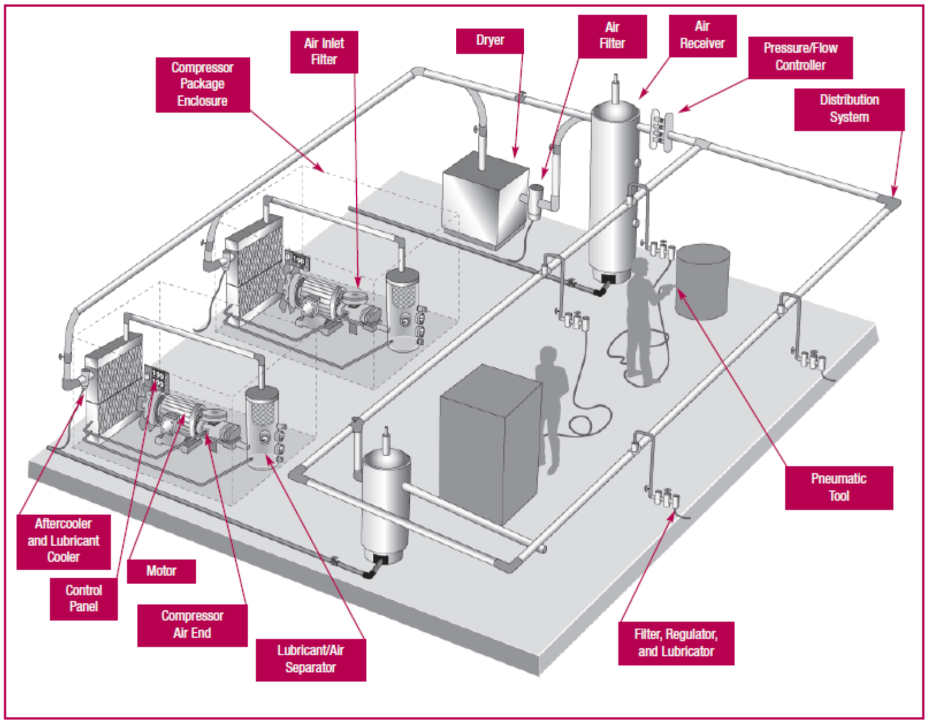
Key terms and definitions:

- 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.
- 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 require that the single compressor be treated as a trim compressor.
- 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.
- 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.
- 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.
- 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.
- 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.
- 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.

- Distribution Piping: Distribution piping includes all piping after the main header and transports air to service lines.
- 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 10-26: Typical Compressed Air System Components



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

Mandatory Measures

Please refer to Chapter 10.8.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Trim Compressor and Storage

Please refer to Chapter 10.8.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Controls

Please refer to Chapter 10.8.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Monitoring

Please refer to Chapter 10.8.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Leak Testing

Please refer to Chapter 10.8.2.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Pipe Sizing

Please refer to Chapter 10.8.2.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Compressed Air System Acceptance

Please refer to Chapter 10.8.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.8.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.8.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Process Boilers

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.

All process piping operating at 105°F or higher shall comply with Section 120.3. Steam pipes, hot water pipes, heated tanks, and vessels shall be insulated in accordance with Section 120.3. New requirements for process pipe insulation are listed in Process Pipe Insulation.

Mandatory Measures

Combustion Air

Please refer to Chapter 10.9.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Combustion Air Fans

Please refer to Chapter 10.9.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Excess Oxygen ≥ 5 MMbtu/h

Please refer to Chapter 10.9.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.9.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Elevators

Please refer to Chapter 10.10.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Mandatory Measures

Reference: Section 120.6(f)

Elevator Lighting Power Density

Please refer to Chapter 10.10.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Elevator Ventilation CFM Fan Performance

Please refer to Chapter 10.10.2.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Elevator Lighting and Fan Shutoff Control

Please refer to Chapter 10.10.2.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.10.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.10.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Escalators and Moving Walkways

Overview

Please refer to Chapter 10.11.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Mandatory Measures

Reference: Section 120.6(g)

Escalator and Moving Walkway Speed Control

Please refer to Chapter 10.11.2.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.11.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.11.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Controlled Environment Horticulture

Overview

Section 120.6(h) sets efficiency standards for controlled environment horticulture (CEH) spaces. These standards are divided into two groups: indoor growing facilities, with a skylight area to roof areas ratio less than 50 percent, and greenhouse CEH facilities with 50 percent or more of the roof is glazed. For indoor growing 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 help reduce the energy usage of horticulture facilities. These requirements are not dependent on the type of crop being grown in the facility.

Mandatory Measures

Reference: Section 120.6(h)

There are three main mandatory requirements in this section:

- Indoor horticultural growing facilities - Section 120.6(h)1 - 2,
- Greenhouse facilities - Section 120.6(h)3 - 4, and
- Horticultural lighting - Section 120.6(h)5

Indoor Growing Dehumidification

Reference: Section 120.6(h)1

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

- Dehumidifiers 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 10-33

Question:

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

Answer:

The Department of Energy's Compliance Certification Management System, https://www.regulations.doe.gov/certification-data/CCMS-4-Dehumidifiers.html#q=Product_Group_s%3A%22Dehumidifiers%22, maintains a database of products that have been certified to meet the federal requirements.

Indoor Growing Electrical Power Distribution

Reference: Section 120.6(h)2

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.

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 are required to have a photosynthetic photon efficacy (PPE) of 2.3 micromoles per joule. This required efficacy level is currently only provided by high efficacy light emitting diodes (LEDs) as legacy high pressure sodium and other sources have lower PPEs and would not comply. Some LED sources, especially those not specifically designed for horticultural use, also have PPEs that are lower than the required 2.3 μm/J. Often the PPE of horticultural luminaires is published. Otherwise, divide the full output photosynthetic photon flux by the rated input watts at full light output. Note that some manufacturers publish the total photon flux from their fixtures; this is not photosynthetic photon flux. Photosynthetic photon flux, as defined by ANSI/ASABE S640, is measured only for the wavelengths between 400 and 700 nanometers. LEDs have become a standard choice as growers have become familiar with their impacts on plant quality and yield while saving energy and reducing internal gains and cooling loads.

Example 10-34

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.

Conditioned Greenhouse Building Envelope

Reference: Section 120.6(h)3

Section 120.6(h)3 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.

Conditioned Greenhouse Space-Conditioning Systems

Reference: Section 120.6(h)4

Section 120.6(h)4 requires that space-conditioning systems used in conditioned greenhouses for plant production must meet requirements applicable to the systems

Horticultural Lighting

Reference: Section 120.6(h)5

Pertaining to Section 120.6(h)5, 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:

- Luminaires with removable lamps shall contain lamps with a lamp photosynthetic photon efficacy (PPE) of at least 2.3 micromoles per joule (μm/J); all other luminaires shall have a luminaire photosynthetic photon efficacy of at least 2.3 μm/J.
- 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.
- Multilevel lighting controls shall be installed and comply with Section 130.1(b).

A lamp and luminaire PPE of 2.3 μm/J will require the use of high efficacy light emitting diodes (LEDs) as legacy high pressure sodium and other sources have lower PPEs and would not comply.

Example 10-34

Question:

How do these Energy Code requirements interact with any CalCannabis requirements?

Answer:

Energy Code 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." Energy Code requirements apply regardless of the type of crop grown in these spaces. The Energy Code 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 Energy Code 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 the Energy Code and are not relevant to Energy Code compliance.

Prescriptive Measures

Please refer to Chapter 10.12.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Section 141.1(c) 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 or 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 must meet the requirements of 120.6(h)3 and 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 Energy Code requirements.

Example 10-35

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)5?

Answer:

Lamp replacements do not trigger the horticulture lighting efficacy alteration requirements. Only replacements 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 10-36

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)5?

Answer:

Lamp replacements do not trigger the horticulture lighting efficacy alteration requirements. Only replacements 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.

Steam Traps

Overview

Please refer to Chapter 10.13.1 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Mandatory Measures

Please refer to Chapter 10.13.2 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Central Steam Trap Fault Detection and Diagnostic Monitoring

Please refer to Chapter 10.13.3 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Steam Trap Fault Detection

Please refer to Chapter 10.13.4 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Steam Trap Strainer Installation

Please refer to Chapter 10.13.5 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Steam Trap System Acceptance

Please refer to Chapter 10.13.6 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Prescriptive Measures

Please refer to Chapter 10.13.7 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Additions and Alterations

Please refer to Chapter 10.13.8 of the *2022 Nonresidential and Multifamily Compliance Manual*.

Process Pipe Insulation

Overview

Section 120.3 applies to a covered process requirement for process piping where the fluid inside the pipe is heated to 105°F or above or chilled to 60°F or below. In addition to space cooling, space heating, and service water-heater systems covered in Section 120.3, all process heating and process cooling piping has been added to pipes requiring insulation as stated in Tables 120.3-A-1 and 120.3-A-2.

Mandatory Measures

Reference: Section 120.3

There are three main mandatory requirements in this section:

- General requirements - Section 120.3(a),
- Insulation protection - Section 120.3(b),
- Insulation thickness - Section 120.3(c)

There are several exceptions to the general and mandatory insulation requirements:

- Factory-installed piping for appliances or space-conditioning equipment.
- Piping that conveys fluids with a design operating temperature range between 60°F and 105°F.
- Where the energy loss from the un-insulated pipe will not increase the building’s energy use.
- Piping that penetrates framing members for the distance of the framing penetration.
- Equipment such as pumps, steam traps, blow-off valves, and piping within process equipment.
- Valves, strainers, coil u-bends, and air separators with at least 0.5 inches of insulation.

General Requirements

Reference: Section 120.3(a)4, Section 120.3(a)5

There are two new requirements for pipe insulation:

- Process heating system piping where the fluid is heated above 105°F. This includes steam, steam condensate and hot water fluid distribution systems for heating a process unrelated to space conditioning or service water-heating.
- Process cooling system piping where the fluid is cooled below 60°F. This includes refrigerant suction, chilled water, and brine fluid distribution systems for cooling a process unrelated to space conditioning.

Insulation Protection

Reference: Section 120.3(b)

Pipe insulation shall be protected from damage to maintain its integrity and its ability to reduce energy loss. The insulation should be protected against weather damage, water, and mechanical damage. Protection, at minimum, should include the following:

- Pipe insulation located outdoors should be protected by outdoor rated insulation cover that is resistant to rain, wind, and solar radiation.
- Cold pipes located outside should be protected by a vapor retarder. All penetrations and joints shall be sealed.
- Underground piping should be mechanically protected against physical damage or getting crushed by providing adequate clearance around the pipe and using physical barriers such as non-crushable casing or sleeves.

Insulation Thickness

Reference: Section 120.3(c)

Pipes should be insulated with a minimum thickness listed in Table 120.3-A-1 and Table 120.3-A-2 for the corresponding pipe diameters and fluid temperatures if insulation conductivity is provided in the relevant temperature range. If the conductivity value is greater than the range given for your application, you must calculate an adjusted required thickness to achieve the minimum required R-value using the following equation.

The following equation is used to calculate the minimum thickness.

$$T = PR[(1 + \frac{t}{PR})^{\frac{K}{k}} - 1]$$

Where,

- T = insulation thickness for material with conductivity K, inches.
- PR = actual outside radius, inches.
- t = Insulation thickness from Table 120.3-A-1 and Table 120.3-A-2, inches.
- K = Conductivity of alternate material at the mean rating temperature indicated in Table 120.3-A-1 and Table 120.3-A-2 for the applicable fluid temperature range, in Btu-inch per hour per square foot per °F.
- k = The lower value of the conductivity range listed in Table 120.3-A-1 and Table 120.3-A-2 for the applicable fluid temperature range, Btu-inch per hour per square foot per °F.

Prescriptive Measures

There are no prescriptive measures for pipe insulation.

Additions and Alterations

All newly installed pipes for process heating or cooling as part of process equipment additions or pipes that are relocated or added as part of an alteration must be insulated with a minimum insulation thickness or R value shown in Table 120.3-A-1 and Table 120.3-A-2.

Example 10-37

A manufacturing facility has newly installed 2” hot water pipe as part of their new Clean in Place (CIP) system. The water is heated to 180°F.

Question 1:

What is the minimum insulation thickness required?

Answer:

The minimum insulation thickness must be 2” or greater.

Question 2:

For the previously described facility, most of the piping is located indoors but some are located outdoors and exposed to the weather. They have installed standard insulation cover that is not outdoor rated. Are they in compliance with the code?

Answer:

No. The sections of the pipe that are located outdoors must have an insulation jacket that is outdoor rated with a water retardant cover and shielding from solar radiation.

Example 10-38

A manufacturing facility operates a boiler at 100 psig. The steam condensate piping passes through the warehouse then goes down the drain. The warehouse is not conditioned. The steam condensate is at 205°F.

Question:

Does the condensate piping need to be insulated?

Answer:

Per Exception 2 above, because the condensate is not recovered and the building is not conditioned, the heat loss of the condensate piping does not increase the building energy use.

Example 10-39

A refrigerated warehouse has 0.75” refrigerated line with refrigerant at -10°F.

Question:

What is the minimum insulation thickness required?

Answer:

The minimum insulation thickness must be 1.0”.

Example 10-40

A facility uses 20°F chilled water to cool their product after it exits the oven. The chilled water pipe is 4" in diameter.

Question:

What is the minimum insulation thickness required?

Answer:

The minimum insulation thickness must be 1.5".

Example 10-41

A facility uses steam at 265°F. They have 200' of bare steam pipe that needs to be insulated. The outside pipe diameter is 1.5". They have an insulation material that they want to use with a thermal conductivity of 0.44.

Question:

What is the insulation thickness required?

Answer:

The conductivity is outside the range shown in Table 120.3-A-1 and Table 120.3-A-2; therefore, the insulation thickness must be calculated.

$$T = 0.75\ [(1 + \frac{4.5}{0.75})^{\frac{0.44}{0.29}} - 1] = 13.6"$$

The minimum required insulation thickness is 13.6."