2028 TITLE 24 REFRIGERATION COMMENT LETTER

June 23, 2026

California Energy Commission

Docket No. 25-BSTD-03

The California Statewide Codes and Standards Enhancement (CASE) initiative presents recommendations to support California Energy Commission (CEC) efforts to update California's Energy Code (Title 24, Part 6), including revisions to simplify and clarify existing code requirements. Three California Investor-Owned Utilities (IOUs)—Pacific Gas & Electric, San Diego Gas & Electric, and Southern California Edison—sponsored this effort.

This letter recommends adjustments to the code language for refrigerated warehouses and commercial refrigeration to address stakeholder concerns, ensure consistent interpretation of the requirements, and support strong compliance. This letter does not recommend changes that will increase the stringency of existing requirements or recommend that existing requirements be applied more broadly. As such, a cost-effectiveness analysis or use of the full CASE report template was determined to not be necessary.

1. Measure Description and Justification

1.1 Proposal Description

1.1.1 Variable Compressor Volume Index (Vi)

Context

New screw compressors at refrigerated warehouses with greater than 150 motor horsepower (hp) must include the ability to automatically vary the Vi in response to operating pressures (2025 Title 24, Part 6, Section 901.1.7.4 [120.6(a)5D]). This capability is known as "variable Vi" and is sometimes called volume ratio. Stakeholders have indicated that motor size is a poor predictor of compressor size and that compressor displacement would better identify where variable Vi is available and cost-effective.

Proposal

1) Change the requirement threshold from motor hp to compressor displacement. The proposed value is 450 cubic feet per minute (cfm), which is effectively equivalent for the application prototypes and compressors considered when the requirement was adopted in 2013.
2) Exempt transcritical CO_2 compressors from the requirement.

1.1.2 CO_2 Gas Coolers

Context

Air-cooled gas coolers are prohibited for refrigerated warehouses in Climate Zones 9-15 and commercial refrigeration in Climate Zones 10-15 (2025 Title 24, Part 6, Sections 901.1.8.1 [120.6(a)8A] and 902.1.5.1 [120.6(b)5A]). Functionally, this is a requirement to use adiabatic gas coolers in these climate zones. This requirement also is assumed to apply to CO_2 condensing units with integral gas coolers, which are a new product. Stakeholders have indicated that this requirement is problematic for multiple reasons, including increased water use in water-scarce areas, increased maintenance costs, questionable operational efficiency, and lack of design flexibility.

Proposal

1) Remove the prohibitions of air-cooled gas coolers (Sections 901.1.8.1 [120.6(a)8A] and 902.1.5.1 [120.6(b)5A])
2) Require air-cooled gas coolers in Climate Zones 9-15 for refrigerated warehouses and Climate Zones 10-15 for commercial refrigeration to have a design leaving gas temperature less than or equal to the design dry-bulb temperature plus [TBD]°F, which shall represent equivalent annual energy performance through added surface area.[1]

[1] The Statewide CASE Team is currently conducting modeling to determine this approach temperature.

Include two exceptions to these requirements:

a. Energy-equivalent alternatives approved by the Executive Director that have been demonstrated to provide at least equal energy savings.
b. Gas coolers serving smaller systems:
    i. Freezers: 20 tons and lower of total heat of rejection
    ii. Coolers: 40 tons and lower of total heat of rejection

1.1.3 Acceptance Testing

Context

Acceptance testing is required for refrigerated warehouse and transcritical CO_2 refrigeration systems (2025 Title 24, Part 6, Sections 901.1.9 [120.6(a)7] and 902.1.6 [120.6(b)6]). The test procedures are outlined in Reference Appendices 7.10 and 7.20. Contractors have indicated that the current refrigeration acceptance tests are impractical to conduct, which may be leading project teams to skip or improperly perform them.

Proposal

1) Align the refrigeration acceptance tests with the existing mechanical system acceptance tests for HVAC in two areas:
    a. Allow functional tests to be executed independent of load and weather conditions.
    b. Allow field or factory calibration for sensors used for control.

1.1.4 Refrigerated Warehouse Condensers

Context

Refrigerated warehouse condensers must meet the efficiency requirements based on condenser type and refrigerant type in Table 901.1-C [Table 120.6-B] (2025 Title 24, Part 6, Section 901.1.6.7 [120.6(a)4G]). Stakeholders have expressed confusion over how to interpret the table.

Proposal

1) Modify the definition of condenser efficiency in Section 901.1.6.7 [120.6(a)4G] to read as follows:

   > Condenser efficiency is defined as the total heat of rejection (THR) capacity divided by all electrical input power including ~~fan~~ power of all fans at 100 percent fan speed, and power of spray pumps for evaporative condensers.

2) Add the modified definition of condenser efficiency as a footnote to Table 901.1-C [Table 120.6-B], so that the table reads as a standalone source of information.

1.1.5 Commercial Refrigeration Heat Recovery

Context

Since 2013, commercial refrigeration systems have been required to use heat recovery for HVAC space heating, using no less than 25% of the design total heat of rejection (THR) of all refrigeration systems that have design heat rejection rates of 150,000 Btu/h or greater (2025 Title 24, Part 6, Section 902.1.4.1 [120.6(b)4A]). This requirement has three exceptions:

1) Stores located in Climate Zone 15
2) HVAC systems or refrigeration systems that are reused for an addition or alteration
3) Stores where the design total heat of rejection of all refrigeration systems is less than or equal to 500,000 Btu/h

Stakeholders have indicated that the requirements are not clear, resulting in increased project cost and risk.

Proposal

1) In the 2028 code cycle, or sooner if the Commission seeks to address this earlier, issue a code clarification to explain when the heat recovery requirement and its exceptions apply and do not apply, with emphasis on scenarios such as the following:
   a. New HVAC equipment is added, but no new refrigeration equipment is added.
   b. New refrigeration equipment is added, but no new HVAC equipment is added.
   c. HVAC equipment and/or refrigeration equipment is retrofitted in a way that changes the system's design total heat of rejection.

    d. New HVAC equipment and new refrigeration equipment are added but the capacity is small enough that 25% of the new refrigeration equipment total heat of rejection cannot be used.
    e. Numerous refrigeration condensing units are added, where each individual unit has a design heat rejection rate of less than 150,000 Btu/h and the total combined design heat rejection rate is greater than 500,000 Btu/h.
    f. A site's refrigeration system serves only a small part of the store.
2) In the 2031 code cycle or later code cycles, reassess this requirement and its intent to update the code to improve feasibility and cost-effectiveness. Example areas for investigation include the following:
    a. Consider allowing heat recovery for domestic hot water heating.
    b. Consider a requirement that requires recovery of all cost-effective heat, not a fixed 25%.
    c. Consider an exception for sites that use heat pumps for space heating and domestic water heating.

1.1.6 WICF Exception

<u>Context</u>

In 2025, the refrigerated warehouse condenser requirements were amended to include exceptions for condensing units that are components of walk-in coolers or walk-in freezers within the scope of the Appliance Efficiency Regulations (California Code of Regulations, Title 20, Sections 1601 through 1608).

These exceptions apply to the following requirements:

1) Section 901.1.6.1 [120.6(a)4A]: Evaporative and water-cooled condenser approach temperatures
2) Section 901.1.6.2 [120.6(a)4B]: Air-cooled condenser approach temperatures
3) Section 901.1.6.3 [120.6(a)4C]: Adiabatic condenser dry-mode approach temperatures
4) Section 901.1.6.7 [120.6(a)4G]: Condenser efficiency limits listed in Table 120.6-B
5) Section 901.1.6.8 [120.6(a)4H]: Air-cooled condenser fin density limit

Equipment that is too large to realistically be *used* for walk-in coolers and freezers could nevertheless be *tested* and *rated* as if it were a federally regulated walk-in cooler or freezer. The current exception language would allow that equipment, just by virtue of being rated for an application it could not serve, to be exempt from five of Title 24's condenser requirements for refrigerated warehouses.

Proposal

1) Consider one of the following options:
    a. Remove the exceptions, as Section 901 [120.6] already, by its own terms, applies to only refrigerated warehouses that are greater than or equal to 3,000 square feet, and walk-ins, by definition, are spaces less than 3,000 square feet.
    b. Add a sizing requirement for the exempted condensing units, such as the exception threshold used in the 2022 Title 24 code that exempted systems with total compressor horsepower less than 100 hp.
    c. Modify the exception language to limit the exception to condensing units that are required to be certified as walk-ins as follows:

        Dedicated condensing units, as defined in 10 C.F.R. 431.302, that are ~~components of~~ subject to the Appliance Efficiency Regulations and required by those regulations to be certified as walk-in coolers or walk-in freezers ~~within the scope of the Appliance Efficiency Regulations~~.

1.1.7 Test Procedures for Evaporator Specific Efficiency

Context

In 2025, a code change was adopted to require fan-powered evaporators used in refrigerated warehouses to meet minimum specific efficiency requirements listed in Table 901.1-B [Table 120.6-A-2] (2025 Title 24, Part 6, Section 901.1.5.5 [120.6(a)3D]). Evaporator specific efficiency is defined as gross total refrigeration capacity divided by electrical input power at 100 percent fan speed, determined "following the [AHRI 420] test procedure." Currently, only one manufacturer appears able to comply. Stakeholders have indicated that this has effectively created a monopoly.

Proposal

1) In the 2028 code cycle, amend the evaporator specific efficiency requirements as follows:
    a. Remove test procedure references.
    b. Modify Table 901.1-B [Table 120.6-A-2] to align with the column headers and structure of the refrigerated warehouse condenser efficiency requirements table (Table 901.1-C [Table 120.6-B]).
    c. Add exception for coolers or freezers for which a licensed engineer has certified that the application has airflow and static pressure requirements that make compliance infeasible.
    d. Include footnote clarification that testing is not required.
    e. Include footnote definitions for rating condition terms.
2) In future code cycles, reassess the feasibility and cost-effectiveness of the evaporator specific efficiency values.

1.2 Proposal Justification

Refrigeration measures were first introduced in 2008 (for refrigerated warehouses) and 2013 (for commercial refrigeration). These provisions are located at Section 901.1 [120.6(a)] and Section 901.2 [120.6(b)], respectively.

1.2.1 Variable Vi

1.2.1.1 Background

Technology

Traditional screw compressors are fixed geometric compression devices that receive a volume of refrigerant gas at suction, trap that gas in the threads of the screw rotor meshing, and reduce the gas volume as the gas traverses the axis of the rotors. Volume index, or Vi, is the ratio of refrigerant gas volume at suction to the volume at discharge. Screw compressor designs can include fixed Vi, field-adjustable Vi, or variable Vi.

Dynamic adjustment of the location of the discharge port allows the compressor to closely match the system's prevailing discharge pressure as load and suction conditions change. This capability is known as variable Vi. It minimizes over- and under-compression of the refrigerant, which both represent thermodynamic irreversibilities. Figure 1 shows a schematic of a screw compressor that uses a traditional slide valve for capacity control and a secondary slide for variable Vi.

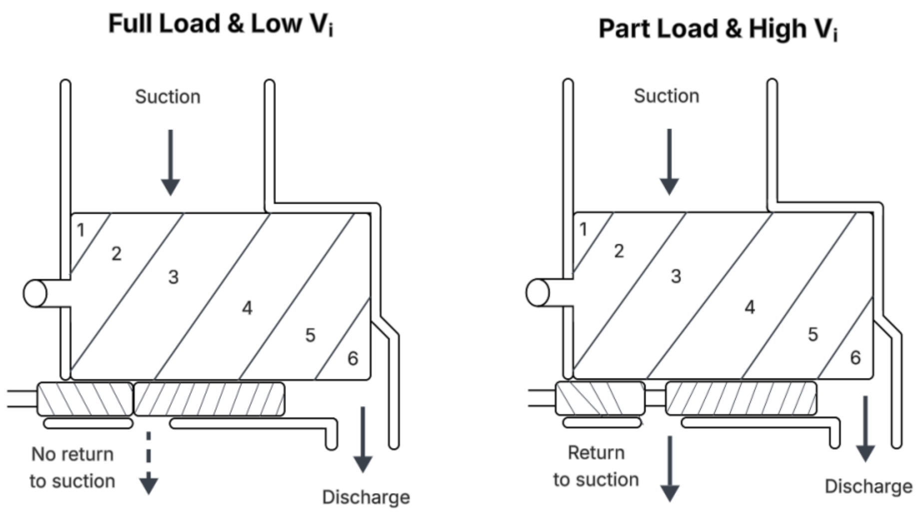

Figure 1: Examples of Screw Compressor Variable Vi and Capacity Slide Positions

Though beneficial in many applications, variable Vi also introduces new clearances into the compressor, which create an internal leakage path. The performance penalty for these clearances is disproportionately higher for smaller compressors.

The diminished net benefit of variable Vi for smaller compressors has become more relevant in recent years as the industry shifts from custom-designed, field-erected central refrigeration plants to modular low-charge package systems. Today the screw compressors in these packaged systems are commonly equipped with motors in the range of 75 to 250 motor hp.

Code

The 2013 CASE Report for the variable Vi measure did no quantitative analysis and claimed no statewide savings.[2] The justification for the measure was based on a survey of four compressor manufacturers about "typical size open drive screw compressors at RWH application conditions," and the report concluded that "most prominent screw compressor manufacturers already offer variable Vi control as a standard feature."

[2] California Utilities Statewide Codes and Standards Team, Refrigerated Warehouse: 2013 California Building Energy Efficiency Standards, Codes and Standards Enhancement (CASE) Initiative (California Statewide Utility Codes and Standards Program, October 31, 2011).

In general, stakeholders have indicated that the current variable Vi requirement is well suited for single-stage ammonia refrigeration systems and correctly targets larger compressors, where variable Vi is more beneficial and more available. However, the requirement has two deficiencies:

1) Motor horsepower is a poor predictor of compressor size,[3] and compressor size is largely what determines whether variable Vi is viable.
2) Transcritical CO_2 screw compressors have high working pressures. Based on stakeholder feedback, the slide assembly required for variable Vi is not feasible in transcritical CO_2 systems.

The proposed code changes address these deficiencies. Using a code trigger based on compressor displacement, which expresses the compressor's physical size in terms of volumetric flow rate of suction gas, would better reflect market availability and cost-effectiveness for variable Vi. In addition, an exception for transcritical CO_2 screw compressors would avoid requiring a compressor design feature where it is technically infeasible and commercially unavailable.[4]

For the application prototypes and compressors considered in the 2013 code cycle, the proposed 450-cfm displacement trigger and the existing 150-hp motor size trigger would have effectively the same applicability, with minimal edge cases. The 450-cfm trigger is more consistent with current equipment and with market changes in the refrigerated warehouse sector. Some large-displacement, low-power compressors (e.g., distributed systems using R-1234yf refrigerant) would come into code coverage and align with where variable Vi is consistently available and beneficial. Some smaller compressors (e.g., distributed low-charge ammonia packages) would fall out of code coverage, which resolves the issue of variable Vi being less available and less beneficial for those applications. This assessment is qualitative and informed by industry stakeholder feedback. It is also consistent with available compressor performance data and how the original 2013 requirement was adopted.

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[3] Note that different suction pressures and refrigerants can result in widely varying input brake horsepower for the same compressor body size. As one industry rule of thumb, relative to screw compressors that use ammonia, screw compressors that use R-1234yf, a low-pressure refrigerant, can nearly double the compressor body size for the same motor horsepower.

[4] Transcritical CO_2 systems also have low pressure ratios, so even if variable Vi were to become feasible in future years, it would have low comparative benefit.

Stakeholders

Highlights from the Statewide CASE Team's interviews with stakeholders about variable Vi are shown in Table 1.

Table 1: Stakeholder Feedback on Variable Vi Requirement

Stakeholder Segment	No. of Firms	Feedback Highlights*
Equipment Manufacturers	4	"For transcritical CO_2, with 1,000 psi differential across the compressor, moving the slides is not feasible, so we use a fixed port that is custom ported to the application." "Horsepower varies a lot by application. It makes sense to use displacement." "A better perspective would be to require a tolerance on actual Vi versus ideal Vi, which could apply to fixed Vi with custom porting and variable Vi." "Variable Vi has an unavoidable geometry penalty, and this penalty gets worse for smaller compressors." "Variable Vi has been overbranded—why do single-stage variable Vi when you could do multistage without it? How much sense does it make for a low-compression-ratio, high-temperature application?" "For low-charge ammonia systems, many compressors don't have variable Vi, and some don't even have it as an available option."

*Interview quotes are paraphrased by the Statewide CASE Team

1.2.1.2 Benefits

Both the efficiency benefit and market availability of variable Vi depend on compressor size, not motor size. Variable Vi introduces additional internal leakage paths. Because manufacturing tolerances do not shrink with compressor size, these leakage paths are proportionally larger and represent a greater geometric penalty in smaller compressors, and larger compressors have a greater net efficiency benefit. As a result, manufacturers tend to offer variable Vi only on larger compressors. Given that displacement is a direct measure of compressor size, the proposed code trigger targets variable Vi where it is both available and beneficial.

In addition, stakeholder feedback has indicated that transcritical CO_2 screw compressors cannot feasibly incorporate variable Vi. Providing an exception for these compressors, which are relatively new to the market, will allow designers the flexibility to use transcritical CO_2 systems for refrigerated warehouses.

1.2.2 CO2 Gas Coolers

1.2.2.1 Background

Technology

In transcritical CO_2 refrigeration systems, a gas cooler performs the same heat rejection role as a condenser in conventional subcritical systems. Because CO_2 has a critical temperature of only 88°F, far lower than most refrigerants, any ambient temperature above roughly 75°F or 80°F forces the high-pressure side of the system to operate supercritically. In this condition, the high-side refrigerant exists as a single-phase fluid that no longer condenses from vapor to liquid. A significant volume of flash gas is generated when the refrigerant is throttled to lower pressures, which lowers system efficiency and capacity. In practice, transcritical operation tends to occur for only several hundred hours per year, depending on climate zone.

To manage the inefficiencies of transcritical operation, manufacturers have several options, such as the following:

1) Ejectors of various kinds, which use gas cooler discharge gas as the motive gas to entrain and partially compress vapor from the outlet of an evaporator or flash tank, recovering expansion work that would otherwise be lost across an expansion valve
2) Parallel compression, which uses a dedicated compressor to draw flash gas—together with any other vapor lifted to the intermediate-pressure receiver, such as by ejectors—and compresses it to the gas cooler pressure over a smaller pressure ratio than the main compressors, reducing overall compression work
3) Flooded evaporators, which raise the suction pressure
4) Air-cooled gas coolers with increased surface area, which lower the high-side pressure
5) Adiabatic gas coolers, which lower the high-side pressure

Adiabatic gas coolers use wetted media to pre-cool the gas cooler's intake air via evaporation of water. During transcritical operation in hot weather, adiabatic operation results in lower high-side operating pressures, leading to less flash gas generation, improved system efficiency, and higher system capacity. In practice, adiabatic gas coolers can have high water use and maintenance costs. Manufacturers frequently recommend that the pads be replaced annually. If the pads are not properly maintained, dust loading can reduce airflow and result in a year-round energy penalty.

Other market options similar to adiabatic pre-cooling are spray and mister technologies in which nozzles spray water into the airstream entering the gas cooler. The water droplets may be large and directly wet the coil, or the water may be atomized into a fine mist that partially or totally evaporates before reaching the coil. Based on stakeholder feedback, these technologies are lower-cost options for achieving partial evaporative cooling, but they can result in higher water drift losses and accelerate fouling from water droplets depositing on the heat transfer surface. The operational and maintenance impacts of these technologies are not well documented.

It should also be noted that certain U.S. cities in dry climates are banning evaporative heat rejection. In Nevada, the City of Las Vegas and the City of Henderson have effectively prohibited evaporative cooling in new commercial and industrial buildings. In the Las Vegas Valley Water District, evaporative cooling is defined as follows:

> . . . any type of cooling technology, device or equipment that utilizes the evaporation of water as part of the cooling process. Evaporative coolers include, but are not limited to, swamp coolers and cooling towers, but do not include misting systems[5]

The practical consequence for refrigeration projects in these areas is that adiabatic gas coolers cannot be assumed permissible. If local water districts in California seek to adopt similar prohibitions to control water use, this would conflict with the current Title 24 prohibition on air-cooled gas coolers.

<u>Modeling</u>

To support this proposed code change, the Statewide CASE Team is currently performing a modeling study to determine what design approach temperature is required for a system with an air-cooled gas cooler to have equal energy performance with a code-compliant adiabatic gas cooler.

This initial scope of work includes the following:

1) Hourly annual analysis in DOE-2.2R with post-processing in Microsoft Excel
2) Climate Zone 12 (Sacramento)
3) One prototype refrigeration system—large warehouse taken from previous code cycles

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[5] Las Vegas Valley Water District, "Service Rules: Effective January 1, 2026," https://www.lvvwd.com/assets/pdf/service-rules.pdf.

4) Load profiles are taken from modeling performed in previous code cycles for the prototype
5) Equipment selection based on 2025 Title 24 requirements

Subsequent work aims to incorporate additional climate zones and the supermarket prototype from previous code cycles, and to abstract the results to smaller systems.

<u>Code</u>

Since 2022, Title 24 prohibits the use of air-cooled gas coolers in transcritical CO_2 refrigeration systems in Climate Zones 9 through 15 for refrigerated warehouses and Climate Zones 10 through 15 for commercial refrigeration. This effectively requires use of adiabatic gas coolers in these climate zones. It is unclear whether spray and mister technologies comply with this prohibition of air-cooled gas coolers.

Figure 2 highlights Climate Zones 9 through 15 for refrigerated warehouses. Note that these Climate Zones represent the Central Valley and the inland portions of Southern California, which are the hottest areas of the state.

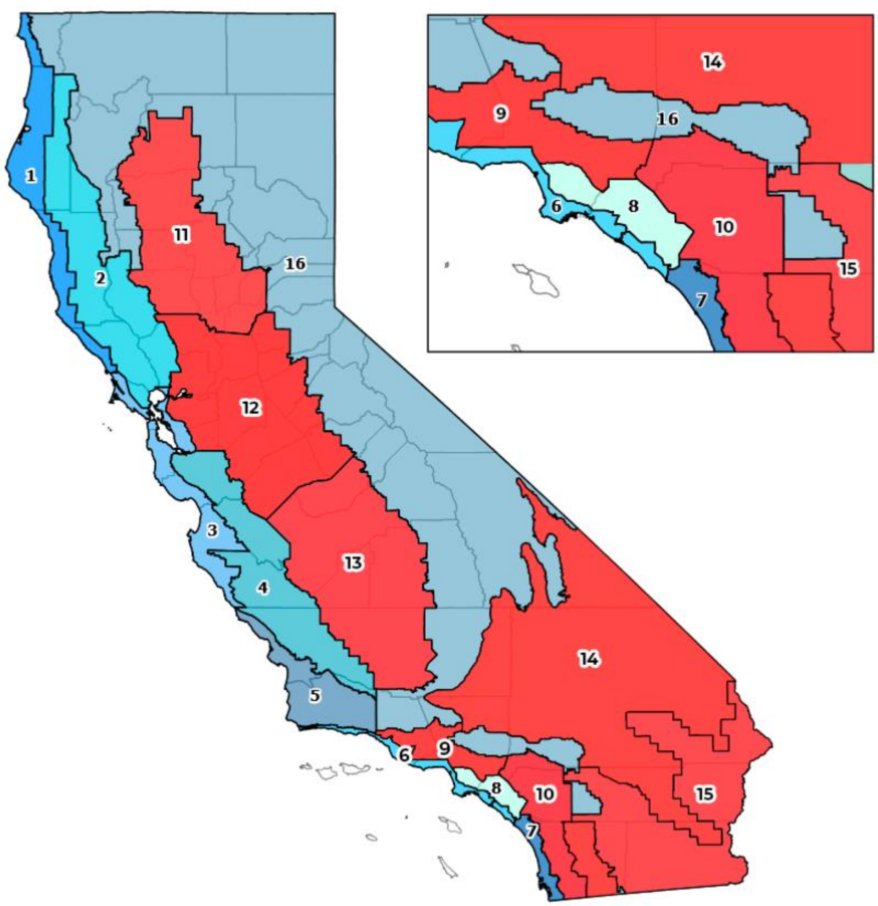

Figure 2: Highlighting of California Climate Zones 9 through 15

Stakeholders

Stakeholder feedback has indicated that the prohibition of air-cooled gas coolers has led to nearly exclusive use of adiabatic gas coolers for new projects in these climate zones. Highlights from the Statewide CASE Team's interviews with stakeholders about CO_2 gas coolers are shown in Table 2.

Table 2: Stakeholder Feedback on Transcritical CO_2 Gas Coolers

Stakeholder Segment	No. of Firms	Feedback Highlights*
Owners & Operators	1	"Our concerns about adiabatic gas coolers include reliability, maintenance, water use, and whether we can trust the adiabatic mode design capacity." "In our experience, most manufacturers that produce transcritical CO_2 gas coolers don't have an adiabatic option."
Equipment Manufacturers	2	"Some customers get adiabatic gas coolers and never turn the water on." "Maintenance question: How often do you run adiabatically? Is it 200 hours per year? Or is it 400 to 600 hours per year? This makes a big difference on how much media loading you get." "You need to consider year-round operation. Air-cooled gas coolers can get great performance at lower ambient temperatures." "Adiabatic is just one option for hot climates. Other options include various kinds of ejectors, parallel compression, increasing the system design pressure, and use of flooded evaporators to raise the suction pressure." "Not all adiabatic gas coolers perform the same. Single-pass systems dump a lot of water to drain. Others spray water onto the coil, which saves the cost of wetted media, but has drift losses and can foul the coil." "Adiabatic only sees savings when the water is turned on, and most of the time it uses more energy than air-cooled." "For a size threshold, 20 tons and above would make sense to me for adiabatic as an option, where that is the sum of the low-temperature and medium-temperature tonnage."

Consulting Engineers	1	"Adiabatic gas coolers get clogged with dust easily, and this can force the system into supercritical operation for most of the year—an unintended consequence. I see this all the time and have many photos."

*Interview quotes are paraphrased by the Statewide CASE Team

Through informal stakeholder conversations, the Statewide CASE Team also made the following observations about CO_2 condensing units:

1) Outside of the climate zones in California where they are banned, many original equipment manufacturers (OEMs), including all European ones that the Statewide CASE Team is aware of, appear to use air-cooled units with increased gas cooler surface area and lower-power fans.
2) Nearly all units use variable-speed compressor and gas cooler fan control—i.e., continuous variable-capacity control.
3) Even small units appear to include the same flash tank control valves and control logic used in large systems.
4) All units appear to have advanced control and monitoring capability, unlike previous halocarbon technologies that were effectively electro-mechanical.
5) Two OEMs use a conventional water spray in lieu of adiabatic precooling wetted media, and the efficiency and efficacy of these spray systems are not well documented.
6) Water-cooled units are available and designed for chilled water production, allowing subcritical operation. Target markets include central chilled water plants at pharmaceutical and hotel sites, among others.
7) Low-temperature units appear to be commonly designed as two-stage systems, using either two compressors or an internally compound two-stage compressor to receive flash gas from the flash tank at an intermediate pressure.

1.2.2.2 Benefits

Allowing other options that have equivalent energy performance and lower water use, such as air-cooled gas coolers with larger surface area or parallel compression or ejectors, gives designers flexibility to minimize costs without sacrificing energy savings. In addition, smaller systems would not be forced to use adiabatic pre-cooling which is often costly or unavailable at smaller sizes.

1.2.3 Acceptance Testing

1.2.3.1 Background

In the 2025 Title 24 Reference Appendices, Sections 7.10 and 7.20 provide acceptance tests for refrigerated warehouse and transcritical CO_2 systems, which include construction inspection and functional test procedures for electrical resistance underslab heating, evaporators, variable-speed screw compressors, and heat rejection systems (air-cooled, evaporative, and adiabatic condensers, and gas coolers). The intent of these procedures is to verify that refrigeration systems deliver expected energy performance. Via informal conversations with stakeholders, the Statewide CASE Team believes that some aspects of the test procedures are not practical for contractors to conduct and may be leading to low compliance.

The Statewide CASE Team proposes that the verification of sensor calibration and controls programming adopt a similar approach to what is currently used in the Title 24 Mechanical System Acceptance Tests (NA7.5), with emphasis in two areas:

1) Verify and document that sensors used for control "have been either factory or field calibrated."

2) Allow simulation of operating conditions to verify controls programming.

The Mechanical System Acceptance Tests are far more practical to conduct and should be used as a guide for revision of the Refrigeration System Acceptance Tests.

<u>Construction Inspection</u>

A sensor reading ±2°F in the right place is far more valuable than a NIST-verified sensor in the wrong place.

The current refrigeration acceptance tests require field technicians to verify sensors against NIST-traceable calibrated instruments, with tolerances of ±0.7°F for temperature and ±2.5 psi for pressure. In most cases, these are specified to be sensors "used for control" or "used by the controller." The Statewide CASE Team believes this is a misplacement of effort. In the Statewide CASE Team's experience, the most damaging sensor errors in refrigeration systems are not calibration errors. They are installation errors.[6] Examples include the following:

[6] To address installation errors, the current refrigeration acceptance tests require that ambient sensors not be placed in direct sunlight (or be provided with a solar shield) and that refrigerated space temperature and humidity sensors be mounted away from direct evaporator discharge air draft.

- Poor placement—for example, an outdoor air temperature sensor placed in direct sunlight or in proximity to machine heat.
- Missing sensor—the sensor was never installed.
- Field wiring issue—the sensor was installed properly but not properly wired.
- Programming issue—the sensor was installed and wired properly but not properly programmed into the control system.

The requirement of field sensor verification with NIST-traceable reference instruments is also not compatible with contractor practices, where it is common to have a single NIST-traceable reference at a contractor shop to which technicians' field measurement devices are then checked for calibration.

In contrast to the refrigeration acceptance tests, the Mechanical System Acceptance Tests require only factory or field calibration, without reference to NIST-traceable reference instruments. Examples include the need to verify and document field or factory calibration of discharge static pressure sensors used in supply fan variable flow controls and supply water temperature sensors used in supply water temperature reset controls. In certain cases, such as carbon dioxide control sensors, only factory calibration is permitted.

Functional Tests

Load and weather requirements are too rigorous for a one-time test.

The functional test procedures for evaporative condensers (NA7.10.3.1), air-cooled condensers (NA7.10.3.2), adiabatic condensers (NA7.10.3.3), variable-speed compressors (NA7.10.4.2), and transcritical CO_2 gas coolers (NA7.20.1.1.2 and NA7.20.1.1.3) all contain language requiring certain ambient conditions, system load, or both. This language includes requirements such as the following:

- "The system cooling load must be sufficiently high, and ambient conditions sufficiently below design . . ."
- "Be cognizant of / account for weather conditions in scheduling testing and, if necessary and possible, arrange to artificially increase or decrease evaporator loads in order to perform the Functional Testing at typical system conditions."
- ". . . ambient conditions sufficiently below the critical point . . ."
- ". . . ambient conditions sufficiently above the critical point . . ."

Achieving and scheduling for such conditions in the field is often onerous and impractical. In addition, the benefits of a one-time test are limited. It is best practice to analyze and optimize a system's setpoints over time, which is an activity that cannot occur until the system is subjected to real load and weather conditions, and this work cannot be captured in a one-time acceptance test.

Given the above, the Statewide CASE Team recommends that the Refrigerated Warehouse Refrigeration System Acceptance Tests and Transcritical CO_2 Systems Acceptance Tests be revised using the current Mechanical System Acceptance Tests as a guide, where test technicians are instructed to directly simulate conditions by manipulating controller inputs. Examples of instruction to simulate inputs in the Mechanical System Acceptance Tests include the following:

- Supply Water Temperature Reset Controls: “Change reset control variable to its maximum value” (NA7.5.6.2)
- Supply Fan Variable Flow Controls: “Simulate demand for full design airflow” (NA7.5.8.2)
- Demand Control Ventilation Systems: “Simulate a signal at or slightly above the CO_2 concentration setpoint required by §120.1(d)4C” (NA7.5.5.2)

1.2.3.2 Benefits

Acceptance testing must be completed for a certificate of occupancy to be issued. Based on informal conversations with stakeholders, the current acceptance test requirements are time-consuming and at times impractical for installing contractors to conduct. In some cases, this may lead project teams to not conduct proper testing or to skip testing. Adopting the proposed changes would make the test procedures more practical and effective.

1.2.4 Refrigerated Warehouse Condensers

1.2.4.1 Background

Section 901.1.6.7 [120.6(a)4G] requires refrigerated warehouse condensers to meet the efficiency requirements based on condenser type and refrigerant type in Table 901.1-C [Table 120.6-B], which is reproduced below in Figure 3.

Table 120.6-B FAN-POWERED CONDENSERS – MINIMUM EFFICIENCY REQUIREMENTS

CONDENSER TYPE	REFRIGERANT TYPE	MINIMUM EFFICIENCY	RATING CONDITION
Outdoor Evaporative-Cooled with THR Capacity > 8,000 MBH	All	350 Btuh/watt	100°F Saturated Condensing Temperature (SCT), 70°F Outdoor Wetbulb Temperature
Outdoor Evaporative-Cooled with THR Capacity < 8,000 MBH and Indoor Evaporative-Cooled	All	160 Btuh/watt	100°F Saturated Condensing Temperature (SCT), 70°F Outdoor Wetbulb Temperature
Outdoor Air-Cooled	Ammonia	75 Btuh/watt	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Drybulb Temperature
Outdoor Air-Cooled	Halocarbon	65 Btuh/watt	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Drybulb Temperature
Adiabatic Dry Mode	Halocarbon	45 Btuh/watt	105°F Saturated Condensing Temperature (SCT), 95°F Outdoor Drybulb Temperature
Indoor Air-Cooled	All	No requirement	No requirement

Figure 3: 2025 Title 24, Table 120.6-B Refrigerated Warehouse Fan-Powered Condenser Efficiency Requirements

In 2025 Title 24, the definition of condenser efficiency appears on the previous page, and stakeholders have expressed confusion over how to interpret the conditions of the input electrical power required to calculate condenser efficiency. The Statewide CASE Team proposes two solutions to lower the burden of code interpretation:

1) Amend the definition of condenser efficiency in Section 901.1.6.7 [120.6(a)4G] to clarify that the input fan power refers to the "power of all fans at 100 percent fan speed."
2) Reproduce this amended definition as a footnote to Table 901.1-C [Table 120.6-B].

1.2.4.2 Benefits

Adopting this clarification will make it easier for end users to use and understand the code.

1.2.5 Commercial Refrigeration Heat Recovery

1.2.5.1 Background

Market

California's retail sector is currently implementing a high volume of refrigeration retrofits and additions. There are two main drivers of this activity:

1) To accommodate growing merchandising space, large retailers are adding numerous supplemental systems.
2) Facing pressure from the 2027 and 2030 refrigerant global warming potential (GWP) requirements, retailers are doing refrigeration system conversions.

Code

It is not clear how the heat recovery requirement should be interpreted for refrigeration retrofits that do not affect the HVAC system, as well as HVAC retrofits that do not affect the refrigeration system. This confusion leads to either (1) overcomplication of projects or (2) falling short of the requirement during permitting. Both are negative outcomes for industry. The Statewide CASE Team recommends issuance of a code clarification in the short term and a more comprehensive analysis of the measure compliance options and cost-effectiveness in the long term.

1.2.5.2 Benefits

Stakeholders have expressed that the heat recovery requirements are confusing, particularly when considering refrigeration upgrades that do not affect the HVAC system or HVAC upgrades that do not affect the refrigeration system. The confusion can lead to project permit delays or unnecessary project cost and complexity. Clarifying the requirements and their exceptions, especially for various retrofit and remodel scenarios, will make it easier for owners to comply with the code, reducing project cost and complexity.

1.2.6 WICF Exception

1.2.6.1 Background

Market

Under current Title 24 language, a builder can install a condensing unit certified for use as a walk-in cooler or freezer (“walk-in”) to become exempt from fan-powered condenser requirements, even when the space is not itself a walk-in, nor the condensing unit part of a walk-in.

Code

In its current form, the Title 24 condenser requirements for refrigerated warehouses allow manufacturers to certify condensing units as “walk-in coolers and freezers” to avoid compliance. Sections 901.1.6.1 through 901.1.6.3 [120.6(a)4A, B, and C] require new fan-powered condensers on refrigeration systems to meet certain design saturated condensing temperatures, varying if they are evaporatively cooled, water-cooled, air-cooled, or adiabatic. Section 901.1.6.7 [120.6(a)4G] requires fan-powered condensers to meet specified efficiency requirements. Section 901.1.6.8 [120.6(a)4H] requires air-cooled condensers to have a maximum fin density.

The Department of Energy (DOE) regulates walk-ins, defined as “an enclosed storage space including, but not limited to, panels, doors, and refrigeration system, refrigerated to temperatures, respectively, above, and at or below 32 degrees Fahrenheit that can be walked into, and has a total chilled storage area of less than 3,000 square feet.”[7] As a general rule, preemption applies to walk-ins in the same way it applies to other “covered products.”[8]

A condensing unit is part of the walk-in’s “refrigeration system.” In practice, there are two types of walk-ins: (1) a manufacturer-assembled walk-in, or (2) a set of components, including refrigeration, doors, lights, windows, or walls, that is assembled on site to build a walk-in. Both are regulated by DOE.

The “basic model” that is certified to DOE is as follows:

> . . . all components of a given type of walk-in cooler or walk-in freezer (or class thereof) manufactured by one manufacturer, having the same primary energy source, and which have effectively identical electrical, physical, and functional (or hydraulic) characteristics that affect energy consumption, energy efficiency, water consumption, or water efficiency; and . . . With respect to panels, which do not have any differing features or characteristics that affect U-factor.[9]

A condensing unit, standing alone, that is not “of a given type” of walk-in is not federally regulated or required to certify. In other words, a condensing unit is subject to DOE regulation only when it is installed as part of a walk-in. This provides a basis for the proposed exception modification options presented in this letter.

1.2.6.2 Benefits

The current exception language allows for equipment that will never realistically be used as components of federally regulated walk-in coolers or walk-in freezers to be exempt from five of Title 24’s condenser requirements for refrigerated warehouses. Having an exception that is broader than needed reduces the savings from Title 24’s refrigeration requirements. Tightening the exception defends the savings that were projected when the measures were adopted. Though the savings were already claimed as part of those measures, this change would still result in increased savings relative to the status quo.

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[7] 10 CFR 431.302; see also 42 U.S.C. 6311(20) (similar definition).

[8] 42 U.S.C. 6316(h).

[9] 10 CFR 431.302 (defining “basic model”); see also 10 CFR 429.53 (certification requirements for walk-ins).

1.2.7 Test Procedures for Evaporator Specific Efficiency

1.2.7.1 Background

<u>Market</u>

In 2025, an evaporator specific efficiency measure for refrigerated warehouses was introduced that requires determining gross refrigeration capacity and electrical input power following the AHRI test procedures. Currently, only one manufacturer appears to comply with this requirement.

Based on stakeholder feedback, the Statewide CASE Team learned the following:

1) The one manufacturer that can comply with the AHRI 420 requirement certifies all evaporator designs to AHRI 420 based on testing in their own facility, with four evaporators tested in the first year and two evaporators per year thereafter.[10]
2) Other manufacturers do not wish to certify. They believe that the code and test method effectively require full certification or 100% testing, which would be prohibitively expensive, and are concerned that AHRI test procedures do not reflect efficiencies at applied conditions—e.g., the test is done with a frost-free coil and no fan external static pressure.
3) Refrigerated warehouse evaporator coils are typically custom designed based on application conditions using manufacturer-specific parametric design software.
4) Test labs may not be willing to run tests on evaporators with ammonia or flammable refrigerants due to safety concerns.

<u>Code</u>

Section 901.1.1.5 [120.6(a)3D] requires refrigerated warehouse fan-powered evaporators to meet the efficiency requirements based on evaporator type and application type in Table 901.1-B [Table 120.6-A-2] , which is reproduced below in Figure 4.

[10] For more background on AHRI 420 certification, see AHRI, "Operations Manual: Unit Coolers for Refrigeration Certification Program," November 2023, https://www.ahrinet.org/system/files/2023-10/UC_OM.pdf.

Table 120.6-A-2 FAN-POWERED EVAPORATORS – MINIMUM SPECIFIC EFFICIENCY REQUIREMENTS

Evaporator Type[1, 2]	Rating Condition	Efficiency (Btuh/Watt)	Test Procedure[3]
Direct Expansion, Ammonia Refrigerant, Cooler/Dock	Dry Coil +25°F saturated evaporating temperature +35°F entering drybulb temperature 0 in. water static pressure	35	AHRI 420
Direct Expansion, Ammonia Refrigerant, Freezer	Dry Coil -20°F saturated evaporating temperature -10°F entering drybulb temperature 0 in. water static pressure	25	AHRI 420
Liquid Overfeed, Ammonia Refrigerant, Cooler/Dock	Dry Coil +25°F saturated evaporating temperature +35°F entering drybulb temperature 0 in. water static pressure	50	AHRI 420
Liquid Overfeed, Ammonia Refrigerant, Freezer	Dry Coil -20°F saturated evaporating temperature -10°F entering drybulb temperature 0 in. water static pressure	45	AHRI 420
Direct Expansion, CO_2 Refrigerant, Cooler/Dock	Dry Coil +25°F saturated evaporating temperature +35°F entering drybulb temperature 0 in. water static pressure	35	AHRI 420
Direct Expansion, CO_2 Refrigerant, Freezer	Dry Coil -20°F saturated evaporating temperature -10°F entering drybulb temperature 0 in. water static pressure	25	AHRI 420
Liquid Overfeed, CO_2 Refrigerant, Cooler/Dock	Dry Coil +25°F saturated evaporating temperature +35°F entering drybulb temperature 0 in. water static pressure	50	AHRI 420
Liquid Overfeed, CO_2 Refrigerant, Freezer	Dry Coil -20°F saturated evaporating temperature -10°F entering drybulb temperature 0 in. water static pressure	45	AHRI 420
Direct Expansion, Halocarbon Refrigerant, Cooler/Dock	Dry Coil +25°F saturated evaporating dew point temperature +35°F entering drybulb temperature 0 in. water static pressure	45	AHRI 1250
Direct Expansion, Halocarbon Refrigerant, Freezer	Dry Coil -20°F saturated evaporating dew point temperature -10°F entering drybulb temperature 0 in. water static pressure	40	AHRI 1250

Notes:

1 Direct expansion: Evaporator in which leaving refrigerant vapor is superheated.

2 Liquid overfeed: Evaporator in which refrigerant liquid is supplied at a recirculation rate greater than 1.

3 Applicable test procedure and reference year are provided under the definitions.

Figure 4: Table 901.1-B [120.6-A-2]—Fan-Powered Evaporator Efficiency Requirements

The 2025 CASE report that proposed this measure (an addendum to the 2022 CASE Report) suggests that the basis for the measure’s feasibility was increased prevalence of manufacturer product selection software, not increased prevalence of laboratory testing. The 2025 results report states the following.[11]

> Evaporator specific efficiency was initially considered in the 2013 Title 24, Part 6 CASE Report, but the measure was ultimately not adopted because the research on evaporator ratings revealed challenges in getting the evaporator capacity and applied fan motor power at rated conditions.
>
> In recent years, more information has become available on evaporators as almost all manufacturers have product selection software, and the capacity ratings are becoming more standardized. Some manufacturers are now providing certified ratings in their product catalogues to provide more confidence in the capacity of the equipment being sold. Additionally, some manufacturers provide the applied fan power at the operating conditions.

The 2025 CASE report states that minimum specific efficiencies at the rating conditions excluded the least-efficient 40% of current coil designs and suggests that applied conditions were normalized to a dry-coil condition at a fixed temperature difference (TD), with additional normalization of fan static pressure for penthouses. The report did not mention normalization for other parameters, such as throw, fins per inch, or airflow to achieve space temperature uniformity. It is also unclear whether the evaluated sample of coils controlled for differences in how entering air temperature is defined (DTM versus DT1).[12]

Given these omissions in the 2025 CASE report, the Statewide CASE Team believes that the current minimum specific efficiency values may conflate inefficient evaporator designs with power-intensive applications, particularly in edge applications that inherently require higher airflow or higher static pressure. An evaluation of the minimum specific efficiency values is beyond the scope of this comment letter and should be considered for future code cycles.

The current condenser requirements for refrigerated warehouses (2025 Title 24, Part 6, Section 901.1.6.7 [120.6(a)4G]) specify minimum efficiencies at rating conditions but have no associated test procedure for determining total heat of rejection capacity or electrical input power. In practice, compliance relies on manufacturer-reported ratings,

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[11] Kyle Larson and Sara Hernandez, 2025 Energy Code Results Report: Refrigeration, Codes and Standards Enhancement (CASE) Initiative, Measure ID 25-NR-PROCESS-33 (California Statewide Codes and Standards Enhancement Program, November 2024), p. 119.

[12] DTM, used by some manufacturers, defines the entering air temperature as the mean room air temperature for capacity calculations. DT1 defines the entering air temperature as the coil inlet air temperature.

which is consistent with industry standard practice where manufacturers use internal models to determine equipment capacity and power requirements. For decades, manufacturers have guaranteed capacity ratings based on these model outputs, providing equipment that meets customer design loads at applied conditions. If the equipment does not perform at the rated capacity, manufacturers must fulfill warranty obligations.

Given the above, the Statewide CASE Team believes the test procedure requirements for evaporators are inconsistent with both industry practice and the approach used for condensers under the same section of Title 24. In addition, requiring physical testing or AHRI 420 and AHRI 1250 certification imposes a compliance burden that most manufacturers cannot realistically meet. In stakeholder discussions, three possible solutions have emerged to address these issues:

1) Remove the evaporator specific efficiency requirements given that feasibility is difficult to assess across applications and given that incremental savings are low following an earlier code cycle requirement for variable-speed fan control.
2) Remove references to test procedures and allow compliance through ratings obtained from manufacturer product selection software and add footnotes to the table to clarify definitions and requirements.
3) Reassess the minimum specific efficiency values and rating conditions for improved measure feasibility and normalization of application conditions.

The Statewide CASE Team recommends the second of these options for the 2028 code cycle, as it largely resolves the issue of market accessibility while preserving the efficiency requirement and its rating conditions until a more comprehensive assessment can be made.

1.2.7.2 Benefits

If adopted, this proposed code change would reopen the California market to more manufacturers of fan-powered evaporators. This would allow designers and contractors to build systems more competitively. Owners can receive more competitive bids, engineers will face less pressure to specify non-compliant equipment, and manufacturer reps in California can maintain sales and staffing.

2. Stakeholder Engagement to Inform Proposal

2.1 Utility-Sponsored Stakeholder Meeting

On March 19, 2026, the Statewide CASE Team virtually held a utility-sponsored stakeholder meeting to present several of the code change proposals: variable Vi, CO_2 gas coolers, acceptance testing, and refrigerated warehouse condenser requirements.[13] During the meeting, the team fielded questions and comments from three industry stakeholders.

2.2 Stakeholder Docketed Comments

Table 3 shows a summary of refrigeration stakeholder docketed comments for the 2028 Energy Code Pre-Rulemaking.[14]

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[13] For the stakeholder meeting posting and notes, see California Energy Codes & Standards, "Compressed Air and Refrigeration – Utility-Sponsored Stakeholder Meeting," March 19, 2026, https://title24stakeholders.com/event/compressed-air-and-refrigeration-utility-sponsored-stakeholder-meeting/.

For the presentation slide deck, see Ryan Swanson, "Refrigeration Comment Letter: Codes and Standards Enhancement (CASE) Proposal," March 19, 2026, https://title24stakeholders.com/wp-content/uploads/2026/03/2028-T24-Stakeholder-Round-2-Refrigeration-Comment-Letter.pdf.

[14] The full docket log can be accessed at CEC, "Docket Log," https://efiling.energy.ca.gov/Lists/DocketLog.aspx?docketnumber=25-BSTD-03.

Table 3: Summary of Stakeholder Docketed Comments

TN #	Individual (Organization)	Date	Issue(s) Summary	URL
269260	Joe Sanchez (BITZER US)	3/20/2026	– Variable Vi losses are worse for smaller diameter rotor screw compressors – A displacement threshold would be more beneficial for variable Vi, as horsepower varies based on application and refrigerant	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269260
269262	Joe Sanchez (BITZER US)	3/20/2026	– Adiabatic gas cooler requirement limits CO_2 adoption – Consider allowing other efficiency options, such as parallel compression, ejectors, and heat exchangers on the discharge of the gas cooler	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269262
269423	Randy Johnson (Walmart)	4/7/2026	– Heat recovery requirements are unclear when applied to existing facilities that may or may not have new HVAC&R systems added or reused – The lack of clarity is disruptive to project execution, cost, and risk	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269423&DocumentContentId=106516
269461	(Bruce V. Nelson Engineering, LLC)	4/10/2026	– Only one manufacturer can comply with the test procedure requirement in Table 120.6-A-2 – AHRI 420 is poorly written and does not provide value to end users	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269461
269497	(SubZero Constructors, Inc.)	4/15/2026	– Not all manufacturers offer variable Vi for all applications, which is problematic for national companies that want to use a single manufacturer – Variable Vi provides little or no benefit for booster and high-stage compressors in two-stage systems	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269497

TN #	Individual (Organization)	Date	Issue(s) Summary	URL
269498	(SubZero Constructors, Inc.)	4/15/2026	– Only one manufacturer certifies RWH evaporator coils to AHRI-420 – Section 120.6(a)3D and Table 120.6-A-2 limit stakeholder ability to build competitively in California	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269498
269537	William Slope	4/20/2026	– AHRI 420 standard rating conditions do not reflect industrial refrigeration applications – Specifying AHRI 420 for "test procedure" in Table 120.6-A-2 effectively creates a monopoly	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269537&DocumentContentId=106629
269827	Danielle Wright (North American Sustainable Refrigeration Council)	5/4/2026	– Mandatory adiabatic gas cooling has practical drawbacks, is ill-suited for smaller CO2 condensing units, and limits the adoption of non-flammable, low-GWP CO2 solutions – Action is needed before the 2028 code cycle, particularly because of state refrigerant regulations and the federal HFC phasedown	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269827&DocumentContentId=106972
269879	Jamon Craig (Heatcraft Refrigeration Products)	5/7/2026	– The adiabatic requirement does not consider factory-built CO2 remote condensing units and other smaller distributed systems and adds cost, maintenance burden, water-use concerns, and design complications – Allowing compliance through equivalent performance would give designers, manufacturers, and end users flexibility to achieve energy goals – Action is needed before the 2028 code cycle	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269879&DocumentContentId=107035

TN #	Individual (Organization)	Date	Issue(s) Summary	URL
269881	Nelson Pinheiro (Rivacold America, Inc.)	5/7/2026	– The adiabatic requirement does not consider factory-built CO2 remote condensing units or other smaller distributed systems which are an important low-GWP solution for small and large businesses – Our company avoids adiabatic solutions due to costs in system manufacturing, installation, and use – Water availability is a huge constraint for project approval in California	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269881&DocumentContentId=107038
269883	Glenn Barrett (DC Engineering)	5/7/2026	– Adiabatic gas coolers with small condensing units are not always economical, have negative impacts on water use and water waste, and carry increased costs when not implemented correctly – Allow an exception for small condensing units based on size or as a percentage of total store refrigeration load	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269883&DocumentContentId=107040
269889	Chris Malt (Carrier Commercial Refrigeration)	5/8/2026	– It does not make sense to require adiabatic in a state where water is scarce – Adiabatic can be unnecessarily burdensome for factory-built CO2 condensing units – Alternative measures can provide more efficiency – Europe is ahead of North America in refrigeration technology and system efficiency strategies, and many European countries have already moved away from, or have restricted use of, water due to environmental concerns	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269889&DocumentContentId=107046
269890	(Refplus)	5/8/2026	– Adiabatic cooling is the best way to achieve efficiency and reliability while keeping design and maintenance simple, and	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269890&DocumentContentId=107047

TN #	Individual (Organization)	Date	Issue(s) Summary	URL
			this is especially true for condensing units in warmer and drier climates – The specific efficiency of 165 Btu/W [*sic*] for adiabatic mode is unrealistic for condensing units and should be investigated	
269932	Kurt Liebendorfer (EVAPCO)	5/12/2026	– Change the Section 120.6(a)5C [*sic*] requirement for screw compressor variable Vi from a threshold of 150 hp to 750 m3/hr, as this would be more accurate for defining compressor size and better suits technology availability for smaller compressors and low-charge ammonia packages	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269932&DocumentContentId=107085
269933	(Omni Mechanical Solutions)	5/12/2026	– Adiabatic requirement can add cost, maintenance burden, water-use concerns, and design complications – Allowing compliance through equivalent performance would give manufacturers, engineers, and end users flexibility to achieve energy goals – CO2 condensing units are an important low-GWP solution and a clear compliance path should be preserved for these systems – Action is needed before the 2028 code cycle	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269933&DocumentContentId=107086

TN #	Individual (Organization)	Date	Issue(s) Summary	URL
269985	Scott Martin, Senthilkumar Shanmugam, John Murray (Hillphoenix)	5/15/2026	– Removal of adiabatic requirement will benefit end users, industry, utilities, and engineering professionals – More municipalities are considering restricting use of adiabatic cooling due to water scarcity and drought, which can delay projects – Allow compliance via equivalent performance – Action is needed before the 2028 code cycle	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269985&DocumentContentId=107136
269997	Ignacio Varela Chaparro (Kysor Warren Epta US)	5/15/2026	– Adiabatic cooling can add cost, maintenance burden, water-use concerns, and design complications – The adiabatic requirement can significantly restrict equipment availability, particularly for smaller capacity ranges – Many end users are hesitant to use water-based solutions for refrigeration due to risk of water leaks and roof damage – Action is needed before the 2028 code cycle	https://efiling.energy.ca.gov/GetDocument.aspx?tn=269997&DocumentContentId=107148
270003	Ron Shebik (Hussmann Corporation)	5/15/2026	– Alternative approaches to adiabatic can be used to achieve strong performance in hot climates: parallel compression, ejectors, subcooling, improved gas cooler design and control – Adiabatic cooling introduces higher first cost, increased maintenance burden, and reliability concerns in areas with poor water quality or seasonal operation – The incremental cost of adiabatic disproportionately affects smaller systems without commensurate benefits – In certain jurisdictions, water considerations may outweigh the energy benefits of adiabatic – Action is needed before the 2028 code cycle	https://efiling.energy.ca.gov/GetDocument.aspx?tn=270003&DocumentContentId=107155

TN #	Individual (Organization)	Date	Issue(s) Summary	URL
270097	Eric M. Smith (International Institute of All-Natural Refrigeration)	5/19/2026	– Proposal to change variable Vi requirement based on displacement is appropriate, and motor power is perhaps not a legitimate metric – Requiring refrigerated warehouse evaporators to comply with AHRI-420 is problematic, as only one OEM claims compliance and use of the test method and the minimum values in the code is questionable – Adiabatic cooling can be problematic in smaller facilities without dedicated maintenance teams, and many CO_2 condensing units that are now available have not incorporated adiabatic precooling, so consider allowing other technologies that meet an equivalent annual energy performance	https://efiling.energy.ca.gov/GetDocument.aspx?tn=270097&DocumentContentId=107261
270659	Trevor Hegg (Evapco)	6/17/2026	– Establishing particular efficiency requirements for system components is not valuable if measured efficiencies are not certified or independently verified. – If it is not required that a component be certified, but certified options are available, code requirements for certified products should be lower in order to encourage companies to certify their efficiency ratings, which would ultimately promote greater efficiency. – The requirements, including definitions and conditions, for determining refrigerated warehouse evaporator specific efficiency must be clearly stated and use common definitions and conditions. This should involve reference to AHRI 420 conditions and methods without actually requiring mandatory testing, certification, or compliance verification.	https://efiling.energy.ca.gov/GetDocument.aspx?tn=270659&DocumentContentId=108248

2.3 Stakeholder Interviews

Table 4 shows a summary of the stakeholder interviews conducted in preparation of this comment letter.

Table 4: Stakeholder Interviews

Stakeholder (No. of Staff Present)	Date	Takeaways
Association #1 **(1)**	5/18/2026	– The 20-ton (freezer) and 40-ton (cooler) thresholds are sound. The real problem is timing. Relief is needed n not next code cycle. – Many CO_2 systems are anticipated to go in before January 1, 2029, (2028 Title 24 code) so small systems need immediate relief. – A lot of near-term activity is small (≤10-ton) as modular add-ons in additions and remodels at existing sites – Early CO_2 adopters span big-box, pharmacy, convenience-store, quick-serve restaurant, and biotech, driven by HFC-reduction goals. – Air-cooled alternatives would help smaller systems reach compliance.
Consultant #1 **(1)**	2/9/2026	– AHRI 420 is poorly written and model-specific (non-parametric)—effectively written by and for a single manufacturer, the only firm in the AHRI certification directory – The parametric AHRI 410 would be a better fit or simply state a required specific-efficiency number and letting manufacturer and client agree. – No other equipment (compressors, condensers) faces an AHRI mandate in Title 24. Evaporators are being singled out. – Real efficiency comes from good design, maintenance, operation, part-load performance, benchmarking, and energy dashboards. – TCCO_2 efficiency is overstated versus low-charge ammonia. Adiabatic media clogs (the "world's best air filters") and can force year-round supercritical operation. – Evaporative gas coolers are better than adiabatic.

Stakeholder (No. of Staff Present)	Date	Takeaways
Contractor #1 **(2)**	4/30/2026	– Whole-system efficiency and COP would be a better basis than per-component rules, because good controls save far more than incrementally better hardware. – AHRI 420 is effectively a near-monopoly. Compressors and condensers face no equivalent test mandate, so singling out evaporators is hard to justify. – On variable Vi, booster compressors shouldn't need it. A swept-volume threshold makes sense, with high-stage treated differently than boosters. – The adiabatic threshold should be higher—closer to 50 tons than 20. Water is scarcer than energy in California, so air-cooled alternatives should be allowed (parallel compression, ejectors). – Many plants are oversized (≈2 times what they need). A load-sizing guideline (IIAR or ASHRAE) would be better than the "350 sf/ton" rule of thumb.
Manufacturer #1 **(1)**	1/28/2026	– Variable Vi carries an unavoidable slider-geometry penalty and doesn't pay off on smaller compressors. It has been overbranded and is of doubtful value for higher-temperature applications. – A motor-horsepower trigger set "without other information" is a poor basis. Displacement is the purest way to describe a compressor's size. Ratings should assume 60 Hz. – A performance-based path would be better than prescribing technology. "Prescribing technology paints a corner that's hard to get out of." – Developing whole-system "digital twin" software to evaluate efficiency across the system is better than component-by-component. "We don't ever want compressors regulated at the energy level."
Manufacturer #2 **(2)**	2/11/2026	– All of their compressors are already variable Vi (ammonia, 1234yf) down to small sizes—no issue there. – TCCO2 screws are the exception: the ≈1,000 psi differential makes moving the slides impractical, so they use a fixed custom port plus variable-speed capacity control. – A dedicated exception for TCCO2 screw compressors is warranted, because project-by-project exceptions are too burdensome. – On the variable Vi trigger, ≈150 hp is about the right cutoff. Displacement could work.

Stakeholder (No. of Staff Present)	Date	Takeaways
Manufacturer #3 **(1)**	2/20/2026	– Adiabatic is a maintenance nightmare. Good in some climates, but not all. – CO_2 is a big California market for them (≥3,000 sq ft), and they may move forward with bolt-on CO_2 kits. – Europe is ahead: where water isn't available, they simply build more efficient systems, such as those using ejectors.
Manufacturer #4 **(1)**	2/23/2026	– Performance-based requirements would be better than prescriptive ones, which limit innovation across the many emerging CO_2 technologies and permutations. – CO_2 efficiency is highly geography-sensitive (dry- and wet-bulb). Adiabatics use less water than some alternatives but still consume it, and water is sometimes more precious than electricity. – Getting too prescriptive on small systems (≤10 tons) makes CO_2 extremely expensive—below 10 tons, conventional refrigerant may cost $1,000-$2,000/ton and CO_2 can be ≈10 times that. – Requirements should be segregated by capacity range: adiabatic grows more attractive and more apples-to-apples with other refrigerants at larger capacities (≈100 tons and up). – Real-world adiabatic problems are common: dirt clogs the wetted media, water distribution across the pads is a perennial concern, and without commissioning the equipment often doesn't operate as intended.

Stakeholder (No. of Staff Present)	Date	Takeaways
Manufacturer #5 **(1)**	3/2/2026	– Adiabatic is just one of several hot-climate CO_2 fixes (ejectors, parallel compression, higher design pressure, flooded evaporators)—and it uses water in water-scarce areas of California. – Not all adiabatic designs are equal: single-pass dumps water, spray-on-coil causes drift and condenser fouling, and some owners never turn the water on. – An adiabatic size threshold around 20 tons (summed low- and medium-temp) makes sense. It has no place on the small systems that stores are increasingly adding. – Title 24 gas-cooler TDs are poorly informed—the 6°F air-cooled TD is too low (2–3°F would be better) and the 15°F adiabatic dry-coil TD is far too high (should match the air-cooled value). – Ejector technology and service expertise are scarce in the U.S., so owners default to standard boosters. – CO_2 performs well under AWEF, which rates at the lower ambient temperatures where CO_2 (which can go down to ≈50°F condensing) performs well.
Manufacturer #6 **(3)**	3/10/2026	– The variable Vi pain point is low-charge ammonia packages (50–300 hp) that already use VSDs but lack variable Vi—an exception should apply when VSDs are used, since speed control is more efficient anyway. – A displacement-based trigger would be an improvement. – On AHRI 420: they are certified, but anyone can self-represent as "rated in accordance with" it—the bigger problem is apples-to-oranges methods, where a European "average room temperature" ΔT inflates ratings versus 420's face-of-coil ΔT. AHRI-420 helps end users compare units (e.g., 3-fan versus 5-fan) on a common basis.
Manufacturer #7 **(1)**	4/22/2026	– Regulating the tolerance of actual versus ideal Vi, or an application COP target, would be better than regulating the hardware itself. – Displacement (swept volume) is a better variable Vi trigger than motor horsepower, which varies too much by application. – Variable Vi matters most for the stage rejecting heat to ambient. The low stage of two-stage and cascade systems are good exception candidates. – Their small screws (71–600 cfm) use stepped Vi, with continuously variable Vi at ≥600 cfm.

Stakeholder (No. of Staff Present)	Date	Takeaways
Manufacturer #8 **(1)**	5/4/2026	– The fan-powered-evaporator specific-efficiency requirement needs to spell out what compliance actually entails (every model tested? certified? audited?). Certification is not the same as Title 24 compliance. – Inlet gas temperature should be added to the gas-cooler specific-efficiency tables, since it anchors the LMTD for a sensible heat exchanger. – Specific-efficiency limits should consider hot climates, where induced-draft fans aren't viable and forced-draft fans use more energy. – AHRI 1250 (the 2020 version DOE uses) would be a cleaner, better-written alternative to AHRI 420. – The market-impact analysis that justified adopting AHRI 420 should be made available. – Adiabatic is problematic in water-scarce areas. In arid climates energy is easier to secure than water, making dry/air-cooled units more beneficial.
Owner #1 **(1)**	3/26/2026	– A minimum size threshold for TCCO2 gas coolers would help—TCCO2 is already costly and adiabatic performance is questionable, and most TCCO2 units lack an adiabatic option anyway. – Their TCCO2 condensing units run 3-4 times the cost of HFC units, so owners won't game a threshold by stacking small units. A 20-ton minimum (low- plus medium-temp) with energy-equivalent alternatives would work, ideally aligned with CARB (pounds of charge). – Adiabatic concerns center on reliability, maintenance, water use, and whether the adiabatic-mode design capacity can be trusted. – The retail heat-recovery requirement is hard to interpret for refrigeration-only retrofits and remodels that never touch the HVAC system. This results in either overcomplicated projects or falling short during permitting. – An interim CEC clarification is needed now. Then do a later code revision to address summer heat rejection not being aligned with HVAC loads, year-round domestic hot water not being a code option, and the challenge that systems are often spread across the store. – Money spent to comply with heat recovery is money not spent on other, potentially better, efficiency measures.

3. Proposed Code Language

3.1 Guide to Markup Language

The proposed changes to the standards, Reference Appendices, and the ACM Reference Manuals are provided below. Changes to the 2025 documents should be marked with dark blue underlining (new language) and ~~strikethroughs~~ (deletions).

3.1.1 Energy Code (Title 24, Part 6)

SUBCHAPTER 9 – PROCESS SYSTEMS AND EQUIPMENT

SECTION 901 – REFRIGERATED WAREHOUSES (NEWLY CONSTRUCTED, ADDITIONS, ALTERATIONS)

901.1 *[Section 120.6(a)]* **Mandatory requirements (Newly Constructed, Additions, Alterations).**

Refrigerated warehouses that are greater than or equal to 3,000 square feet and *refrigerated spaces* with a sum total of 3,000 square feet or more that are served by the same refrigeration system shall meet the requirements of Section 901.1 *[Section 120.6(a)]*.

Refrigerated spaces that are less than 3,000 square feet shall meet the requirements of the *Appliance Efficiency Regulations* for walk-in *coolers* or *freezers* contained in the *Appliance Efficiency Regulations* (California Code of Regulations, Title 20, Sections 1601 through 1608).

…

901.1.5 *[Section 120.6(a)3]* **Evaporators.**

New fan-powered evaporators used in *coolers* and *freezers* shall conform to the requirements of this section.

…

901.1.5.5 *[Section 120.6(a)3D]* **Specific efficiency.**

Fan-powered evaporators utilizing volatile refrigerants shall meet the applicable efficiency requirements listed in Table 901.1-B *[Table 120.6-A-2]*.

Evaporator specific efficiency is defined as the gross total refrigeration capacity (Btu/h) divided by the electrical input power (Watts) at 100 percent fan speed at the rating conditions listed in Table 901.1-B *[Table 120.6-A-2]* ~~following the test procedure listed in Table 120.6-A-~~2. Except as defined in Table 901.1-B [*Table 120.6-A-2*], all conditions and evaporator coil construction shall be as furnished for, and consistent with, the application.

EXCEPTION 1 to Section 901.1.5.5 *[Section 120.6(a)3D]*: Evaporators designed solely for the purpose of quick chilling/freezing of products, including but not limited to spaces with design cooling capacities of greater than 240 Btu/hr-ft2 (2 tons per 100ft2).

EXCEPTION 2 to Section 901.1.5.5 *[Section 120.6(a)3D]*: Coolers or freezers within refrigerated warehouses for which a licensed engineer has certified that the application has airflow or static pressure requirements that make compliance infeasible.

Table 901.1-B *[120.6-A-2]* FAN-POWERED EVAPORATORS – MINIMUM SPECIFIC EFFICIENCY REQUIREMENTS

Evaporator Type[1, 2]	Refrigerant Type	Minimum Specific Efficiency (Btuh/Watt)[3]	Rating Condition[4, 5, 6, 7]	~~Test Procedure~~[3]
Direct Expansion, ~~Ammonia Refrigerant,~~ Cooler/Dock	Ammonia	35	Dry Coil, +25°F saturated evaporating temperature, +35°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Direct Expansion, ~~Ammonia Refrigerant,~~ Freezer	Ammonia	25	Dry Coil, -20°F saturated evaporating temperature, -10°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Liquid Overfeed, ~~Ammonia Refrigerant,~~ Cooler/Dock	Ammonia	50	Dry Coil, +25°F saturated evaporating temperature, +35°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Liquid Overfeed, ~~Ammonia Refrigerant,~~ Freezer	Ammonia	45	Dry Coil, -20°F saturated evaporating temperature, -10°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Direct Expansion, ~~CO2 Refrigerant,~~ Cooler/Dock	CO_2	35	Dry Coil, +25°F saturated evaporating temperature, +35°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Direct Expansion, ~~CO2 Refrigerant,~~ Freezer	CO_2	25	Dry Coil, -20°F saturated evaporating temperature, -10°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Liquid Overfeed, ~~CO2 Refrigerant,~~ Cooler/Dock	CO_2	50	Dry Coil, +25°F saturated evaporating temperature, +35°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~

Liquid Overfeed, ~~CO2 Refrigerant,~~ Freezer	CO_2	45	Dry Coil, -20°F saturated evaporating temperature, -10°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 420~~
Direct Expansion, ~~Halocarbon Refrigerant,~~ Cooler/Dock	Halocarbon	45	Dry Coil, +25°F saturated evaporating dew point temperature, +35°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 1250~~
Direct Expansion, ~~Halocarbon Refrigerant,~~ Freezer	Halocarbon	40	Dry Coil, -20°F saturated evaporating dew point temperature, -10°F entering drybulb temperature, 0 in. water static pressure	~~AHRI 1250~~

Notes:

1. Direct expansion: Evaporator in which leaving refrigerant vapor is superheated.
2. Liquid overfeed: Evaporator in which refrigerant liquid is supplied at a recirculation rate greater than 1.
3. ~~Applicable test procedure and reference year are provided under the definitions.~~ Testing of evaporators is not required for compliance with the specific efficiency values in this table.
4. Dry coil is the condition in which no condensate or frost forms on the coil.
5. Saturated evaporating temperature is the saturation temperature corresponding to the refrigerant pressure at the evaporator coil outlet.
6. Saturated evaporating dew point temperature is the dew point temperature corresponding to the refrigerant pressure at the evaporator coil outlet.
7. Entering drybulb temperature is the air drybulb temperature at the evaporator coil inlet.

901.1.6 *[Section 120.6(a)4]* **Condensers.**

New fan-powered *condensers* on new refrigeration systems shall conform to the requirements of this section.

…

901.1.6.7 *[Section 120.6(a)4G]* **Condenser efficiency.**

Fan-powered *condensers* shall meet the *condenser* efficiency requirements *listed* in Table 901.1-C *[Table 120.6-B]*. *Condenser* efficiency is defined as the *total heat of rejection* (THR) capacity divided by all electrical input power including ~~fan~~ power of all fans at 100 percent fan speed, and power of spray pumps for evaporative *condensers*.

Exception 1 to Section 901.1.6.7: *Adiabatic condensers* with ammonia as refrigerant.

Exception 2 to Section 901.1.6.7: Condensing units that are components of walk-in *coolers* or walk-in *freezers* within the scope of the *Appliance Efficiency Regulations*.

TABLE 901.1-C FAN-POWERED CONDENSERS – MINIMUM EFFICIENCY REQUIREMENTS

CONDENSER TYPE	REFRIGERANT TYPE	MINIMUM EFFICIENCY	RATING CONDITION
Outdoor Evaporative-Cooled with THR Capacity > 8,000 MBH	All	350 Btuh/watt	100°F Saturated Condensing Temperature(SCT), 70°F Outdoor Wetbulb Temperature
Outdoor Evaporative-Cooled with THR Capacity < 8,000 MBH and Indoor Evaporative-Cooled	All	160 Btuh/watt	100°F Saturated Condensing Temperature(SCT), 70°F Outdoor Wetbulb Temperature
Outdoor Air-Cooled	Ammonia	75 Btuh/watt	105°F Saturated Condensing Temperature(SCT), 95°F Outdoor Drybulb Temperature
Outdoor Air-Cooled	Halocarbon	65 Btuh/watt	105°F Saturated Condensing Temperature(SCT), 95°F Outdoor Drybulb Temperature
Adiabatic Dry Mode	Halocarbon	45 Btuh/watt	105°F Saturated Condensing Temperature(SCT), 95°F Outdoor Drybulb Temperature
Indoor Air-Cooled	All	No requirement	No requirement

Footnotes to Table 901.1-C *[Table 120.6-B]*:

1. *Condenser* efficiency is defined as the *total heat of rejection* capacity divided by all electrical input power including power of all fans at 100 percent fan speed, and power of spray pumps for evaporative *condensers*.

…

901.1.7.4 *[Section 120.6(a)5D]* **Compressor volume ratio control.**

New screw compressors with ~~nominal electric motor power greater than 150 HP~~ compressor displacement of 450 cfm or greater shall include the ability to automatically vary the compressor volume ratio (Vi) in response to operating pressures.

Exception to 901.1.7.4: *Transcritical CO_2 refrigeration systems.*

901.1.8 *[Section 120.6(a)8]* **Transcritical CO_2 gas coolers.**

New fan-powered *gas coolers* on all new *transcritical CO_2 refrigeration systems* shall conform to the requirements of this section.

901.1.8.1 Air-cooled gas coolers. *[Section 120.6(a)8A, 8C]*

~~Air-cooled *gas coolers* are prohibited in *Climate Zones* 9 through 15.~~

The design leaving gas temperature for air-cooled gas coolers shall be less than or equal to the design dry-bulb temperature plus the value specified for the applicable climate zone:

i. Climate Zones 1, 3, 5, 6, 7, and 16: 6°F.

ii. Climate Zones 2, 4, and 8: 8°F.

iii. Climate Zones 9 through 15: [TBD]°F.

Exception 1 to Section 901.1.8.1: The requirement of Section 901.1.8.1 does not apply to transcritical CO_2 refrigeration systems in Climate Zones 9 through 15 that achieve at least equal energy savings through alternative means approved by the Executive Director.

Exception 2 to Section 901.1.8.1: The requirement of Section 901.1.8.1 does not apply to air-cooled gas coolers on systems with a total heat of rejection of:

i. 20 tons (240,000 Btu/h) or less for systems serving freezers; or

ii. 40 tons (480,000 Btu/h) or less for systems serving coolers.

~~**901.1.8.2** *[Section 120.6(a)8B]* **Design leaving gas temperature – air-cooled gas cooler.**~~

~~Design leaving gas temperature for air-cooled *gas coolers* shall be less than or equal to the design dry-bulb temperature plus 6°F.~~

~~**Exception to Section 901.1.8.2:** Design leaving gas temperature for air-cooled *gas coolers* in *Climate Zones* 2, 4 and 8 shall be less than or equal to the design dry-bulb temperature plus 8°F.~~

901.1.8.~~3~~2 *[Section 120.6(a)8C]* **Design leaving gas temperature – adiabatic gas cooler.**

Design leaving gas temperature for adiabatic *gas coolers* necessary to reject the design *total heat of rejection* of a refrigeration system assuming *dry mode* performance shall be less than or equal to the design dry-bulb temperature plus 15°F.

901.1.8.~~4~~3 *[Section 120.6(a)8D]* **Variable speed fans and fan controls.**

All *gas cooler* fans shall be continuously variable speed, with the speed of all fans serving a common *condenser* high side controlled in unison.

901.1.8.~~5~~4 *[Section 120.6(a)8E]* **Gas cooler pressure control.**

While operating below the critical point, the *gas cooler* pressure shall be controlled in accordance with Section 901.1.6.6 *[Section 120.6(a)4F]*.

901.1.8.~~6~~5 *[Section 120.6(a)8F]* **Gas cooler pressure reset.**

While operating above the critical point, the *gas cooler* pressure setpoint shall be reset based on ambient conditions such that the system efficiency is maximized.

901.1.8.~~7~~6 *[Section 120.6(a)8G]* **Condensing temperature setpoint.**

The minimum *condensing temperature* setpoint shall be less than or equal to 60°F for systems utilizing air-cooled *gas coolers*, evaporative-cooled *gas coolers*, adiabatic *gas coolers*, air or water-cooled *fluid coolers* or cooling towers for heat rejection.

Exception to Section 901.1.8.~~7~~6: *Transcritical CO_2 refrigeration systems* with a design intermediate saturated suction temperature greater than or equal to 30°F shall have a minimum *condensing temperature* setpoint of 70°F or less.

901.1.8.~~8~~7 *[Section 120.6(a)8H]* **Gas cooler efficiency.**

Fan-powered *gas coolers* shall meet the *gas cooler* efficiency requirements *listed* in Table 901.1-D *[Table 120.6-C]*. *Gas cooler* efficiency is defined as the *Total Heat of Rejection* (THR) capacity divided by all electrical input power (fan power at 100 percent fan speed).

TABLE 901.1-D [Table 120.6-C] TRANSCRITICAL CO_2 FAN-POWERED GAS COOLERS – MINIMUM EFFICIENCY REQUIREMENTS

CONDENSER TYPE	**REFRIGERANT TYPE**	**MINIMUM EFFICIENCY**	**RATING CONDITION**
Outdoor Air-Cooled	Transcritical CO_2	160 Btuh/watt	1400 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature
Adiabatic Dry Mode	Transcritical CO_2	90 Btuh/watt	1100 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature

SECTION 902
COMMERCIAL REFRIGERATION
(NEWLY CONSTRUCTED, ADDITIONS, ALTERATIONS)

902.1 *[Section 120.6(b)]* **Mandatory requirements (Newly Constructed, Additions, Alterations).**

Retail food or beverage stores with 8,000 square feet or more of *conditioned floor area*, and that utilize either refrigerated display cases, or walk-in *coolers* or *freezers* shall meet all applicable state and federal appliance and equipment standards consistent with Section 101.2 *[Section 110.0]* and Section 101.3 *[Section 110.1]* or, for *equipment* not subject to such standards, the requirements of Sections 902.1.1 through 902.1.4.

…

902.1.5 *[Section 120.6(b)5]* **Transcritical CO_2 Gas Coolers.**

New fan-powered *gas coolers* on all new *transcritical CO_2 refrigeration systems* shall conform to the requirements of this section.

902.1.5.1 *[Section 120.6(b)5A]* **Air-cooled gas coolers ~~prohibition~~.**

~~Air-cooled *gas coolers* are prohibited in *Climate Zones* 10 through 15.~~

The design leaving gas temperature for air-cooled gas coolers shall be less than or equal to the design dry-bulb temperature plus the value specified for the applicable climate zone:

i. Climate Zones 1 through 9 and 16: 6°F.

ii. Climate Zones 10 through 15: [TBD]°F.

Exception 1 to Section 902.1.5.1: The requirement of Section 902.1.5.1 does not apply to transcritical CO_2 refrigeration systems in Climate Zones 10 through 15 that achieve at least equal energy savings through alternative means approved by the Executive Director.

Exception 2 to Section 902.1.5.1: The requirement of Section 902.1.5.1 does not apply to air-cooled gas coolers on systems with a total heat of rejection of:

i. 20 tons (240,000 Btu/h) or less for systems serving freezers; or

ii. 40 tons (480,000 Btu/h) or less for systems serving coolers.

~~**902.1.5.2** *[Section 120.6(b)5B]* **Design leaving gas temperature - air-cooled gas coolers.**~~

~~Design leaving gas temperature for air-cooled *gas coolers* shall be less than or equal to the design dry-bulb temperature plus 6°F.~~

902.1.5.~~3~~2 *[Section 120.6(b)5C]* **Design leaving gas temperature - adiabatic gas coolers.**

Design leaving gas temperature for adiabatic *gas coolers* necessary to reject the design *total heat of rejection* of a refrigeration system assuming *dry mode* performance shall be less than or equal to the design dry-bulb temperature plus 15°F.

902.1.5.~~4~~3 *[Section 120.6(b)5D]* **Variable speed fans and fan controls.**

All *gas cooler* fans shall be continuously variable speed, with the speed of all fans serving a common *condenser* high side controlled in unison.

902.1.5.~~5~~4 *[Section 120.6(b)5E]* **Gas cooler pressure control.**

While operating below the critical point, the *gas cooler* pressure shall be controlled in accordance with Section 902.1.1.2, 902.1.1.3, or 902.1.1.4 *[Section 120.6(b)1B, 120.6(b)1C, or 120.6(b)1D]*.

902.1.5.~~6~~5 *[Section 120.6(b)5F]* **Gas cooler pressure reset.**

While operating above the critical point, the *gas cooler* pressure setpoint shall be reset based on ambient conditions such that the system efficiency is maximized.

902.1.5.~~7~~6 *[Section 120.6(b)5G]* **Condensing temperature setpoint.**

The minimum *condensing temperature* setpoint shall be less than or equal to 60°F for air-cooled *gas coolers*, evaporative-cooled *gas coolers*, adiabatic *gas coolers*, air or water-cooled *fluid coolers* or cooling towers.

Exception to Section 902.1.5.~~7~~6: *Transcritical CO_2 refrigeration systems* with a design intermediate saturated suction temperature greater than or equal to 30°F shall have a minimum *condensing temperature* setpoint of 70°F or less.

902.1.5.~~8~~7 *[Section 120.6(b)5H]* **Condenser efficiency.**

Fan-powered *gas coolers* shall meet the *condenser* efficiency requirements listed in Table 902.1-B *[Table 120.6-E]*. *Gas cooler* efficiency is defined as the *total heat of rejection* (THR) capacity divided by all electrical input power (fan power at 100-percent fan speed).

TABLE 902.1-B [Table 120.6-E] TRANSCRITICAL CO_2 FAN-POWERED GAS COOLERS – MINIMUM EFFICIENCY REQUIREMENTS

CONDENSER TYPE	REFRIGERANT TYPE	MINIMUM EFFICIENCY	RATING CONDITION
Outdoor Air-Cooled	Transcritical CO_2	160 Btuh/watt	1400 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature
Adiabatic Dry Mode	Transcritical CO_2	90 Btuh/watt	1100 psig, 100°F Outlet Gas Temperature, 90°F Outdoor Dry bulb Temperature

3.1.1.1 *Reference Appendices*

NA7.10 Refrigerated Warehouse Refrigeration System Acceptance Tests

Refrigerated Warehouse Refrigeration System acceptance tests shall be in accordance with *either* NA7.10.1 or NA7.10.2.

NA7.10.1 Refrigerated Warehouse Refrigeration System Acceptance Tests, Option 1

…

NA7.10.1.1Electric Resistance Underslab Heating System

NA7.10.1.1.1 Construction Inspection

…

NA7.10.1.1.2 Functional Testing

…

NA7.10.1.2 Evaporators and Evaporator Fan Motor Variable Speed Control

NA7.10.1.2.1 Construction Inspection

…

NA7.10.1.2.2 Functional Testing

…

NA7.10.1.3 Condensers and Condenser Fan Motor Variable Speed Control

NA7.10.1.3.1 Evaporative Condensers and Condenser Fan Motor Variable Speed Control

NA7.10.1.3.1.1 Construction Inspection

…

NA7.10.1.3.1.2 Functional Testing

…

NA7.10.1.3.2 Air-Cooled Condensers and Condenser Fan Motor Variable Speed Control

…

NA7.10.1.3.2.1 Construction Inspection

…

NA7.10.1.3.2.2 Functional Testing

…

NA7.10.1.3.3 Adiabatic Condensers and Condenser Fan Motor Variable Speed Control

…

NA7.10.1.3.3.1 - Construction Inspection

…

NA7.10.1.3.3.2 Functional Testing

…

NA7.10.1.4 Variable Speed Screw Compressors

…

NA7.10.1.4.1 Construction Inspection

…

NA7.10.1.4.2 - Functional Testing

…

NA7.10.2 Refrigerated Warehouse Refrigeration System Acceptance Tests, Option 2

~~The measurement devices used to verify the refrigerated warehouse controls shall be calibrated once every two years using a NIST traceable reference. The calibrated measurement devices to be used in these acceptance tests are called the "standard"~~

~~and shall have the following measurement tolerances: The temperature measurement devices shall be calibrated to ±0.7°F between -30°F and 200°F. The pressure measurement devices shall be calibrated to ±2.5 psi between 0 and 500 psig. The relative humidity (RH) measurement devices shall be calibrated to ±1% between 5% and 90% RH.~~

NA7.10.2.1 Electric Resistance Underslab Heating System

NA7.10.2.1.1 Construction Inspection

Prior to functional testing, verify and document the following for all electric resistance underslab heating systems:

(a) Verify that summer on-peak period is programmed into all underslab heater controls to meet the requirements of Section 120.6(a)2.

NA7.10.2.1.2 Functional Testing

Step 1: Using the control system, lower slab temperature setpoint. Verify and document the following using an electrical test meter:

(b) The underslab electric resistance heater is off.

Step 2: Using the control system, raise the slab temperature setpoint. Verify and document the following using an electrical test meter:

(c) The underslab electric resistance heater is on.

Step 3: Using the control system, change the control system's time and date corresponding to the local utility's summer on-peak period. If control system only accounts for time, set system time corresponding to the local utility's summer on-peak period. Verify and document the following using an electrical test meter:

(d) The underslab electric resistance heater is off.

Step 4: Restore system to correct schedule and control setpoints.

NA7.10.2.2 Evaporators and Evaporator Fan Motor Variable Speed Control

NA7.10.2.2.1 Construction Inspection

Prior to functional testing, document the following on all evaporators:

(e) All refrigerated space temperature sensors used for control ~~are verified to read accurately (or provide an appropriate offset) using a temperature standard~~ have been either factory or field calibrated.

(f) All refrigerated space humidity sensors used for control ~~are verified to read accurately (or provide an appropriate offset) using a humidity standard~~ have been either factory or field calibrated.

(g) All refrigerated space temperature and humidity sensors are verified to be mounted in a location away from direct evaporator discharge air draft.

(h) Verify that all fan motors are operational and rotating in the correct direction.

(i) Verify that fan speed control is operational and connected to evaporator fan motors.

(j) Verify that all speed controls are in "auto" mode.

NA7.10.2.2.2 Functional Testing

Conduct and document the following functional tests on all evaporators.

Step 1: Measure current space temperature or humidity. Program this temperature or humidity as the test temperature or humidity setpoint into the control system for the functional test steps. Allow 5 minutes for system to normalize.

Step 2: Using the control system, lower test temperature or humidity setpoint in 1 degree or 1% RH increments below any control dead band range until:

(a) Evaporator fan controls modulate to increase fan motor speed.

(b) Evaporator fan motor speed increases in response to controls.

(c) Verify and document the above.

Step 3: Using the control system, raise the test temperature or humidity setpoint in 1 degree or 1% RH increments above any control dead band range until fans go to minimum speed. Verify and document the following:

(d) Evaporator fan controls modulate to decrease fan motor speed.

(e) Evaporator fan motor speed decreases in response to controls.

(f) Minimum fan motor control speed (rpm or percent of full speed).

Step 4: Restore control system to correct control setpoints.

NA7.10.2.3 Condensers and Condenser Fan Motor Variable Speed Control

NA7.10.2.3.1 Evaporative Condensers and Condenser Fan Motor Variable Speed Control

NA7.10.2.3.1.1 Construction Inspection

Prior to functional testing, document the following:

(a) Verify the minimum condensing temperature control setpoint is at or below 70°F.

(b) Verify the master system controller saturated condensing temperature input is the temperature equivalent reading of the condenser pressure sensor.

(c) Verify all drain leg pressure regulator valves are set below the minimum condensing temperature/pressure setpoint.

(d) Verify all receiver pressurization valves, such as the outlet pressure regulator (OPR), are set lower than the drain leg pressure regulator valve setting.

(e) Verify all condenser inlet and outlet pressure sensors ~~read accurately (or provide an appropriate offset) using a pressure standard~~ have been either factory or field calibrated.

(f) Verify all ambient dry bulb temperature sensors used by controller ~~read accurately (or provide an appropriate offset) using a temperature standard~~ have been either factory or field calibrated.

(g) Verify all relative humidity sensor used by controller ~~read accurately (or provide an appropriate offset) using RH standard~~ have been either factory or field calibrated.

(h) Verify all temperature sensors used by the controller are mounted in a location that is not exposed to direct sunlight.

(i) Verify that all sensor readings used by the condenser controller convert or calculate to the correct conversion units at the controller (e.g., saturated pressure reading is correctly converted to appropriate saturated temperature; dry bulb and relative humidity sensor readings are correctly converted to wet bulb temperature, etc.).

(j) Verify that all fan motors are operational and rotating in the correct direction.

(k) Verify that all condenser fan speed controls are operational and connected to condenser fan motors to operate in unison the fans serving a common condenser loop.

(l) Verify that all speed controls are in "auto" mode.

NA7.10.2.3.1.2 Functional Testing

~~Note: The system cooling load must be sufficiently high to run the test. Artificially increase evaporator loads or decrease compressor capacity (manually turn off compressors, etc.) as may be required to perform the Functional Testing.~~ Note: Where a step calls for a change in operating condition, the technician may simulate that condition by adjusting the control system setpoint or by simulating the associated sensor input to the controller.

Step 1: Override any heat reclaim, floating suction pressure, floating head pressure and defrost functionality before performing functional tests.

Step 2:

(a) Document current outdoor ambient air dry bulb and wet bulb temperatures, relative humidity and refrigeration system condensing temperature/condensing pressure readings from the control system.

(b) Calculate and document the temperature difference (TD), defined as the difference between the wet bulb temperature and the refrigeration system saturated condensing temperature (SCT).

(c) Document current head pressure control setpoint.

Step 3: Using the desired condenser fan motor cycling or head pressure control strategy, program into the control system a setpoint equal to the reading or calculation obtained in Step 2. This will be referred to as the "test setpoint." Allow 5 minutes for condenser fan speed to normalize.

Step 4: Using the control system, raise the test setpoint in 1 degree (or 3 psi) increments until the condenser fan control modulates to minimum fan motor speed. Verify and document the following:

(a) Fan motor speed decreases.

(a) All condenser fan motors serving common condenser loop decrease speed in unison in response to controller output.

(b) Minimum fan motor control speed (rpm or percent of full speed).

(c) If the refrigeration system is already operating at minimum saturated condensing temperature/head pressure, reverse Steps 4 and 5.

Step 5: Using the control system, lower the test setpoint in 1 degree (or 3 psi) increments until the condenser fan control modulates to increase fan motor speed. Verify and document the following:

(d) Fan motor speed increases.

(e) All condenser fan motors serving common condenser loop increase speed in unison in response to controller output.

Step 6: Document the current minimum condensing temperature setpoint. Using the control system, change the minimum condensing temperature setpoint to a value greater than the current operating condensing temperature. Verify and document the following:

(f) Condenser fan controls modulate to decrease capacity.

(g) All condenser fans serving common condenser loop modulate in unison.

(h) Condenser fan controls stabilize within a 5 minute period.

Step 7: Using the control system, reset the system head pressure controls, fan motor controls and minimum condensing temperature control setpoint to original settings documented in Steps 3 and 6.

Step 8: Restore any heat reclaim, floating suction pressure, floating head pressure and defrost functionality. Reset the minimum condensing temperature setpoint to the value documented in Step 6.

NA7.10.2.3.2 Air-Cooled Condensers and Condenser Fan Motor Variable Speed Control

Conduct and document the following functional tests on all air-cooled condensers.

NA7.10.2.3.2.1 Construction Inspection

Prior to functional testing, document the following:

(a) Verify that the minimum condensing temperature control setpoint is at or below 70°F.

(b) Verify that the master system controller saturated condensing temperature input is the temperature equivalent reading of the condenser pressure sensor.

(c) Verify all drain leg pressure regulator valves are set below the minimum condensing temperature/pressure setpoint.

(d) Verify all receiver pressurization valves, such as the outlet pressure regulator (OPR), are set lower than the drain leg pressure regulator valve setting.

(e) Verify all condenser inlet and outlet pressure sensors ~~read accurately (or provide an appropriate offset) using a pressure standard~~ have been either factory or field calibrated.

(f) Verify all ambient dry bulb temperature sensors used by controller ~~read accurately (or provide an appropriate offset) using temperature standard~~ have been either factory or field calibrated.

(g) Verify all temperature sensors used by the controller are mounted in a location that is not exposed to direct sunlight.

(h) Verify that all sensor readings used by the condenser controller convert or calculate to the correct conversion units at the controller (e.g., saturated pressure reading is correctly converted to appropriate saturated temperature, etc.).

(i) Verify that all fan motors are operational and rotating in the correct direction.

(j) Verify that all condenser fan speed controls are operational and connected to condenser fan motors to operate in unison the fans serving a common condenser loop.

(k) Verify that all speed controls are in "auto" mode.

NA7.10.2.3.2.2 Functional Testing

~~Note: The system cooling load must be sufficiently high to run the test. Artificially increase evaporator loads or decrease compressor capacity (manually turn off compressors, etc.) as may be required to perform the Functional Testing.~~ Note: Where a step calls for a change in operating condition, the technician may simulate that condition by adjusting the control system setpoint or by simulating the associated sensor input to the controller.

Step 1: Override any heat reclaim, floating suction pressure, floating head pressure and defrost functionality before performing functional tests. Document current outdoor ambient air dry bulb temperature and refrigeration system condensing temperature/condensing pressure readings from the control system.

Step 2: Calculate and document the temperature difference (TD), defined as the difference between the dry bulb temperature and the refrigeration system saturated condensing temperature (SCT). Document current head pressure control setpoint.

Step 3: Using the desired condenser fan motor cycling or head pressure control strategy, program into the control system a setpoint equal to the reading or calculation obtained in Step 2. This will be referred to as the "test setpoint." Allow 5 minutes for condenser fan speed to normalize.

Step 4: Using the control system, raise the test setpoint in 1 degree (or 3 psi) increments until the condenser fan control modulates to minimum fan motor speed. Verify and document the following:

(a) Fan motor speed decreases.

(b) All condenser fan motors serving common condenser loop decrease speed in unison in response to controller output.

(c) Minimum fan motor control speed (rpm or percent of full speed).

(d) If the refrigeration system is already operating at minimum saturated condensing temperature/head pressure, reverse Steps 4 and 5.

Step 5: Using the control system, lower the test setpoint in 1 degree (or 3 psi) increments until the condenser fan control modulates to increase fan motor speed. Verify and document the following:

(a) Fan motor speed increases.

(b) All condenser fan motors serving common condenser loop increase speed in unison in response to controller output.

Step 6: Document current minimum condensing temperature setpoint. Using the control system change the minimum condensing temperature setpoint to a value greater than the current operating condensing temperature. Verify and document the following:

(a) Condenser fan controls modulate to decrease capacity.

(b) All condenser fans serving common condenser loop modulate in unison.

(c) Condenser fan controls stabilize within a 5 minute period.

Step 7: Using the control system, reset the system head pressure controls, fan motor controls and minimum condensing temperature control setpoint to original settings documented in Steps 2 and 6.

Step 8: Restore any heat reclaim, floating suction pressure, floating head pressure and defrost functionality. Reset the minimum condensing temperature setpoint to the value documented in Step 6.

NA7.10.2.3.3 Adiabatic Condensers and Condenser Fan Motor Variable Speed Control

Conduct and document the following functional tests on all adiabatic condensers.

NA7.10.2.3.3.1 Construction Inspection

Prior to functional testing, document the following:

(a) Verify the control system minimum Saturated Condensing Temperature (SCT) setpoint is at or below 70°F.

(b) Verify the control system maximum SCT setpoint (if used) is at or near the system design SCT.

(c) Verify accuracy of refrigerant pressure-temperature conversions and consistent use of either temperature or pressure for the controlled variable setpoint in the control system.

(d) Verify the discharge pressure sensor (or condenser pressure if used) ~~reads accurately, using a National Institute of Standards and Technology~~

~~(NIST) traceable reference pressure gauge or meter. At the minimum, the discharge pressure sensor accuracy shall be verified at two different pressures within the typical operating range. Calibrate if needed. Replace if outside manufacturer's recommended calibration range.~~ has been either factory or field calibrated.

(e) Verify the ambient dry bulb temperature ~~using a NIST traceable instrument, including verification of at least two different ambient readings. Calibrate if needed. Replace if outside manufacturer's recommended calibration range.~~ sensor has been either factory or field calibrated.

(f) Verify all ambient dry bulb temperature sensors are not mounted in direct sunlight or is provided within a suitable solar shield.

(g) Verify that all sensor readings used by the condenser controller convert or calculate to the correct conversion units and are displayed at the controller (e.g., observed pressure reading is correctly converted to appropriate saturated temperature, etc.).

(h) Verify that all fan motors are operational and rotating in the correct direction.

(i) Verify that all condenser fan speed controls operate automatically in response to changes in both pressure (SCT) and ambient temperature.

NA7.10.2.3.3.2 Functional Testing

~~Note: The system cooling load must be sufficiently high, and ambient conditions sufficiently below design, to operate with all condenser fans in operation and observe controls in average conditions. Be cognizant of weather conditions in scheduling testing and, if necessary and possible, arrange to artificially increase or decrease evaporator loads in order to perform the Functional Testing at typical system conditions.~~ Note: Where a step calls for a change in operating condition, the technician may simulate that condition by adjusting the control system setpoint or by simulating the associated sensor input to the controller. The functional test shall be performed in dry mode.

Step 1: Verify mechanical controls and other strategies will not affect tests.

(a) Verify condenser pressure low-limit holdback and/or bypass regulating valves, if any, are set below the minimum SCT setpoint. Condenser pressure controls valves will cause fans to operate at 100% speed if they are not set below the minimum SCT value. ~~In warm weather, this may require setting out of range, and deferring valve settings until cold weather allows valves to be adjusted.~~

(b) Turn off any heat reclaim controls and any intermittent defrost pressure offset strategies that would affect condenser setpoint control.

(c) Document adiabatic mode switching setpoints, if necessary for test temporarily change the adiabatic mode setpoint such that the condenser operates in dry mode. Verify that the adiabatic pads are completely dry before beginning tests.

Step 2: Operate in control range and verify

(a) Verify the condenser control value is operating in the variable setpoint control range, i.e., above the minimum SCT setpoint and below the maximum SCT setpoint.

i. If necessary, increase or decrease the system load.

ii. If necessary, during low load or low ambient conditions with system observed at the minimum SCT, temporarily adjust the minimum SCT to a lower value, if the refrigeration system design will allow, or increase the control TD to result in a higher control value.

(b) Observe control operation for at least 30 minutes to confirm stable control operation, as shown by condenser fan speed varying as compressor capacity changes, and not ranging from maximum to minimum fan speed or constant "hunting". If required, adjust control response setpoints to achieve stable operation. Since condenser control settings require fine-tuning over time, this is often accomplished using control system history or visual trends, showing one hourly and daily operation.

Step 3: Identify control Temperature Difference

(a) Record the current outdoor ambient air dry bulb and refrigeration system condensing temperature/condensing pressure readings from the control system. Note whether discharge pressure or a dedicated condenser pressure sensor is used for condenser pressure control.

(b) Document current head pressure control setpoints, including the Temperature Difference (TD) setpoint.

(c) Calculate and record the actual observed TD, defined as the difference between the dry bulb temperature and the refrigeration system SCT.

(d) Confirm agreement between the current control system TD setpoint and the observed TD. If values are different, address and correct controls system methods.

Step 4: Test adjusted control Temperature Difference (Setpoint1).

(a) Enter a smaller TD value into the control system sufficient enough to cause an observable response, such as 1 to 2 degrees smaller, but not small enough to cause the system to operate continuously at 100% fan speed. Record this value as TD Test Setpoint 1.

(b) Observe change in control system operation which should include an increase in fan speed and a decrease in condensing temperature.

(c) Allow time for the control system to achieve stable operation.

(d) Document current head pressure control setpoints, including the TD setpoint.

(e) Calculate and record the actual observed TD, defined as the difference between the wet bulb temperature and the refrigeration system SCT.

(f) Confirm agreement between the current control system TD setpoint and the observed TD. If values are different, address and correct control system methods.

Step 5: Test adjusted control Temperature Difference (Setpoint2) Enter a TD value into the control system that is different from TD Test Setpoint1, sufficient enough to cause an observable response. Record this value a TD Test Setpoint2.

(a) Observe change in control system operation which should include an increase in fan speed and a decrease in condensing temperature.

(b) Allow time for the control system to achieve stable operation.

(c) Record the current outdoor ambient dry bulb temperature.

(d) Record the current refrigeration system condensing temperature/condensing pressure readings from the control system.

(e) Document current head pressure control setpoints, including the TD setpoint.

(f) Calculate and record the actual observed TD, defined as the difference between the dry bulb temperature and the refrigeration system SCT.

(g) Confirm agreement between the current control system TD setpoint and the observed TD. If values are different, address and correct control system methods.

Step 6: Document current minimum condensing temperature setpoint. Using the control system change the minimum condensing temperature setpoint to a value greater than the current operating condensing temperature. Verify and document the following:

(a) Condenser fan controls modulate to decrease capacity.

(b) All condenser fans serving common condenser loop modulate in unison.

(c) Condenser fan controls stabilize within a 5 minute period.

Step 7: Using the control system, reset the system head pressure controls, fan motor controls and minimum condensing temperature control setpoint to original settings documented in Steps 3 and 6.

Step 8: Restore any heat reclaim, floating suction pressure, floating head pressure and defrost functionality. Reset the minimum condensing temperature setpoint to the value documented in Step 6.

NA7.10.2.4 Variable Speed Screw Compressors

Conduct and document the following functional tests on all variable-speed screw compressors.

NA7.10.2.4.1 Construction Inspection

Prior to functional testing, document the following:

(a) Verify all single open-drive screw compressors dedicated to a suction group have variable speed control.

(b) Verify all compressor suction and discharge pressure sensors ~~read accurately (or provide an appropriate offset) using a standard~~ have been either factory or field calibrated.

(c) Verify all input or control temperature sensors used by controller ~~read accurately (or provide an appropriate offset) using temperature standard~~ have been either factory or field calibrated.

(d) Verify that all sensor readings used by the compressor controller convert or calculate to the correct conversion units at the controller (e.g., saturated pressure reading is correctly converted to appropriate saturated temperature, etc.).

(e) Verify that all compressor speed controls are operational and connected to compressor motors.

(f) Verify that all speed controls are in "auto" mode.

(g) Verify that compressor panel control readings for "RPMs," "% speed," "kW", and "amps" match the readings from the PLC or other control systems.

(h) Verify that compressor nameplate data is correctly entered into the PLC or other control system.

NA7.10.2.4.2 Functional Testing

~~Note: The system cooling load must be sufficiently high to run the test. Artificially increase or decrease evaporator loads (add or shut off zone loads, change setpoints, etc.) as may be required to perform the Functional Testing.~~
Note: Where a step calls for a change in operating condition, the technician may simulate that condition by adjusting the control system setpoint or by simulating the associated sensor input to the controller.

Step 1: Override any heat reclaim, floating suction pressure, floating head pressure and defrost functionality before performing functional tests.

Step 2: Measure and document the current compressor operating suction pressure and saturated suction temperature.

Step 3: Document the suction pressure/saturated suction temperature setpoint. Program into the control system a target setpoint equal to the current operating condition measured in Step 2. Allow 5 minutes for system to normalize. This will be referred to as the "test suction pressure/saturated suction temperature setpoint."

Step 4: Using the control system, raise the test suction setpoint in 1 psi increments until the compressor controller modulates to decrease compressor speed. Verify and document the following:

(a) Compressor speed decreases.

(b) Compressor speed continues to decrease to minimum speed.

(c) Any slide valve or other unloading means does not unload until after the compressor has reached its minimum speed (RPM).

Step 5: Using the control system, lower the test suction setpoint in 1 psi increments until the compressor controller modulates to increase compressor speed. Verify and document the following:

(a) Any slide valve or other unloading means first goes to 100 percent before compressor speed increases from minimum.

(b) Compressor begins to increase speed.

(c) Compressor speed continues to increase to 100 percent.

Step 6: Using the control system, program the suction target setpoints back to original settings as documented in Step 3.

Step 7: Restore any heat reclaim, floating suction pressure, floating head pressure and defrost functionality.

Transcritical CO_2 Systems Acceptance Tests

NA7.20 Transcritical CO2 Systems Acceptance Tests

Acceptance tests for Transcritical CO2 Systems shall conform with *either* NA7.20.1 or NA7.20.2.

NA7.20.1.1 Transcritical CO2 Systems Acceptance Tests, Option 1

NA7.20.1 Transcritical CO_2 Gas Cooler and Gas Cooler Fan Motor Variable Speed Control for Refrigerated Warehouses and Commercial Refrigeration

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NA7.20.1.1.1 Air-Cooled and Adiabatic Gas Coolers and Gas Cooler Fan Motor Variable Speed Control

…

NA7.20.1.1.1.1 Construction Inspection

…

NA7.20.1.1.1.2 Functional Testing (Option A: Subcritical Operation)

…

NA7.20.1.1.1.3 Functional Testing (Option B: Supercritical Operation)

…

NA7.20.2 Transcritical CO2 Systems Acceptance Tests, Option 2

NA7.20.2.1 Transcritical CO_2 Gas Cooler and Gas Cooler Fan Motor Variable Speed Control for Refrigerated Warehouses and Commercial Refrigeration

The purpose of these tests is to confirm proper operation of gas cooler control, including variable speed fan operation and variable setpoint control logic, which are both important elements of floating head pressure control, with the intent to operate with the lowest total system energy (considering both compressors and gas cooler fan power) through the course of the year.

Note: Transcritical CO_2 refrigeration systems are unique in that they can operate in one of two modes: subcritical operation and supercritical operation. Subcritical operation generally occurs during periods where ambient conditions are below 75°F to 80°F, where high pressure CO_2 vapor will condense in the gas cooler. Supercritical operation generally occurs during periods where ambient conditions are above 75°F to 80°F, where the high pressure CO_2 vapor will not condense (or partially condense) in the gas cooler, and pressure and temperature can vary semi-independently during the heat rejection process. Because these two modes of operation are based on ambient conditions, it may not be possible for the field technician to observe both subcritical and supercritical control strategies during a single acceptance test.

The field technician shall perform either the functional test outlined in NA7.20.1.1.2 or NA7.20.1.1.3 depending on the ambient conditions and resulting system operating mode at the time of the test. The construction inspection must be completed regardless of ambient conditions.

The following test methods are general in nature, with the understanding that refrigeration systems are commonly custom designed, with many design choices, as well as varying load profiles. For all of these reasons, a thorough understanding of both refrigeration system design and refrigeration control system operation is necessary to effectively conduct these tests.

~~The measurement devices used to verify the refrigeration system controls shall be calibrated to a NIST traceable reference, with a calibration reference dated within the past two years. The calibrated measurement devices to be used in these acceptance tests are called the "standard" and shall have the following measurement tolerances: The temperature measurement devices shall be calibrated to ±0.7°F between -30°F and 200°F. The pressure measurement devices shall be calibrated to ±7.5 psi between 0 and 1500 psig.~~

NA7.20.2.1.1 Air-Cooled and Adiabatic Gas Coolers and Gas Cooler Fan Motor Variable Speed Control

Conduct and document the following functional tests on all air-cooled and adiabatic gas coolers.

NA7.20.2.1.1.1 Construction Inspection

Prior to functional testing, verify and document the following:

(a) Verify the control system minimum saturated condensing temperature (SCT) setpoint is at or below 60°F. If the design saturated suction temperature (SST) of the intermediate suction group is greater than or equal to 30°F, verify the control system SCT setpoint is at or below 70°F.

(b) Verify accuracy of refrigerant pressure-temperature conversions and consistent use of either temperature or pressure for the controlled variable setpoint in the control system.

1. The condensing temperature has an equivalent pressure during subcritical operation.

2. Either pressure or temperature may be used in the control system as the controlled variable to maintain gas cooler pressure (condensing temperature) during subcritical operation, as long as the setpoint value is similarly expressed in pressure or temperature.
3. Documentation may be achieved through pictures of control system screens or control system documentation, supported by sample calculations of observed pressures or temperatures and associated conversion values, as available in the control system interface.

(c) Verify the gas cooler outlet temperature sensor ~~reads accurately, using a NIST traceable instrument, including verification of at least two different gas cooler outlet readings. Calibrate if needed. Replace if outside manufacturer's recommended calibration range.~~ has been either factory or field calibrated. If multiple gas coolers are installed in parallel, ensure sensor is installed on the common header.

(d) Verify the discharge pressure sensor (or gas cooler pressure if used) ~~reads accurately, using a NIST traceable reference pressure gauge or meter, and with pressure checked for at least two pressures within the typical operating range. Calibrate if needed. Replace if outside manufacturers recommended calibration range~~ has been either factory or field calibrated.

(e) Verify the ambient dry bulb temperature ~~using a NIST traceable instrument, including verification of at least two different ambient readings. Calibrate if needed. Replace if outside manufacturer's recommended calibration range.~~ sensor has been either factory or field calibrated. If the ambient dry bulb temperature sensor is installed between the adiabatic pad and the gas cooler coil for adiabatic gas coolers, verification must be performed when operating in "dry" mode.

(f) Verify the ambient dry bulb temperature is not mounted in direct sunlight or is provided with a suitable solar shield.

(g) Verify that all sensor readings used by the gas cooler controller display correct values at the controller, as well as derived values (e.g., observed pressure is correctly converted saturation temperature for CO_2).

(h) Verify that all fan motors are operational and rotating in the correct direction.

(i) Verify that gas cooler fan speed controls are operational and controlling all gas cooler fan motors in unison.

(j) Verify that all speed controls operate automatically in response to changes in pressure, gas cooler outlet temperature, and ambient dry bulb or precool air temperature.

(k) Verify the installation of the gas cooler holdback valve, which may be located near the inlet of the intermediate pressure vessel or near the outlet of the gas cooler.

NA7.20.2.1.1.2 Functional Testing (Option A: Subcritical Operation)

~~Planning: The system cooling load must be sufficiently high, and ambient conditions sufficiently below the critical point, to operate subcritically with all gas cooler fans in operation and observe controls in average conditions. Account for weather conditions in scheduling testing by, if necessary, artificially increasing or decreasing evaporator loads in order to perform the Functional Testing at typical system conditions.~~ Planning: Perform this test when the system is operating subcritically. Where a step calls for a change in operating condition, the technician may simulate that condition by adjusting the control system setpoint or by simulating the associated sensor input to the controller.

Step 1: Verify mechanical controls and other strategies will not affect tests:

(a) Turn off any heat reclaim controls and any intermittent defrost pressure offset strategies that would affect gas cooler setpoint control.

(b) If testing an adiabatic gas cooler, adjust setpoints to ensure that the gas cooler stays in "dry" mode or "precool" mode consistently throughout the test.

Step 2: Operate in control range and verify stable control:

(a) Verify the gas cooler control value is operating in the variable setpoint control range, i.e., above the minimum SCT setpoint and below the maximum SCT setpoint.

- If necessary, increase or decrease the system load.
- If necessary, during low load or low ambient conditions with system observed at the minimum SCT, temporarily adjust the minimum SCT to a lower value, if the refrigeration system design will allow, or increase the control TD to result in a higher control value.

(b) Observe control operation for at least 30 minutes to confirm stable control operation, as shown by gas cooler fan speed varying as compressor capacity changes, and not ranging from maximum to minimum fan speed or constant "hunting". If required, adjust control response setpoints to achieve stable operation. Note: Since gas cooler control settings require fine-tuning over time, this is often accomplished using control system history or visual trends, showing one hourly and daily operation.

Step 3: Identify control TD:

(a) Record the current outdoor ambient air dry bulb or precool air temperature and refrigeration system condensing temperature/condensing pressure readings from the control system. Note whether discharge pressure or a dedicated gas cooler pressure sensor is used for gas cooler pressure control.

(b) Document current head pressure control setpoints, including the TD setpoint.

(c) Calculate and record the actual observed temperature difference (TD), defined as the difference between the ambient dry bulb temperature or precool air temperature and the refrigeration system saturated condensing temperature (SCT).

(d) Confirm agreement between the current control system TD setpoint and the observed TD. If values are different, address and correct control system methods.

Step 4: Test adjusted control TD:

(a) Enter a smaller TD value into the control system, sufficient to cause an observable response, such as 1-2 degrees smaller, but not small enough to cause system to operate continuously at 100% fan speed. Record this value as TD Test Setpoint 1.

(b) Observe change in control system operation which should include an increase in fan speed and a decrease in condensing temperature.

(c) Allow time for the control system to achieve stable operation.

(d) Document current head pressure control setpoints, including the TD setpoint.

(e) Calculate and record the actual observed temperature difference (TD), defined as the difference between the ambient dry bulb or precool air temperature and the refrigeration system saturated condensing temperature (SCT).

(f) Confirm agreement between the current control system TD setpoint and the observed TD. If values are different, address and correct control system methods.

(g) Perform the above test sequence with a second TD value, recorded as TD Test Setpoint 2, and record the same values above to confirm agreement between the current control system TD setpoint and the observed TD. If needed perform corrective actions and repeat testing until variable setpoint control can be confirmed and documented.

Step 5: Verify and document all fans operate in unison down to minimum SCT:

(a) Document that all fans are in operation, fan speed, actual SCT and control system minimum SCT setpoint, by recording control system screens or trends along with observations.

1. In cool weather and/or light loads, this may be the observed operation during testing without need to manipulate system setpoints.
2. In warmer weather and/or higher loads, the control system minimum SCT value can be increased slowly to a value equal to, and then above, the current operating condition, in order to observe the fans operating in unison and fan speeds dropping as the minimum SCT setpoint is achieved.

Step 6: Restore setpoints:

(a) Restore any heat reclaim or defrost functionality that was turned off to allow testing.

(b) Reset the minimum condensing temperature setpoint if it was adjusted during Step 5.

(c) Reset adiabatic mode controls to original values.

NA7.20.2.1.1.3 Functional Testing (Option B: Supercritical Operation)

~~Planning: Ambient conditions must be sufficiently above the critical point to operate supercritically. Account for weather conditions in scheduling testing by, if necessary, artificially increasing or decreasing evaporator loads in order to perform the Functional Testing at typical system conditions.~~ Planning: Perform this test when the system is operating supercritically. Where a step calls for a change in operating condition, the technician may simulate that condition by adjusting the control system setpoint or by simulating the associated sensor input to the controller.

Step 1: Verify mechanical controls and other strategies will not affect tests:

(a) Turn off any heat reclaim controls and any intermittent defrost pressure offset strategies that would affect gas cooler setpoint control.

(b) If testing an adiabatic gas cooler, adjust setpoints to ensure that the gas cooler stays in "dry" mode or "precool" mode consistently throughout the test.

Step 2: Operate in supercritical mode and verify pressure control:

(a) Observe operation for at least 30 minutes or reference control system history or visual trends to verify the gas cooler holdback valve modulates its opening in response to changes in ambient dry bulb or precool air temperature resulting in a change in gas cooler pressure. Reference the original equipment manufacturer operating manual or sequence of operation descriptions to confirm the observed variation in the pressure setpoint is consistent with the design control strategy.

Step 3: Restore setpoints:

(a) Restore any heat reclaim or defrost functionality that was turned off to allow testing.

Reset adiabatic mode controls to original values.